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ALCO 251

REVIEW OF DESIGN FROM 1951 THRU 1971

PREPARED BY
F. C. TROMEL

AUGUST 1972

FIG 2

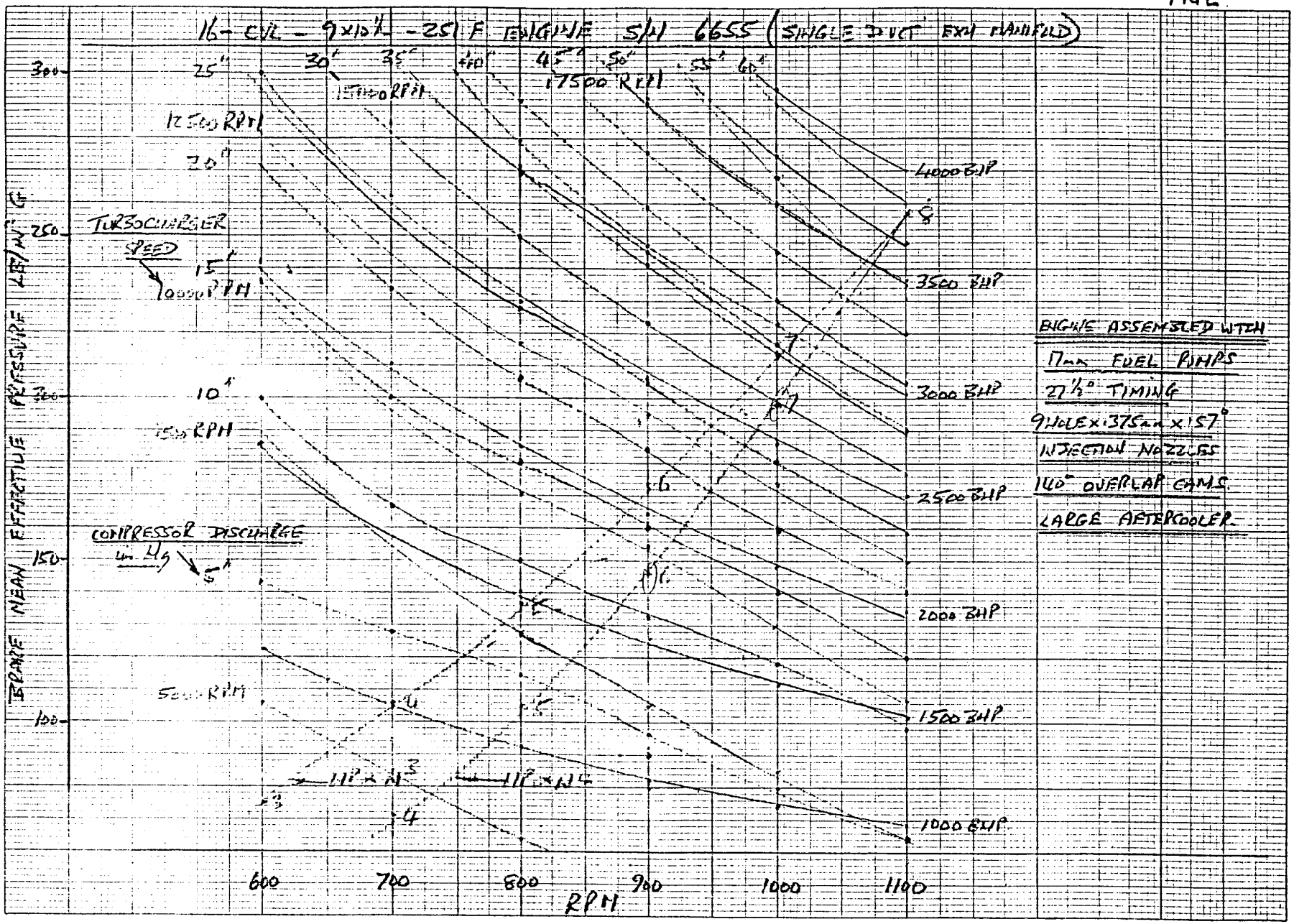

16 - CYL - 9 x 10½ - 25 F ENGINE S/N 6655 (SINGLE DUCT EXH MANIFOLD)
25"
30"
35"
40"
45"
50"
55"
60"
1500 RPM
1750 RPM
2000 RPM
TURBOCHARGER SPEED
15
10
5
COMPRESSOR DISCHARGE in Hg
4000 BHP
3500 BHP
3000 BHP
2500 BHP
2000 BHP
1500 BHP
1000 BHP
BRAKE MEAN EFFECTIVE PRESSURE LBS/SQ IN
300
250
200
150
100
600
700
800
900
1000
1100
RPM
ENGINE ASSEMBLED WITH
17mm FUEL PUMPS
27½° TIMING
9 HOLE x .375mm x 157°
INJECTION NOZZLES
140° OVERLAP CAMS
LARGE AFTERCOOLER

FIG 3

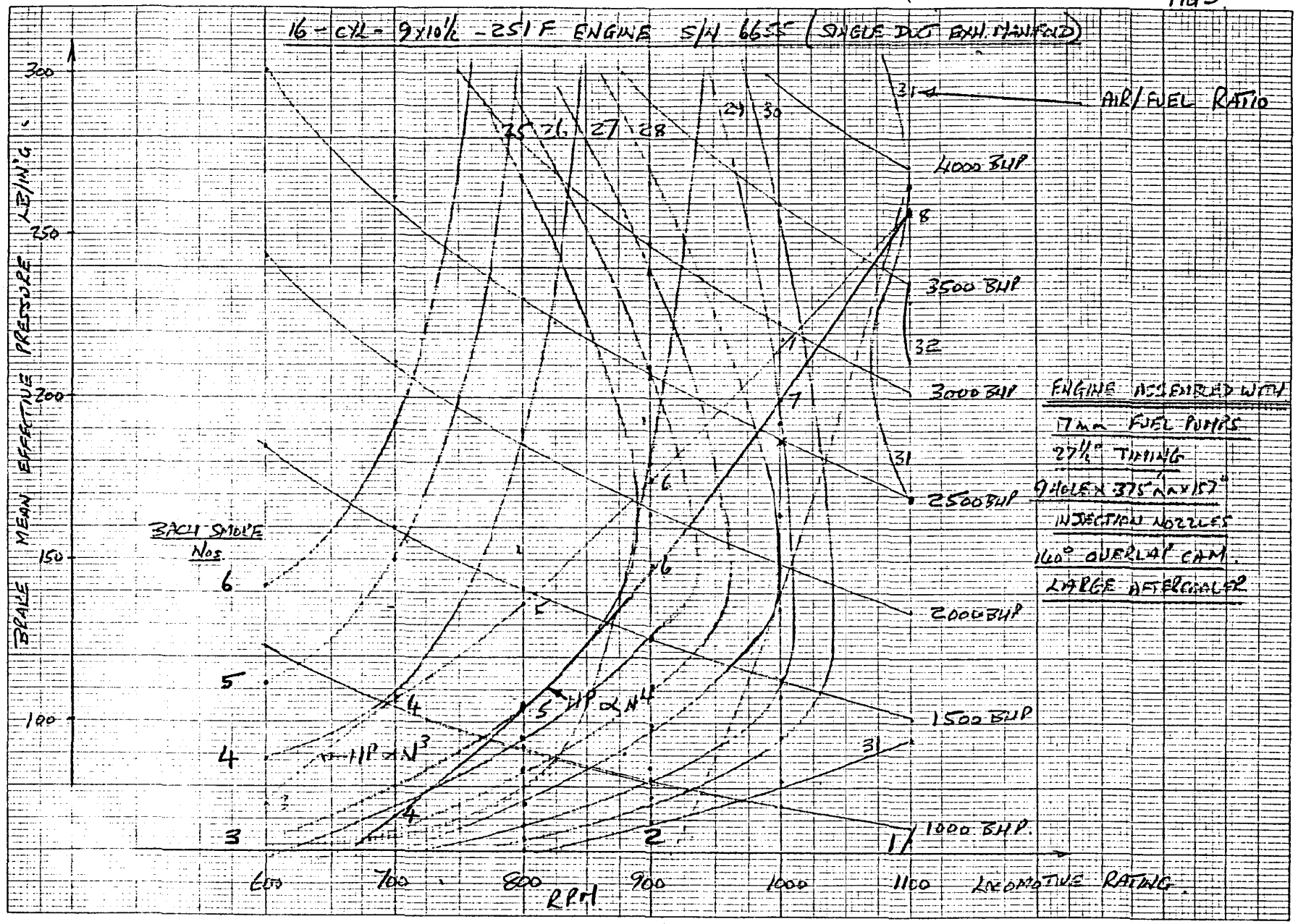

16 - CYL - 9 x 10½ - 251 F ENGINE S/N 6655 (SINGLE DUCT EXH. MANIFOLD)
BRAKE MEAN EFFECTIVE PRESSURE LB/IN²G
300
250
200
150
100
AIR/FUEL RATIO
4000 BHP
3500 BHP
3000 BHP
2500 BHP
2000 BHP
1500 BHP
1000 BHP.
EACH SMOKE Nos
6
5
4
3
HP ∝ N³
HP ∝ N⁴
600
700
800
900
1000
1100
R.P.M.
LOCOMOTIVE RATING.
ENGINE EQUIPPED WITH
17mm FUEL PUMPS
27½° TIMING
9 HOLE x .375 mm x 157° INJECTION NOZZLES
160° OVERLAP CAM.
LARGE AFTERCOOLER

FIG 5

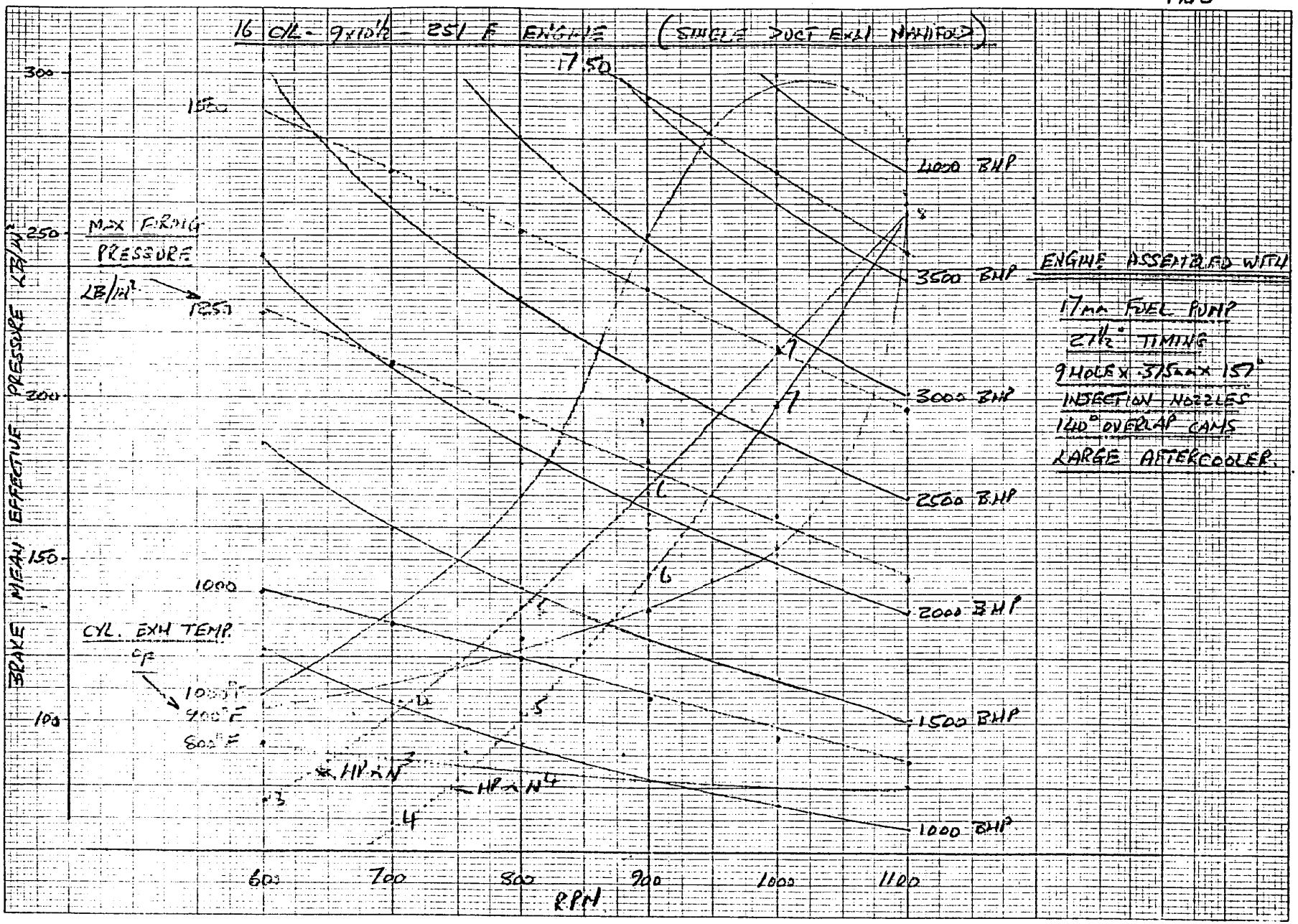
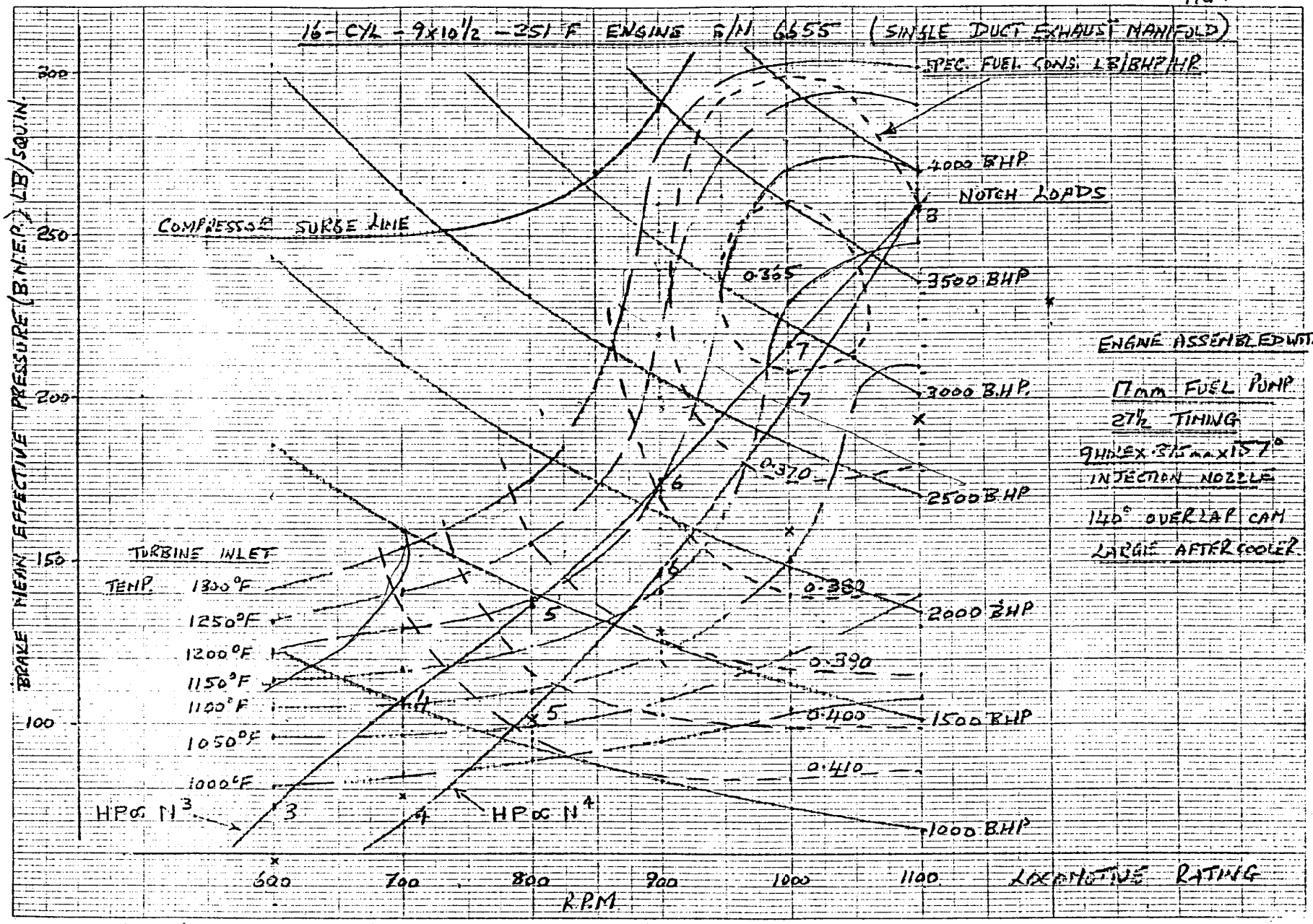

16 CYL - 9×10½ - 251 F ENGINE (SINGLE DUCT EXH MANIFOLD)
1750
1550
4000 BHP
3500 BHP
3000 BHP
2500 B.HP
2000 BHP
1500 BHP
1000 BHP
MAX FIRING PRESSURE LB/in²
1250
CYL. EXH TEMP. °F
1000°F
900°F
800°F
* HP ∝ N³
HP ∝ N⁴
BRAKE MEAN EFFECTIVE PRESSURE LB/in²
300
250
200
150
100
600
700
800
900
1000
1100
RPM
ENGINE ASSEMBLED WITH
17mm FUEL PUMP
27½° TIMING
9 HOLE x .375mm x 157° INJECTION NOZZLES
140° OVERLAP CAMS
LARGE AFTERCOOLER.

FIG 1



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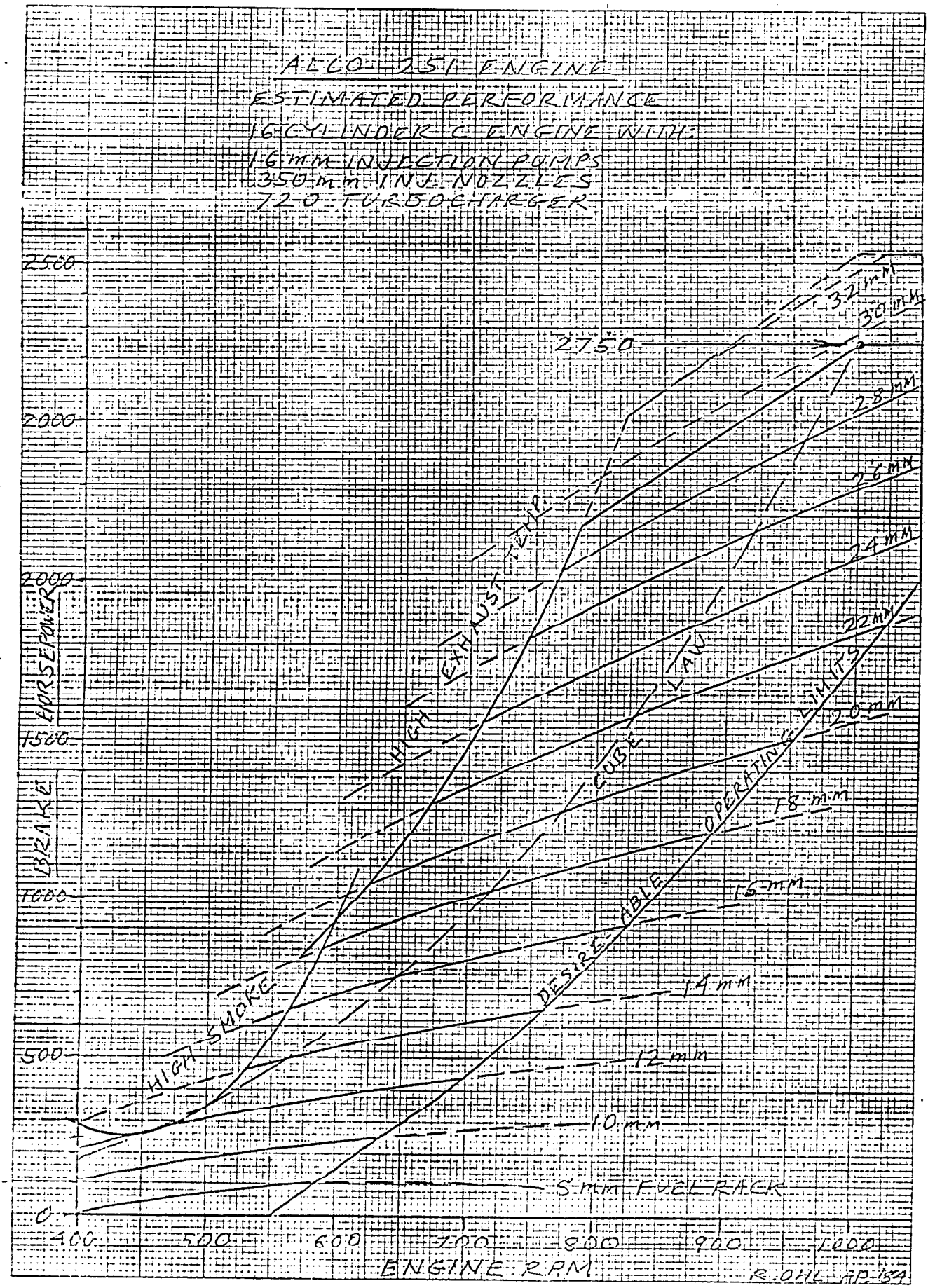
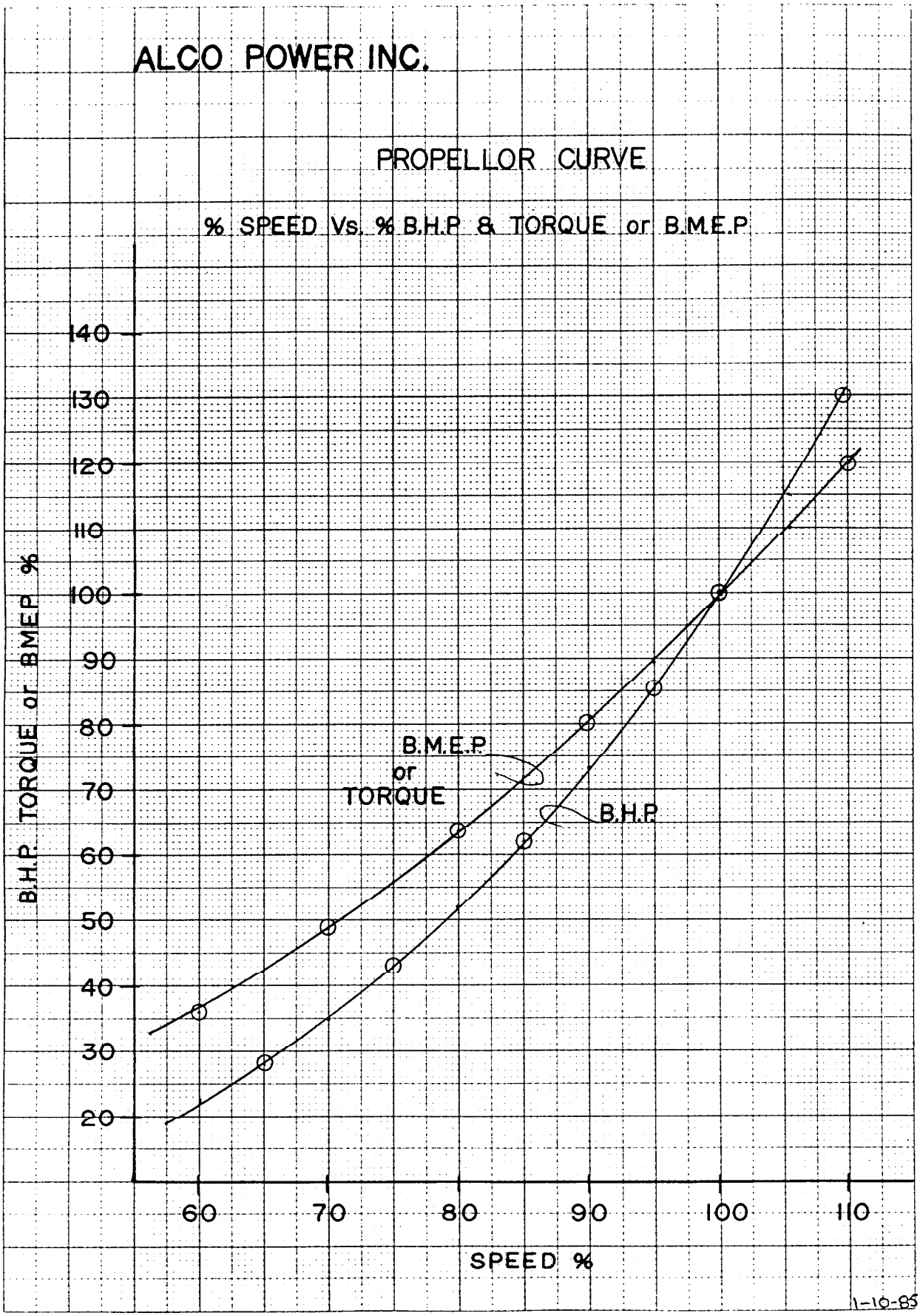

ALCO 251 ENGINE
ESTIMATED PERFORMANCE
16 CYLINDER C ENGINE WITH:
16 mm INJECTION PUMPS
350 m.m. INJ. NOZZLES
720 TURBOCHARGER
2500
2000
1500
1000
500
0
BRAKE HORSEPOWER
2750
32 m.m.
30 m.m.
28 m.m.
26 m.m.
24 m.m.
22 m.m.
20 m.m.
18 m.m.
16 m.m.
14 m.m.
12 m.m.
10 m.m.
8 m.m. FUEL RACK
HIGH EXHAUST TEMP.
CAUSE
HIGH SMOKE
DESIRABLE OPERATING LIMITS
400
500
600
700
800
900
1000
ENGINE RPM
R.OHL AP-134



46 1320
K•E 10 X 10 TO ½ INCH 7 X 10 INCHES
KEUFFEL & ESSER CO MADE IN USA

FOREWORD

This review is intended for use and distribution within Alco's engineering department, for wider circulation it would require substantial editing.

Its purpose is to explain original design intent of the various parts and reasons for changes made over the years, point out areas were deficiencies have occurred if and how they were attacked, also whether the attack was brought to a successful conclusion or not.

The objective is to help those who lack the familiarity with the 251 engine and its background to answer questions from customers and licensees, to assist in failure analysis, in correcting complaints both familiar ones and new ones which may crop up, and also to present a foundation on which extension of existing or developments of new designs may be based.

The review is written with the outlook of a designer who was intimately involved in all phases, excepting the turbocharger,of the 251 engine design. Consequently the review does not include more than passing references to engine performance and development work.

While files, prints, and test data have been examined in preparing this review, a great deal is based on the writer's memory; which he hopes need not be regarded as convenient memory. A few errors of fact may have crept in but not many. Points with some degree of uncertainty or difficult to prove have been flagged by expressions such as "by memory".

In contrast to facts presented,comments on them reflect the writer's opinion and value judgements.

F.C. Tromel
August, 1972

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EXPLANATION OF NUMBERS USED AS REFERENCES

MEL 3436 Typical Mechanical Lab Report
MET 2640 Typical Metallurgical Lab Report
14AA71527 Typical Drawing (Letter can be D, C, B, A, EE, AA)
19LYA1234 Typical Layout 20LYA126 Typical Graph
863361 Typical Development Project
01P4570, 31P5302 Typical Process, Product Std. Sheets
80370 Typical Material Std. Sheet
MI Maintenance Manual

INTRODUCTION

Original Design Objectives

In the late Forties Alco built locomotives equipped with 12 and 16 cylinder Vee model 244 engines with 9" bore and 10½" stroke. One 6 cylinder engine of this type had also been built a couple of years earlier and operated as a switcher in the Schenectady Plant, but was not put into production. The 244 engine did not have a good reputation, it had gone through costly cylinder block, frequent crankshaft failures and recurrent bearing failures.

In preparation for a new design a few 244 engines were built with certain changes to prove out the feasibility to base a new design on them. These changes included a reduction in crankpin diameter from 7½" to 6" and the adoption of 60° serrations for the joint between main bearing cap and saddle instead of tongue and groove. At least two engines so equipped, were operating in regular railroad service while design work on the future engine was under way. By recollection these engines were brought back after two years for examination which proved completely satisfactory.

The new engine model 251 (2 standas for the 9" bore, 51 for 1951 the year when the design was started) was to be strictly a locomotive engine. The instruction to the engineering department was: "You are not to add a dollar's worth of equipment that is not essential to a locomotive engine. If the engines can be applied for other uses it will be desirable but do not compromise requirements for locomotive service in any way."

The 251 must lend itself to Inline and Vee engine execution and the number of not interchangeable parts should be kept to a minimum. The 251 should have the bore and stroke of the preceeding 244 engine, but must be much more reliable and less costly to produce. At that time the probability of future up-rating beyond the then current 192 MEP was not emphasized.

SUMMARY OF DESIGN HISTORY

After a false start on a 6 cylinder engine with a removable large casting containing camshafts, rocker mechanism and fuel pump which turned out to be much too expensive in spite of embodying all features manufacturing then thought desirable for low cost, the 251 design proper was started.

It was based on a rather flexible crankshaft design, which has proven eminently successful, and in all other aspects aimed at reliability of operation, accessibility for maintainance, simplicity and low cost of parts even at the expense of weight, bulk, and performance.

The first experimental engines were 6 and 8 Inline engines. Both types were built for the laboratory. The eight cylinder had an early crankshaft failure since the Houdaille Hershey dampers could not suppress severe torsional vibrations. It also soon became clear that a locomotive chassis suitable for the long eight cylinder Inline engine would cost more than a chassis for a higher powered shorter 12 cylinder engine.

The eight cylinder Inline prototype was soon scrapped and never again considered. Its existence did however, lead to some undesirable over dimensioning of the turbocharger intended for the Inline engines.

The first production engines to go to the field were six cylinders installed in G.E. built locomotives for Cuba; two prototype 12 cylinder engines were in railroad operation for about 3 years under continuing observation by Alco laboratory personnell. In the Fifties a large number 6 and 12 cylinder engines were sold. Some, but comparatively few 16 cylinders went into operation until an order for some 50 16 cylinder locomotives for freight service on the Santa Fe was put into the field in the early Sixties.

In contrast to the 6 & 12 cylinder engines which performed creditably from the start the 16 cylinders required several serious modifications to make them satisfactory. Amongst them were turbocharger, exhaust manifold, serration failures, and camshaft bearing support failure in the cylinder block, failure of camshaft gear train, and piston failures.

Some of these weaknesses were corrected fully at the time, some reappeared again at higher engine ratings, and required additional corrective work.

Other deficiencies on smaller parts, which have been persistent at some ratings in some applications but less so in others, relate to water pump seal leakage, air valve wear, valve guide breakage or sticking, bushing wear on fuel pump rockers and valve levers, fuel nozzle breakage, cylinder head cracking, for long time not serious, should now be added.

These two lists together cover about all the areas which were likely to cause complaints in the past. Note that this list does not include crankshaft, bearings, base, front end gear train, generator mounting, liners, oil pump fuel injection pumps, (the baffle ring was a time limited, fully corrected mistake), the aftercoolers on 16 F engine have recently acquired a tendenacy to fail only on MLW or Goodwin built locomotives.

With the development of the 6D engine in 1960 started the gradual uprating of engines Under section #30 the differences in components are listed which were made to permit increases in power of the various models.

While the engines made by Alco up to the late Sixties were preponderantly locomotive engines, some other applications were also made.

In the Fifties one Snorkel 16 cylinder engine with engine driven positive displacemen blower was built for the Navy and a long endurance test on this engine in the lab helped in general development of the 251.

Dual fuel engines were designed and built; between Alco and MLW some 50 such engines were put in operation, mostly for pipe line pumping. While the controls are over designed and pilot fuel is in the range of 7%, performance seems satisfactory to customers and complaints are non existent.

Beginning with 12 and 16 cylinder engines for MSTS in 1958, many marine engines have been delivered; the largest contract for 153 LST engines was received in 1966. These required substantial design development to meet High-Shock specifications.

Diesel generator set have been built in slowly increasing numbers, in more recent years they are mounted on skids as self contained units including coolers, filters, small fuel tank , and other auxiliaries. Most of these installations also employ the fully engine supported generator originally developed for locomotive service.

The review of design details of the 251 engine on the following pages deals primarily with the locomotive engine, which is as close as one can come to a "basic" design. Some of the more significant modifications for other applications are mentioned briefly.

The review is arranged in the grouping of engine components established at Alco's forerunners many years ago for drawing room and specification work.

Bearings:Main, Connecting Rod

The bearings of the 251 engine in general have shown outstanding performance. Into their design and specifications went all the knowledge gained both by Alco and Clevite during the frustrating years of bearing development for the 244 engine, which originally had the reputation of having totally unsatisfactory bearings. The various oil chamfers, thickness relief at the parting line, reduction of height at lock tabs, as shown on the drawings, are not arbitrary but results of prior experiences. Note the relatively few significant changes made on bearing drawings.

The fact that the connecting rod bearings and the center main bearing never had oil grooves in the most heavily loaded areas is one of the most important factors in their satisfactory operation. The center main bearing of the 6, 12, and 16 cylinder engines is actually grossly over designed, extra width, no oil grooving, no diversion of oil to connecting rods, oil supply at the split line by a rather complicated drilling in the cap. The extra width of the bearing, requiring extra wide saddle and cap forgings with an extra wide cylinder spacing which influences camshafts, exhaust manifolds, tooling for blocks bases, and piping, presents a cost penalty, most of the extra cost being hidden. But a cost reduction program to eliminate the extra width is not feasible at this stage where all tooling exists, and where the present style parts must be kept in production to protect thousands of engines in operation. The concept, however, should not be carried over into new designs; it was not used on the later eight cylinder engine; the eighteen cylinder engine does not have a center bearing.

The 251 engine from the beginning had hardened crankshaft bearing surfaces, either by induction or nitriding. This permitted the use of the intermediate copper lead layer where the lead is dispersed in essentially globular form, which makes the bearing more resistant to acid corrosion. See Alco standard sheet 31P5703 for specification of material and typical examples of micro structure. The specifications demand the use of a nickel dam between bronze and overlay; its primary purpose is to prevent tin migration, the secondary purpose will be mentioned later. The 251 bearings were designed to be made by the method of casting the intermediate layer on to the flat steel backing strips as developed by Clevite. The casting of the copper lead is not spelled out in the Alco specifications, so that sintering method used by other bearing manufacturers (e.g. Glacier) is not ruled out, but it was never proven out by Alco.

Up to 1958, that is before production quantities justified the substantial tooling expense for strip stock bearings, all bearings were made of tubing. Even today the thrust bearing with its heavy flanges is made of tubing. The thrust bearing in its present form is not the original 251 design but had to be designed when the original half thrust rings in the center bearing proved unsatisfactory, see section Crankshafts. The .001" thick overlay is precision plated, no machining after plating. It has occasionally small blow holes for which an aribitrary standard of acceptance (50C72096) has been set up and observed. Over the years there have been a few instances of overlay not being properly bonded to the intermediate layer. Alco chemical laboratory has developed a test procedure.

There is no overlay on the flat thrust faces of the thrust bearing. These surfaces like all other surfaces of all bearings are flash plated as a corrosion preventive. This flash plating, which is only fractions of one mil thick, gets worn off easily from the thrust faces thereby exposing the copper lead color. This leads to occasional false alarms of "worn out" thrust bearings.

Since the change from 6 pitch helical camshaft gears to 4 pitch straight gears in 1960 no thrust force is developed within the engine. In locomotive application thrust forces do occur as shock loads when the masses of generator armature and crankshaft

are accelerated in bumping of cars and switching and as steady loads when operating on grades. To resist these forces one thrust shell in the saddle (formerly half thrust rings in the center saddle) are adequate. The flanged thrust bearing being much more expensive than the straight "intermediate" shell, a straight shell is used in the cap of the locomotive engines. There have been a very few cases of thrust face damages on locomotive engines, which could not be explained. Suspicions that a D.C. generator might exert a magnetic axial force were always denied by generator builders. While such cases seemed baffling a new bearing shell and smoothened shaft surfaces seemed to correct the conditions, in no case did it lead to prolonged investigations.

To provide increased thrust load capacity for conditions mentioned, all stationary and marine engines are specified with thrust shells both in cap and saddles. Spherical roller bearings in the generator, where the outer race must slide in the housing to compensate for thermal expansion, may exert substantial thrust loads onto the engine thrust bearing. (The locomotive generators are equipped with straight roller bearings which slide longitudinally without end thrust).

Some of the couplings mounted on the flywheel also may create end loads, as also marine engines installed at a rake. A common occurrance on bearings removed after long operation are cavitated areas. They appear in the region between the split line to about 30° above and below. They are irregular spots where the overlay has disappeared, most usually the exposed layer is gray, that is , it is the Nickel dam which acts as an effective barrier to the spread of cavitation into the intermediate layer. As long as the Nickel dam is not broken thru and the extent of the cavitation is within reason, say 3/4 to 1" in approximate diameter, the bearing may be reused.

One type of catastrophic failure occurred on the first marine installations in 1958 on several engines and many of their bearings. Ref. 14-1, occurred in a few instances later.

The complete failure is characterized by a shell having broken in two with a circumferential break in the oil groove. The cause of the failure is massive dirt and mechanism is:

1. The bearing surface gets over heated most likely due to dirt.

2. The combination bronze and steel acts as a bimetal strip, permanently deforming the bearing in hour glass like deformation.

3. Due to the deformation contact between shell and bore is lost in the center. The unsupported portion of the shell is flexed under the cyclic bearing load.

4. Fatigue failure in an area of stress concentration, e.g., oil groove results.

As proof of this mechanism, the shell can be placed in a good bore or a scale can be laid against the outside of the shell. If there is a hollow spot (light shines through or a paper can be pulled through) between scale and shell in the center indicating the hour glass deformation, the cause of failure as over heating is reasonably established.

In the particular case of the marine engines CUNO edge filters were specified as lube oil strainers, with by pass filtration. The CUNO elements were probably not cleaned soon enough, in any case when they were turned by force, they bent the disks so that large quantity of dirt passed into the engines. Alco since that time has refused the use of edge filters for lube oil and on the basis of well documented, Ref. 14-2, description of the strainer failures and been granted exception from specifications calling for edge type strainers. CUNO, incidentally, has since changed the design of their disk and spacer mountings in a way which probably prevents recurrance of the failures.

Since Alco began to run at 1100 RPM and MEP of around 250 psi, bearings removed from laboratory engines after 500 to 3000 hours in 1967 have shown more polished appearance in the highly loaded areas than was seen before, one bearing in particular showed actual wear. (Ref. 14-3) It may be another consequence of increased lube oil contaminmation due to either low compression flat top pistons and/or use of high dispersant oils. No field reports have indicated deteriorati of bearing performance to date.

Bearings Commercial Aspects

The main supplier of bearings for the 251 engine has been Clevite, who owned the Nickel dam patent. An agreement existed for many years according to which American Bearing was permitted to fill up to a certain percentage (25% by recollection) Alco's requirements of bearings with the Nickel dam. Later when Alco's requirements dropped, the agreement with American Bearing lapsed. American Bearing is still Alco's second source, the Nickel dam patent has expired. Both companies have never sold bearings in the aftermarket directly. The aftermarket in bearings has been an important source of Alco income.

For a long time no other manufacturers were either able or interested in making Trimetal bearings and Alco with its two suppliers, Clevite and American Bearing, enjoyed a virtual monopoly.

Manufacturers of aluminum bearings, Johnson Bronze and Bohn, tried hard to interest Alco and a program to test steel backed aluminum bearings was started, bearings procured; in the end Alco decided to drop the program because if the bearing would prove satisfactory, Alco either would have to use them in substantial quantities or repeat again previous errors of helping to set up "pirates" in the aftermarket.

In recent years true Trimetal bearings can be made by MIBA Austria, Unity Japan, Glacier England, the latter using the sintering process for the copper lead layer and no Nickel dam. All three manufacturers quote bearings at lower cost than Clevite and American Bearing. Alco has been, for several years, engaged in on again, off again, negotiations with Glacier and MIBA; trying on one hand to get less costly bearings but trying on the other to discourage any one of the suppliers to enter the aftermarket directly. Information has recently been received that MIBA has received and accepted a substantial order for Alco engine bearings from Mexico. MIBA does make bearings from tubing rather than from flat stock, but experience cited earlier, proves that bearings made from tubing are just as satisfactory.

Main Engine Bases

Background: The desirability of the 4 point suspension was first recognized on one locomotive with a 539 engine. In this case the engine base was bolted to the chassis with flanges continuing for the whole length of the engine. At operating speed the rotating internal moments caused the engine to deflect cyclically and this deflection induced intolerable bending vibrations in the locomotive chassis. The condition was then overcome by bolting heavy cast stiffening rails to the side of the base. On the subsequent 244 engine the 4 point support which cannot transmit bending deflections into the chassis beams was developed. On this engine the rear mount was part of a separate bell housing; on the 251 the engine and lower generator mounts are part of the base. The rear engine and lower generator mount is chocked in all directions on the girders of the chassis. The forward foot is chocked to prevent lateral movement but there is enough clearance between hole and mounting bolt to permit thermal expansion of the base. This expansion and contraction is so definite a movement that the mounting pad on the bases wore enough to cause mislignment of the Christmas tree (the shafting to brake air compressor, fan drive, originally also traction motor blower drive). Wear pads of alloy steel are now welded to the mounting feet. The choice of steel for the pads, electrode and method of welding were trickey; any changes should be made with care. On the bases for the 6-cylinder the distance from crankshaft center to interface with the support is 1/8 (the thickness of the wear plate) longer at the front foot then on the back end. On the Vee engines machining of the front foot was revised to bring the foot with wear plate in line with the rear mount which has no wear plate.

The longitudinal location of the mounts was chosen so that the operating deflection of the engine free end should be about equal the maximum deflection between the mounts. This at least was the design intent.

The placing of the bulk heads was dictated by desire to make as few different screen sizes as possible, yet allow the main bearings to be dropped next to the bulkhead. The early bulkheads had a reinforcing rib welded to one side of the round cutout.

It was desirable to make the bases so that they would drain with the locomotive at level grade so the bottom therefore received a slight slope; there was also some specification which called for the ability to withstand crankcase explosions of 20 psi (by recollection). To meet this the bottom was rolled to a large arc. Both, to make the bottom slope upwards from the center and to use available rolls of limited length, the bottom was spliced. The splice is made against a bulkhead so that a full penetration weld can be obtained. After making the weld the cutout in the bulkhead, to let the oil flow from one compartment to the other, must be cut down to within the height of a fillet weld from the bottom surface. More than once the shop failed to cut this dam back and angry complaints of having to sop out the base were received.

In most installations the bottom of the base is inaccessible yet the drain connections must be made there. If the connection to a drain pipe should leak, there is no chance to correct it until the engine is lifted. Therefore in all cases where the drain is on the bottom, the hole is closed by a pipe plug installed thru a base door from the top. The pipe plug has a hex bar extension to bring it above oil level.

The thickness of the end plates is chosen to permit fastening all pumps and other components with capscrews in blind holes. The side sheets were originally made 7/8 thick in the thought to machine them to 5/8 in the area of the doors. This was not carried thru; for long time doors were supposed to seal against the unmachined plate with a thick gasket, unless adjacent welds had distorted the sealing area to uneveness beyond 1/32 in which case this door opening only should be machined. The shop always objects to "if required" instructions and now machines routinely the areas around all door openings by a minimum of stock removal.The plate thickness was reduced to 3/4 several years ago.

The top rails are definitely considered strength members intended to resist horizontal bending of the engine due to free moments. To make them act in concert with the cylinder block,closely spaced fairly large (7/8") bolts are used for this joint. It is a metal to metal joint and evidence of fretting between block and base is not normally seen.

The oil suction pipe has its inlet opening near the longitudinal center of the base. This is a must for locomotive engines which must contend with bumping cars. On a 539 model the suction opening was at the front end near the pump and the low oil pressure shut down cut off the engine whenever the brake was sharply applied because the oil slushed away from the suction opening. To prevent the suction pipe from trying to carry loads for which it is not designed, it is rigidly fastened only at the end plate and supported by deformable hangers rather than rigid welds against bulkheads.

The dipstick is one item which has caused an unproportional and recurring amount of difficulties. It should be readable with engine idling and with exhauster motor operative; if one is not careful in venting it, the pipe which protects the dipstick from misleading oil splash will act like a manometer. It is rather ridiculous but virtually every time a base was redesigned for any purpose whatever, the dipstick at first gave wrong readings.

The dipstick conditions are aggravated by the larger variations of oil level between an operating engine and one at stand still. Depending on locomotive piping systems where filter tanks (65 gallons for the smallest one), coolers and strainers may drain back more or less quickly, it was very difficult to provide room in the bases to accommodate such drain back without flooding the screens and thereby nullifying their chief function. Without screens the drain back level is determined by the door openings. It must be possible to open any door without having to contend with oil running out.

Base screens were vitally important on the 244 engine; it had frequent main or conn rod bearing failures. By making routine examinations of the base screens thru the doors, flakes of bearing metal caught on the screens gave valuable warning of impending bearing failures. Related frequent bearing replacement also was helped by screens. On designing the 251 base screens were provided, although of simpler design as on the 244. Even though the 251 proved to have very few bearing failures, it took a long time to muster the courage to leave the screens out. It was finally done a few years ago when the bases with larger oil capacity were introduced. From that time on screens were left out of all engines but the suction pipe opening was protected against foreign material by screen compartments of various designs. For flow capacity thru screen see Ref. 14-4.

All bases have a duct on the inside of the drive end plate; for discussion of its purpose see section Generator Connection, Auxiliary Devices.

The Alco "244 and 251"engines have always had the reputation of having crankcase explosion only rarely (believed due to the use of constant speed motor driven crankcase exhausters). This is about the only excuse for having up to 1965 something called safety covers which were anything but safe. The two small openings above the rear engine mount were closed by a sheet metal with large perforations, about 1/2" dia. holes, a diaphragm, made of a Vellumoid type gasket and a thin slotted sheet for outside protection. In case of an explosion, the gases were supposed to break thru the diaphragm bend open the slotted outside sheet metal. It is recalled that during a crankcase explosion in the laboratory, a couple of the regular base doors flew off but the safety cover was still closed. In the whole the safety cover which did not open was probably safer than one which did open to admit fresh air and triggered a secondary explosion. These "safety" covers were finally replaced by solid covers and one of the doors was replaced first by a homemade spring loaded safety cover and later by Bicera doors. For marine and some stationary applications the specifications call for some reasonable relation between crankcase volume and safety cover area which is met by using several Bicera covers (depending on the engine size). On all other engines the use of the single Bicera door is rather inadequate and rationalized only for locomotive and similar applications where personnel is hardly ever next to an engine while in operation.

The regular base doors, installed with long side vertical on the In-line engines and horizontal on the Vee, started out as sand castings became permanent mold, than stamped steel. Leakage at the gasket and at the knob was a frequent complaint. Cork and sponge rubber gaskets and a sealing washer at the knob seem to have brought the number of complaints within reason.

The preceding discussion applies pretty well to all bases. Over the years a number of redesigns of details, including welding details for cost reduction only, were carried thru without affecting vital parts of the bases. No failures on any part of any 251 base is remembered. Failures had been quite frequent on the preceding 244 engine, which encouraged some over-designing on the 251 bases.

The original flat locomotive bases could accept operation at 5° pitch, limited by oil reaching the generator oil seal; 10° roll for the 6 and 12-cylinder, 5° roll for the 16-cylinder limited by loosing submersion of the oil suction pipe opening.

BASES WITH INCREASED CAPACITY

In the late Sixties the demand for a longer interval between replenishing the oil in locomotive bases became persistent. On study it was found that all locomotive chassis had placed a pan under the whole engine, sufficiently far away from the engine base to permit installing a 2" pipe for draining the bases. By eliminating the drain pipes and shaping the bottom of the base to follow the contour of the sloping under pan with 3/4" clearance the lube oil quantity between high and low level was increased by 80 gallons for the 16 cylinder base. Instead of sloping the bottom to the center, it is now sloped down to the front end and the drain opening is at the lowest point of the end plate.

At the same time screens were eliminated and the lube oil header increased to reduce the rather high oil velocity. Both the 16 cylinder and the 12 cylinder bases are now being specified to this design, they are considered interchangeable for all locomotive applications.

Marine Bases

Bases to permit operation at 15° pitch and 30° roll have been in use for marine engines. On the 12 and 16 cylinder engines the whole base including the side plates is made deeper in the center to keep the lube oil suction pipe safely below the oil level.

There is also added a second set of screens covering the lowered part of the base. This screen should be considered a baffle preventing too violent motion of the oil in heavy areas rather than a screen in the usual sense. In the more recently designed marine base for the 18 cylinder engine the suction pipe end is placed in a pot with sloping sides welded to the bottom of the base. In designing the pot attention was paid to details of welds and construction. The engine is always to be considered a beam cyclic bending, the bottom sheet of the base is at the extreme fiber and any changes in contour or sharp discontinities will act as stress raisers.

This base was designed to operate at 15° pitch, 45° roll, refer to layout 19LYA1724.

Bases For Special Uses

For some boats dry bases were made, they are the flat locomotive type bases with a larger opening at each end which are to be connected to an oil tank located below the base, the oil suction line is brought to a flange in the base side plate for outside pipe connection to the tank.

Alco built prototype hydraulic locomotives. These had special bases in which the bottom plates were made 3/4" thick and shaped to half surround the cardan shaft which transmits power to the trucks. The 3/4 plate was to act as armour in case of shaft breakage. To make up for oil volume lost by the cardan shaft covering, "saddle bag" like compartments were added to the side sheets. These 6 engines recently reached the second hand market.

The bases for the LST engines are completely special. The specifications called for high shock, severe pitch and roll and minimum oil capacity of one gallon per each 10 rated H.P. The location of the mounts was based on shock calculations. They are not over designed as demonstrated by their rather severe deformations in the barge shock tests. To meet the large oil volume and pitch and roll requirement a deep base with double bottom was designed. A quarter scale model was made on which indlinations as well capacity could be demonstrated. Both ends of this base are less deep than the main section. The depth of these narrow

end compartments was determined by a man's ability to reach the bottom joints with a welding electrode. The other deeper compartments are wide enough to get head and shoulders inside.

REFERENCES - BEARING BASES

14-1 Circumferential Breaks - Correspondence Folder - Avondale/Bethlehem 1958/1959.

14-2 Cuno Filter - MEL 3268 and CHEM 187.

14-3 Bearing Appearance on High Load Engines - Correspondence folder - Bearings 1967/1968 - letters to from Clevite.

14-4 Flow Capacity Thru Screen CHEM Lab Report 853.

STANDARD PARTS

THREADS BOLTS NUTS

For common bolting Alco uses SAE grade 5 material specifications.

Formerly bolts for which drawings were made were specified as AISI 1035 steel which is virtually obsolete. Requests for material substitutions are therefore frequent; setting up a suitable new bolting material specification has been initiated recently. Alco has always avoided the use of resulphurized (free machining) steel designated 11XX to avoid sulphur stringers which might initiate failures. Material for higher strength bolting is chosen on individual requirement.

On sizes 1/4 thru 1" Alco uses series UNC for coarse threads, UNF for fine threads. For sizes over 1" the 8 thread and 12 thread UN series are used.

5/16 diameter bolts are used, no other sixteenth sized diameter is permissible.

On all general threads class 2A and 2B are now being specified, formerly class 3 was in general use and will be found on many drawings. For studs Alco uses a modified class 5; it is based on the original 1944 edition of hand book H28. Principally Alco reduced the tolerances on stud diameter, possible by using only rolled threads. For detail see a report by V.V. Collings of 12-16-69 on Interference Screw Threads.

In general studs have been eliminated where ever feasible; in the camshaft couplings stud bolts with loose fit are used as studs with ~~LOCKTITE~~. Note the detailed instructions on assembly drawings, e.g. 16371052, to insure adequate locking; they were established after finding that bolts with ~~LOCKTITE~~ casually applied did not lock.

LOCKTITE

The shop has often requested that the cylinder head stud fit in the block be changed from the modified class 5 to something loose preferably class 3 with LOCKTITE. It was carefully considered in 1969 but rejected. Unfortunately the thread (also the one for the generator mounting bolts) is 12 threads per inch which is somewhat fine in the low strength plate of the cylinder block deck. If studs are removed in reworking a cylinder block, or if threads were damaged, oversize studs are used contravening all principles of strict interchangeability. There are several oversize steps of a few thousands available, there is also available a 1/8 stud with 1/8" oversize stud end. This breach of interchangeability has not caused field complaints of any significance.

In rebuilding engines the questionable practice of building up stud threads by copper plating is permissible and has not resulted in complaints.

For salvaging some new LST blocks oversize head studs were not acceptable to government inspectors and a bushing was developed which could be used more generally. On the 244 engine damaged stud holes were welded up, redrilled and retapped. There was one stud fairly close to the edge of the top deck where cracks originated in welded up holes and caused severe engine damage. Welding up of stud holes has since been avoided unless followed by stress relief.

For threads tapped into cast iron coarse thread is considered a must. In recent years fine threads have been specified less and less frequently, they should be used when needed for adjustment, or for maximum resistance to loosening and in rare cases when the presumed improvement in strength of fine threads is of real significance.

Alco now produces virtually all external threads by rolling. For this reason entrance and exit chamfer of 40° included angle are desirable. The blank diameter should be determined by trial; the pitch diameter is usually a good starting point for the blank diameter.

Locknuts with nylon inserts should not be used inside of the engine, because nylon is likely to soften at high oil temperatures. Bolts with inserted nylon pellets and nylon filled slots were tried, one set to hold the bolts of the water pump seal plate, that is temperature was definitely limited. After a few hundred hours they had lost absolutely all locking effect.

Up to recently Huglocks were normally specified as locknuts, they are now out of business and an equivalency list was prepared. It was necessary to watch for height difference between locknuts made by different manufacturers.

Regular lockwashers are used extensively even inside of the crankcase, no complaints on loosing bolts are remembered.

Shops, field and licensees have insisted more and more on having tightening torque specified on virtually every bolt, not only more important ones. The published torque recommendations show so many variations that Alco prepared a practice sheet 01P4101 which for uniformity's sake should be followed unless special conditions obtain, which they often do, especially in respect to rigidity of the parts to be joined.

C-1

CAMSHAFT

General Consideration

The general design concept of the engine calls for camshafts to be installed axially from the front end of the engine. They would have to pass the 4.5" diameter bearings; they would have to be divided into sections short enough to insert them within the limited length in front of the engine.

To meet this as well as machine tool limitations it was established that all camshaft sections would be 2 cylinders long (on the 18 there is a one cylinder long section).

The couplings are made with close fitting male and female spigots, with 8 bolts or studs ½ dia on 3 9/16 BC; torque is transmitted by friction due to these bolt. There is at each joint one straight dowel to establish location, it is used in machining and dimensioning as basis for angular relations of cams, keyways, etc. Originally this indicating hole would be in line on all sections of camshaft; it is still that way on all engines except the 16 cylinder, where on certain sections the holes at the two flange ends are offset by 180° due to change in firing order to be discussed later.

All flange faces are ground at a slight taper to help in keeping sections in alignment to each other. The shaft sections have a 3/4 dia rifle drilled hole with 1/4 outlet holes at each bearing for lubrication. The oil enters thru a larger cross drilled hole matching a circumferential groove in bronze casting at the free end which serves as a "thrust" bearing.

There is no positive lubrication on the cams themselves. Observation at engine idling and at the beginning of a run showed fuel cam dry. Design attempts to provide a drip edge on the control compartment sheet were dropped when needed weldment changes proved formidable. A complicated system of oil sprayers to the fuel cams was designed and installed on river boats, its effectiveness was never clearly established. Ref. 16-1.

Since the camshaft is slid in from the end provisions were made to take up the weight and prevent the cam noses from "broaching" the bushings. All bushings have a 7/16" long steel extension, in which the bronze lining is stopped off. As an assembly tool a set of troughs can be jacked between these bushing extensions and the block rail. The shafts are then slid on these troughs. The gear must be brought sideways into the cylinder block and the camshaft is pushed into the taper bore of the gear to which the key has been permanently attached. The shaft is pulled into the gear by the end nut. In most cases the end of the camshaft is not accessible due to a flywheel or a generator. Therefore, the shaft nut serves also a second purpose to jack off the gear from the shaft. The gear bore is extended and enlarged to receive a shoulder (hardened) of the nut which pushes against a snap ring in the gear. A rigid open end wrench on the shaft nut, together with sledge or a hoist on the other end of the wrench breaks the gear loose. Few complaints about this procedure were heard, but nobody was successful in making a torque wrench.

When examining gear end sections it will be noted that the keyway location was altered, and that corresponding changes were made for lift check points on the assembly drawings in 1963. This relates change from 6 pitch to 4 pitch gears discussed section gears which introduced a small timing discrepancy later corrected by keyway location changes.

All fuel cams carry a notation to polish the rising ramp, all assembly instructions call for the use of Molycote on the fuel cams. Both are the result of repeated observations on engine test and in the field which indicates that the first few hours of running may damage the cam, and roller, sufficiently to cause early failures. Appearance of cams treated as indicated showed substantially better conditions than unpolished and not Molycoted cams.

Material and Hardening

The shafts are machined from round bar stock by turning and grinding operation.

Except for a few low carbon, carburized and hardened shafts made before heating coils were available, all camshafts are made of Alco 81602 steel which is AISI 1050 modified by the addition of .15 to .30 chrome, all cams and bearings are induction hardened to 60 to 62RC .100 to 150" deep.

Over the years the shop had several periods where scrapping of shafts due to hardening or grinding cracks became serious problems, and correspondence files are full of reports on the subject.

The reasons which occur singly or in combination are:

A. Material deficiencies, steel from some suppliers e.g. CRUCIBLE has consistently given less scrap than from others. We do not remove as much stock from the OD as recommended, but any additional cost for larger stock and additional machining would more than offset normal scrap losses.

B. Inadequate benching of the ends of the cams.

C. Deficiencies arising in the somewhat overage induction hardening equipment and controls.

The cam detail drawings now carry a .030 tolerance for the dimension over nose and base circle of each cam. The theoretical profile matches the low end of the tolerance. The shop is expected to work to the high end, if magnaflux indications or other deficiencies are found the cams can be reground still within the tolerance. For restoring cams on used shafts a salvage drawing permits another regrinding to.080 under the minimum new dimension. The depth of the induction hardened case is deep enough.

Over the years it was suggested many times to improve wear characteristics by material changes. The only actual trial was made in 1970 with 2 shaft sections made of AISI 1080. Ref. 16-2. It was based on various statements found in literature that resistance to contact stresses depends not only on high hardness, but also on high carbon content. For years it had been maintained that high carbon steel could not be induction hardened without cracking. When first eccentrix and then true camshafts sections were attempted minor difficulties were experienced in drilling with standard tooling and standards. The steel had been stress relieved before machining; heating cycle for induction hardening was slightly modified. Hardness on cams after tempering was 62 to 63RC compared to a general hardness level of 58 to 60RC on regular production shafts. This difference in hardness could be very significant.

The two sections were installed in Canada 16B73227-2 in two 630 locomotives sometime in 1970. A report of satisfactory operation was received, which does not mean too much since thousands of cams of 1050 and Cr are also in satisfactory operation.

It is, however, an approach to improving fuel cam wear which has now been proven possible and might be quicker and easier to attain than design changes.

While the files contain very many reports relating to cams, MET 3208 shows, typical hardness pattern, also one typical spalling of a fuel cam.

PRODUCTION CAM PROFILE DEVELOPMENT

Soon after the first runs with the 251 experimental engine were made, it was recognized that the air and exhaust cams designed for a reasonable tappet clearance of .012" caused the valves to open near firing top center when the heavy loads on the fuel cam during injection deflected the camshaft upwards. It must be noted that the arrangement of the fuel pump lever is peculiar to the 251 engine and pushes the camshaft in the opposite direction of the one normally expected. The tappet clearance was therefore increased to .034". No adverse effect of this large clearance has been recognized, probably because the spring under the valve yoke keeps all parts of the linkage, aside from the contact between yoke and valves themselves, without clearance together. This point needs consideration when thinking about leaving out the yoke spring.

For all cam calculations, see ref. 16-3.
The profiles for all cams were based on earlier experience with cams on the 244 engine. While the air and exhaust cams there were developed from sinoid curves, unequal for acceleration and deceleration, the 251 cams (still in use in 1972) are based on equal parabolas for acceleration and deceleration. The parabolas permitted accurate and easy integration (before availability of computers) to obtain velocity and lift. The acceleration parabolas were cut off at the point were profile velocity equalled the desired ramp velocities, .0025"/deg. for opening, .0012"/deg. for closing. At the nose, the acceleration parabolas were permitted to drop to zero, thus allowing the ready interposition of various dwells. Graph 20 LYA 107 illustrates these profiles. These graphs may appear misleading because the deceleration parabola is drawn above the abscissa to get it on the paper, if drawn correctly it would be below the abscissa. Refer also to Ref 16-3.

The other air cam profile in use in 1972 was developed as a compound polynomial acceleration cam to provide as much margin as possible of the spring forces over acceleration requirements to counteract exhaust valve sticking. It is illustrated on 20 LYA 125. This cam was built and run experimentally as exhaust cam, for performance see latar. Production air cam 16-EE-72048 uses above profile at the opening flank but as closing flank profile per 20 LYA 107 is used. In this case the purpose of the polynomial case is the reduced lift on the opening side which gives more clearance between piston and air valve.

A typical diagram illustrating the relation of piston and valve movement is shown on 20 LYA 127; representing the former standard 123-3/4 overlap cams, showing both the regular production exhaust cam and the polynomial experimental exhaust cam. Whenever cam shapes were changed by reducing lift for one purpose or another, the area during valve overlap was left as undisturbed as possible. This graph shows the attempt to provide more distance between piston and exhaust valve which might stick and not follow the cam, while the air cam on the opening flank needs less clearance from the piston. The piston shown is the 12.5:1 compression ratio piston.

Graph 20 LYA 137 shows condition with 140° overlap. To get piston to valve clearance, a change was made from the original parabolic air cam 16-A-72036 to the polynomial opening flank of 16-A-72048. It also shows relation of piston height (compression ratio) with valve clearance.

PRODUCTION CAM PROFILE DEVELOPMENT - cont'd

The cam profile drawings show actual valve opening and closing positions, which represent the points at which the lift from the base circle equals the tappet clearance, cold without taking any deflections into account. These points are referred to when citing degrees of valve opening or closing on assembly drawings. Being on very slow parts of the profile, they are subject to substantial variations in measuring them both on the engine or on test fixtures.

The 251 engines do not have timing disks to discourage the attempts to find and report inaccurate valve opening and closing points (graduated timing disks are also expensive). To check correct relation of cams to crankshaft, all camshaft assembly drawings give, with tolerance, the push rod lift, also fuel tappet lift, when the crankshaft is set at top center of the cylinder being examined. Top center, and fuel injection points are marked on flywheels or fans; if neither is accessible marked camshaft gears are used.

The fuel cam adopted in 1954 and still in use in 1972 was developed in several stages, it was desired to close the ports at rising velocity and to open them while velocity was still high, and of course to minimize Hertz stresses. The pertinent dynamics are shown on 20 LYA 103. The initial part of the profile has a reentrant radius of curvature.

The profiles discussed above are those used on production engines. On the next page is a tabulation showing profiles and events also typical drawings. On line 1 the camshafts of the first 251 production engine is listed together profiles and graphs; they are earlier but very similar forms of the parabolic air and exhaust cam, the fuel cam has a slightly smaller nose radius. Lines 1 thru 5 cover all 251 production engines to date. The succeeding lines show various experimental arrangements, but not those made with loose camshafts.

Note that all graphs are based on 1000 RPM engine speeds; also spring force relation. Unless specifically marked, they refer to the cam roller; the air and exhaust valve lift ratio is 1.28, the fuel lever ratio is 1.33.

FUEL CAM TIMING

Disregarding the early shaft for the 251-A engine, all fuel cams up to the adoption of the 140° valve overlap shafts were timed so that port closing of the fuel pump at .1095" lift from the base circle would occur at 25-1/2 degrees before T.C. On all shafts with 140° overlap, the fuel cams were advanced 1 cam degree so that port closing lift occurs now at 27-1/2 degree before T.C.

In the past, injection timing was often advanced to 27-1/2 degrees as the maximum advance with sufficient margin for the pump plunger not to strike the delivery valve seat. With all 140° overlap shafts, the same margin will be obtained with 29-1/2° timing.

TABULATION OF TYPICAL CAMSHAFT ASSEMBLIES, TIMING PROFILES, APPLICATION

	Used On Engines	PROFILE AND CURVE NUMBERS Fuel	Exhaust	Air	Fuel Port Closing At Lift, Deg TC	EXHAUST Opens Before BC	EXHAUST Closes After TC	AIR Opens Before TC	AIR Closes After BC	Typical Assembly	Typical Section	Approx. Year Introd.	Date Super-Sedure	Remarks
1	6A	16-A-72021 20 LYA 98	16-A-72015 20 LYA 85	16-A-72015 20 LYA 85	.210" 25.5°	41°	53°	53°	41°	16-B-71023	16-B-73072	1952	1954	No after-cooler
2	6E, 12B&C 16B,C,E	16-A-72028 20 LYA 103	16-A-72024 20 LYA 107	16-A-72023 20 LYA 107	.1095" 25.5°	41°28'	60°	53°48'	29°40'	16-B-71030 16-B-71032	16-B-73100	1954 1955		
3	6D	16-A-72028 20 LYA 103	16-A-72036 20 LYA 103 Long Dwell	16-A-72023 20 LYA 107	.1095" 25.5°	57°28	60°	63°48	29°40	16-B-71048	16-B-73195	1963		
4	16F 16E	16-A-72028 20 LYA 103	16-A-72036 20 LYA 107 Long Dwell	16-EE-72036 20 LYA 107 Long Dwell	.1095" 27.5°	57°28'	60°	80	37°28'	16-B-71052 -1&2	16-B-73211 -1	April 1967	Oct. 1967	
5	6F, 8F, 12E,F 16EF, 18F	16-A-72028 20 LYA 103	16-A-72036 20 LYA 107 Long Dwell	16-EE-72048 20 LYA 126 and 107	.1095" 27.5°	57°28'	60°	80	37°28'	16-B-71052 -3&4	16-B-73211 -2	Oct. 1967		MEL
6	6B,D 16B	16-A-72028 20 LYA 103	16-A-72035 20 LYA 126 Polynomial	16-A-72023 20 LYA 107	.1095" 25.5°	41°28	60°	63°48'	29°40'	98-B-16512 1&2	98-B-16516	Testing 1960	Only 1963	3359 3375J 3384
7	6D	16-A-72028 20 LYA 103	16-A-72036 20 LYA 103 Long Dwell	98-A-16531 Slow Ramp	.1095" 25.5°	57°28'	60°	63°48	48°38'	98-B-16532	98-B-16533	Testing 1963	Only	3375P
8	16	98-EE-32777 98-EE-32776	16-A-72036	16-EE-72048		57°28'	60°	80	37°28'	98-B-16573 1&2	98-B-16570	Testing 1968-69	Only	3524S last page
9														
10														
11														

Firing Order

The camshaft assembly drawings list the firing orders of the various engines. On all Vee engine, right and left bank fire consecutively except for the eight cylinder where the banks fire non-consecutively. The firing orders were established primarily on the basis of torsional vibration requirements, although on the 18 cylinder distribution of cylinder discharges into the exhaust pipes of a simple manifold design was considered important.

On the 16 cylinder the firing order was changed in 1961, that is some 6 years after the first engines had been put into service. The change was made on all engines then in existence to reduce torsional vibrations of the crankshaft and related failures in cylinder block serrations. See discussion cylinder block.

This was one of the rare cases where a major change can be affected with great ease. All that was necessary was to rotate the two middle camshaft sections, cylinders 3, 4, 5, 6 relative to the outer sections 1, 2, 7, 8 by 180°. The indicating holes in the four flanges had to be redrilled and the old holes plugged; this is the reason why some of the indicating holes on the 16 cylinder camshafts are now not in line.

The firing orders for all production engine are listed:

6 Cyl 1, 4, 2, 6, 3, 5
8 Cyl 1R, 4L, 2R, 3L, 4R, 1L, 3R, 2L
12 Cyl 1R, 1L, 4R, 4L, 2R, 2L, 6R, 6L, 3R, 3L, 5R, 5L
16 Cyl original FO: 1R,1L, 5R, 5L, 7R, 7L, 3R, 3L, 8R, 8L, 4R, 4L, 2R, 2L, 6R, 6L
16 Cyl current since 1961: 1R, 1L, 4R, 4L, 7R, 7L, 6R, 6L, 8R, 8L, 5R, 5L, 2R, 2L, 3R, 3L
18 Cyl 1R, 1L, 8R, 8L, 5R, 5L, 3R, 3L, 9R, 9L, 6R, 6L, 2R, 2L, 7R, 7L, 4R, 4L

POLYNOMINAL EXHAUST CAMS (REF 16-4)

Around 1960 exhaust valve sticking was a problem of epidemic proportions. One approach to combat this was the thought to make more spring force available to overcome friction by reducing the force required to counteract inertia. Other cam conditions should be changed as little as possible. Graph 20 LYA 125 shows how and to what degree this was accomplished. The adverse effect was a lowering of valve opening when approaching the highest lift.

A secondary objective was that the new profile should be so that existing camshafts could be reground to new profile; this objective was met.

The profile was worked out by more sophisticated mathematics than used up to that time. To design this profile one must be sure of knowing the required duration because any introduction of dwell either defeats the desideratum of not having abrupt changes in acceleration or the whole cam must be calculated. The fact that the individual who did develop this cam was not a regular employee and had become unavailable to Alco contributed to the later decision not to pursue it fully although one half of the profile is in current use on the air cam.

This exhaust cam profile was first tried on the 6B engine tabulation line 6 shows the pertinent data. At 210 MEP, 1028 RPM, the performance could be considered virtually a standoff with the production shaft, line 2, in table. This trial was followed up in 1962 by operation in the 16 cyl. lab engine where at 193 MEP, 1000 RPM, no performance difference was detected. In March 63 the decision was made not to adopt this cam in production. The shaft with these cams was continued in the lab engine til the end of 63. It appears that reasons not to pursue this profile included adoption or intended adoption of flat top pistons which would not bend the valves if they did strike, heavier valve springs, (46-D-72023, 30% more force) probability of need to increase cam length, the D engine then already used the longer duration exhaust cam, cost of changing all masters.

In the light of the current (1972) persistent complaints about valve lever bushing and shaft wear which definitely are related to spring forces, which in turn need to be high to match the increase in engine speed, reconsideration of cams using this design approach might prove worthwhile.

Under line 7 of the table a shaft is listed on which the closing ramp of the air cam was extended to reduce the ramp velocity from 0.0012"/cam degree to 0.0007"/cam degree. This was undertaken in the then current belief that the excessive air valve wear which was becoming objectionable with the 243 MEP at 1100 RPM on the 6D had something to do with valve seating. Performance was unchanged per MEL 3375-P. By recollection no improvement was achieved and no record has been found. With later interpretation of wear as being related to deflections between insert and valve under dry condition, the ramp change would not be likely to reduce wear.

CYCOIDAL FUEL CAM

The persistent recurrence of occasional fuel cam failures whose cause cannot be clearly established leads to the request to design a cam with more margin against failure under any conditions, that includes excess pressure due to plugged nozzles, jumping due to excess speeds, jumping due to deficiencies in the cam profile. The resultant development, started in 1967, of the cycloidal 66° cam profile 98-A-32776 based on theoretical considerations attempted to meet the following specifications:

a) The current cam has a reentrant portion of a 16" radius of curvature. This severely limits grinding speed since contact between wheel and profile becomes long. It is believed to tend to cause profile waviness which later initiates roller vibrations and jumping. It was therefore postulated that the new cam must show no reentrancy.

b) It is believed that abrupt changes in acceleration may induce false roller motion, the new cam therefore should have smooth acceleration.

c) Hertz stresses should be lowered.

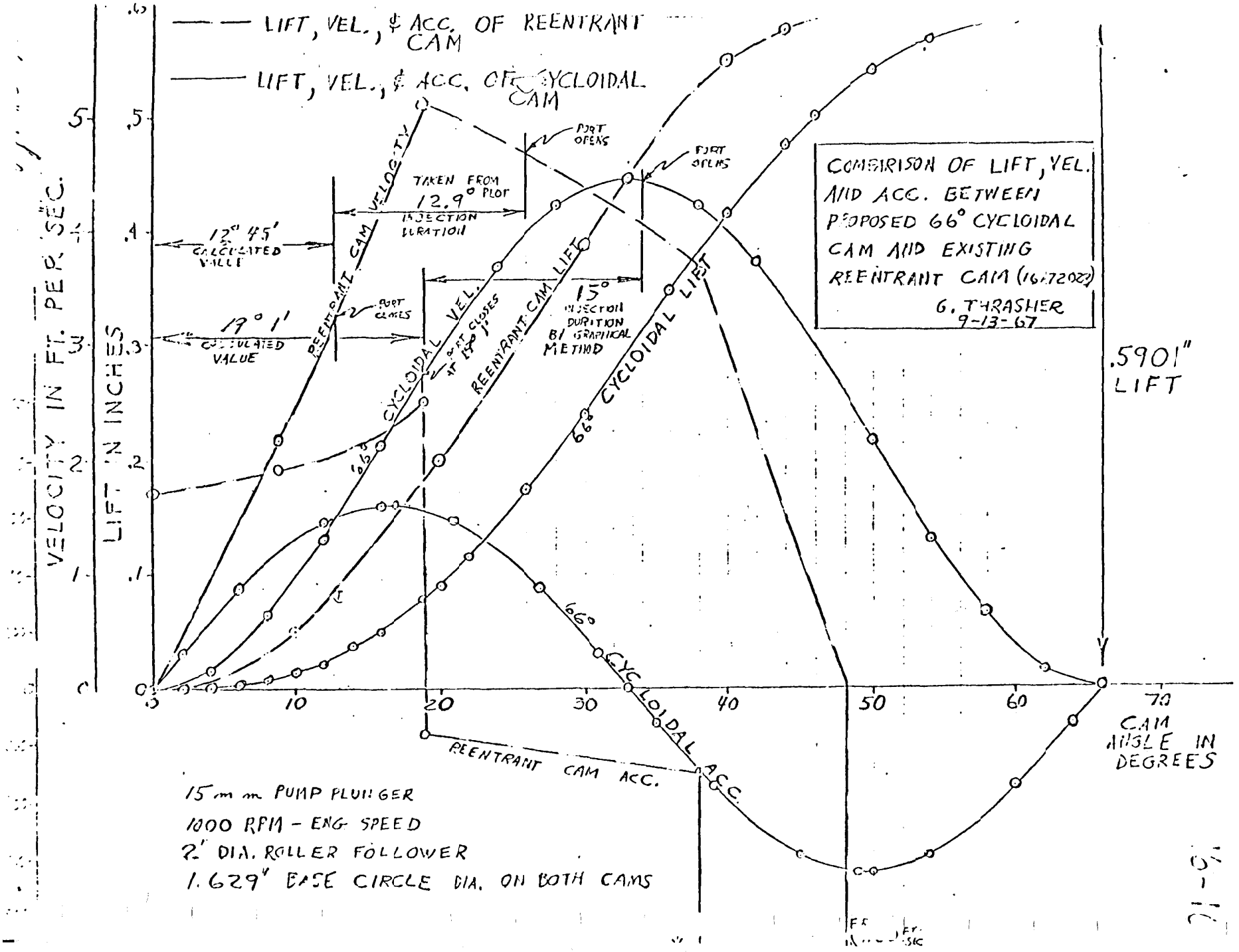
d) Injection characteristics should not be affected adversely.

The cam was computer designed, the motion curves compared with the curves of standard cam are shown in sketch 16-10. A cam to this profile was made and injection characteristics were determined by American Bosch in comparison to the current cam Ref 16-5. The new cam has slightly reduced line pressures and increase in injection duration of about 2°.

Engine test was conducted on the 16 cyl. engine up to 263 MEP, 1100 RPM. When replacing the cams without any other changes fuel consumption increased by 3% smoke worsened by one point. When changing the 17 mm pump and .375 mm nozzles to 18 mm and .400 nozzles, original performance was virtually restored. The same conditions obtained at reduced loads and speeds. The need to use larger pumps to retain engine performance offset any gain in Hertz stress due to the "kinder" profile.

Reduction of laboratory personnel forced discontinuation of testing. MLW at this time complained of fuel cam damage, which appeared to be caused by rollers sliding on cams and roughening them up, not spalling. The sliding probably was due to excessive overspeeds during transitions and allegedly disappeared when later changes in electrical controls prevented the overspeeding. At the time of the complaint MLW accepted our suggestion to install the camshaft with the special fuel cams for a field test. The shaft was sent, MLW changed their mind and the shaft was returned to Alco Auburn early 1972.

The fact that the new cam increases fuel consumption by 3% if no other changes are made should not be considered sufficient reason to scrap the program. A comparison of profile surfaces by Tallyround can indicate if waviness has been improved. If the fuel pump drive mechanism is instrumented with strain gages, comparison of the cams may show a gain even if the larger pump must be installed.


LIFT, VEL., & ACC. OF REENTRANT CAM
LIFT, VEL., & ACC. OF CYCLOIDAL CAM
VELOCITY IN FT. PER SEC.
LIFT IN INCHES
COMPARISON OF LIFT, VEL. AND ACC. BETWEEN PROPOSED 66° CYCLOIDAL CAM AND EXISTING REENTRANT CAM (1647202)
G. THRASHER
9-13-67
.5901" LIFT
CAM ANGLE IN DEGREES
REENTRANT CAM ACC.
66° CYCLOIDAL ACC
66° CYCLOIDAL LIFT
REENTRANT CAM LIFT
CYCLOIDAL VEL.
REENTRANT CAM VELOCITY
PORT OPENS
PORT CLOSES
TAKEN FROM 12.9° PLOT
INJECTION DURATION
12° 45' CALCULATED VALUE
19° 1' CALCULATED VALUE
15° INJECTION DURATION BY GRAPHICAL METHOD
15 m m PUMP PLUNGER
1000 RPM - ENG SPEED
2" DIA. ROLLER FOLLOWER
1.629" BASE CIRCLE DIA. ON BOTH CAMS

CAMSHAFT TORSIONAL STIFFNESS

The joint of the earliest camshafts, 6-cylinder with 3 cylinder long sections, example 16-B-71028, used long stud bolts with holes passing the hardened bearing area. Aside from too long expensive drilling, it showed that holes near the surface interfere badly with the induc tion hardening process. This lesson was observed in the latest design illustrated on 16-B-71052.

While even the first camshafts were designed primarily with bending stiffness in mind, various torsiograph results seemed to emphasized the desirability for maximum torsional stiffness.

The 12-D experimental engine (about 1961) for which the camshafts were designed for maximum stiffness, regardless of availability of tooling (multiple spindle drill) and practicality of hardening, seemed to show the possibility of substantial reduction of amplitudes of torsional vibrations; ref. 16-8. Actually the 12-cylinder camshafts had never experienced gear failures due to torsional vibrations. But with these test results as inducement, the 16-cylinder shaft and later all other shafts were redesigned for maximum practicable stiffness. The difference in the joints shows up on assembly drawings 16-B-71045 of original flanged joint with simple bolting, and 16-B-71052 with joints for increased stiffness using special nuts to permit minimum wrench clearance. For interchangeability, the original stud bolts with Class 3 thread on both ends are used as studs requiring Locktite. A comparison of sections 16-A-73174 original and 16-B-73214 also shows that the length of the 3" dia. has been shortened in favor of the full 4-1/2 dia. in two places by 19/32. Quoting from memory the torsional stiffness of a whole shaft was increased about 15%.

Under otherwise equal conditions, the stiffening of the camshafts, ref. 16-9, resulted in reducing the 8th order amplitude insignificantly from 0.181 to 0.160 degree half double amplitude, but brought the 8th order frequency from 900 to 1000 RPM. The 4th order amplitude has been reduced from about 0.33 to 0.20 degree half double amplitude in the operating speed range; but the 4th order vibration has never been considered to have caused gear failures. The decision to adopt the stiffened shaft, fell into a time when the vast improvement of gear strength and wear resistance by nitriding had not yet been proven, and when pressure of customer complaints made a recognizable change desirable. The stiffer shafts are fully interchangeable with any individual section of the older shafts. This comparison test which casts doubt on the wisdom of using the stiffer shafts was made with dampers at the camshaft ends. The stiffer shaft without a damper brings the peak of the 8th order safely away from 900 RPM synchronous speed as discussed later.

When fuel cam loading was increased, both, by use of larger plunger diameter pumps and adoption of 1100 RPM operating speeds, the fuel cam width was increased from 1-1/4 to 1-1/2. This was done in 1963 on all shafts regardless whether the wider roller was actually used or not.

LOOSE CAM CAMSHAFT

Much development work, especially determination of timing events, was done with camshafts on which cams of various profiles could be assembled. The angular location of these cams could be varied in the engine. The shaft design is shown on Layout 19-LYA-843 for the 16-cylinder engine. The laboratory had such a shaft also for 6-cylinder engine; for the 8-cylinder the shaft for the 16-cylinder was modified.

Originally the laboratory had a collection of cams, many of them made as tangential cams with different duration. There were also some cams with reduced lift which were tried in the 6-D engine with results so bad that no formal report seems to exist. It is known that many of these cams were inadvertently scrapped when dismantling the laboratory; but the shafts themselves exist with some cams attached. Cams can be made by sawing off and boring out scrapped production camshafts.

These shafts were last used in preliminary testing of 140° overlap on the 16-cylinder engines in early '67. Lab reports 3524-A and on 8-cylinder in 1968.

It was always laborious to develop the instructions to the laboratory to set particular cams at particular timing. A method to calculate the settings is given. Ref. 16-6 and 16-3.

Instructions for setting were then formalized in sketches such as 98-C-16428.

CAMSHAFT COST REDUCTION

Material is round stock requiring substantial stock removal by rough turning. Use of forgings was examined; to make sense the forgings would have to have profiles forged into them. The needed orientation and different length of forgings requires too many different forgings; chilled cast shafts, as well as nitrided shafts, were tried and rejected due to failures after very short running many years ago.

The only serious C.R. efforts attempted to reduce the enormous number of different shaft sections which we currently require. It is easy to count 54 different sections, needed for production and renewal parts, without even considering opposite rotation engines; which add another 37 different sections. It takes, under normal conditions, about three months to put a camshaft section thru the shop. It has always been necessary to start some excess material to protect against scrappage at a late operation. All this leads to the need of having an unproportionally high amount of labor and material tied up in camshaft production. While such costs are very real, they are hidden and do not show up in any cost estimates normally made and, therefore, handicap design attempts at cost reduction.

Any thought of cams shrunk on a thin shaft are inapplicable when taking torsional stiffness requirements previously discussed into consideration.

Considered was one basic approach, Ref. 16-7, illustrated on assembly 98-B-16544 and 98-A-16546. It is based on using one cam cluster; the same cluster would be used for any cylinder on any engine with the same direction of rotation (for the opposite rotation one other cluster is needed). Between the clusters, spacers with journal surfaces will be installed. The timing to be done by suitable orientation of dowel holes in spacers and clusters. Clusters and spacers have rabbet fits.

Spacers and clusters to be slipped over a long stud in the cylinder block. A hydraulic jack is used to stretch the shaft while tightening the one end nut. The same drawback of too many different longitudinal spacing conditions between clusters and bearings, which handicaps the current shafts, would require a large number of different spacer pieces.

Another drawback and the one causing rejection of this approach is the high cost of the many joints; which exceeded labor cost of current shafts (hidden cost remained hidden). From a functional point of view, it is expected that the central highly elongated stud can exert enough friction force to make the joints torsionally stiff.

CAMSHAFT SALVAGE

The only authorized salvage methods are shown on 50-B-73701 which allow for grinding a cam undersize .040" on the radius. They also allow restoring bearing surfaces by chrome plating.

It is known that outsiders (e.g. Chrome-Crankshaft Co.) offer cam restoration to Alco customers by hard facing of some sort. Alco trials found them completely unsuitable, see reports MET 3000 and MEL 3302.

Torsionally Stiffened
12 Cylinder, 251 Standard Rotation 140° - DRP-14792

	Drawing Number	Part Number
Assy., R.S.	16 T 71056-1	21600221
Sect., R.S.	16 B 73248-1	21610235
Sect., R.S.	16 B 73247-1	21610236
Sect., R.S.	16 B 73246-1	21610237
16 Studs		2151682
24 Nuts		2151375 ✓30
2 Dowels		2151054
1 Pipe Plug		16393715
Assy., L.S.	16 T 71056-2	21600125
Sect., L.S.	16 B 73251-1	21610140
Sect., L.S.	16 B 73250-1	21610141
Sect., L.S.	16 B 73249-1	21610142
16 Studs		2151682
24 Nuts		2151375 ✓30
2 Dowels		2151054
1 Pipe Plug		16393715

Torsionally Stiffened
12 Cylinder, 251-F Reverse Rotation 140° - DRP-15215

	Drawing Number	Part Number
Assy., R.S.	16 T 71064-1	21600230
Sect., R.S.	16 B 73283-1	21610252
Sect., R.S.	16 B 73282-1	21610251
Sect., R.S.	16 B 73281-1	21610250
16 Studs		2151682
24 Nuts		2151375 ✓30
2 Dowels		2151054
1 Pipe Plug		16393715
Assy., L.S.	16 T 71064-2	21600133
Sect., L.S.	16 B 73286-1	21610156
Sect., L.S.	16 B 73285-1	21610155
Sect., L.S.	16 B 73284-1	21610154
16 Studs		2151682
24 Nuts		2151375 ✓30
2 Dowels		2151054
1 Pipe Plug		16393715

Torsionally Stiffened
12 Cylinder, 251 Standard Rotation 123° - DRP-15501

	Drawing Number	Part Number
Assy., R.S.	16 T 71069-1	21600235
Sect., R.S.	16 B 73312-1	21610266
Sect., R.S.	16 B 73311-1	21610267
Sect., R.S.	16 B 73310-1	21610268
16 Studs		2151682
24 Nuts		21513753 ✓30
2 Dowels		2151054
1 Pipe Plug		16393715
Assy., L.S.	16 T 71069-2	21600136
Sect., L.S.	16 B 73309-1	21610164
Sect., L.S.	16 B 73308-1	21610165
Sect., L.S.	16 B 73307-1	21610166
16 Studs		2151682
24 Nuts		21513753 ✓30
2 Dowels		2151054
1 Pipe Plug		16393715

Torsionally Stiffened
12 Cylinder, 251 Reverse Rotation 123° - DRP-11607

	Drawing Number	Part Number
Assy., R.S.	16 T 71072-1	21600238
Sect., R.S.	16 B 73323-1	21610273
Sect., R.S.	16 B 73322-1	21610274
Sect., R.S.	16 B 73321-1	21610275
16 Studs		2151682
24 Nuts		21513753 ✓30
2 Dowels		2151054
1 Pipe Plug		16393715
Assy., L.S.	16 T 71072-2	21600139
Sect., L.S.	16 B 73326-1	21610171
Sect., L.S.	16 B 73325-1	21610172
Sect., L.S.	16 B 73324-1	21610173
16 Studs		2151682
24 Nuts		21513753 ✓30
2 Dowels		2151054
1 Pipe Plug		16393715

REFERENCES TO GROUP 16 CAMSHAFT

16-1 Special fuel cam spray drawing 32-EE-72248-4.

16-2 High carbon steel shafts, Project folders 863675 and 863678, Camshaft file 1970.

16-3 Cam calculations are preserved quite complete in two grey calculation binders marked - Cam Graphs, are filed in vault under grouping 20-LYA.

16-4 Polynomial exhaust cam, see Project folders 863162, 863181, MEL reports 3359, 3375J, and 3384.

16-5 Cycloidal fuel cam, Bosch report EJL5072 of 10-23-68, correspondence folder - Injection Equipment 1968, engine test MEL 3524B,last pages.

16-6 Loose cam camshaft, various instructions, former stock of cams- See Project folder 863571.

16-7 Cost Reduction Project P-8106, Estimate P-1307, Assembly 98-B-16544, Details 98-B-16538 thru 44; also 98-C-16545 and 98-A-16547.

16-8 Comparison of camshaft vibrations on 12-C engine, MEL 3294-B and on 12-D engine 3390-B.

16-9 Comparison of original and stiffened camshafts under equal conditions, on 16-cylinder engine MEL 3512-A and memo Freeman to Tromel 11-20-67, correspondence file.

16-10 12-Cylinder camshaft MEL-3294-B, 3297-A; 3549.

16-11 16-Cylinder Camshaft MEL 3294-A, Project 863413,Memo Freeman to Tromel 11-20-67. Zotta to E.C. Roberts 2-15-71 GO-58536; Ruiter to Smalley 2-23-71.

16-12 18-Cylinder camshaft MEL 3527, containing Freeman memos of 10/2 and 10/29/68.

16-13 Lateral vibration, Project file 863414.

16-14 (2) engines on American Commercial Barge Line, DE-35220, Engine SN 9917 and 9918.

CONNECTING ROD

The connecting rod design for 6" crankpin was developed from the previous rod with 7½" crankpin for the 244 engine which is very similar and which, by recollection, had been fatigue tested in tension and compression. The 6" pin design was subjected to brittle lacquer and a static strain gage test, Ref. 17-1. The imposed tensile load of 20,500 lbs. is still a good figure for 92 lbs. reciprocating weight at 1100 RPM but the compressive load of 62,600 lbs. used in the test is about a half of the loads currently imposed. The most highly stressed point due to tension is at the transition of the bolt head seat fillet to the side face. The drawing always carried a special rounding off instruction which, in the past, has been observed carefully in the shop. The higher firing pressures do increase the stress range, e.g. at gage #2 (shank stress) to about +5000 to -31000 which is not exactly low.

Over the years very few connecting rods have caused failures; in the beginning there were a few failures due to inadequate benching of the intersection between oil hole and wristpin bore. Rods in which a broken drill got stuck in the oil hole, which was then removed by an electric disintegrator, did fail by fracture in the zone affected by the heat. A couple of failures occurred about 4 or 5 years ago within a short space of time. In the oil hole of one rod a piece of round mild steel was found; on the other rod no foreign material was found but suspicious indications. It was believed that the assembly practice of bringing a pallet with a set of pistons and rods assembled so that the big end pointed upward and leaving it standing uncovered next to the partially assembled engine, made it possible to have foreign material enter the rods. A couple of more failures are remembered in which there were unexplained cracks in the shank at right angle to the exit, Ref. 17-a. The instances cited are very near to being all that occurred during the past 18 years on thousands of engines.

Every time a new forging supplier was introduced, controversies arose concerning forging weight. The drawing now specifically spells out that weight tolerances prevail over dimension balances.

Material specification, 31P-5100, include Izod impact test requirement: no one for years past remembered how this came into the specifications unless it is a carry-over from the side rods of steam locomotives. The virtual absence of connecting rod failures in spite of high stresses discouraged any changes in specifications.

The 251 rod from early production on, had a partial oil groove in the upper bearing shell which required ducting the oil to the center hole in the rod. For reasons of cost and difficulty of attaining burr-free intersection, a partial groove is milled in the bore rather than using holes drilled at same angles. The original groove was 5/16 wide, which did lead to deforming of the back of the shell into the groove, and would presumably have led to fatigue failure of the lining. The groove width was, therefore, changed to 3/16; while the groove outline is now still visible in the bearing back, there is usually no noticeable deformation. The center hole was originally ½", was changed to 3/8" partly because manufacturing wanted it partly to give better support to the shell. When more oil flow was wanted for the steel cap piston, the 1/2 hole size was reintroduced.

While connecting rod production was still in Schenectady (1958), Industrial Engineering felt that a serrated joint would be less costly. Product Engineering felt that a serrated joint which could transmit side loads in both directions, would be more rigid. Several rods, per 17-A-71008, were made and operated in laboratory engines. With the move of production to Auburn, interest in the serrated joint disappeared and the whole project was dropped, but some of the rods accumulated many hours of operation in laboratory engines.

The significant point of this trial is that the serrations were not 60° but 90°. It was felt that 90° serrations would have less tendency to yield and close in, especially when the joint area is as limited as it is on the connecting rod. While these rods were not particularly watched, they were installed and removed many times. They were finally scrapped when the lab was dismantled in 1970. These rods never caused any complaints. If any future development should require serrations, it is strongly recommended that 90° serrations be considered and evaluated.

CONNECTING ROD BOLT

The thread of the bolt is tapered so that the pitch diameter near the nut seat is smaller than at the cotter pin end, thereby causing a more even distribution of load on the threads. No primary bolt failures seem to occur, unless bolts were not properly tightened. Conforming to Alco practice, highly stressed bolts are not heat treated to give exceptionally high tensile strength but rather a substantial elongation, and hardness suitable to thread rolling. See specification 86800 ac. Con rod bolts once were returned by a customer who claimed that they were grossly deviating from specified dimensions. They actually were 7/16" too long. A trial on new bolts showed that they could be stretched that amount without breakage with normal wrench.

Up to closing the Schenectady shops, the bolts were made there of hot upset forgings. Since that time negotiations and trial concerning "warm" cold upset forgings have been going on; they are approved in principle; it is not known if any production shipment could be accepted to date (March 1972) Ref. 17-4.

WRIST PIN BUSHING

The wrist pin bushing is steel-backed bronze rolled from flat stock with interlocking joint. Originally two different bushings were used, one type for production had excess stock on the I.D. and was bored in place to a .001" tolerance, the other type was precision bored by the vendor and was used for field installation; of necessity it is subject to a larger tolerance .003" when assembled in the rod.

Many years later there was a serious complaint about piston skirt wear near the ends of the skirt due to rocking and a test with different wrist pin to rod bushing clearances was run in a laboratory engine. The evidence seemed to favor larger clearances and it was decided that the pre-bored bushing could be used safely in all instances. (There never had been any reports from field operation which would indicate that the "bored in place" bushing was superior). This decision also fell into the time when the shallow groove in the small end bore of the rod was introduced to increase oil flow. Boring the bushing in place presented the danger that chips might get into the groove, without reliable ways of getting them out again.

An early defect of the bushing was fatigue cracking near the oil hole in areas about 90° from the split line. This could be traced to inadequate contact between bushing OD and rod bore. This was due to the method of manufacturing the bushing. The hole was punched and did deform an adjacent kidney shaped area which was not fully supported but permitted minute flexing which led to the cracks. In forming the split bushing it was made round by closing a split round die around the bushing, with the bushing split placed at 90° from the die split. The die split again caused deformities leading to lack of support.

Remedial action was:

1. Drill the hole instead of punching

2. Push a hard mandrel through the bushing bore while the outside was confined in a ring.

3. Slightly increase interference in the rod bore.

Bushing thus made performed well until one case of bushings with fatigue cracks and questionable back contact occurred on a river boat in 1970. At that time a grinding operation was added to the bushing O.D.

Occasionally inaccuracies in sizes but more in parallelism or taper of the bores occur. They are not of large magnitude but outside of tolerances. On request of the shop, trials were made by brush plating the bore with nickel "DALIC" process. Two rods with plating thickness of .0015 were run in an engine 549 hours. A subsequent test of a rod plated .005" thick was terminated after 79 hours when the laboratory was dismantled. Engineering considers the process an acceptable salvage method. Ref. 17-5.

REFERENCES - CONNECTING ROD GROUP (17)

17-1 Stress determination MEL X 2114.

17-2 Failure Reports MET 3379, 3274, 3276.

17-3 Serrated joint rods - Project Folder 863370.

1 -4 Metallurgical Report on cold headed bolts - MET 6813.

17-5 Bore Salvage Project Folder - 801016.

February 29, 1972

CRANKSHAFT FOR 251 ENGINES

The basic design, 8 1/2" dia journals 6" dia. pins, 3/8 open fillets with hardened surfaces was never changed. Power and vibration torque is transmitted by friction at flanges, not by dowels.

Number of unexplained failures is extremely small. The theory of the design is discussed in Ref. 1, typical stress values are shown in Ref. 2. Shaft stresses are primarily influenced by firing pressures; some components of the bearing loads are strongly influenced by speed and counter weight arrangements.

MANUFACTURING PROCESSES

The shafts were made originally by National Forge by completely machining a slab forging. All such shafts were nitrided, their counter weights were hand welded to the webs .

Alternately and concurrently shafts especially 6 & 12-cylinder were made by Park Drop Forge from a closed die. Counterweights were resistance welded to the webs. About 100 16 cylinder shafts were made by resistance welding two drop forgings together in the center main journal. All machining as well as Tocco hardening (and joining the two 16-cyl values) was done by Ohio Crankshaft.

About 1962 National Forge introduced their press (Ref 3) which forges each throw consecutively, thereby becoming independent from limitation of die length.

MATERIALS

National Forge used 4130, Alco 31P5204, for nitrided shafts, while Park Drop used 5046, Alco 31P5203, for the Tocco hardened shafts. National Forge made a few hundred shafts of 5046 with Elotherm hardening, see appendix for details.

The eight cylinder (short) shafts are made of 4135 quenched and tempered 31P5207 to get higher core strengths, a few 16-cyl were tried, but straightening was too difficult. We now use 4137, Alco 31P5208, to meet special specification, these would be suitable for everything except unnecessary cost.

Originally 5046 Tocco hardened and 4130 nitrided were considered a virtual standoff; the Tocco hardened shaft lending themselves more readily to reconditioning by plating (no deterioration of properties due to removing a "compressive case") while nitrided shafts were slightly higher in cost and has about 10% higher endurance limit.

In recent years the 10% gain in endurance limit was considered mandatory for the higher rated engines and N.F. became virtually the sole supplier. To date, 1972, no other source of nitrided shafts of comparable size seems to exist anywhere.

STRAIGHTENING

All Alco shafts carry the notation NO COLD STRAIGHTENING PERMITTED. This instruction is believed to be an essential ingredient to the successful performance of these shafts. It is based on experiences with the preceding 244 engine shafts.

All straightening after heat treating must be done by hammer peening

which introduces only compressive stresses. This permits only limited straightening and is expensive.

Straightening by sledge hammer, which also introduces only compressive stresses was done at some of the salvage platersand Alco did not protest.

THRUST BEARING (Ref. 4)

The 251 engine design intended to insure the elimination of failures which had occurred on the earlier 244 engine. Therefore, the penalty in overall design of making the outermost and innermost webs thicker than the rest and of making a wider center main bearing, without oil grooves and oil drilling,was accepted. This actually led to the fact that the center main bearing is over designed; oil film thickness is about 3 times that of the other bearings.

Originally engine thrust was taken on two bronze half rings located in counterbores of the center main saddles. Severe longitudinal vibrations occurred on 16-cylinder marine engines (MSTS) with overhanging heavy generator. The thrust bearing was relocated at the last engine bearing which became an expensive flanged bearing in the saddle, the lower bearing remained plain on locomotive engines. Upper end lower shells were made flanged on all stationary and marine installations after s ome unforeseen thrust conditions from couplings were found.

It took a long time, especially on the 16-cylinder, to make the relocation of the thrust bearing effective. (Stock condition). It required increasing the collar next to the split gear to be increased in diameter from 9-3/4 to 10-1/2. Being an after thought, the thrust collar with 3/8 thickness is marginal. All shafts are still being machined to accept thrust bearing at the last bearing or thrust collars at the center bearing. The 12 and 16 cylinder blocks were machined both places for a long time, but this was discontinued as cost reduction around the middle sixties.

The 6-B engine is still being built with the less expensive half thrust collar at the center main bearing. On the 6-D with higher loads and heavier generator indications of longitudinal vibrations were found and the thrust bearing at #7 was adopted.

The collars are occasionally being cracked at crankshaft installation. If the shaft is not kept horizontal but tipped while it is being lifted, the collar may crack. (This was actually proven by doing it on purpose). No salvage method is known for a cracked collar nor of a deeply scored thrust face. Such shafts can be used in block which still have the center saddle machined for the thrust half collars.

Before crankshafts suitable for end thrust bearings could be procured, a number of marine and stationary installations were delivered in which the thrust was taken either in the generator or the cradle bearing, while the thrust half collars were made thin to provide only a stop during shipment or assembly. Note that end clearance of crank webs in block needed checking and length of center main bearing cap needed reducing.

On the LST engines the collar thickness was increased accepting an axial misalignment of the camshaft and crankshaft gear. On the eight and eighteen cylinder engines the coll is thickened and block dimensions were made to suit.

COUNTERWEIGHTS

6, 12 and 16-cylinder shafts were originally made with 4 counter weights welded to the center and end webs. Counter weights serve to reduce internal moments and bearing loads. After experiencing failures in the serrations of the main bearings, the counter weight shafts for the 16 and 12-cylinder engines were developed, see ref. 2 for magnitude of

load reduction. The 8 weight shafts are particularly important at speeds over 1000 RPM.

The 18-cylinder shaft was originally designed for 6 counter weights; resulting in the, at first, not recognized excessive load on the serration. Two stand-by engines were released with such shafts. After this, shafts were made with 18 weights of two different sizes. The original heavy weights needed for 1100 RPM engines brought a critical speed with a very large heavy generator to very near the 900 RPM synchronous speed. The weights were therefore reduced in mass, to get the critical speed away. The lighter counter weights are, however, considered insufficient for 1100 RPM engines.

The load considered critical is the horizontal component of the bearing load acting to shear off the serrations between bearing and cap. From experiences on the 16-cyl. engines, then operating at about 192 MEP, this load should be kept below 42,000 lbs.

STRESSES

PORTER'S Variousanalyses show stresses in terms of safety factors, see ref. CR5 for their discussion. By Alco experience this factor should not drop below 2. All 251 crankshafts in virtually all foreseen conditions have factors of 2 or slightly larger.

On the former 244 shafts this factor was slightly below 2 and crankshaft failures were frequent enough to be one serious inducement for starting the 251 design.

Bearing loadings are expressed in terms of an average minimum film thickness. Alco practice considers .26/1000 as about the minimum acceptable figure. Calculated film thicknesses near this minimum do occur at the #7 and #3 main bearings on the 16 cylinder engine. The engine does not have frequent bearing failures but if any failures occur #7 & #3 are the most likely position.

On the 244 engine connecting rod bearing which originally had an oil groove for the full circumference with a calculated average minimum film thickness of .162/1000 no satisfactory service was obtained until partial grooves were designed which raised this thickness to .398/1000. The 251 engine started out with partial oil grooves and has satisfactory bearing life.

Crank deflection specifications are based on bending stress in the shaft and load on the last bearing. The numbers specified are based on ref. CR6.

Oil Drilling

All holes intersect with the bearing surfaces at angles not too far from 90° to permit induction hardening without creating thin hardened areas likely to break out. The gradualness of hardness transition in the oil holes is important and defined in specifications. For nitriding these considerations are not applicable.

The oil holes are arranged to feed equally well in both directions of rotation.

The previously mentioned effort to optimise the center main bearing led to avoiding any oil drilling in this bearing (a short hole is drilled for the bearing shell removal tool). This resulted in feeding the pistons from the other bearings and in particular of feeding both number one pistons and also

the vibration damper from the first main bearing.

While it was long suspected that the number one pistons were being short changed in cooling oil, early tests of oil pressure in the main bearing grooves did not confirm this suspicion. Only when in the course of developing the steel cap pistons, oil flow meanvements were made it was proven that these pistons were starved. Consequently the oil pumps from the first bearing to the damper was plugged and the damper oil is now being supplied thru a lantern ring arrangement on the extension shaft. The 18 and 8 cylinder crankshaft design were made after this point was recognized. Damper oil is fed from #1 main, piston 1R & 1L are fed from #2 main. No wide center bearing exists on these shafts.

White Layer

The nitriding process is likely to produce a coating on the nitrided surfaces called white layer; its thickness is in the range of one mil, it is glass-like in hardness and transparency. It can not be distinguished visually. If left on journal bearing surfaces it will in time crack off and act as a sharp destructive grinding or lapping compound. The crankshaft manufacturer must remove any trace of the white layer by a lapping process.

The existence of white layer can be proven by swabbing the surface with copper sulphate solution, for details see crankshaft material specifications. If white layer exists the surface will stay clean, if no white layer exists the copper will plate out.

At one time a number of bearing failures occured (Santa Fe Railroad proven, suspected in other places) due white layer coming off some crankshafts. At that time Alco inspection included copper sulphate examination of bearing surfaces, so that undetected occurrance of white layer was alarming.

The reason was found to be: it is apparently a common shop trick when plating some steel with copper sulphate (for layout purposes or similar) to add a small amount of acid. This had been done innocently to the copper sulphate solution for detecting white layer. The small acid addition acted to breakdown the white layer and made the test meaningless.

SUPPLEMENT I --N.F. Induction Hardened Shafts

Around 1966 to 1968 National Forge intended to reduce cost by using 5046 Elotherm hardened shafts. Some 300 to 400 such shafts were produced. Failures occurred on such shafts primarily due to improper, sharp hardness transition and hardening cracks in oil holes, perhaps also because minor flaws which are harmless in nitrided shafts (nitriding puts a compressive layer around any exposed inclusion) were not detected by personnel used to work with nitrided shafts.

Hardening, heating cycles were changed, inspection procedures were sharpened and all shafts were finally accepted with warranty obligations extended to include some contingent damage. A confidential file is in Mr. A.L. Williams keeping. At this time, February, 1972, it appears that the precautions and corrections then made have been effective and that no rash of failures developed. Since then no more Elotherm hardened shafts were accepted.

SUPPLEMENT #2 - LST SHAFTS

The shafts for LST engines (8 and 16) have part of the OD of the large flange nitrided and polished to provide a suitable surface for a lip seal. The bronze housing for the lip seal serves also as an abutment for radial deflection during high shock; this is the reason for the excessive thickness of the bronze casting. During shock test on the 8-cylinder, this abutment was substantially deformed, thereby, proving it that it is needed.

The LST shafts can be used interchangeably with other shafts, but not vice versa.

SUPPLEMENT #3 Salvage of Crankshafts

Alco did at one time chrome plate shafts for salvaging. Specification 31-P-5205 covers the procedure which was then used. It is understood that Precision Engineering, who is now doing the salvage plating, follows this method more or less. If licensees should want instructions these specifications should be reviewed, but are close to being correct.

Before chrome plating became fully acceptable, Alco used to grind shafts undersize and furnish undersize bearings. This was discontinued because bearings bought in small quantities and kept in stock for years proved to be uneconomical.

Due to the inability to salvage damaged thrust collars, a thrust bearing with oversize length was frequently discussed but never actually made.

SUPPLEMENT #4 - "UNFINISHED" SHAFTS

To take advantage of lower import duties into Canada, some shafts have the holes in the free end flange drilled but not tapped. This makes them "unfinished" by customs definition. They are identified by special line numbers.

GENERATOR CONNECTION, BASIC

The currently used mounting of overhung generators to the engine is an outgrowth of the design originally developed for the 244 engine but with significant departures. The typical original assembly is shown on 19-A-73009. Cylinder block and base were bolted to a large casing which contained a compartment for the camshaft gear drive; another compartment serving as bell housing piloting the generator, as air scroll for the generator cooling fan, and as support for the oil catcher. The casing also embodie the mounting pads for the back end of the engine including chocking areas and means to receive the flange bolting of the generator.

After at first designing and building one prototype generator to fit the original design concepts for the 251 Vee engines, that is a generator with integral mounting brackets, as it became the production version for the 6-B, market reaction (railroads) insisted that the new engine should be able to be coupled with their existing generators. The 251 design was worked out accordingly, changes which had to be made to the generators were: opening up some of the flange stud holes to accept 1-3/8 bolts jig drilling and reaming two holes for dowels in accurate relation to the pilot fit of the generator, redrilling and tapping the holes in the rotor to accept 1-1/4 bolts. It was also necessary to change out the generator fan, if this still had a 3/8" thick back plate. A few holes for attaching air duct work had to be drilled. 50-C-73284 shows the changes needed, this reworking is usually done in G.E. service shops and is still occasionally required.

While the pilot fit on the 244 generator adapter insured alignment of generator and engine, the accurately maintained location of the two dowel holes in block, generator and support spacer have maintained alignment of generators and 251 Vee engines in thousands of units with hardly any difficulties. The originally daring design to hang a heavy generator on three individual not interconnected spacers and a few bolts has been proven reliable and no single case of loosening, even in collisions of locomotive: has been reported.

Another departure from then usual practice was the omission of dowels or shear bushings from the joint between crankshaft and rotor.

Print 19-EE-73171 is used a sreference for the discussion below. The mounting was developed by Engineering cooperation between Alco and G.E. Both,the joint of the rotating parts as well as that of the stationary parts had to be sufficient to transmit by friction a flashover torque of 284,000 lbs. inch in addition to engine torque without slipping. The connection of the stationary members also had to carry the weight of the overhung generator about 12000, with an acceleration factor of 2.5 to 3 g. In all calculations the dowels were considered locators, and not included in the strength calculations. Ref. 19-7.

The original generator rotor joint was based on 12 studs 1-1/4 dia. and the pilot fit. In the new application 6 bolts 1-3/8 diameter and two 1-1/4 diameter straight dowels do the same job. The location of the 4 uppermost studs matched the heavy top deck section of the cylinder block. A spacer bracket, originally welded now cast, makes up the space between the end of the block and the generator flange. This bracket is held during assembly with two 5/8 capscrews to the block; once the assembly is completed, long generator mounting bolts, not studs, and the straight dowels pass, thru generator flange, then spacer past a long counterbore into the female threads in the block. The hollow dowels are kept from coming out by long capscrews threaded into the block.

On the lower joint only two of the original 6 stud locations are used. The reason for this is that the width between portals of the Alco special purpose planer, it can plane four 12-cylinder or three 16-cylinder bases at the same time; limits the width

of the bases to within less than one inch of the 55" dimension over the feet of the weldment. The feet on the base include a block of steel large enough to accept the thread for the generator capscrew and the engine mounting bolt. The foot also leaves exposed a flat area below the generator flange which is used as one of the surfaces against which the chocks against logitudinal motion of the engine abut. The spacers between base and lower generator flange is now made integral with sheet metal shields bolted to base and cylinder block to cover an exposed part of the generator fan and to provide mounting areas either for air ducts, used with engines inside of streamliner cabs, or for expanded metal protective guards. Adequate unrestricted passages for generator cooling air was one of G.E.'s insistent demands.

Proper crankshaft deflection is attained by raising the generator bearing above the crankshaft bore. For this purpose 3/32 thick shims are permanently welded to the two lower feet of the generator, while an adjustable shim is sandwiched between upper generator flange and spacer blocks. By reducing the upper shim thickness by 1/32 or somewhat more, the generator is slightly tilted and the angle of shaft deflection due to rotor weight is made zero at the last crankshaft bearing.

The reason for using capscrews rather than studs for generator mounting is: when attaching generator to engine, thegenerator is suspended from the crane. Two headless dummy pins are screwed into the rotor flange. The rotor rests on fiber shims in the stator. By maneuvering the crane the dummy pins are entered into crankshaft flange holes, a few bolts are entered and the joint is made finger tight. Now only are the long entrance tapers on the dowels at the upper spacer with crane maneuvering used to line up stator to engine. If studs were used, maneuvering of the suspended generator would be too much restricted.

From a strength standpoint and reduction of probability of thread damage, studs would be preferable. Damaged internal threads are not completely unknown and the block salvage drawing includes instructions to bush such holes. The hardness of the top deck plate of the cylinder block has been increased in recent years to improve thread conditions at this point as well as the head studs. Except for MLW complaints, this has not been a serious deficiency.

The torque value of 1150 ft. lbs. on these bolts is an experience figure. It was 700 ft. lbs. originally, but it was found that crankshaft deflection would change slightly when lifting engines from the assembly line to the test bed and then again when mounting them on the chassis. Trials with raising the torque to different values established the 1150 value as sufficient to keep deflection unchanged when lifting the engines. With the size restrictions on the wrench, this torque value is rather high and needs care so men do not slip or wrenches break; one or two minor mishaps occurred in the shops.

The torque instructions for the bolts between crankshaft and rotor of tightening until the crankshaft turns appear strange. This torquing must be done thru the very small opening in the base and use of a torque wrench is out of the question. 850 ft. lbs. torque was the design figure and specified for many years, it caused innumerable questions. It does take about the weight of a man, say 150 lbs., on a 6 ft. lever to break the assembled engine loose, so the crude torque instruction approaches design intent . The same instruction is used on the 6-cylinder engine, where the tightenin torque, of course, becomes smaller, but so does the torque transmission requirement.

MEMO TO FILE:

This memo is written in response to A/R No. A0264157 to provide clarification regarding the measurement of diesel engine generator (DEG) crankshaft deflection. This memo summarizes information obtained from conversations with representatives of GE Locomotive, the DEG supplier.

ALCO maintenance instruction MI-11069F, contained in the DEG Vendor Instruction Manuals (DC-663082-80 and DC-663082-81), describes the removal, inspection, maintenance, and installation of the DEG crankshaft. This MI also provides guidance for measuring crankshaft deflection after crankshaft installation.

Ideally, crankshaft deflection would be zero inches. In reality, however, this would be difficult to achieve and, therefore, a tolerance is allowed. ALCO maintenance instruction MI-17219D and machinery arrangement drawings 19AA73275 and 30AA73162 state that the maximum allowable crankshaft deflection is $\pm$.0015 inches. This tolerance applies during initial alignment of a unit after recoupling its engine and generator. If this tolerance is exceeded, MI-11069F provides instruction to correct the condition by applying or removing shims at the top bolting pads of the generator.

Subsequent measurements of crankshaft deflection may differ from measurements taken during initial alignment due to settling of the unit. Therefore, crankshaft deflection of up to $\pm$.002 inches is allowed for a previously aligned unit to allow for settling (Ref. Chron 177572). The increased deflection tolerance is applicable to previously aligned units which have not been subsequently uncoupled.

Crankshaft deflection will vary depending on the condition of the engine (cold or hot). Deflection limits specified by ALCO are based on cold DEG conditions (see attachment 1). After recoupling, when the generator is initially aligned to the engine, cold crankshaft deflection is measured. A DEG in the cold condition means that there is no preheating and circulation of engine jacket water and lube oil. On the other hand, hot crankshaft deflection would be measured after a unit has been operated and brought up to normal operating temperatures. Once DEG temperatures have stabilized at normal operating levels, a normal shutdown of the DEG is performed and hot crankshaft deflection is measured. It should be noted that there is no vendor specification which requires the measurement of crankshaft deflection with the DEG hot. Hot crankshaft deflection can be measured and trended if desired but no limits have been specified for hot crankshaft deflection. As long as cold crankshaft deflection is maintained within the vendor's specifications satisfactory operation of the DEG is anticipated.

Currently, the crankshaft deflection tolerance specified in the Vendor Instructional Manuals (DC-663082-80 and DC-663082-81) concurs with machinery arrangement drawings 19AA73275 and 30AA73162. The level of detail provided in MI-11069F regarding the measurement of crankshaft deflection is consistent with the level of detail provided in other maintenance instructions. Therefore, NECSME concludes that revision of the vendor manuals is not required to clarify crankshaft deflection measurement.

1 of 1

MEMO TO FILE:

This memo supplements information provided in an earlier memo on the same subject (Ref. Memo to File 129.10, dated 6/25/92, Chron #191965). Collectively, these memos address A/R No. A0264157 which requests information concerning the measurement of diesel engine generator (DEG) crankshaft deflection. The information contained in this memo is based on telephone conversations with representatives of GE Locomotives, the DEG supplier.

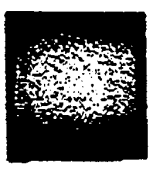
DEG crankshaft deflection readings are to be recorded when the DEG is cold. This means that the following conditions should exist:

1. There is no preheating and circulation of engine jacket water.

2. There is no preheating and circulation of engine lube oil.

3. The generator space heaters are off.

The following information clarifies the steps outlined in ALCO maintenance instruction MI-11069F (Ref. DEG Vendor Instruction Manuals DC-663082-80 and DC-663082-81) for measuring crankshaft deflection. The deflection gauge is positioned at the center punch marks between the webs of the crankpin nearest the power take-off end of the shaft. The deflection gauge must be positioned parallel to the shaft. With the deflection gauge positioned at the bottom center of the shaft and the gauge's dial indicator set to zero, the shaft is rolled over. As the shaft is rolled, the deflection gauge travels up the side of the shaft. The dial indicator moves away from zero, in either the positive or negative direction, if there is misalignment between the engine and generator. As the deflection gauge travels up the side of the shaft, the dial indicator is monitored to determine the maximum positive or maximum negative reading. The maximum positive or negative dial indicator reading is recorded along with the corresponding shaft position. When shaft rotation reaches its limit, as determined by connecting rod interference, the shaft is then rolled over in the opposite



direction. As the shaft is rolled over in the opposite direction, the deflection gauge travels down the side of the shaft toward the bottom center position and the dial indicator approaches zero. The dial indicator should return to zero at the bottom center position. The deflection gauge travels up the other side of the shaft as shaft rotation is continued past the bottom center position. As the deflection gauge travels up the other side of the shaft, the dial indicator is monitored to determine the maximum positive or negative reading and the maximum reading is recorded along with the corresponding shaft position.

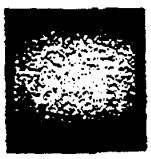
Utilizing the process described in the preceding paragraph, the direction and quantity of crankshaft deflection can be determined. The direction of crankshaft deflection, which depends on the actual configuration of vertical and/or horizontal misalignment between the engine and generator (i.e. generator is high and/or to the left of the engine, or vice versa), is related to the positive or negative sign associated with the dial indicator readings. For example, if the dial indicator reads positive as the deflection gauge travels toward the top of the shaft, the crankshaft webs are spreading apart. This indicates that the generator is sitting lower than the engine. The quantity of crankshaft deflection is obtained by taking the difference between maximum and minimum dial indicator readings. The following examples illustrate how the quantity of crankshaft deflection is determined from dial indicator readings:

1. The maximum dial indicator reading observed while the deflection gauge travels up one side of the shaft is +.001. As the deflection gauge travels up the other side of the shaft, a maximum negative (or minimum) reading of -.001 is observed on the dial indicator. Crankshaft deflection is the difference between the maximum and minimum dial indicator readings which is +.001 - (-.001), or .002.

2. The maximum dial indicator reading observed while the deflection gauge travels up one side of the shaft is +.001. The maximum reading observed while the dial indicator travels up the other side of the shaft is +.00125. In this situation the maximum reading is +.00125 and the minimum reading is zero (dial indicator reading at the bottom center position). Crankshaft deflection is .00125.

3. The maximum negative dial indicator reading observed while the deflection gauge travels up one side of the shaft is -.00075. The maximum negative reading observed while the dial indicator travels up the other side of the shaft is -.0015. In this situation the maximum reading is zero (dial indicator reading at the bottom center position) and the minimum reading is -.0015. Crankshaft deflection is .0015.

In accordance with ALCO maintenance instruction MI-17219D and machinery arrangement drawings 19AA73275 and 30AA73162, crankshaft deflection is not to exceed ±.0015 inches during initial alignment of a DEG after recoupling its engine and generator. For a previously aligned DEG which has not been subsequently uncoupled, crankshaft deflection is to be less than ±.002 inches (Ref. Chron 191964). These crankshaft deflection limits could be more simply stated as .0015 and .002 inches. The ± symbol in the vendor specified deflection limits simply represents the fact that the direction of crankshaft deflection will depend on the actual configuration of vertical and horizontal misalignment between the engine and generator.



crankshaft deflection measurement.

GENERATOR CONNECTION, VARIATIONS

As engine output was increased, generator weight increased also. In 1963 an increase from the 12000 lbs. range to about 18000 had to be accommodated. Friction force from the original bolting did not suffice anymore. An additional pair of bolts was required, preferably near the lower bolts. PRint #19-EE-74730 shows a typical arrangement. The previously mentioned limitation on base width prevented a bolt in the existing generator flange location to go thru into the base. The spacer however could receive such a bolt, and the spacer itself could be fastened more securely to the base by two smaller 1" dia. bolts. These bolts had to be countersunk to retain the flat abutting surface for the chocks. Note: The chocks were never made to conta a part of the generator, because it was considered mandatory to be able to remove the generator in the locomotive without disturbing the engine mounting.

Described above are the basic arrangements used on Vee engines. Special conditions exist on marine and stationary engines when generators were built to match engine conditions; such generators are extended to meet the end of the cylinder block without spacers. They may have one more or less uninterrupted flange to fit on the engine. Additional bolts are provided in the top, they anchor in a back-up block in the air manifold of the cylinder block. If matched with a marine base, bolts or taper dowels can be made to meet the base end plate between the base mounting feet.

GENERATOR CONNECTION 6-CYLINDER

When first designed no generator to 6-cylinder existed which would impose the interchangeability requirements prevailing on Vee engine generators. Brackets welded to the generator skirt extended beyond the fan and met the end face of the engine directly. Print 19-A-73033 illustrates the conditions. The size of the generator also matched the width of the base easily. The upper mounting bracket is high enough to permit placing the bolts in the top deck of the cylinder block. Undesirable was and is the fact that alignment of generator to engine is done by ream bolts in the base while the cylinder block establishes crankshaft location.

When MLW first started to build 6-cylinder engines, Canadian G.E. did not wish to tool up for the small GE-584 generator but wanted to use the basic configuration of the Vee engine generator such as the 581. An adaptation design, refer to print 19-EE-73138, was worked out involving a spacer on top; note use bolts with substantial stretch length. On the lower outrigger spacers were designed as steel castings. These spacers are to be assembled to the base, followed by reaming with a jig, not with the generator, to establish generator and engine alignment.

When the 6-D engine was introduced with a rating in excess of the capacity of the GE-584 generator, the design developed for MLW became the general 6-D design.

Several paper studies were made to incorporate the offset spacers into the base but none appeared feasible.

On the 6-cylinder engines misalignment between generator and engine cropped up at least twice thru shop carelessness. A fit involving bolts with excentric bushings was devised and used in the field (overseas!).

GENERATOR CONNECTION, VARIOUS AUXILIARY DETAILS

The oil catcher was originally tested (perhaps also developed) by G.E. for the 244 engine arrangement. It was taken over on the 251 at first without any changes and is being used in the same form on the 6-B engine. On the 244 the oil drain from the catcher was made as a real trap; there was the apprehension that, at a crankcase explosion all the base oil would be pushed into the generator. On the 251 the trap was eliminated but the drain was led in a channel carefully below the oil level. Whether or not this channel is needed, was never proven out because it would be difficult to fix up a channel in a complete engine if a trial without channel would prove negative. It could be tried if a pipe coupling were welded into the channel so that the hole could be plugged with a pipe plug if necessary. The thin wall channel has been a nuisance in making base weldments. Direct connection of the drain with the crankcase would tend to aggravate low crankcase vacuum complaints.

In time it was found that oil leaked between the end plates of base and block, this led to incorporating wings on the oil catcher castings covering this joint.

On the Vee engine only, probably due to the larger fan diameter, the back plate of the fan acted somewhat like a fan itself and sucked air and with it oil thru the catcher clearance out of the crankcase. To break this suction an additional annular chamber was built on the outside of the catcher, the chamber being vented thru a duct extending to about the O.D. of the fan. The duct is cut off so that any circulating air might tend to be caught and driven inwards.

Leakage complaints, at one time were very severe, but it is likely that the oil catcher was blamed for oil passing between block and base joint, oil leaking out of the Woodward governor, both, in service and when filling; note the oil tray under the governor, added when Woodward did not cooperate in getting the governor tight. The crankcase exhauster also is not reliably oil tight. Any leakage would be picked up by the air stream and make the whole area unsightly.

All locomotive generators until the advent of the A.C. alternators had electric starter windings used to start the engines. All these generators, except the largest D.C. machine the 598, had an integral cooling fan. Its original thickness of 3/8" had to be increased long ago to get out of axial vibrations to 5/8" thickness. This thickness permitted the machining in of a circumferential groove at a given distance from the machined O.D. of the fan. One side of the groove is slightly tapered. The claw-like end of bar 29-C-72070 can be entered in the groove and leaning against the fan O.D.'s can be used to rotate the engine in a motion reminiscent of a can opener. If the locomotive stands at a working platform so that a 4 to 6 ft. extension can be used on this turning bar, the device is simple and usable; if it has to be used inside a cab or from the runway next to the engine, it requires the strength of 2 men to move the 16-cylinder engine. To reduce this effort a treadle per 29-A-72233 was developed, It takes very little force to use it, but it is rather awkward to handle.

The large 598 DC generator had a separate blower for cooling air and therefore no fan disk. For this installation a barring gear engaging the camshaft gear has been developed; it also includes a timing pointer. Its use should be discouraged because it puts high stresses on the teeth of the camshaft gear.

When the AC alternators appeared, use of air starter motors became the universal mode of starting. The fans were equipped with ring gears. Originally they were of one piece, piloted on the fan O.D. and bolted on. Air pressures in locomotive

tanks were frequently low and starters were kept on grinding without the engine starting. This damaged the ring gear and led to the demand for easy replaceability. The fact that the fans had 33 vanes and that a third of a ring gear was more easily entered between the generator spacers, led to making the ring gears in three pieces. At 1100 RPM and a diameter of over 50", it was considered necessary to hold the gear segments by a tongue and groove construction against centrifugal force. The gear segments are fully interchangeable.

Once the ring gear became available, all barring is done by a small pinion mounted on the opposite engine side of the starter motor. There are two generations of the barring device, the earlier is mounted on an eccentric, it is engaged by rotating the eccentric, a weighted lever keeps the eccentric out of engagement when not needed (it does not disengage itself). The later device is engaged by moving pinion axially against a spring which always tends to pull it out of engagement.

Both starter motor and barring devices are mounted on suitably altered lower spacers between generator and base mounts.

Current A.C. generators used by MLW and G.E. are using one of the auxiliaries to start the engines, again electrically. Alco has not yet built such engines; if a ring gear is not required for starting, its expense can hardly be justified just for barring the engine. The camshaft gear engaging device is probably the sensible if undersirable choice.

Small hollow holes with suitable stamping; one for each pair of cylinders, are drilled in fan discs or ring gear circumferences and serve as timing marks. Suitably located, are double ended pointers, one end matching the indicator hole when the respective cylinders are at top center; the other at begin of injection. The distance between the two pointer ends depends on specified injection timing.

NON-LOCOMOTIVE APPLICATIONS

On generator sets with overhung generators, most of the preceding discussion applies in principle. The fan disc may be rather a flywheel but it will carry ring gear and timing marks. More than one starter may be used. Shim adjustment to get crankshaft deflection remains as described.

In many applications thin flywheels are mounted at the crankshaft flange and one half of a coupling is piloted and mounted on the flywheel, ring gear, starter, barring device follow principles described. In this case crankshaft deflection is non-adjustable and flywheel weight must be limited to permissible crank deflection.

Mentioned but not discussed because it was done too infrequently is the mounting of a substantial steel casting at the end of the base to provide a cradle bearing, a typical installation is shown on 19-EE-73081.

Alco uses generators, mostly locomotive types for loading and testing all engines. Engines destined for other applications still have all standard provisions to mount generator, even if needed only for testing.

CRANKSHAFT VIBRATION DAMPER

Since the early Sixties all Alco 251 engines have been equipped with the "paddle wheel" hydraulic vibration damper. This was developed by Alco in the Forties and used on 244 engines for sometime. As a cost reduction item it was replaced by the sealed Houdaille-Hershey dampers, about 1950. All the early 251 engines were built with H.H. dampers, but it is significant that all bases were so designed that room was left for installing the wider Alco damper. The H.H. damper consists of a housing rigidly connected to the crankshaft which forms an annular space with about 2 ft. OD; in it fits a ring of rectangular cross sections with relatively small clearance. There is a cover plate which closes the annular space, it is permanently fastened to the housing either by welding or rolling. The small clearance between housing and the ring, which serves as a flyweight mass is filled with a viscous Silicone fluids.

The damping is effected by the relative motion of and attendant fluid friction between the flyweight mass attempting to rotate at constant speed and the housing which is subject to the variable speed caused by crankshaft torsional vibrations.

There were several generations of H.H. dampers. At first it was a simple weldment per print 19-B-74284 and 19-B-74271 which (by recollection) seemed to work on Vee engines satisfactorily. On the in line engine seizures between flyweight and annulus occurred leading to changes from the mild steel stamped annulus to malluble iron castings, then to the introduction of bronze rings and bushings to prevent seizures on either the circular fits or the rides of the flyweight.

Except for a classical torsional failure of an eight cylinder in line crankshaft in the laboratory within weeks of starting to operate it, no torsional crankshaft failures in the field occurred. Work on the eight cylinder in line was soon after discontinued. The question was then often half seriously raised whether the dampers were actually effective or superfluous. There was no way by which to determine whether a damper was operative. H.H. put every new damper prior to shipment to Alco on a vibration tester and obtained a record of dampening characteristics. Alco established the practice of returning dampers at engine overhaul to H.H. for re-vibration testing. The new observations were then compared to the original records and they very rarely corresponded. At about the same time some serious accidents with H.H. dampers not on Alco engines, were reported to have occurred. Local excess temperature, due to seizures were said to have caused transformation of the Silicone fluid into highly explosive compounds. The various changes in construction, previously mentioned had raised the price of the dampers enough to counteract the original cost advantage over the Alco damper.

In the late Fifties the Alco "paddle wheel" hydraulic damper was re-evaluated and redesigned then soon reintroduced in production. Ref. 19-8 gives basic performance characteristics of the dampers, note in particular the insensitivity of the damper to changes in axial clearance between .031" and .0095".

Print 19-A-74400 is used as illustration. The damper consists of external gear surrounded by an internal gear of equal number of teeth and very large backlash. The fact that both the internal and external protinsious have the shape of gear teeth is irrelevant.

Using a Fellows gear shaper permitted accurate machining of the minor diameter of the external gear and of the major diameter of the internal gear. On the smaller vibration damper for the camshaft, which use the same design principles, the members are shaped as a spline with excessive circumferential clearance,

rather than gears since their size permitted accurate machining by broaching.

The large backlash between the outer and inner member is filled by lube oil fed into the central cavity from whence it passes by centrifugal force thru restricted holes into both sides of each gear tooth.

The external gear, called the spider is rigidly mounted on the crankshaft, sandwiched between crankshaft and extension shaft.

The internal gear together with the two disks bolted too it surrounds the spider and acts as flyweight. The flyweight attempts to rotate at constant speed while the spider follows the vibratory speed variations. Any relative motion between the two must be accompanied by increasing the volume of oil on one side of each tooth at the expense of the volume on the other side of the tooth. This displacement of oil can only be effected by the oil passing thru the radial and axial clearances between the teeth. Some of the oil may pass in and out of the radial holes in the spider. Oil which is squeezed out against centrifugal force between spider and flyweight rings, at the ring I.D. is replaced by feeding into the central cavity. This oil leakage is one limitation on the use of this damper since it requires it to operate inside the crankcase or a smililar oil space.

In the outer ring there are four tooth spaces with a small drilled hole in the middle. Their intended design function is to prevent the teeth of the flyweight from remaining in contact with the teeth of the spider. If the teeth were in contact it is intended that the leakage thru these four uncovered holes would cause oil pressure difference tending to center the teeth in the tooth space. Actual tests with these holes plugged showed no difference in damper performance than with the holes open as designed. They were omitted in the design of the camshaft dampers.

There were a few minor design changes. The bolts heads originally were in counter bores as now, for cost reduction the counter bores were changed to a turned relief. Then came the requirement for maximum flyweight mass, the counter bores were reintroducted and the diameter of the damper was increased to 24½ reducing clearance between turbo support and damper to a minimum. This required care in retrofitting the large dampers to older engines, some turbo supports had to be ground to give clearance. Replacing of old H.H. dampers by Alco dampers was done very frequently and occasional complaints of lack of clearance did arise. It is believed that virtually all H.H. dampers are by now replaced.

In production dampers and extension shafts are bolted to the crankshaft prior to setting the block onto the base. There is enough, but not more, clearance in the base to move extension shaft and gear axially to disengage it from the damper, provided the damper remains in its location on the crankshaft. After the extension shaft has been lifted the damper can also be removed. A combination of jackscrew and clearance holes helps in separating the parts in the sequence described. Turbo support and lub. and water pumps must have been removed before hand.

Materials used are cast iron for the spider, steel plate for all other parts. On dampers for high shock specifications the spider is made of ductile iron. Oil grooving on the spider is on one side only, but was found sufficient in trials.

Lube oil supply to the damper was investigated many years ago and found not to be very critical; on the early 244 engine oil was led at reduced pressure thru the extension shaft. When the H.H. were superseded tests were made to indicate whether or not it would be permissible to take oil from the number one main bearing for the damper in addition to the oil taken for the connecting rods and pistons of the number one crank.

This test was based on measurements of oil pressure fluctuations in the oil groove of the number one main bearing. The pressure readings indicated no substantial pressure loss and dampers as well as pistons were fed from the first main bearing for years. When oil flow measurements were started in connection with development of the steel cap pistons it was found that the pressure readings had been completely misleading and that the damper robbed most of the oil from the pistons on the first crank. Since the means of measuring the oil flow affected the flow itself, no clear flow figure can be given. It is believed that the damper reduces flow to the number one piston to perhaps one fourth.

To avoid inadequate cooling of the pistons on thefirst crank a design of separate oil supply to the damper print 19-A-74805, has been developed. It applies to 6D, 12, and 16 cylinder engines and can be retrofitted. Oil drilling in shafts for the 8 & 18 cylinder engines which were developed after the robbing of oil was recognized is arranged so that number one main bearing supplies oil to the dampers and number 2 main bearing supplies oil to the pistons on the first crank.

In contrast with the old H.H. damper it is very easy to determine whether an Alco damper is functional or not. By reaching into the crankcase one tries to rotate the damper flyweight either direction, if the damper is functional a total circumferential movement of about one inch can be attained, but it takes substantial force to move the damper.

engine design, however, does not permit locating balance shafts at the height of the crankshaft centerline. The balance shafts are therefore placed about 18" below the center line. This permits dropping the main bearing cap without interference with the balance shafts. The horizontal distance between the two shafts has been made as small as feasible, still maintaining the significant relation between the balance shaft and engine center lines. The final arrangement is illustrated in the right hand figure of sketch 19-1. While the horizontal components of engine free force and balancing forces are equal and in opposite direction, they are off set by the moment arm and a free secondary moment tending to rock the engine about its longitudinal axis has unfortunately been introduced.

It has been questioned whether or not the complication of the mechanically bothersome balance shaft arrangement is really necessary or if it could be omitted. The engine was therefore operated with and without balance shafts,both,on the laboratory test bed and also in a locomotive. Operation without balance shafts was unacceptably rough while operation with balance shafts was reasonably smooth; these trials were made at speeds up to 1100 RPM. General vibrations in excess of that found on other 251 engines did show up in that several failures of the generator brush holders occurred while operating the engine on the laboratory test bed; This normally is a very infrequent type of failure. These failures, general observations together with calculated high torsional stresses in the #4 crankpin when the engine was running at no load and 1100 RPM, led to the decision to limit operating speed of this engine to 1000 RPM. Over twenty such engines have been in operation in C-415 locomotives since 1966 at 1700 GHP 1000 RPM rating and no complaints on vibrations have been received. From the over 50 LST generator sets operating at 900 RPM, a few instances of vibration complaint were heard, but they seemed not to be systemic and rather due inadequate tightening of mounting bolts.

It is understood that eight cylinder engines in MLW built light locomotive chassis do experience excessive vibrations and that flexible mounts are now being worked on. The laboratory engine was operated without balance shafts but mounted on Korfund dampers; operation at all speeds and loads was surprisingly smooth, only at stopping and starting was a substantial torque reaction. (Quick movement at the base mounts was about 1/8"). All earlier balance shafts were calculated to match aluminum pistons, only the latest design with wide bearings is dimensioned to match the heavier steel cap pistons. No particular readings were taken when aluminum pistons were replaced by steel cap pistons but engine balance was not noticeably affected. For laboratory observations on vibrations, see Ref. 19-10. The current design of the right balance shaft represents about the maximum unbalance which can be machined in this size shaft and with the length available between bearings. Even so, the drive gears for these shafts are drilled to provide some additional unbalance which permits bringing the C.G. of the shaft unbalance into the center of the engine.

With oil drilling at the side of and at a right angle from the flats, also with oil supplied to a circumferential groove, the latest balance shaft arrangements are suitable for both directions of engine rotation.

Balance Shaft Design Execution

The balance shafts are made of the same material (SAE 1050 + some chrome) and general size relations as the camshafts; this permits use of existing stock and equipment such as drill jigs, induction coils. Originally the standard camshaft bearing bushings were used but these proved inadequate for the service. The balancing function is achieved by milling flats on the shafts between bearings and couplings. As shown in preceding chapter, the balance weight of the right side shaft must be very much larger than that of the left shaft. Therefore, the flat is milled deepest into the right shaft making it in overall weight, somewhat paradoxically, the lighter shaft. The bearing loads of this lighter, but more unbalanced, shafts are significant and much higher than those of the left shaft. Left shaft bearing failures do not occur unless as secondary failures caused by the, originally not infrequent, failures of the right shaft bearing, particularly the most heavily loaded bearing #4. For bearing loads aligned and misaligned, also bending stresses in the shafts, see Ref. 19-9. Bearing failures did occur and caused several redesigns of the affected components.

The main causes of these failures were believed to be related to insufficient rigidity of bearing supports in the base, unfavorable oil supply to the bearings, and excessive sensitivity of the bearings to dirt. As more failures occurred and were analyzed, it seemed that none of the design weaknesses alone would cause a failure unless dirt was also present. The earliest bases had the bearing blocks laid on the bulkheads and fillet welded to them. Under high bearing loads this would tend to tip the bearings causing corner loading; it was corrected by welding ribs, to bearing block, bulkhead and side plates. On the succeeding designs, both LST and current wide bearing design, the bearing blocks are inserted in the bulkheads to support the bearing loads centrally in the bulk heads. On the #1 bearing, which is not highly loaded, this feature is somewhat compromised.

Relocating the bearing blocks from lying on the bulk heads to being inserted in them changes the longitudinal location of the bearings and makes the balance shafts not interchangeable between the early locomotive and the LST engines.

The flexibility of the bulk heads and their influence on the bearings should not be underestimated as the following instance shows. All 251 bases originally had screens over the whole base, as the LST engines still have. For reasons of cost reduction the full screens were superseded by box shaped screens over the oil suction pipe opening. On the first execution of this box, the flange and 4 bolts fastening it to the bulkhead were 1/2" thick and 1/2 diameter respectively. Tightening this box to the bulkhead caused enough distortions to lock the balance shafts from turning although they turned freely when the box was not tightened. (The final box fastening was made more flimsy)

In arranging for lubrication of the balance shaft bearings both the very satisfactory arrangement of supplying oil thru the drilled camshaft was considered, as well as that a suitable oil wedge for a shaft with a rotating load would be created by a radial oil feed hole in the shaft, preceding the load at some optimum angle. It took a long time to find out what this optimum is under actual conditions rather than what hydrodynamic theory says it should be. The bearing supplier's recommendations, Clevite's, which had been obtained at every change were often quite wrong until much later when in 1971 one of their more service oriented engineers proved that the intersections of the oil holes in the shafts were inadequately benched, leaving small slivers which might break off and the holes themselves were inadequately cleaned of small metal chips so that new engines went out with a storage of potentially damaging dirt particles in the oil system.

Up to this time all shafts were drilled and oil was supplied to one end of these shafts. In section 35 covering gears the changing of the balance shaft drive from one end to the other and back again is discussed. These changes required several relocations of lub oil supply.

True to the principle of trying the simplest solution first and abandoning it only if it proves inadequate, the first design was a cap over the generator end of the balance shaft and leading oil into this cap. It resulted in thrust bearing loads of over 120 psi which these bearings did not accept. From then on short bronze tubes were pressed into the ends of the shafts; the tubes protrude into the bores of fittings into which oil is fed. To account for bearing clearances and misalignment of the fittings, the clearance between tube and fitting is rather large allowing oil escapement. A check of oil pressure in the central hole at the opposite end of the oil feed was made on an LST engine recently and proved that oil pressure there is fully adequate and need not be considered the point of concern which it was up to then.

The oil feeds were originally at the generator end. When the drive gear was mounted at the generator end of the right side shaft, the oil supply for this shaft only was relocated to the free end. Later on the LST engines with front end drive, the feed for both bearings went back to the generator end. Note that for some, not remembered, reason the bronze tubes are not interchangeable between the several shaft executions, but of different diameters.

The evolution of the oil feed in the bearings themselves is shown on Sketch 19-2. The original design figure 1 experienced an early failure on the lab engine. In this, as in nearly all later failures, the most heavily loaded #R4 bearing, scored, froze to the shaft and turned in the bore. The bore gets damaged and likely causes distress in #L4 bearing. The base had to be rebored oversize and bushed (salvage drawings for this procedure exist).

The failure was blamed on flexibility of the base, insufficiently tight fit of the bushing and inadequate oil supply. (The trailing oil hole was considered wasting oil in the wrong place). Disregarding the local detrimental effect on the oil wedge when the load passes the stationary oil hole, oil to the heavier loaded R.S. shaft was supplemented as shown on figure 2. The first batch of about 20 locomotive engines went out that way. Some failures, but not many did occur.

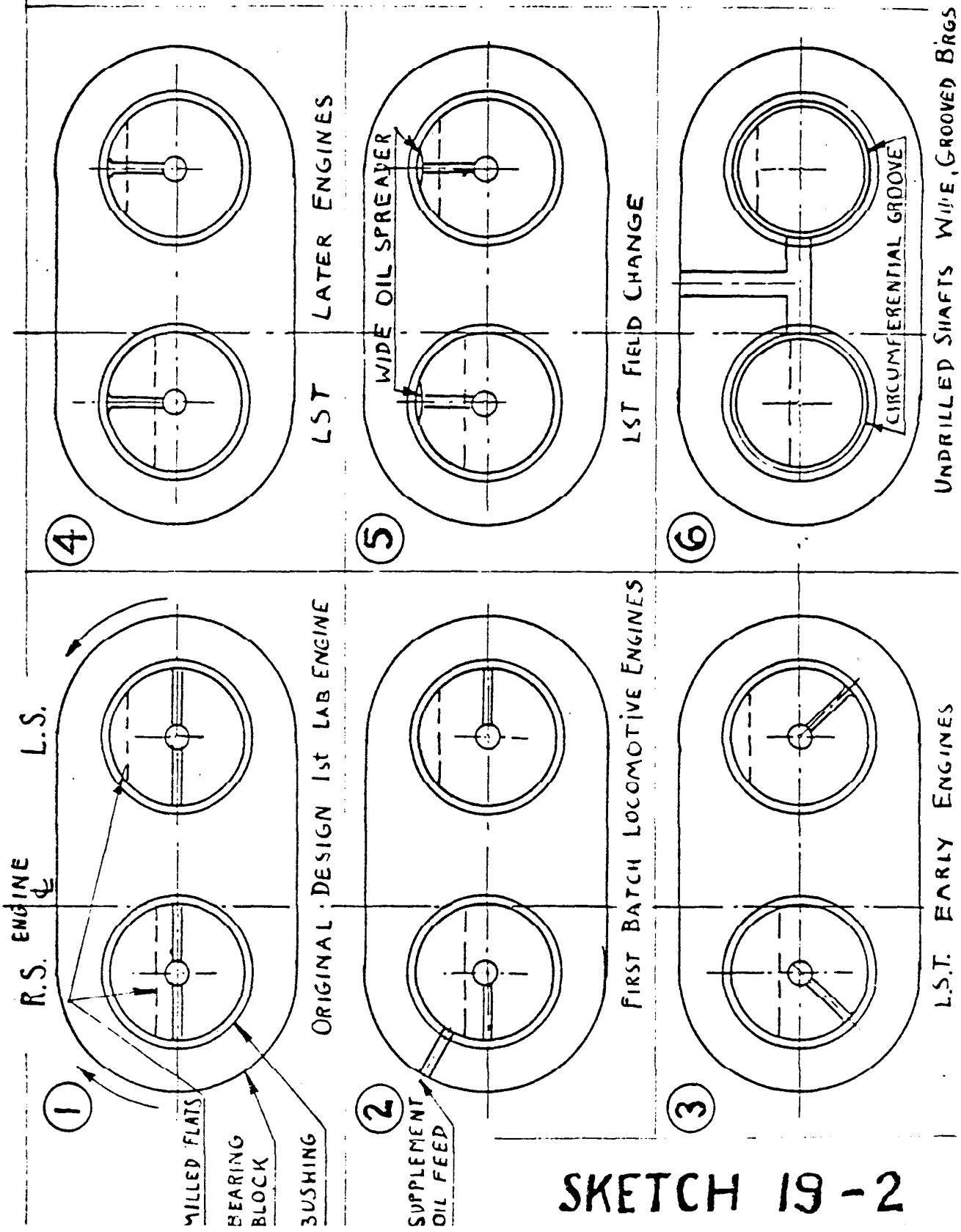
When the LST order was to be built, the bearing blocks were inserted. The stationary feed hole was considered basically wrong, so the feed hole was placed into the hydrodynamically optimum position figure 3. The bushings O.Ds. were ground to improve the fit in the bore. They were also installed with Locktite and material was changed from SAE-792 to the higher lead 794, in an effort to improve embeddability. By that time the importance of dirt had been recognized and extreme precautions for filtering the oil before it entered the shaft were taken, temporary run-in strainers between header and shaft feed were installed. Even so two failures occurred at Alco on engine test.

This led to another review with Clevite and special consultants, the result was drilling per figure 4. This was tried by operating an engine equipped with thermocouples in the bearings #4R successively with drilling per figure 3 and 4. Metal temperature was reduced some 20°F (by memory). The change in drilling was made but no reliable record when it actually took effect on the LST engines exists. Over half the engines of this order went out with the earlier drilling.

Several bearing failures occurred on LST engines after short time of operation; reliable assessment of the causes was made difficult by suspicion of having strainers by passed and similar conditions. On at least two engines all balance shaft bearings showed deep dirt scratches while main and connecting rod bearings showed no indication of dirt. At that point the inadequacy of cleaning and benching the shaft holes was discovered. A field reworking was initiated including cleaning of oil holes with wire

19-19

8 CYL BALANCE SHAFT BEARING
EVOLUTION OF OIL FEEDING SCHEMES



SKETCH 19-2

brushing and of milling large oil spreading depressions into the journals, figure 5. These depressions aim at two effects, first by spreading the oil for a greater length of the bearing, the flow of oil thru the bearing and availability of oil to form a wedge for the whole bearing is increased; second, the length of path for any dirt to be harmlessly washed out of the bearing before being dragged into the oil wedge is greatly shortened. It is assumed that by now most of the LST engines have had this work done. No report of a failure on reworked shafts has been received. *

When MLW received an order for some 50 locomotives with 8-cylinder engines, the whole subject was reviewed in the light of accumulated service experience. It was decided to abandon the principle of a drilled shaft and to provide instead a stationary oil feed in the bearing block, figure 6, into circumferential central grooves, both, in the bushing and the shaft. To compensate for the reduction of film pressure by the central groove the length of the bearing was increased from 2-1/4" to 4". The increase in bearing length has one adverse effect in that original hardening specifications, probably too severe because originally developed for cam surfaces, had to be relaxed to permit induction hardening within the capability of Auburn's equipment. No service experience on this version exists to date.

* The basic reason for large oil spreading depressions is to provide copious oil for quenching hot spots created by introduction of foreign material. The material has a chance to imbed or work out before overheating with resulting seizure occurs. The small holes thru the shaft used previously seriously limited flow only thru the opening formed by the bearing clearance and the hole circumference.

  This same reasoning brought about the full annular groove with external oil supply described in paragraph 2. The field fix for the LST had not yet been conceived. DE-35264 was recently released with the modified LST version rather than expending the money to provide a new deep sump design with annular grooved bearings.

R.D.S. 8/25/72

References To Group 19, Crankshaft And Related Part

19-1 Crankshaft Stress Analysis and Bearing Load Carrying Capacity, F.P. Porter, ASME 68-DGP-8 presented May '68, Houston, Diesel and Gas Engine Power Conference.

19-2 Alco Internal Report by F.P. Porter of August 1965, Summary of Crankshaft Stress Analysis.

19-3 Precision Forming of Diesel Engine Crankshafts by R.M. Spencer 1962, ASME 62-OGP-3.

19-4 Alco Internal Report by F.P. Porter, Axial Vibration, November, 1957.

Correspondence file folders "Avondale and Bethlehem" also "Thrust Bearing Relocation" about 1960/62. Screening item 1944.

19-5 Alco Internal Report F.P. Porter 1962, Crankshaft Stress Analysis and Bearing Load Carrying Capacity. Also, Torsional Stress Analysis of Change in Firing Order Mech. Lab Report 3378.

19-6 Alco Internal Report F.P. Porter Jan. 25, 1957, Crankshaft Deflections, also Mech. Lab Reports 2974, 2974A.

19-7 Generator Connection Calculations, Calculation Books Group 19, Review GMV 5-2-63, Also Correspondence File Generator 1966 to 1969 Ruiter to McKean 2-13-69.

19-8 MEL 3102, 3113, 3124, 3144B. Damper Performance.

19-9 Calculations of Balancing Arrangement 8-Cylinder Engine. Design Report 8-251 by G. VanDeMark, also Calculation binder 8-251.

19-10 8-Cylinder Vibrations - MEL-3531 - Also letter to MLW of early 1972 listing test data of engine on Korfund dampers, see Correspondence folder.

CYLINDER LINERS

The liner design was established very early and has experienced only minute changes. In the very first execution the liner did not have the baffle ring which forms the distinct annulus in the water space just below the flange. It was added together with the cut-out to increase turbulence and thereby improve liner cooling near the top of the ring travel. It has the one undesirable feature of being dependent on correct installation by orienting the hardly readable stamp marking "Fuel pump side" towards the outside of the engine. No failures due to wrong installation have been reported, but the lack of fool proving the assembly is deplorable; no reasonable solution was ever found. The four lugs on the liner flange serve as drivers during machining.

The thickness under the baffle ring was increased to compensate for anticipated increase of firing pressures when uprating from the 190 MEP level. The seal between liner and head has always been a lapped fit; between liner and block the seal is conceived as a lapped metal to metal fit but for insurance the small Silicone "O" ring was added in the chamfer. The "O" ring deforms permanently on short time, but leakage at this point is virtually unheard of.

Leakage at the lower seal prior to using the insert was a serious complaint for many years. It was closely allied to fretting between O.D. at the bore in the middle deck. The fit at this point was gradually increased from .002 tight .001 clear to .005 to .002 tight which helped but did not solve the problem. The "O" rings were tried in Silicone which became badly erroded by water, various Bunas were tried with indifferent success. The surfaces in the middle deck opposite of the gaskets remained reasonably smooth, but opposite the contact with the liner they became so badly fretted that many gaskets were inadvertently ruined at installation by cutting.

These deficiencies disappeared with a simultaneous change (about 1961) of adopting Viton "O" rings and fitting the middle deck of the blocks with cast iron inserts. Note that only the "O" ring next to the water is of Viton, the one next to the oil has remained Buna (Viton rings cost about 6 times as much as Buna).

The inserts were originally made of piston ring quality iron with very heavy interference, subsequently ductile iron rings were installed with less interference, observing the ratio of moduli of elasticity. The high interferences cause the liner bore to go out of round opposite the insert. This is due to the middle deck having very different cross section and therefore resistance to deformation longitudinally and across the engine. Interference was therefore reduced somewhat with introduction of Locktite in the joint.

The dimensions of the insert casting seem absurd. It is apparently very difficult to make a ring casting with a sound core. Core soundness is necessary, early inserts leaked water. By giving the casting a very high machining allowance on the I.D. only the outer dense layer of the casting appears in the final insert, porosity having been bored out.

While the insert in the middle deck of the cylinder block was an unqualified success, inserting the upper seat has been unsatisfactory as discussed in section cylinder block salvage.

In shipping and storing, the lower end of the liner may go out of round, it can be made round by tapping it with a wooden mallet, This is the significance of the straightening note on the print.

Liner Finishes

The majority of 251 liners are chrome plated. The plating increases the liner cost by about $35.00. Alco specifies unplated liners only in the standby units where quick ring seating is significant but not liner wear, and on drill rig engines. The original rationalization was that drill rig engines have much light load operation and will be completely rebuilt before the liners are worn out. The practice has not resulted in complaints and is still being followed. Some railroads, e.g. N.Y. Central, did not use water treatment and claimed that cavitation damage to the outside of the liner established life rather than bore wear and therefore specified unplated liners. Unplated liners are honed to 40 to 80 RMS, 8 to 12" crisscross. While early 251 liners were plated by VanDerHorst, liner scoring on production test was intolerable, it was claimed due to inability to clean the depressions in the plating from honing debris. It was followed by what was then called open chrome plating. The honed liner was grit blasted before plating so deeply that the pits were reproduced in the plate. This type of plating was often very satisfactory in operation, but control of the amount of porosity in a finished liner varied from 20 to 50%.

When the idea of strictly controlled pit marks of Honey chroming appeared around 1959, it sounded very desirable. It was tried in limited production for a long time and use was gradually increased to full production. In more recent years the performance of Honey chrome has been seriously questioned in respect to oil control and ring wear. At this time, 1972, it is still Alco standard.

Exfoliation of the Chrome plating is not a frequent occurrence, but a rash of cases did come up at one time. It was caused by cost reduction in honing at Alco, a double honing was reduced to a single honing, which created, by heavier pressure, a smeared overlay of metal to which the chrome did not bond properly. It was corrected by more intense blasting of the bore surface prior to plating. See Metallurgical Report #3098.

CYLINDER HEAD CASTING

Work on the head for the 251 started in 1950 and a head very similar to the present one except without any provision to create swirl was designed and made. The preceeding 244 head had just "grown" with adding a piece to the pattern here or lining a corebox there so that the head itself was so different from the drawing that the only way a pattern could be made in a new foundry was to cut a head and transfer, whatever was found on the casting, to paper; the still current 244 head drawing 22-EE-71092 is the result. When the 251 was designed it was postulated that every wall or duct should be clearly geometrically defined. A further demand was the absolute separation of fuel ducting from the oil space.

The test engine began to run in spring, 1951, with completely unsatisfactory combustion. For over a year frantic development of different head and piston shapes was undertaken. These included a set of heads with an attempt at duplication of the 244 air duct. Performance did not come near duplicating 244 performance, probably because the air swirl which does exist on the 244 head is greatly dependent on the 90° air elbow which on this head is an integral part of air duct which could not be duplicated on the 251 head.

Two abortive trials were made with two valve heads, one of them a semispherical combustion chamber.

The different heads were matched with a variety of piston crown shapes, "Mexican Hat" dished top, flat top. See reference 22-1.

Finally efforts on the original head were renewed and swirl was introduced by partially blanking off the outboard air valve port which more or less contributed a breakthrough; it gave for the first time a "commercially acceptable" combustion; while full load combustion in time became really good, part load combustion, up to the present date, never got beyond "commercially acceptable."

Various ways were tried to convert the makeshift valve port insert to a production design. Early in 1952 the final shape of the sloped air valve recess was developed. The efficacy of this valve recess has been questioned many times. At some later date heads without the recess were again tried on a lab engine (this by recollection) and this trial also confirmed the absolute need for swirl producing means. The recessed valve port forms part of a patent, which proved not strong enough to deter Hunt Spiller from selling Japanese made heads in the U.S. From about 1952 on to this date, 1972, no changes to the head proper have been made which could be shown to have affected combustion. A substantial number of changes for various other reasons were made. The most important one was the early addition of stock at the valve port, to permit use of valve seat inserts. About 1956 the original wood pattern was replaced by metal core boxes; the horizontal sections of air and exhaust ducts were raised about 1/4" to allow sufficient sand in the water passages under the ducts even with the added metal to back up the seat inserts. Core holes were added and some existing ones

increased so that every part of the jacket can be seen and examined for remnants of core sand.

In 1963 the exhaust duct was again changed to bring cooling water closer to the valve guide boss. Short bosses for the exhaust valve guides were added to prevent the valve guide bores from being distorted by the original angular exit of the guide bore through the upper duct surface. Ref. 22-2. This also was done in preparation for the short valve guides.

The last pattern change was the addition of three grooves next to the gasket surface for the valve lever cover to permit this cover to have an oil drip edge to improve sealing. (It helped a great deal.)

Virtually all changes to the pattern were documented by changes in pattern numbers. The question is frequently asked whether a head with a certain pattern number should be reused in rebuilding or scrapped. Heads with pattern number 822-5738 or higher include the last significant change in the exhaust duct and can be reused on any engine except for considering possible deterioration due to age. Heads with pattern number 822-5735 or 822-5737 are from the metal pattern and have adequate backing for the inserts. It is recommended that their use be restricted to C ratings; consideration of deterioration due to age is advisable.

Another frequently asked question concerns a thinning of the horizontal wall, refer to 22-EE-71112, section F-F, where the print actually calls for a local 5/32" wall thickness. On print 22-EE-71111, in "section above air and exhaust passages," is shown in construction lines the extent and location of this thinning of the baffle (on the print it is labeled loci of 1/8" radius...). Note that the thinning ends in holes in the deck; compare this to drilling a hole to stop a crack from propagating!

This horizontal wall is intended as a baffle to the flow of cooling water; it forces the water which enters at the two bolt flange pad to flow vigorously around the junctions of the walls of the air and exhaust ducts with the bottom deck. This baffling is the only function of the wall, which should be prevented from taking part in resisting the various forces from bolting or firing pressures. To make it impossible for this wall to participate in carrying structural stresses, weak areas have been intentionally introduced. If cracks occur in this area either in the foundry or in operation, they do not propagate into the structural members of the casting.

Cracking during foundering has actually not been experienced but openings due to not filling have occurred rarely in some periods of time, frequently in others. Print 50-B-72232 shows the extent of permissible, if unspecified, openings in the water baffle. If recent investigations have shown that the water does not flow around the ports as intended, this does not affect the water baffle in principle but may alter the judgement on which print 50-B-72232 is based.

CYLINDER HEAD, FOUNDRIES, MATERIAL, QUALITY

All original development was made with CWC foundry, they first made a wood pattern then a metal pattern toward the later fifties. (The wood pattern was scrapped.) Their method was the conventional wire reinforced sand cores which needed a great battery of core dryers. They discontinued making heads for Alco in 1966, restarted about 1970 for a year but have since broken off making any head castings for Alco. The pattern equipment is being considered for use by other foundries. In the early or mid-sixties, Motor Casting was developed as a second source; they have since become the sole source. They started with a completely new pattern, utilizing CO_2 cores. By recollection the CWC pattern with all the driers cost about $42,000 before 1960, Motor Castings patterns were around $22,000. Every time a new foundry was brought into the picture (including 244 heads), it took at least one year from placing the order to the first trickle of production castings.

One other pattern for the 251 head exists, it is controlled by Hunt-Spiller and is now in Japan; they offered to Alco heads also machined in Japan but not unmachined castings. Alco examined such a head in detail in an abortive attempt to establish patent infringement. The head met Alco requirements in all respects except that lock spring grooves for the inserts were omitted.

The material for the 251 head is flake iron per specification 31P5400. Test pieces cut from cylinder heads give approximately 40,000 psi tensile strength. The Alco spec does call for 1% minimum nickel. Both during the Korean war and during the long strike a few years ago copper was substituted for nickel. It has never been established if this substitution affected head performance in any way. For safety's sake nickel was always reintroduced as soon as it again became available.

All heads are water tested after machining. Water glassing per applicable Alco specification is permissible but actually rarely required. The only area prone to porosity is in the valve guide bores. Permission to sleeve such porous bores was frequently requested, but never permitted by engineering.

Quite frequently the shop complains of difficulty in machining heads, soft skins, too many carbides, etc.; metallurgical report Met 3400 is typical. There are many in the files. They do give the impression that machining procedures, quite intensively automated, are overly sensitive to minor casting variations. Frequent controversies and rejections on this score are believed to have contributed noticeably to the loss of CWC as a supplier.

Some experimental 251 heads were cast in ductile iron. Original straingage investigations by Lessels Associates indicated that the 40,000 tensile flake iron would be adequate. Concurrent experience

with ductile iron 244 heads, thermal cracks due to poor conductivity, recurrent foundry difficulties did not encourage use on the 251 head.

The failures, which do occur on the 251 head, would not necessarily be prevented by use of ductile heads, at least not without substantial design changes.

Various customers have tried salvage welding heads, including building up of valve seats. To Alco's recorded knowledge no such attempt ever produced a durable head.

CYLINDER HEAD CRACKING

Occasional reports of cracked cylinder heads have always existed. At one time heads were cracked in operation due to overtightened injection nozzle studs. Instruction to finger tighten both nuts then tighten only the one next to the exhaust manifold to 50 ft. lbs. overcame this type of failure.

In 1964 when the 16 cylinder engines were first uprated to 3050 MEP at 1050 RPM reports of cracked heads became frequent and typical enough to start a strain gage investigation. It had to be done statically so that influence of stud tightening torque and simulated firing pressures up to 2000 psi on the stresses in the combustion deck and in the cooling water compartment could be determined. Ref. 22-3.

Most of the cracks seem to start on the water side in the top deck near the valve ducts or in the ducts themselves. For a description of a typical failure, see Ref. 22-4. Strain gage results showed that the sequence of stud tightening, much suspected at that time was irrelevant, but that tightening torque and subsequent static loading caused stresses of dangerous magnitudes. Reducing the then specified torque from 700 to 500 ft. lbs. reduced the mean stress at 2000 in cylinder pressure near outboard air valve port from -6000 to -3000 and alternating stress from 11000 to 6000. The torque reduction could be made without incurring any complaints of gas leakage between liner and head. In 1968 there came reports of numerous (85) head failures in India, all cracked in the outer wall of the one sleeved stud loss, all after four to five years' service. A few similar cases had become known domestically including one on a head of the 6 cylinder lab engine which had seen much service. It was investigated, ref. 22-5, without coming to an explanation. Recently suggested explanation of damage initiated at rolling in the tube and very slow propagation in service seems to fit facts, especially since rolling in procedure was very questionable up to 1966.

The strain gage test of 1964 was followed up by determining metal temperatures ref. 22-6 on heads operating both on the 16 and 6 cylinder engines at 1100 RPM at up to 245 MEP. The temperatures measured reach 647° at full load with water flow reduced 33% under normal; with normal flow temperature is 633. The data are quite extensive (loading unfortunately is in locomotive notches), no concerted effort had been made to determine probable stresses from the temperatures measured. While this test series included different water temperatures and reduced water flow, simulation of steam or gas bubbles in the cooling water was not attempted. Neither at this nor at any other time was direction of water flow inside the 251 head examined by itself nor its influence on metal temperatures.

One change made in 1966 may have affected water flow more seriously than anticipated. Looking at the underside of the head there is a 3/4" radius boss right next to the stud hole between air and exhaust flanges. Up to then a 3/8" diameter hole was drilled into the water jacket. The secondary purpose of the hole was the opportunity it

offered to probe that the water baffle above this point was open as intended. The primary purpose was to provide an opening for a vent connection from the cylinder block. In the Vee engine block the highest point in the annulus between liner and block was vented through an angularily drilled passage, which in turn was connected to the head by a special connector. This connector, virtually inaccessible under the head and below the exhaust manifold was a continuing cause of leakage complaints. Since all views of this connection have been deleted from pertinent drawings a cut out from an old print showing the details is here included as appendix 22-1. When one of the railroads advised that they had plugged this connection without apparent ill effect, a trial was made in the lab by leading the vent from the block per transparent tube to the expansion tank, also by letting it discharge into a cup, no steam bubbles were found and the vent connection was plugged in all production engines and probably on most engines in the field. In retrospect it is suspected that removing the jet-like stream from the vent in the particular point of the water jacket may have affected the water circulation adversely. In reviewing background for this section, it was noted unexpectedly that the railroad which had reported the lack of adverse effect of plugging the vent was the N.Y. Central which had the highest rate of head replacement as noted below. Early pertinent experience with 244 locomotives, unfortunately forgotten or disregarded by locomotive cooling system designers in the sixties, clearly showed that if one head would crack and let combustion gases into the water system, all heads would crack within a few hours of operation unless provisions were made to collect and expel such gases from the cooling system. At that time a vertical oil cooler was used which had a large domed head. By using this head as a steam dome, vented by fairly large piping to the expansion tank, the head cracking epidemic disappeared over night. This vertical oil cooler was cost reduced into a horizontal box which not only lost the steam feature but in addition reduces the need for cooler cleaning by raising the oil velocity so high that it overloads the engine lube oil pumps by having about 40 psi pressure drop. These remarks have been added at this point to emphasize the need to look also at the various installation features before blaming detail design features of the head for all complaints. Head cracking reports on non-locomotive engine seem rather rare.

Another influence on head performance appears to lie in maintenance and/or service conditions. Ref. 22-7 shows that yearly head replacement on various railroads ranges from a horrible 23% on 12 B and C engines on the New York Central to the excellent .44% on the N & W which had a total of 3308 heads in service at the time a great many on 16 cylinders up to the E rating. The next worst is the Santa Fe with 16.4% on seven year old 16 cylinder at 2400 HP, the next best the Pennsylvania Railraod with 1.04% on a variety of engines many over ten years old.

ALTERNATE DESIGN CYLINDER HEAD

In 1970 a feasibility study was made on the drawing board of a head made of steel rather than cast iron to get away from cracking. It was conceived as a combination of simple steel castings, pressed steel parts, sheet metal welded or brazed together. The result is shown on 19 LYA 1728, which shows this attempt completely unusable and does not even suggest any other more hopeful approach.

REFERENCES CYLINDER HEAD GROUP #22

22-1 Early combustion chamber shapes 98-A-22070, 98-A-22105, 98-B-22111.

22-2 Layout of exhaust duct change 1962. Up to change 19LYA1263. After change 19LYA1256, 1257. 22-EE-7111-X of 3/28/63.

22-3 Cylinder head stresses MEL 3464, 3464B.

22-4 Head failure from water side MET 6810.

22-5 Stud boss cracking strain gage check see project folder 863892-19.

22-6 Cylinder head deck temperatures MEL 3464D.

22-7 Head replacement by different railroads 1967 cylinder head file, E. C. Roberts to F. X. Gaenger of November 7, 1967.

SECTION R-R

SCALE - FULL SIZE

CHANGE 71087 REV H-H

CHANGE NO.

VALVES, GUIDES, AIR AND EXHAUST

So much work on valves, guides and seats, much of it abortive, has been done over the years that, to keep it manageable, this discussion will be confined to parts used in production engines. Ref. 23-2 gives leads to look for other information.

The 251 head started out with 45° valve seats directly machined into the head. In 1955 the need for valve seat inserts became obvious. Even then it was recognized that air valves and their seats were wearing while exhaust valves and seats would pit a little but hardly wear at all. It would, of course, have been flying in the face of all preconceptions and usual practice to insert air valves and not exhaust valves; for this reason all valve seats in the head were equipped with inserts made of Stellite 3. Inserting heads in production was a new departure and a locking device was felt to be a must. The wavy ring of stainless steel, located half in a groove in the head and half in a groove in the insert, was then adopted. It is in no way proven that this ring is required, but proof that it is not needed does not exist either.

The Stellite 3 insert was superseded in about 1964 as a cost reduction by a Welltite insert; the Stellite insert was readopted in 1969. Lab data showed the Welltite insert at least equal to Stellite but all field data seemed to show Welltite to be inferior. Ref. 23-1. There was one pecularity, Welltite definitely needed a wider seat contact. The O.D. of the seat on Stellite inserts is 2-15/16" while on the Welltite it is 3-1/16" the same as the valve O.D. Height of the inserts also differs by 1/16. This should be kept in mind when looking at the condemning instructions in various manuals.

All Alco valves were procured from Eaton or TRW. Eaton has never entered the after market directly while TRW has always actively solicited direct business from Alco customers. This obviously made Eaton the preferred OEM source.

All valves are made of two pieces welded together. The valve head is of heat resistant material, the stem of hardenable and hardened steel, the tip of the stem has higher hardness than the rest. Up to recently Eaton used flashbutt welding, they are now in the process of switching to friction welding. Valve quality is defined and inspected 100% per Alco Specification 31-P-5651. After a few defects were found some time ago, Eaton currently inspects all valves additionally by supersonic means.

Up to 1969 it was an unbroken rule that air and exhaust valves should be identical. At that time the rule had to be broken to provide valves which would live at the increased engine ratings.

The first 251 valves were made of 2112 heads (carried over from the 244 engine) and stems of 3140 nominally, for this 8645 was substituted, later it became 4140 or its equivalent which is still used in 1972. At one time only, Silchrome Xbe was tried on some 2000 stems to permit exposure of the stem material to exhaust gases, but frequent breaks near the weld forced withdrawal of these valves.

The original 2112 valves wore badly and were replaced in late 1954 by valve 23-C-71045 on which the seat was hard faced with Stellite #6 and the stems were chrome plated to .0005" thickness on the diameter.

This plating initiated a serious number of failures. The end of the plating coincided with the end of the ground surface, beyond which was a slightly reduced diameter, the plating apparently spilled over the small step. In any case the failures started at this point, the valves were used up by buffing the transition area; but this incident has discouraged chrome plating of valve stems since that time.

The Stellite 6 hard facing did not wear particularly well. Head material 214N then came on the market, it not only resisted wear better than Stellite 6 but, of course, was also much less expensive. Stellite faced valves remained available for engines operating on special fuel, particularly those containing Vanadium. The plating of the stem was dropped.

For all other applications 21-4N became standard head material. It remained so to this date 1972, except that in engines operating at very high temperatures or sustained high load 21-4N has been replaced by Inconel X-750 but only for exhaust valves. On air valves 21-4N is way superior to Inconel in respect to wear.

In 1957 the long time standard valve 23-C-71050 was adopted. It has 21-4N head, straight stem and weld line 3-3/4" from the head, As long as engine rating did not exceed 200 MEP these valves gave completely acceptable wear on locomotive engines. Wear complaints began to be serious when enginesat this rating were operated at sustained loads, say in some marine applications.

Persistent complaints during this period concerned valve sticking which led to many changes on valve guides, some on valve stems, abortive trials with stem seals, stem umbrellas, lub oil supply to stems.

The production valve guide 23-C-71031, cast iron with type A graphite, has a bored finish intentionally not too smooth. It is lubrited to permit pressing it into the head without galling. Since lubriting is difficult to mask the bore also gets lubrited, which is not really desired, it dulls any reamers but tolerated. This guide has a shoulder which, however, must not be pressed against the head counterbore but must be kept away about 1/32 to prevent deformation of the bore. The print reflects the uncertainty about the influence of shielding the stem on valve sticking. The shielding started out by a counterbore 1-5/8 deep was changed in 1954 to 1/2" deep and changed back again to 1-5/8 in 1962. The lower part of the guide protruded into the valve duct; it broke off rather frequently leading to turbo failures due to "foreign material". (30% to 50% of turbo failures were for long time ascribed to foreign material and the number of broken off valve guides was significant). The counterbore did shield the lower part of the valve stem material from direct impact of exhaust gases. The intended stem-guide clearance was specified as .004 to .007 which allowed for .001 closing in of the bore due to press fit and lubriting. While reaming of the guide after pressing in was then considered desirable, EMD did not call for it so Alco could not either, but both resorted to the weasel wording "ream if necessary".

In the early sixties the 6-D engine was developed and with it a new approach to guide and valve stem was tried. The valve stem became tapered to keep clearance small at the top, where oil would tend to run in and manifold air would tend to leak out, but to provide larger clearance at the bottom where thermal expansion of the stem of the exhaust valve would reduce clearances. In addition the guide bores were made undersize and were tapped with slightly oversize 5/8-18 tap. This tap created a shallow spiral groove, intended for lubrication, to improve on the coarsely bored finish. After pressing the guides in they had to be reamed and reaming debris wire brushed out of the oil groove. This awkward procedure was specified on the first few 6-D production engines.

This development did lead to the adoption of the tapered stem concept, retained to date (1972) on exhaust valve stems.

In 1965 an interrelated group of changes was made, the exhaust duct in the head (see head section) received bosses for valve guides and cooling of the inboard guide was improved. At the same time the guide with all counterboring was cut off so it protrudes just slightly beyond the bosses in the exhaust duct. The guides have become straight tubes without shoulder and depend on being pressed into the head to a certain height. The guides at this time are still of cast iron unhardened. The shorter guides were, of course, checked out in the laboratory, but shouldered, cut off guides of hardened cast iron with honed bores had been furnished by TRW directly to the Santa Fe RR and were reported to have cured valve sticking. Note that all short guides have the edge at the lower end of the bore left sharp to scrape off any deposits.

The short guides at first included the oil distributing thread. In service it was found that the thread transferred itself by some fretting mechanism to the stems of the air valves, exhaust valve stems were not affected. The threads were therefore abandoned in 1966 in favor of a coarse phongraph finish in the bore. The various guide valve head combinations are shown on Ref. 22-2. The valves were made with tapered stem and the weld location was raised to bring the joint within the shortened guide. Drawings 23-C-71071 and 23-C-71072 show the production guide and valve respectively.

The shortened guides certainly eliminated turbo damage by broken off guides, valve sticking seemed to have disappeared, but changes in gum content of fuel oils took place at the same time which clouds the issue somewhat.

In order to get the shortened guides quickly into the field valves and guides were applied to the exhaust side as soon as they became available while stock of earlier valves and guides (both the ones from the 6D and from regular production) were used up on the air side. It took longer than one year to use them up. Attempts to develop hardened guides by Alco some years ago had then not led anywhere due to lack of cooperation by manufacturing. When, however, the combination of soft-short guides and tapered valve stems led to excessive stem wear, hardened, honed guides from TRW had to be specified in spite of substantial increase in cost over Alco made soft guides. They are now specified on higher rated engines.

In the meantime seat wear on air valves became a major deficiency on higher MEP engines. Extensive wear testing was carried thru on the 6 cylinder lab engine at 1100 RPM at 245 MEP. Much of this is reported in ref. 23-2.

The following conclusions from these tests and other reports are stated from memory.

1. No better air valve material than 21-4N was found.
2. Wear is much greater at sustained high loads than at varying loads as in locomotive operation.
3. 30° air valve seats reduce wear significantly (in the 30 to 50% range) but do affect combustion cleanliness adversely.
4. Geometric changes in transition from head to stem are not significant.
5. Oil mist introduced into the air manifold reduces wear very much.
6. Field reports (Jackson Marine per Boynton) indicate need for retention of oil mist device and 30° air valves for marine application.

7. Inconel exhaust valves are more resistent to guttering at high temperature operation, but don't use them for air!

8. Once it is accepted that air valves differ from exhaust valves, air valves should not retain tapered stems. Small clearance for the whole length will give better control over oil and scavenging air escaping past the valve stems to raise crankcase pressure.

9. If air valves must not get installed on the exhaust side, any distinguishing difference in appearance is desirable. Therefore the thickened stem of air valve 23-C-71076 should be retained even if it has no other merit.

These conclusions form the basis of current (1972) head assembly specifications. While the premium head normally specified for new production used 21-4N exhaust valves, Renewal Parts sell only premium heads containing Inconel valves. except MLW — soft guides, 1 1/2 valve, and same air as 4th valve 21-4N

REFERENCES - VALVES - 23 - GROUP

23 - 1 Valve seat insert review, see Valve file 1971
F.C.T. to Smalley 12-7-71.

23 - 2 Valve wear testing MEL-3525.

Project folders GO-863630, 631, & 634.

Valve temperatures, see Eaton Report of 5-23-67
in GO-863630 folder.

Valve guide temperatures early MEL - 3380
late MEL - 3524B

Valve Material Combinations MEL - 3375 N

Oil Mist Development MEL - 3494, 3494A

VALVE GEAR

In reviewing the design of the valve gear, one is surprised at the very few changes made from the first production engine around 1952 till 1972. The only parts changed are the valve lever shaft, the springs and the cover.

The shaft changes, numerous in recent years, are all related to excess bushing wear and the earlier cost reduction changes tended to increase or create wear complaints. The original shaft had a central long oil hole with cross drilled holes in the center of the bearings. It probably was ground on centers; bearing finish had to be improved to 20 RMS even during lab testing. After that, performance was satisfactory until about 1960. Cost reduction changes then eliminated the center hole, fed oil only to the top of the shaft, reduced finish allowance to the point that surface decarb prevented adequate hardening. Centerless grinding created lobing and waviness. It took a few years until the combination of these changes, including perhaps the spring changes mentioned below, created a serious wear problem on engines in service. To date it is hoped that change to 51200 steel will result in consistently adequate hardness and that improved grinding practices will avoid lobing and waviness. It is not certain that these changes alone will correct wear complaints.

The springs started out with loads 94 lbs. installed, 193 lbs. valves open; they are now 139 lbs. installed, 250 lbs. open. The increase was needed to compensate for increase in engine speed from 1000 to 1100 RPM. The increase of load in "installed condition" is undesirable because the increase of load on the yoke spring, which presses the linkage together even in "valve closed" position, makes it harder for oil to enter between the various contacting surfaces (bearing, pushrod, ball joints). In this respect the increase in spring load tends to aggravate bushing wear.

An experimental spring set with increased load on the valve spring and very low load on the yoke spring "installed" is in use in several locations, including Australia. It has been reported as correcting bushing wear but the danger of using wrong spring combinations mitigates against adoption in production.

A complaint first heard about three years ago is excessive wear of push-rod ends and the matching ball ends of adjusting screws. This wear appears to be caused by lub oil contamination related to the change from 12.5:1 compression ratio dished top pistons to 11.5:1 flat top pistons (Australia).

Reported for the first time only in 1971 and since then frequently repeated are breakage of the valve yoke; the tube breaks off tee bar. Related to it is excess wear on other parts of the valve gear including cams. Current explanation is excessive valve sticking due to use of fuel with excess gum content. This complaint is limited to the mining railroads in North Western Australia.

Leakage complaints between head and valve lever casing were a constant irritation. The adoption of a pressed steel cover with an oil drip edge protecting the gasket from the oil which creeps down the walls has corrected the complaints.

WATER PUMP

Water Pump General

The water pumps were originally designed and built by Morris Machine Works for the 244 Vee engines which established the principal design features. When the 251 engine came along two significant changes were made. On the 244 the drive gear was mounted in two ball bearings in a separate casting. The gear had a hub with an internal spline. The pump shaft ended in matching external spline. On the 251 the pump shaft was made with straight fit onto which the gear was pressed and secured by a nut. On the 244 the bronze impeller was pressed on a straight fit, also driven by a key. One frequent defect was loosening of impeller on the shaft, at first this was corrected with brazing and reboring. Later the impellers were bored out tapered to receive a thin slotted sleeve pushed in by the original nut with a thick Belleville washer to protect the 3/4 stud and from over-stressing. When the 251 pump was made this tapered sleeve was incorporated from an early date but the taper was put on the shaft and the impeller bore became straight. The stud end was increased and the Belleville washer replaced by a plain washer.

The pump for the 6 cylinder utilized most of the Vee engine parts, but volute and impeller are made narrower. All pumps were designed from the beginning to accept either a 9" diameter impeller or a 10". The impellers of the standard pumps have back swept curved vanes resulting in very flat delivery curves, which makes the flow very dependent on the resistance in the cooling system.

There are two pecularities incorporated in all the pumps, the outer bearing has a specially long inner race which serves to provide a hardened surface with excellent finish to operate the oil lip seal on. At present this bearing is difficult to procure; also it has to be replaced many times not because the bearing had become defective, but rather because the seal had worn a ridge in the inner race. This bearing should be considered obsolete. (The newly drawn up radial vane pumps have separated the function of bearing and companion surface for the lip seal.)

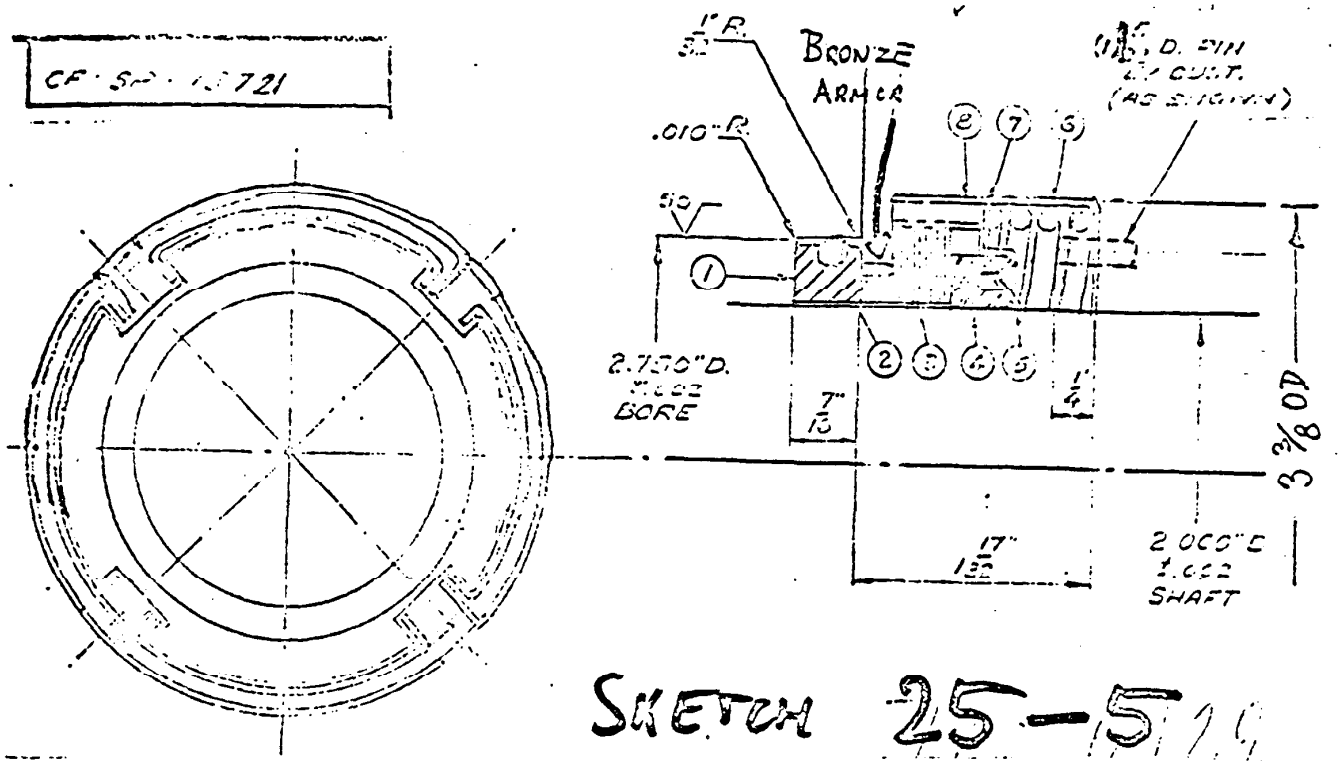
The second peculiarity is the strange cross drilled hole thru the drain hole from the seal compartment, which is also blocked off by a rib. This was invented to prevent repetition of loosing a complete engine because somebody tried to "fix" a leaking water seal by plugging the drain hole and thereby forcing the leaking water thru the oil lip seal into the crankcase. Note also the slinger which will prevent a stream of water from a leaking seal to hit the oil seal.

About 1958 Alco copied the pump design and started manufacturing in house. Sometime later the bronze impellers were changed to cast iron for cost reduction.

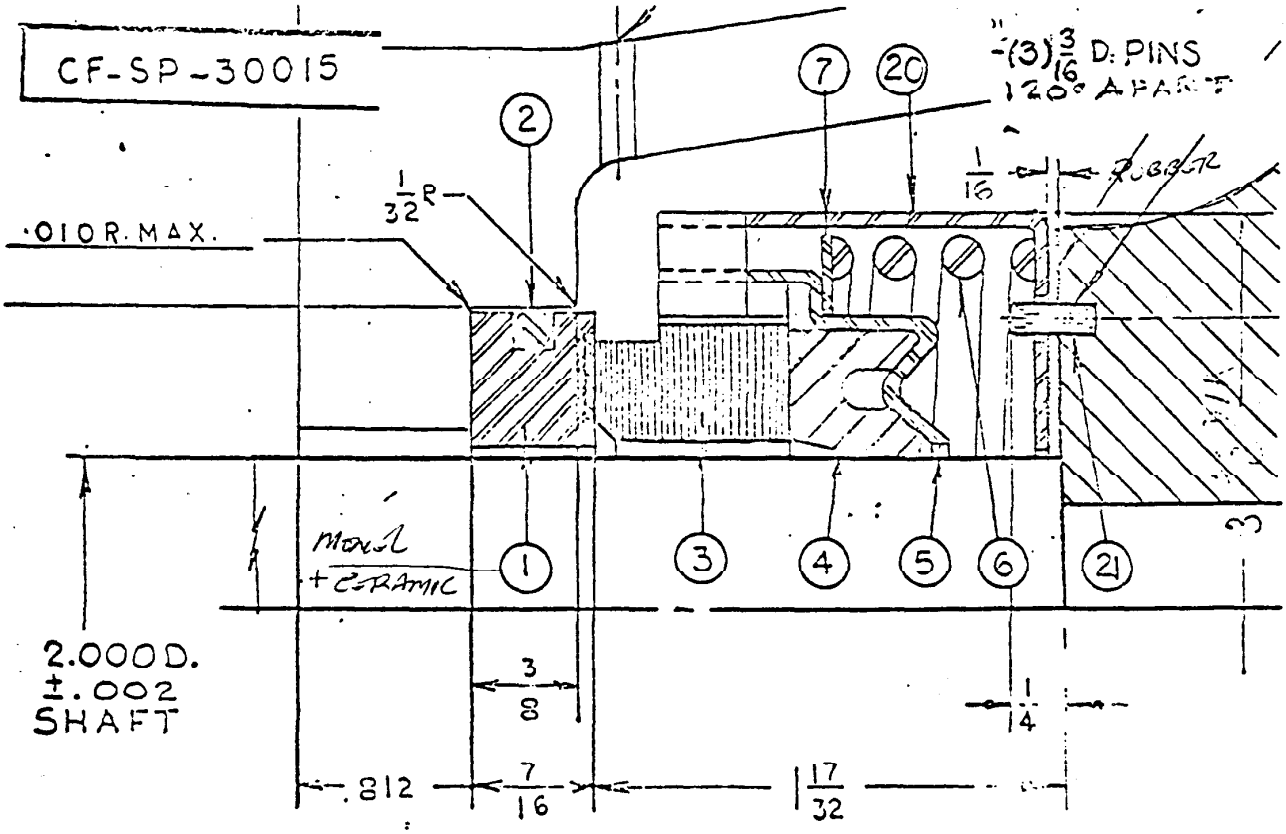
For performance data see ref. 25-1 for a recent full scale section of the current pump see 19LYA1799.

NOMINAL WATER FLOWS IN GPM

Eng. Speed	Cylinder # 6	8	12	16	18
720	210	360	430	430	500
900	260	450	540	540	630
1000	290	500	600	600	700
1050	300		630	630	
1100	330		660	660	770
Gear Ratio	1.72:1	1.61:1	1.72:1	1.72:1	1.975:1



SKETCH 25-5



SKETCH 25-6

The LST seal sketch 25-6 began as a duplicate of seal 25-5 except that it was to be executed in sea water resistant materials, that is Monel, and the seat is ceramic coated Monel. After the 1000 hr. acceptance run during which no leak was discovered, the drive hole in the seal casing of the raw water pump was found to be worn elongated and on the jacket water pump the pin had disappeared (by memory). No elongation of these holes seems to occur on locomotive pumps. The radial location of the pin hole was found to force the spring into a slightly excentric position. It was even possible to assemble a seal with the spring riding on the end of the pin. There was some difference in dimensions for the Monel and the stainless steel springs which caused the difference.

To correct the deficiency found during the acceptance test, the one pin was changed to 3 somewhat larger pins at a reduced radial distance. To counteract the tendency to vibratory wear a 1/16 thick rubber sheet was placed as a damper between seal and impeller. Not many complaints have been received from the LST's. It and its locomotive execution is probably the best seal Alco can presently furnish.

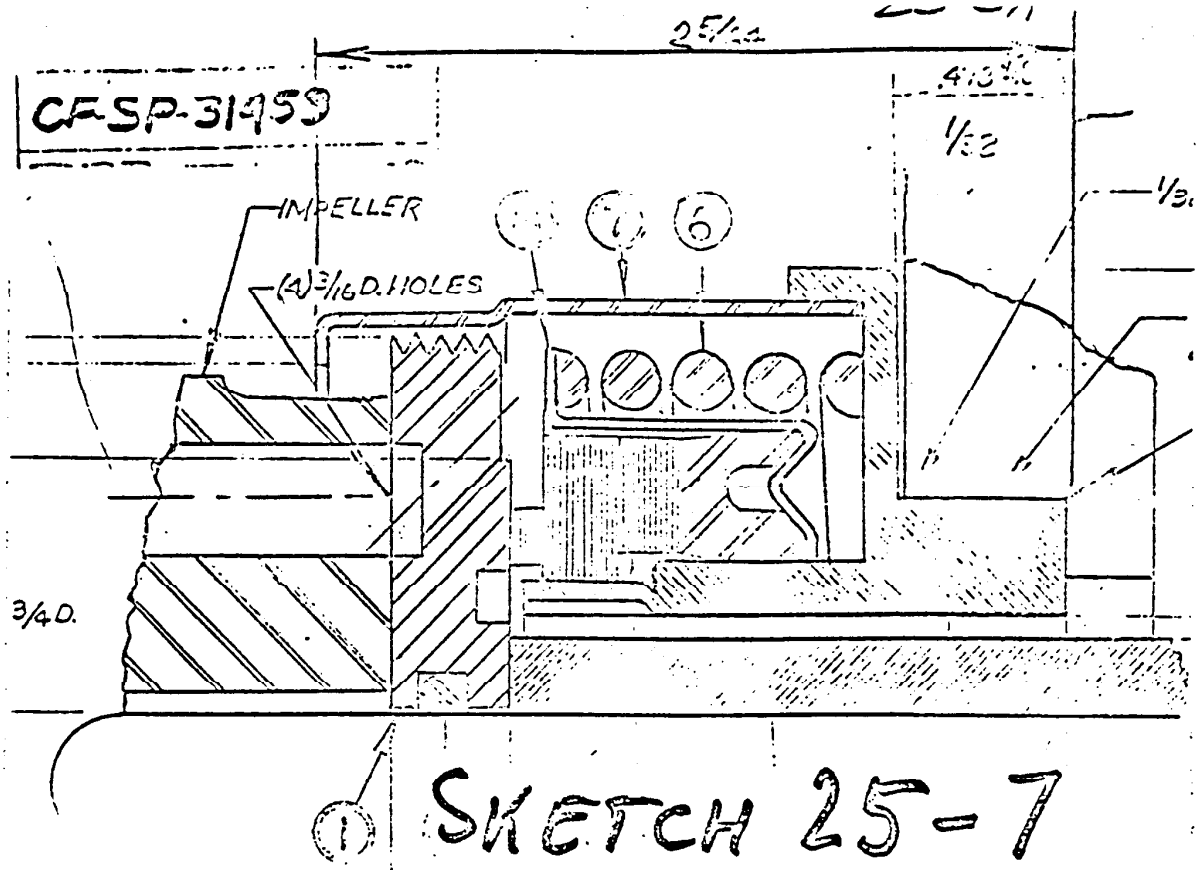
In the meantime seals with still greater sophistication were under giving lab. and field tests and as usual reports from the test seals (Canada) were outstandingly good. The seals actually tested in the field and adopted as production seal is shown on 25-8, 25-7 shows virtually the same seal except that a chevron is used instead of a bellows. By recollection the bellows was adopted because they came in first and had accumulated substantial mileage before the chevron seals became available.

These are the first seals imposing requirements for significant re-machining of pump parts. The shaft sleeve has to be cut off and three holes for water circulation must be drilled in the impeller hub.

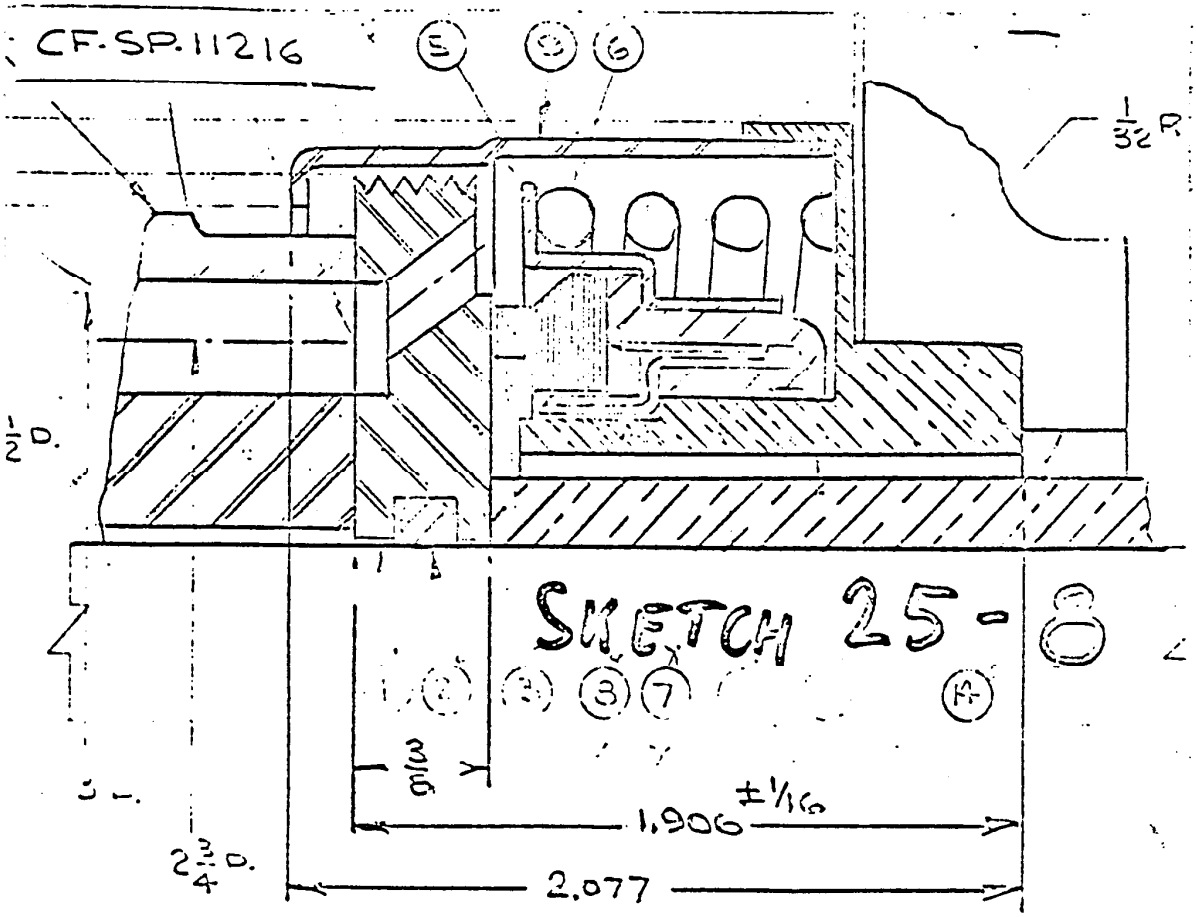
The design intent of the seal is to flush away from the face seal area wear producing contaminants. Actually it would appear that the contaminants are invited to enter thru the large holes in the impeller but prevented from getting out of the seal case by the small clearance of the labyrinth. Difficulties with the first production seals included warping of the exceptionally large seat plates, slipping of these heavy seat plates relative to the shaft. The shaft nut works thru the expanding sleeve and only a fraction of the end load serves to clamp the seat plate. The casing for the prototypes was machined accurately on a lathe, later a stamping did not provide an equally accurate bore to match the labyrinth on the seal plate.

A large number of these seals is in Alco stock, it has been recommended to scrap them they are being installed in production engines pending a clear decison concerning their disposition. They are made to fit the Alco pump and have no scrap value whatever.

Review of files shows a large number of other seals which were tried and discarded, amongest them, a magnetic seal (MEL 3177) a seal in which bellows or chevron was replaced by "O" rings; a fairly recent test with a lip seal furnished by Johns Manville which stood up surprisingly well, it started to leak because the mating sleeve surface deteriorated, contrary to normal expectations this seal merits further consideration.



SKETCH 25-7



SKETCH 25-8

WATER PUMP REFERENCES

25-1 Pump performance curves made on Morris test stands are in correspondence file, folders--water pumps prior 1960.

Pertinent lab reports are:

9" versus 10" impeller on 12 cylinder engine MEL 3052 comparison of Alco versus Morris built pumps for 6 cylinder MEL 3153-A, for 12 cylinder MEL 3153. Performance 9" versus 10" impellers on 16 cylinder engine 3433; also 9" impeller on MLW built pumps 3433 B.

25-2 Early chevron and 12A field experience Field Test Report F-13

25-3 12T seal failure description MEL 3042 C
Spring cocking MEL 3333
MEL 3042, 3042, A, B, C.

25-4 Alco designed seal 98-C-25095
Lab test MEL 3197, 1397 A, B, C.

TURBOCHARGERS GENERAL

The turbocharger is so essential a part of the 251 engine, as it also was on the 244 engine, that none of these engines was ever run by Alco without a turbocharger or in the one single case of the Snorkel engine without a gear driven charging air compressor.

Not only is the turbocharger a component essential to deliver the power output of these engines, it is also close to being the most failure prone item on the engine. Turbocharger design and technology is quite specialized and differs markedly from that applicable to the rest of the engine. The writer of this review, however, has been involved only off and on in turbocharger work, mostly as a pinch hitter, and is therefore not competent to deal with Alco's turbocharger history in anything but a superficial way. All statements in this section should be considered subject to correction.

Alco claims to be the first U.S. manufacturer to have equipped production engines with turbochargers. The early ones were built to Buechi patents by Brown-Boveri but Alco soon started to design and build turbochargers under license and guidance of Dr. Buechi. These were fairly large slow running machines with all housings split in two halves along the shaft center line. These turbos were applied to the 12½ x 13 model 539 and 540 engines, in rather large numbers.

In the early stages of the 244 engine G.E. claimed to have amassed knowledge and experience on turbochargers for aircraft engines, then believed immediately applicable to turbos for railroad diesel engines. The direct adaptation of aircraft turbos to the experimental 9 x 10½ engine proved hopeless and a special turbocharger called RD1 for the 12 cylinder, and RD2 for the 16 cylinder engine was designed. It was air cooled, the turbo shaft axis was placed athwart of the engine. Exhaust manifold entered the inlet casing tangentially. The shaft ran on antifriction bearings. For several years the 244 engines were equipped with these turbos. They had started out with single duct exhaust manifold which proved hopeless as far as acceleration smoke was concerned, and a clumsy first version of a jet assist, a two inch pipe blowing brake air tangentially on the turbine wheel was useless. These experiences led to the early development of a multiduct exhaust manifolds and suitable turbo inlet casings. These turbochargers had unsatisfactory reliability, G.E. came out with a redesigned water cooled turbo; but not until Alco had done considerable testing with Elliot built turbos, which also were substantially less expensive than the new G.E. types RD3 & 4 of which a few went to the field.

500/700 TURBOCHARGER

For production the Elliot or Elliot type turbo was adopted. By some arrangement between Alco and Elliot, probably influenced by the fact that Alco was Buechi's first licensee, reproducibles for all details of the 500/700 series turbochargers were obtained from Elliot. These prints were given Alco numbers without checking for conformance with Alco practice. Since Elliot's prints left much to the discretion of pattern makers and shop personnell, many misunderstandings did arise. By now the drawings have under gone so many changes and redrawings that this source of friction with the shop has been largely eliminated.

Alco started with buying complete turbos from Elliot, then followed with making parts from Elliot furnished castings and forgings and ended up with making the complete turbos. The result was that Alco and Elliot made turbos were at one time completely interchangeable in every detail; they still retain many interchangeable parts.

The 700 series turbos fit the 16 cylinder engines and the 500 series the 12 cylinder engines. For general reference see sketch 26-1.

Gas inlet casings must be specified to match exhaust manifolds, nozzle rings and diffusors differ on turbos of the same model depending on engine rating. In addition to these parts there are only three pieces of hardware which differ between the 700 and 500 series.

The impeller of the 700 turbo is slightly wider than that of the 500, the inducer is a different casting, and the air inlet casing is machined to match the impeller

In about 1963 it was intended to obsolete the 500/700 series turbochargers and replace them by model 600/800. When the new models proved after extended field service in many installations not satisfactory, the 500/700 series turbo was upgraded in material and reintroduced in production.

In the succeeding paragraph many failures, the parts causing them as well as corrective action taken are discussed, but when considering turbo failures in general it should always be remembered that even a perfect turbo can be destroyed easily by parts of foreign material hitting the nozzle ring. At one time probably in the mid sixties causes of failure on turbos returned for rebuilding, were recorded. Only such cases where an actual foreign material part could be found in the turbo were considered proven foreign material failures and recorded as such. The incidence of these proven failures did not fall under 30% and occasionally reached 50%. Typical parts to be found in the turbos were piston ring groove inserts, piston rings, molten aluminum coating the nozzle ring, broken off valve guide ends, parts broken off the exhaust manifold, broken off injection nozzle tips.

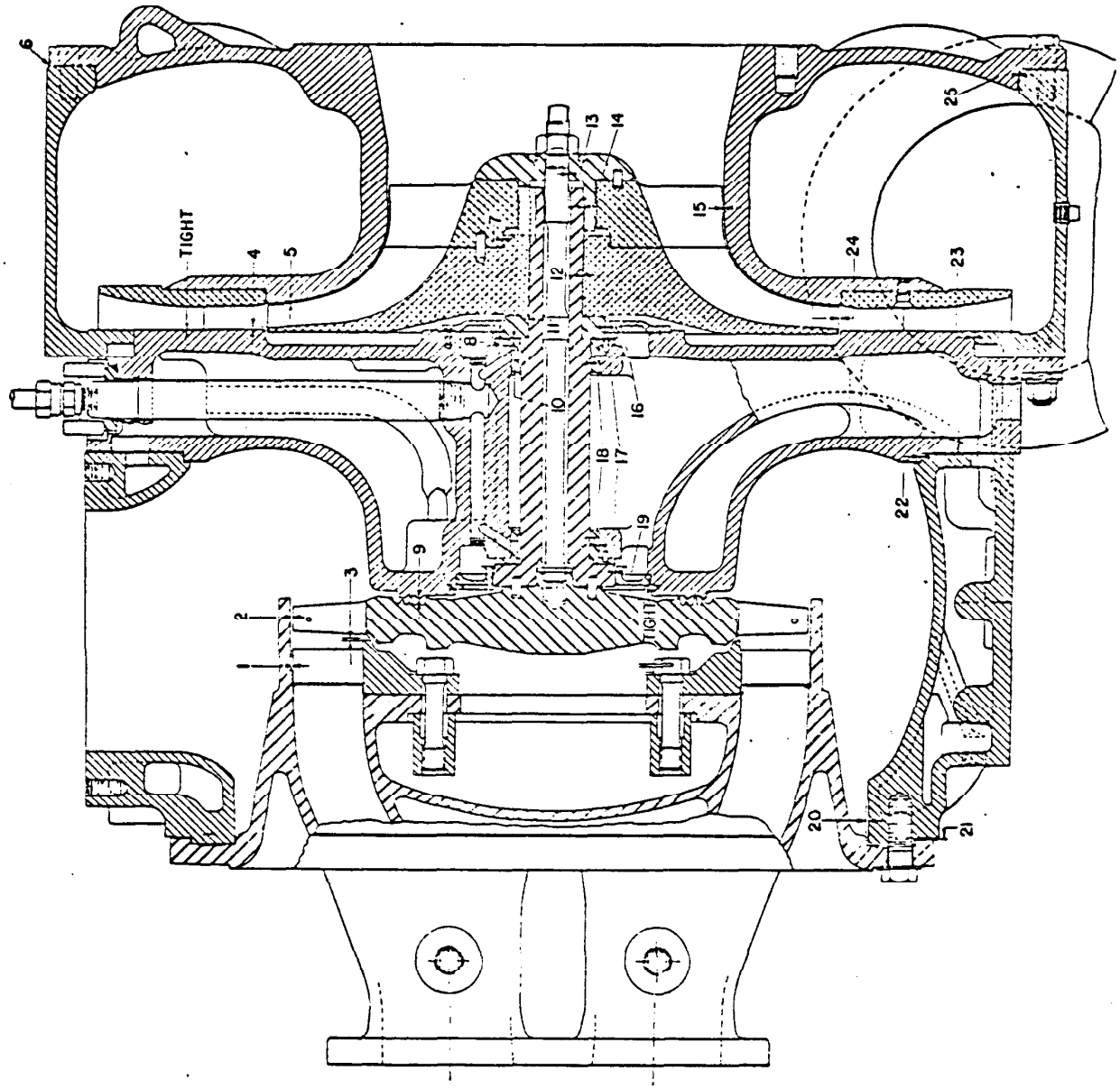
One frequently encountered report was that a failure had occurred on a turbo which had operated satisfactorily for many months. The railroad replaced the turbo by turbos rebuilt by Alco and every one of them failed within a few hours or days. The reason for the early repeat failures could usually be traced to the fact that the exhaust manifold had not been removed or even seriously inspected for containing additional loose parts of foreign material.

The nozzle ring was originally made by Elliot with sheet metal vanes cast into an inner ring and outer ring the latter fastened by a bayonet type joint to the gas inlet casing. This was soon replaced on Alco made turbos by a nozzle ring comprised of an inner ring made of a 410 type ferritic chrome steel casting

500/700 TURBO

ALCO

TYPICAL CROSS SECTION



SKETCH 26-1

with cast in profiled vanes of 18-8 chrome nickel steel. The inner ring is fastened to the gas inlet casings by four bolts. While thousands of nozzle rings of this type have gone into service, periods were frequent when nozzle rings failed epidemically with vanes broken off next to the ring casting.

The 18-8 steel of the vanes actually has a lower melting point than the 410 inner ring. A wash and temperature control during pouring was relied on to prevent local melting of the vanes. If they did melt a brittle alloy resulted locally which would lead to early service failures. If on the other hand vanes were loose in the casting this also resulted in failures. A turning operation was added to the ring to expose a part of the cast in root of the vanes to inspection for partial melting. Fairly recently even the most carefully controlled nozzle rings of this type failed in commuter train service, involving steadily repeated stopping and acceleration to full speed. A change to making the nozzle rings of precision integral castings has finally overcome this problem.

Up to the early Sixties turbine buckets were Hydrotel machined out of twisted 19-9DL bar stock, the same material from which disks also were made. Buckets were and are joined to the disks by broached fir tree roots. Longitudinally they were held by peening on the portion of the disks between the brackets. In the general upgrading, buckets became precision forged of Inconel X-750 and the disks Incolloy 901. Fastening by peening was changed to fastening by bent over tab locks.

The prevalent mode of failure of the original turbine wheels was thermal growth in diameter until the buckets rubbed in the inlet casings. Root failures were virtually unknown. The gas inlet casings were originally made of one of two types of low chrome-Moly steels, 5CR - .5MO and 2.25CR - 1MO. They had the great advantage of being readily salvage weldable, but they did erode thermally badly, also distorted and cracked.

Note that it is essential to reasonably efficient operation that the tips of the turbine buckets do not have much clearance from the surrounding shroud of the gas inlet casing. Note further that the rotor is held in bearings located in the water cooled casing while the gas inlet casing is uncooled and should be kept in line with the axis of the main casing. This aims to be accomplished by a double pilot fit; in cold condition the male pilot on the water cooled casing fits the recess in the uncooled inlet casing with minimum clearance, while substantial clearance exists between the bore in the cooled casing and the male fit in the uncooled casing. As the turbocharger changes from cold to operating temperatures the loose clearance on the inside becomes tight, and the tight clearance on the out side becomes loose.

The importance of holding tip clearance over the buckets small while being dependent on the strange piloting of the casings, emphasised the need for distortion free inlet casings; materials were changed from Chrome-Moly steel to Niresist #3, to Ductile Niresist 3, to stainless steel castings, also stainless sheet steels; cracking occurred to greater or lesser degree on all these materials.

Recently a water cooled gas inlet casing was adopted, it has not yet been proven in service.

The inducers originally had circumferential shrouds which rotated in a recess bored in the air inlet casings. The inside of the shroud collected by centrifugal action a tight layer of dirt; when the layer became thick enough centrifugal force on the shroud rim exceeded the beam strength and shrouds failed between the vanes. Consequently, the inducers were made without shrouds but the recess machined into the air inlet casing had to be filled up. The obvious method of boring out the recess and pressing a bushing in required substantial machine work on a large lat

An insert split like an internal piston ring was therefore developed which permitted rotor change out in the field without maching.

For a long time the intermediate casing was plagued with thermal cracking, and no gasket between this casing and the turbine casing would live. One contributing cause was cooling water boiling off on the higher elevations of the cross continental runs. This was overcome by adding a 7 psi pressure cap to the locomotive cooling system. The other cause was related to the unclear drawings taken over from Elliot in that the intermediate casing had large steam pockets. An elaborate harness of vent pipes from the intermediate casing and various points of the turbine casing eliminated this failure type.

A related condition was that with the original piping the turbo cooling water outlet was connected to the engine outlet headers. It was found that under one condition, which in locomotive operation in mountainous service can be quite frequent, cooling water flow thru the turbo was interrupted for a minute or even longer. This occurred if the engine operating at full load up a grade getting the turbocharger very hot was notched back to idle speed. Water pressure at the reduced speed was not enough to overcome steam formation on hot metal parts of the turbocharger. This was corrected for this turbo and all later ones by connecting the water outlet from the turbo directly to the suction side of the water pump. This together with providing a small vented collecting chamber at the outlet from the turbo prevents interruption of the cooling water flow. On later turbos the function of the collecting chamber is accomplished by slanting the top flange of the turbine casing, the high point is vented, the water outlet to the suction side of the pump is placed at the low point.

There are two cases where this arrangement has not been used. On 6B and 320 turbo the water outlet from the turbo has a very short connection to the engine outlet header and was left that way.

On the LST engines with high shock requirements it was judged better to avoid having the shipyard make the connection at the pump suction; the outlet from the turbo was led directly into the cross over outlet pipe from the engine. Since these engines would be operating with cooler water than locomotives and would not usually go from flank speed to idle this was considered acceptable.

Bearing failures were and probably still are too frequent. One type seemed due to oil whirl, characterised by a loud horn like howl lasting several seconds alternating with several seconds of quiet running. It usually occurred only at full engine load and could be stopped by dropping load somewhat. No clear reason for such occurances could be detected; if the turbo was disassembled and reassemb with the same parts the noises may or may not have stopped. This occurred frequently on engine or locomotive test. It seems to have become very rare in recent years; since no changes were made which would obviously influence this condition, the greater care in balancing and holding tolerances may have correcte this. NOTE: Whirl is still a 520/720 problem - about 1 in 10 FRS engines. (RDS)

The thrust bearing surface consists of a number of narrow flats which join with very smooth transition oil wedge producing tapers. A few years ago (LST turbos may have been affected). American Bearings furnished such thrust bearings in which the transition between taper and flat had a step, less than one thousands, but enough to casue very early failures. It could never be established how many bearings with this condition went into the field before it was discovere

One of the perennial complaints on this turbo is the inadequacy of the oil seals. If the seal clearance is too small the seal will squeal if it is too loose oil w pass and carbon up the labyrinth in the turbine disk and also let too much seal. air into the crankcase.

These seals are threaded labyrinth seals; on these turbos the rotating shaft is smooth and the stationary bore is threaded. On the 320F the same complaint was overcome by reversing the condition and placing the thread on the rotating shaft and making the bore smooth.

It should be noted that a fairly complete redesign of the 500/700 turbo, which included the reversed labyrinth together with other more important items like unloading the thrust bearing, more stable journal bearing was developed in 1969. One such turbo was briefly tested at the time and required some rework, this turbo is believed now ready for application. Material for a field trial of 9 additional turbos was procured. It was never machined and the whole project remains in limbo. The only part of this redesign now incorporated in production turbos is the water cooled inlet casing together with a nozzle ring designed to fit in it. Unfortunately the redesign is largely not interchangeable in many respects. * See Note.
There probably are still many more 500/700 turbos now in the field on Alco engines than any other type. No data are at hand to show if reliability and warranty costs have become acceptable.

One point which further depressed the performance record of the 720 was its installation on the 16E engines rated at 222 MEP. This engine was intended from the beginning to be equipped with a 165 turbo but getting the 165 into production was delayed for over a year and all original 16E engines went out with a 720 turbo known to be operating beyond its design capacity.

* NOTE: This paragraph is in reference to the 154 turbo. The first of nine turbos has been run on a lab engine - it whirled.

R.D.Smalley
8/25/72

320 TURBOCHARGER

When the 251 engine was in the design stage turbo chargers type 500/700 for the Vee engines were available; but for the inline engine no turbos at least none for manufacture in house were ready. Development of a turbo called 320 intended specifically for the inline engines was started. Unfortunately the 8 cylinder engine was very much in the picture and the 320 turbo was dimensioned with this engine or rather with an eight cylinder at increased ratings in mind. The resultant turbo has a rather large frame and used on the 6 cylinder engine at 192 MEP, has strangly short buckets and nozzle ring vanes. Built for one engine type the turbo casting embodies a bracket to bolt the turbo to the cylinder block without any other supporting structure. It has its shaft axis athwart the engine axis and of all turbos made by Alco it is the only one rotating clockwise, looking from the compressor end; all others turn anticlockwise. This is a significant point because it prevents using any of the precision forged buckets made in larger quantites on this, now, small volume machine.

The 320 turbo is a perfect example of relation of load factor and reliability of a piece of machinery. As long as the engine was sold domestically in fairly large numbers for switcher locomotives the 320 of original design appeared a perfect machine of which one never heard. As soon as these engines were sold abroad and used at the same rating but in main line service they became completely unsatisfactory. It took redesigns numbered from A thru F to come up with a machine that lives in heavy service. The final F version see sketch 26-2 seems to be fully satisfactory. The 6-B engine and with it the 320 turbo was considered obsolete several times but revived as many times when substantial foreign orders were received.

The original version had the turbine disk and shaft made of an integral 316 stainless steel forging. Slipped over the shaft was a case hardened steel sleeve to provide suitable bearing surfaces. The impeller with pressed on inducer was clamped between the end nut and a collar with 90° male cone fitting into a 90° recess in the impeller. Any transient temperature differences could result in increasing bursting stresses in the impeller. Bursting impellers with consequent bursting of the main casing became so frequent at on time on a certain mountain ra line in Greece that in order to avoid accidents service personell were expressly forbidden to take turbo speed readings with a FRAM tachometer.

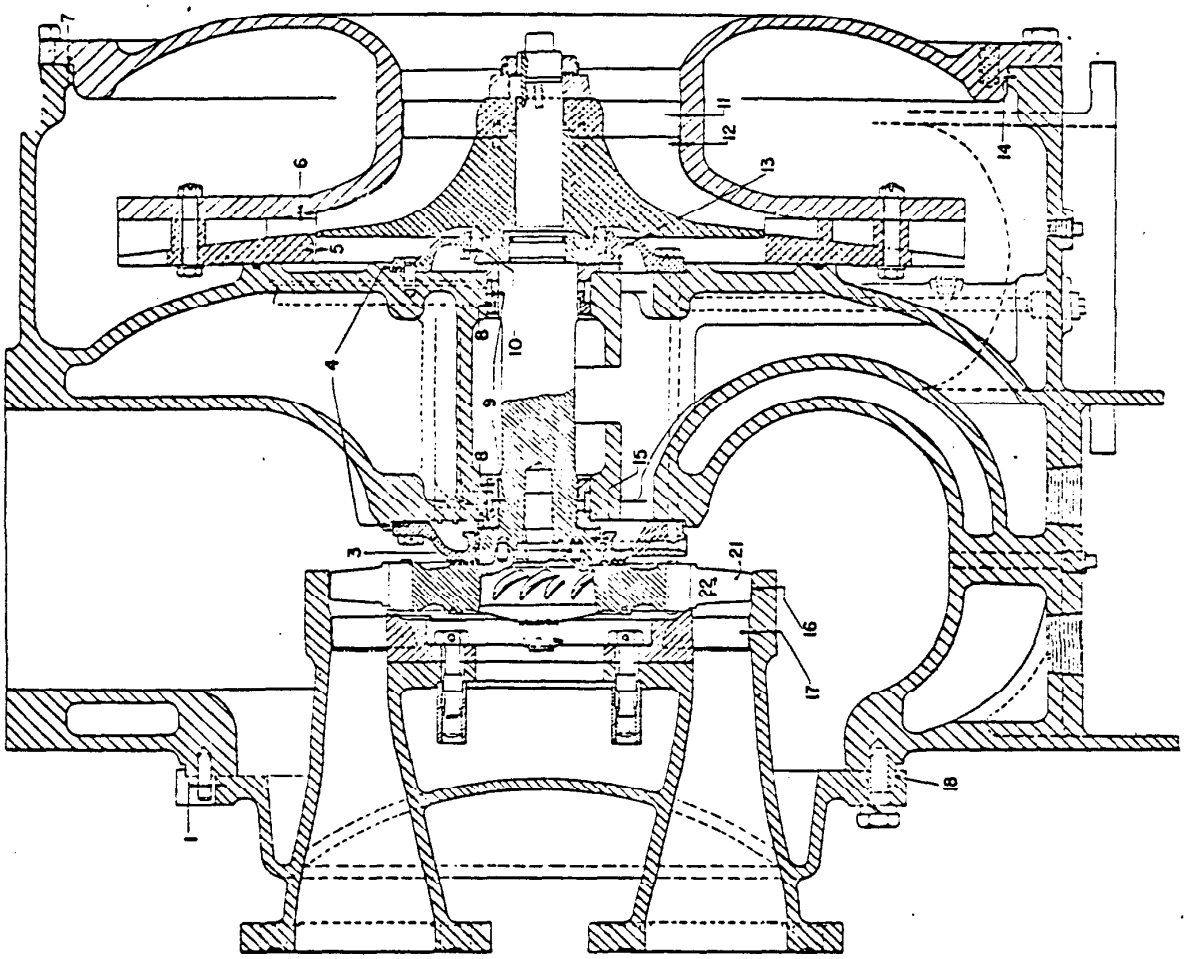
The next to last version, the 320 E, had a rotor which copied all significant features of the 500 but acceptable reliability was not attained. It would take careful examination of drawings to trace the whole 320 development which hardly is worth while since any rebuilding of 320 turbo would be made to the F design. One deficiency which may still cause failures of rebuilt machines, if not recognized should be mentioned.

The 320 was the first Alco built turbo using one large casing in contrast to the three casing construction of the 500/700. Engineering did not specify or the shop disregarded drawing requirements, in any case the counter bores locating the seal plates are not necessarily concentric with the main bearing bore, one of them was made after turning the casting around without suitable precautions. It took several failures until this fact was recognized. Since already many casings were then in service the correction (sic) was to reduce the seal plate OD so it would obviously not be centered in the recess and to line up the seals with an arbor at assembly and locate them with two dowels.

Seal leakage at one time was a major complaint item and for a while brake air was piped into the drilled holes for the sealing air in the main casing; this was

26-6A

320 F TURBO



SKETCH 26-2

always one of two alternates, the other is to plug the connection to the outside piping and open the connection to the high pressure area of the compressor. Since operators never did like the constant bleed off of brake air for this purpose and since the seals are presumed to be now satisfactory it is believed that most locomotives disconnected the line to the brake air tank.

The early 320 were very prone to intermittent howling as described before. The fact that the hardened sleeve pressed onto the shaft necessarily tended to introduce some excentricity may have contributed to these occurrances.

350 - 600 - 800 TURBOCHARGER

In the early sixties a separate R & D group was charged with developing a new series of turbo chargers to overcome all the short comings of the 320 and the 500/700 series. These turbos were developed, the 350 turbo to fit the 6D, the first engine uprated to over 240 MEP, the 600 to replace the 500 series on the 12 cylinder and the 800 to fit the 16 cylinder engine. For general reference see sketch 26-3 , a section through the 350 turbo.

The new series of turbo was built and put into the field. Some clear mistaken judgement had been made but also some important improvements. If the engineers responsible for this development has remained with the company, one quit, the other retired, the short comings could probably have been corrected rather than to scrap the design altogether and restart with the 131/165 types.

The 350 turbo was applied and is still operating on all but the last dozen 6D engines built. A substantial number of 600 turbos did go into the field and were not recalled. For new production they were superseded by the upgraded 520, a decision which did not result in improved engine performance. Only a few 800 turbos went to the field, they failed quickly and were recalled and replaced by 720.

This series of turbos does have one clear improvement in using floating steel backed bushings as bearings, eliminating oil whirl complaints. These bushings cut to half their length were retained for the 131/165.

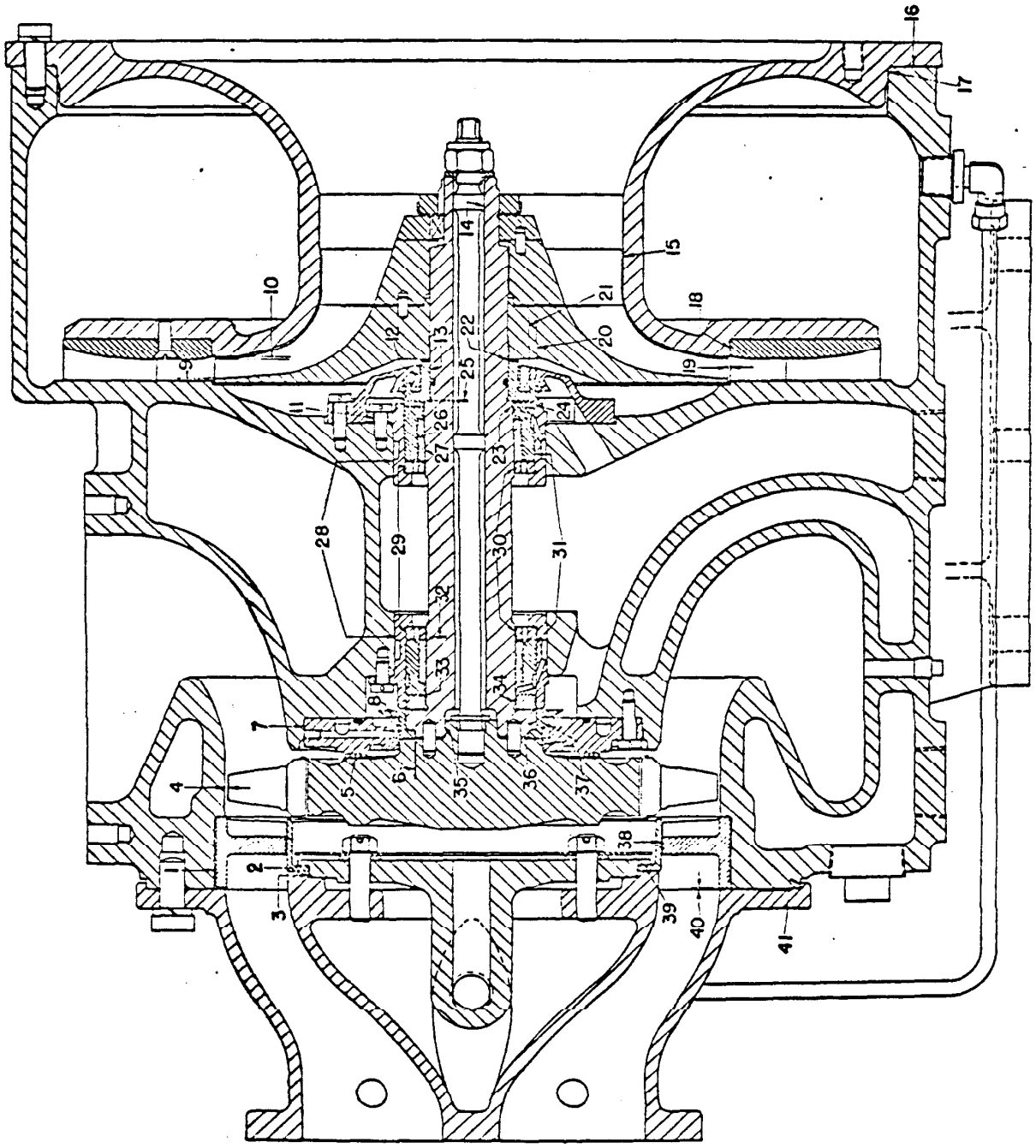
These turbos utilize large casings which include the air compressor housing (scroll is not quite descriptive because they are of plain cylindrical shape).

In contrast to the 500/700 which were designed for universal mounting conditions and therefore require a not too desirable mounting at three points, in the middle and on two sides, the 350/600 machines have a straightforward decently spread four bolt mounting. To fit this mounting, suitable supports were incorporated in the aftercooler cover, while the bridge supporting the 500 was eliminated. More difficult to handle in the field (for a while a campaign to replace 500 turbos by 600 was under way) was the location of the compressor air discharge side ways inclined about 20° from horizontal. This required welding a large flanged pipe to the turbo support and installing a substantial discharge elbow for connecting the turbo to the support casing.

The most fundamental maljudgement was in the choice of material for disk and buckets. Upgrading was clearly desirable but the step, from 19-9 DL to A286, was too small to be of any significance at expected operating temperatures and any advantage was lost if the turbo operated at higher than expected temperatures as it may happen due to manifold leaks or faulty injection equipment. A286 buckets could still be machined from bar stock by milling.

CROSS SECTION 350 TURBO

600/800 TURBOS SIMILAR



SKETCH 26-3

The following data illustrates the point: Stress to produce rupture (psi).

	19-9DL	A286	Inconel X	Incolloy 901
in 100 hrs. at 1200°	40,000	61,000	80,000	81,000
in 100 hrs. at 1500°	13,300	13,800	28,000	19,000
in 1000 hrs. at 1200°	34,000	46,000	69,000	64,000
in 1000 hrs. at 1500°	10,000	7,700	18,000	----

Another serious shortcoming which killed the 800 outright and caused failures on the 350 especially if turbo speed at higher elevations was not well controlled is insufficient capacity of the floating thrust washer. On the 131/165 the bearing capacity has not been much increased but the load has been reduced by partly balancing the air pressure on the impeller.

In the preceding section 500/700 the importance of maintaining small tip clearance was discussed. On the 350/600 a not quite successful attempt to control this was made. The turbine wheel supported by bearings in the main casing will always stay concentric with the bore of the casing, thereby eliminating one source of tip rubbing, but a much more significant source of tip clearance variation was introduced. In order to have adequate clearance with a wheel at operating temperature and a water cooled housing, clearance with the wheel cold must be made so large that acceleration of the turbo from low power is seriously affected.

Another point difficult to handle was that the nozzle ring which is subject to large temperature and therefore size variations is also confined in the water cooled casing. It led to collapsing the inner shroud until large clearances were provided. The large clearances however made it difficult to hold the nozzle ring from rotating until some substantial stops were added.

In both the 350/600 and the 131/165 series turbo designers insist on the need for leading cooling air through the gas inlet casing to the center of the turbine disk. This makes for complication of inlet casing and piping. The need for it should be proven in field service. Temperature measurement taken at one time on a turbine disk did not show a startling reduction when air was applied. Under frequent idling these passages are likely to become plugged by oil and soot in any case.

For the 350 a watercooled inlet casing has been developed and is being marketed. It is believed to have improved reliability of this turbo markedly.

131 AND 165 TURBOCHARGERS

When the 800 turbo did not become operational the need for a turbocharger for higher rated 16 cylinder and for a contemplated engine with a larger number of cylinders became acute. The perfectibility of the 350 turbo and its adaptability to a fairly high rated 8 cylinder also was considered questionable. Therefore the design from scratch of two presumably similar turbos of sizes suitable for these purposes was started.

The smaller 131 turbo was pushed ahead and applied in 1966 to production 8 cylinder locomotive engines. A substantial number of these machines have since been put in operation on lower rated 8 cylinders, on 12 cylinders operating at 900 RPM or less, also on a few 6D. In general few field complaints arose then or now.

For reason of company policy or politics, work on the 165 was delayed for over a year. When it did get into the field, it encountered severe failures in several areas.

With the 131 and 165 turbo this design review is brought up to current time. Informed and experienced personnel are still in the department and working on these problems and qualified to discuss them in detail. This write up is confined to listing weaknesses and to mention corrective action as remembered by a bystander.

These series of turbos have the following characteristics. Buckets are precision forged Inconel X-750, disks are Incolloy 901. Fir tree roots are different in detail from the earlier turbos; buckets are held by tab locks. Buckets originally were drilled for and assembled with damping wires; distortion of the wires brought unbalance, buckets broke out of the drilled holes. Wiring and drilling was eliminated without incurring bucket vibrations. Elimination of damping wires was then also carried back to the 500/700 turbos.

The disks are held to the shafts by a short stud and expansion compensating sleeve, an arrangement first used for the redesign of the 320. This is a great improvement over the long stud and small spud of the 500/700 where separation of shaft and disk is not too infrequent.

Gas inlet casings are of watercooled low carbon steel castings. Inadequate rounding of cast intersections caused frequent failures by cracking. Suitable changes have been made in the castings. Nozzle rings have sheet metal vanes cast in a solid inner and a split outer ring. Tip clearance is held small and constant by a thin-walled shroud ring designed in a way to permit nearly free thermal expansion and to maintain the same temperature as the buckets under all operating conditions. The bearings are narrow floating bushings. The thrust bearings are located on the compressor side for two reasons; to get them away from the hottest area, also to use their influence of damping oil whirl on the compressor side, where whirl seems to originate more readily than on the heavier turbine side. On the 131 the

thrust bearing is between journal bearing and compressor impeller, where it also was on the early 165. This makes for rather short distance between the journal bearings and was believed to have contributed to instability of the heavy rotor of the 165. Therefore on this machine only the thrust bearing was moved to between the journal bearings next to the compressor end bearing. This increases the distance between the journal bearings substantially, at the cost of more awkward assembly.

The load on the thrust bearings is reduced by a labyrinth seal between impeller back wall and main casing so that this space is not exposed to full charging pressure and in addition this space is connected by a large cored passage to the space between turbine disk and main casing. This space again is sealed off by a labyrinth. The two areas on wheel and impeller on which air pressure acts are not of the same diameter, thrust is not eliminated but reduced.

Considerable shaft length has been devoted to make sufficient space available for oil seals. Since the floating bushings create double bearing clearance, seal clearance must be increased accordingly. The problem of too much sealing air discharge into the crankcase has not been improved compared to earlier turbos.

Both machines were originally equipped with precision cast aluminum inducers, these failed badly on the 165, much less on the 131. They were changed into steel castings for both models but the failure rate on the 165 hardly dropped. Changes in heat treatment, better control of dimensions together with removal of vanes in the air inlet funnel as possible source of vibration exitation seems to have alleviated if not eliminated these failures. * See Note.

The aluminum impellers had a tendency to a small relative rotational movement between shaft and impeller, leading to fretting and ultimately to unbalance. Again pronounced on the 165, hardly noted on the 131. It seems to have been overcome by inserting a steel sleeve with heavy interference in the impeller bore.

The 131/165 are the first Alco built turbos with a true compressor scroll, it can be rotated relative to the main casing to fit installation conditions.

Both turbos can be built with different length buckets to fit the various engine requirements. The 165 also can be equipped with two different compressors and scrolls.

The larger of these scrolls used on the 18 cylinder engine is arranged to accept four nozzles for a "jet assist" arrangement to improve acceleration of the turbo and load acceptance of the engine.

* NOTE: All 165 inducer failures have been in Canada. Extensive testing with Ontario Research indicates only cause to be cold weather surging. Work is under way to increase surge margin.

Current inducers are turfftrided to improve fatigue resistance.

R.D.S.
8/25/72

Exhaust Manifold

The struggle to produce exhaust manifolds giving satisfactory engine performance and adequate life began with the first 9-X-10½ 241-engine in the early Forties and is continuing with many failures and some successes to this writing. (1972)

Ref. 1 recounts development, mostly from a mechanical and metallurgical point of view from 1944 to 1953. The report was written to help management to defend itself from accusations by directors and customers about lack of effort to produce a satisfactory manifold.

Shrouded Manifolds

The last 244 production manifolds in this review, both for the 12 and the 16 cylinder, are those on which the design of the first 251 manifolds were based. On the 244 the exhaust parts in the cylinder heads were so arranged that the exhaust elbows of each pair of opposite cylinders came very close together carrying some design difficulties

For this reason air and exhaust port flanges on the 251 head were reversed in comparison to the 244, giving more even spacing of all exhaust elbows. Bolting to cylinder heads was made more positive for the 251.

Ref. 1 shows the many approaches, sensible and otherwise, which have been considered; many of them were actually tried.

Very brief summary of the lessons learned:

1. Standard weight stainless steel pipe, welded with stick electrodes, is prone to failures in the heat effected zone due to intergrannular corrosion. (The welds between the long pipes on the current 18-cyl manifolds may become subject to such failures).

2. Sliding joints can only work with leakage, if sliding joints are used leakage must be collected and disposed of.

3. Single ply bellows won't live.

4. Fairing of duct intersections need not be over emphasised, although later experience with original square entry, single duct in comparison with the subsequent "streamlined" manifold showed substantial and measurable improvement.

The 251 manifolds remained in production without essential changes till about 1962; they are described in ref. 2 and shown on sketches 28-1 & 2. They were also patented in a rather ineffective effort to keep Hunt-Spiller off the profitable replacement market.

Based on much earlier performance tests, the 12 cylinder manifold pipes were based on 3½" ID; the 16 cylinder pipes on 3-3/4. The difference can be rationalised by considering that one firing interval in the 4 duct 16 cylinder is 45° while it is 240° in the 12; double puffs in the 16 Cyl. pipes resulting in.

On the 12 cylinder manifold the joints gradually grew; the bells might occasionally split but on the whole the manifolds lived long enough to justify salvage trials by Alco, actual routine salvaging by some customers by weld deposit and remachining of the male ends to over size and boring out the bells to a larger dimension. On all later manifolds the male ends were slightly crowned to accommodate some misalignment.

On the 16 cylinder engines the joints hardly stayed long enough in service to wear

to the point of considering salvage.

All these shrouded manifolds were assembled on a fixture completely and then applied to the engine as one unit. Principally there were individual elbows attached to the cylinder head, the flange at the other end being in one horizontal plane; these flanges or their mates on the tees above them had a projection slightly longer than the thickness of the "base plate". This served as base for the shroud but also to locate the many individual components above.

A very definite distinction must be made between manifolds for the 6 & 12 cylinder engines and those for the 16 cylinder engines when discussing adequency of design life.

The 6 & 12 cylinder manifolds ,which never required the collection of gases from cylinders on the opposite banks but rather allowed collecting gases from three adjacent cylinders on the same bank ,permitted much simpler designs. While the sliding joint manifolds on 6 & 12 cylinder engines, the majority of Alco production in the Fifties and early Sixties, were not really satisfactory, the number and severity of complaints was in a different order of magnitude from that of the 16 cylinder manifolds.

When ref. 1 was compiled it was hoped that re-availability of high Nickel containing irons,Niresist initially, later ductile Niresist 3 would eliminate growth and binding of the sliding joints; actually it reduced the problem somewhat but especially on the 16 cylinder manifold, to a very limited degree.

One pecularity was that the elbows attached to the cylinder heads never failed in any form whatever, in spite of being made of virtually plain cast iron. The base plate was finally made of low alloy steels; on disassembly they were usually found cracked and needed replacement.

On the 12 cylinder manifolds there was originally frequent ratcheting of the long pipes joining the "offset" tee of cylinders #4 to the turbochargers, these failures were nearly eliminated by firmly anchoring the pipe at the turbo inlet casing. The "offset" tee was pretty well kept aligned by the tee of #5 cylinder.

In contrast the 16 cylinder manifold with short "tees" some quite high, connected by pipes of varying length to other "tees", it was never consistently possible to avoid cocking between pipe and tee at the joint which was supposed to slide, while the other end of the pipe did not consistently make a rigid joint with the "tee". The usual mechanism of failure was mostly freezing a joint, ratcheting out, cocking to the point where the bells cracked. None of the many changes on details made any significant improvement. The final advent of the single duct manifold constitute somewhat of a break through which permitted increase in sales and power of the 16 cylinder engine.

It is interesting that some marine engines both 12 and 16 cylinder engines were built i the late Fifties in which the glass-wool insulated shrouds were replaced by water blankets of stainless steel, the manifolds themselves remaining the same as those considered standard at the time. From all reports these water jacketed manifolds, both on 12 and 16,performed without significant complaints for a number of years.

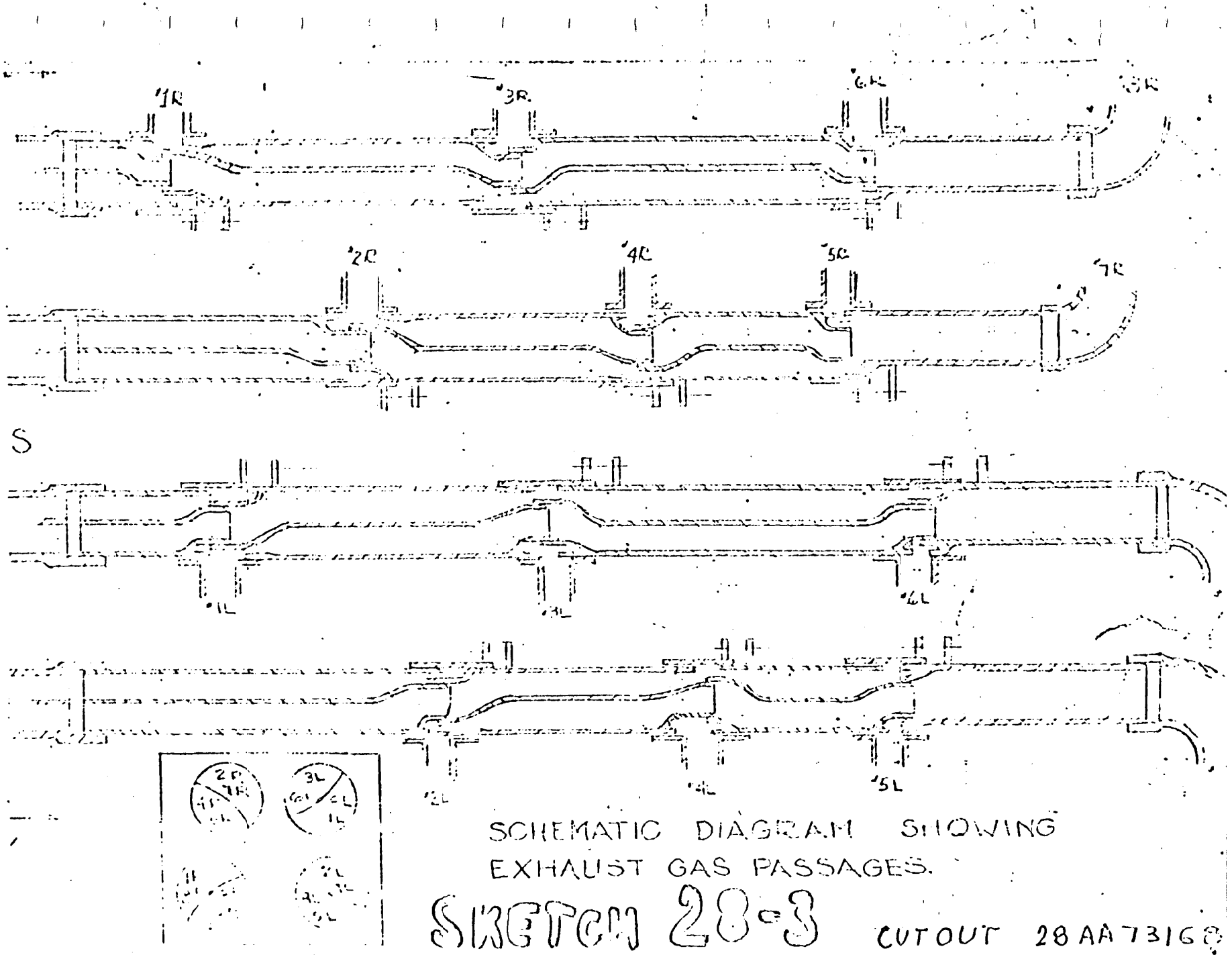
This suggests that some cooling of the internal of the shrouds might have improved performance. As told in ref. 1 the shrouds did start out with strong ventilation; cooling air discharged from the generator was originally piped into the shroud. Unfortunately, feeding fresh air into the shroud, in which not completely burned lube or fuel oil was also discharged in the presence of hot metal parts, led to frequent severe explosion. The large 6" diameter vent from the shrouds is a carry over from that time.

In the later Fifties some dual fuel engines were sold, which accentuated the need to reduce differences in cylinder exhaust temperatures. An eight duct exhaust manifold with sliding joints was designed, see sketch 28-3. The eight duct design was accomplished in principle by starting out with the four duct design and cylinder distribution and dividing each of the four pipes in two halves, one would be fed by two cylinders on the right bank and the other by two cylinders on the left bank. One strange result, not recorded in the lab reports nor on the assembly drawing, was the need to make a restricted connection at the generator end of all pipes between the two halves to obtain reasonable acceleration. The detail drawings of the end pipe show this by-pass to be two 1¼" diameter holes.

On the dual fuel engines at 154 MEP these manifolds, inspite of the awkward design, gave reasonably satisfactory performance for a number of years. Their application on locomotive engines proved disastrous. About 20 engines were so equipped and all had to be changed to four ducters as they failed mechanically in even shorter time than the four ducters.

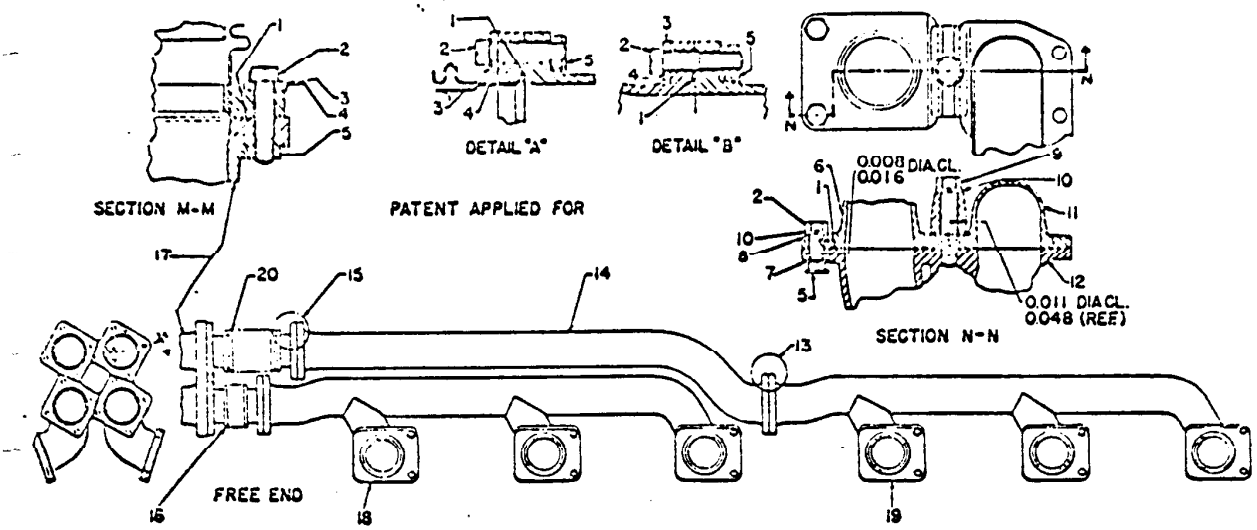
The drawing number 28AA73168 for this 8 duct manifold shows the dual fuel version in which the manifold and turbo charger are raised 6 5/8" compared to the standard manifold to make room for the dual fuel gas valves built on top of the air inlet elbows.

This drawing also indicates an "Foreign material catchers". These were made of high temperature precision castings. They never became a production item but a number of them were field tested mostly in 12 cylinder manifolds. They reflect the fact that a larger percentage 30 to 50% of turbocharger failures were due to foreign material. The catchers probably did catch broken off valve guide extensions and some manifold parts. Piston ring fragments passed through molten aluminum from pistons passed through. After service exceeding one year the catchers themselves showed cracks. The short comings of the catchers themselves, their resistance to flow and concerted efforts to prevent failures on other parts, led to abandoning such approaches.



SCHEMATIC DIAGRAM SHOWING EXHAUST GAS PASSAGES.

SKETCH 28-3 CUT OUT 28AA7316



1. Gasket
2. Capscrew
3. Lockplate
4. Spl. Washer
5. Spl. Nut
6. Exhaust Elbow
7. Washer, Flat
8. Spl. Sleeve
9. Strongback Bolt
10. Lockwasher
11. Air Elbow
12. Cylinder Head
13. See Detail "B"
14. Upper Pipe
15. See Detail "A"
16. Lower Connector - Bellows Type
17. Turbo Gas Inlet Casing
18. Front L.S. Manifold Section
19. Rear L.S. Manifold Section
20. Upper Connector - Bellows Type

FIG. 2 - PHOTO 5371
EXHAUST MANIFOLD ASSEMBLY
LEFT BANK - SHOWN - RIGHT BANK - OPPOSITE

SKETCH 28-34-19-A

another case of the impossibility to weld a thin stiff tube against a solid flange. A half hearted attempt at making the joint more flexible by turning a semicircular groove into the flange was ineffective. The tubes broke next to the flanges even after a support had been introduced to stop the tube end and bellows flange from vibrating. The support, a thin stainless steel sheet fastened to the cylinder block, flexible longitudinally but rigid vertically and laterally, has proven effective in stopping vibrations. In production the cast pipe was reintroduced, but a substantial number of thin walled tubes were continued in foreign service after reinforcing the flange joint per 50B73955.

This manifold has been in commercially satisfactory operation since about 1963. A heat shield was designed for this manifold but locomotive users comments led to omitting it. A suitable design is available for installations where it is needed.

Recently a new manifold ref. 4 has been designed to permit the turbocharger to be raised, to make room for a larger aftercooler. It has the pipe size increased from 3½ to 3 3/4 ID. The legs are made longer because the end flanges had to be raised. It is used on the 12 F engine but no service experience exists to date.

The 6 cylinder engine continued for long time with the sliding joint shrouded manifold. It had operated mostly in foreign parts, and few complaints were received. Salvaging by weld deposit of joints was routine. In retrospect it is probable that manifold leaks contributed to if they did not cause the frequent failures of the 350 turbo.

The 6 cylinder engine is not (since 1969) equipped with a manifold with bellows based on the 12 cylinder design; Ref. 5.

The 18 cylinder manifold ref. 6 also is patterned after the 12 cyl. but no parts are interchangeable the cylinder center distance is different. It has standard weight pipes butt welded to cast flanges connecting the intermediate and generator end castings to the turbo. Bellows at the turbo end absorb longitudinal expansion, supporting members at the bellows to pipe flange joint support the pipes. The current design of the support is second generation, the first one having failed. On this manifold bellows a weakness was discovered. The free turning flange is slipped over a ferrule originally made by spin forming a tube section in the center of a round flange. In spinning the metal is thinned in the transition between tube and flat section. After a failure metal thickness was found about 1/32, sectioned manifold bellows of the 16 cylinder parts used for years without failures were also thinned. Fabrication has been changed to stamping which minimized such thinning. Bellows for the 6 and 18 cylinder are procured as triple laminations from Flexonics. There is no compelling reason for this, they are probably somewhat more expensive.

The eight cylinder, the only Alco Vee engine with non consecutive firing order for the cylinders of the two banks connects each pair of cylinders on the same crank with one of the four exhaust pipes. Each elbow on the left cylinder is bolted by four bolt flanges to a Tee on the respective right bank cylinder, the other end of the Tee leading thru a bellows to the turbo.

About 20 or 30 eight cylinder engines have been in locomotive service since 1966, quite a few of the 51 LST engines have seen many months of service; if manifold failures have occurred they were not considered serious enough to bring to Engineering Department attention.

All gaskets used on exhaust manifold joints are spirally wound made by Flexitullic or probably in smaller number by Johns Manville.

Fabricated Single Duct Manifold for 16 Cylinders: First Version, Square Entry of Legs into Main Pipe

Hanlon and Wilson, originally a supplier of various parts to railroads, had entered the aftermarket with fabricated manifolds, thin wall tubing with welded on bellows, for Alco engines. Their manifold for the 12 cylinder engine was a serious competition for both the 244 and 251 engines, but it was expensive and the Alco manifold had sufficient advantages, e.g., possibility of replacement of parts to combat the competition.

Hanlon and Wilson then gradually developed a 4 duct manifold for the 16 cylinder engine. Mechanically it was as sound a design as could be made. It was much more reliable than Alco's cast manifold. Negotiations to let H & W become a O.E.M. supplier led nowhere, mainly due to Alco's reluctance to let a "pirate" force its way in.

At about the same time, while trying on a Canadian Railroad to use 6 cylinder switcher locomotive bump cars without too much smoke, some unnamed mechanic had kept the fuel pumps racks manually at about half of full stroke and reduced smoke substantially. On the basis of this trial the first acceleration control device, controlled by air manifold pressure, was developed by MLW with Alco's knowledge and cooperation.

This was the background which led Alco management to appropriate funds to develop a single duct manifold at what appeared an outright gamble. A single duct manifold should be simpler, therefore, more reliable than the four ducter, it would not be a copy of the H & W manifold and an acceleration control device which just might overcome the impossible smoke conditions of the single duct manifolds in the mid Forties seemed right.

In designing the first single duct manifold, mechanical reliability was considered the foremost objective. Thermal expansion in longitudinal direction would be taken up by bellows while the legs would be made in shape and length flexible enough to withstand lateral expansion. A prototype manifold was built in a local shop using American Boa bellows and Marman clamps. Sketch 28-5, the illustr tion of the patent granted in 1965 is based on the prototype; all joints were MIG fillet welds, flanges were hogged out of solid. The size of the main pipe, 7½" dia, was chosen by the highly scientific approach of duplicating the cross sectional area of 4 ducts of 3 3/4 dia.

The first run on the laboratory engine showed performance including acceleration with a control device (smoke pot) promising. The laboratory report 3370 includes a clear photo of this manifold.

The first run in the field resulted in a failure of the weld joining a ferrule to the flanges at the cylinder head. This was quickly changed to the swivel joint shown as fig. 4 on sketch 28-5. After that the manifold and some 10 duplicates performed without failures for long time. Ultimately all prototypes were with-drawn because the fillet welds between the thin pipe and the heavier flanges and spacers under the bellows cracked.

Based on the design and field experience of the prototype manifolds a production design was worked out with Flexonics. The main difference was the use of resistance seam welding where ever possible, forming flanges for the Merman clamps out of sheet metal so that all components have comparable flexibility. The bellows are made by Flexonics, not because American Boa bellows were deficient, but only to keep responsibility for all parts with the same supplier. For assembly drawing see Ref. 8.

A few systemic failures occurred and needed corrections; it was at first not appreciat that the weight of these manifolds was so low that one could dispense with lifting fixtures. The early manifolds (see also photo of the prototype) had lifting loops

April 13, 1965 F. C. TROMEL 3,177,649

EXHAUST MANIFOLDS

Filed March 8, 1963

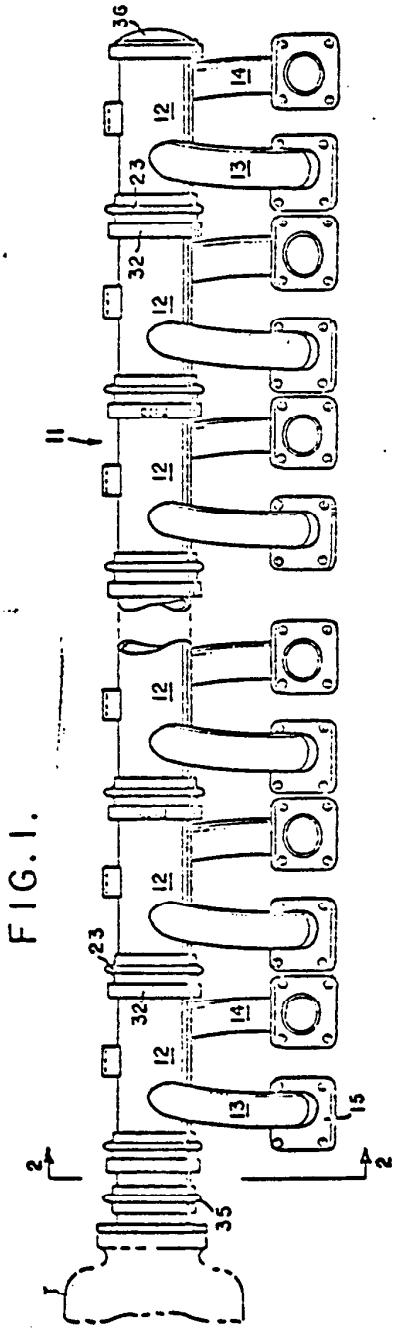


FIG. 1.

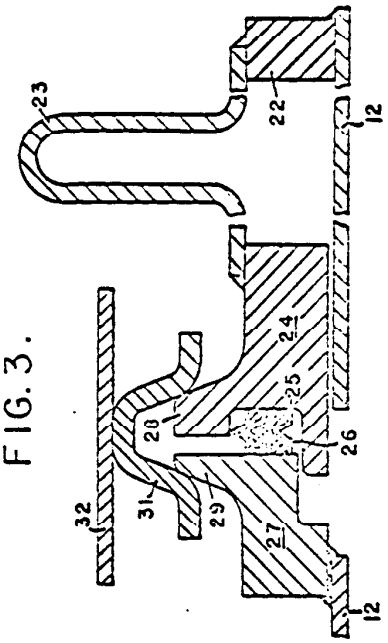


FIG. 3.

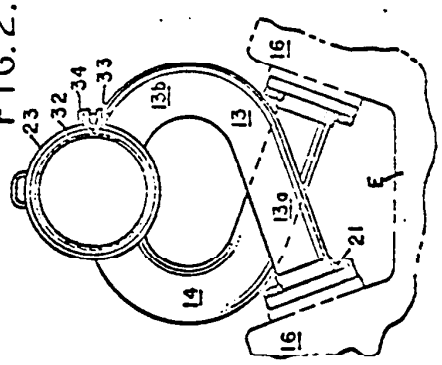


FIG. 2.

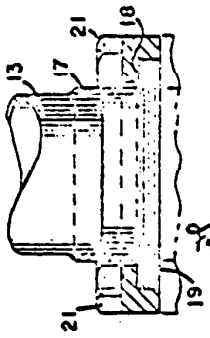


FIG. 4.

INVENTOR

Frederic C. Tromel

Sheet 2 of 5

FABRICATED SINGLE DUCT MANIFOLDS FOR 16 CYLINDER
SECOND VERSION: STREAM LINED

When G.E. did come out with their version of a single duct manifold they had paied considerable attention to fairing and acceleration smoke was considerably better than Alco's. (The CB-CE engine seems to have less acceleration smoke than the Alco even disregarding the manifold). For competitive reasons the manifold design had to be restudied. Based on Alco layouts Flexonics made a prototype, streamline manifold. To meet space limitation the size of the leg pipes 3 7/8 ID on the original square entry manifold was reduced to 3½ ID on the new design. To avoid time consuming layout and prototype tooling without knowing that anything substantial would be gained, a manifold of 8 identical sections, in appearance similar to those used on the subsequent production manifold for cylinders 2 thru 8 was made and the turbocharger moved forward so that the cylinder #1 legs could still be fed into the main pipe. Results on the test bed were sufficiently encouraging to proceed with design and fabrication of a turbo-charger gas inlet casing and into which cylinder #1 pipes could be entered directly, thereby retaining the established turbocharger location. Arrangement of hatches, air filters expansion tank, etc. in the locomotive prevented any thought of moving the turbocharger.

The first attempt was to retain the successful sheet metal inlet casing by welding two suitable nozzles with Vee band flange connectors for cylinder #1 pipes. The pipes themselves were light weight tubing with integral bellows. They vibrated and moved under firing pressures, and as usual, restraint on the thin pipes could not stay in place. Neither pipes nor the nozzles on the fabricated inlet casing could be made to live on some 20 manifolds in the field. The thin walled inlet casing had to be abandoned and after some additional trials cast stainless steel inlet casin and pipes came into use. Two horizontal small bellows, flanged to the inlet casing and Vee band connected to the pipes were put in production. Principally the combination of a flange and a Vee band coupling has these advantages. The Vee band coupling takes less room (very close quarter between bellows an main pipe and leg #1 pipes) and the gasket can be made tight without straining the bellows.

The flange bolts can then be used to stretch the bellows and make the second gasket tight.

The production manifold is shown on 28-AA-73652, the first few orders of manifolds (orders were placed in lots of 100) used saddles to join the legs to the run, on the print shown as lines 1 and 2. The cut out in the main pipe was sometimes considerably smaller than the flange of the saddle. This led to vibration of the unsupported material between cutout and saddle flange which led to cracking. It was not an epidemic but occurred often enough to cause concern. Flexonics, therefore, worked out a method to flue out the main pipe to fit the legs without intermediate saddle. The flueing out would cause some thinning of the metal; to compensate and retain as a minimum the basic wall thickness of .060 even in the reduced portions the wall thick-ness of the main pipe was increased to .080. This version is shown on lines 3 to 6 of the print. Five manifolds to this design were originally built, three vent into the field, two on LST engines, one for the high shock the other for the endurance run. The saddle-less version was used on all LST engines and went into general production late in 1968.

No significant number of failures on Flexonics, Illinois built manifolds of this type occurred. Epidemic like weld failures on this type of Flexonics, Canada built manifolds occurred, some were admittedly due to faulty fit up; others were not convincingly explained. In any case MLW and Flexonics Canada reverted to the earlier design.

8 DUCT MANIFOLDS WITH BELLOWS JOINTS FOR 16 CYLINDER ENGINES

In September 1964 the development of an eight duct manifold was authorized "to overcome intolerable acceleration and objectionable middle notch smoke". Application to locomotive engines with the severe limitations on overhead space, due to brake resistors then still mounted over the engine, was still the primary objective. The design concept intended to duplicate the approach used on the single duct to make all parts as uniformly flexible as possible. Ref. 28-12 The pipes would connect cylinder 1-5, 2-6, 3-7, 4-8 on each bank respectively. There would be per pipe one bellows at the turbo and one in between the cylinders. The first approach also provided a dummy bellows and a closed pipe between the back tee and a rigid abutting bracket at the generator end. To maintain the maximum number of equal parts no attempt at streamlining was made. The manifold consisted of 4 groups of tees of different length, each group had 4 tees, then bellows and pipes of different lengths; all parts were joined together by Vee bands. The gas inlet casing for the then current 710 turbo was a rather complicated uncooled stainless steel casting. Three such manifolds were made. The first run in April 1965 showed engine performance about as expected except that acceleration smoke was a disappointment, the way it still is in 1972. Visual observation, later reinforced by stroboscope determination of frequencies made it obvious that the manifold as designed would not live.

Some of the pipes vibrated violently in longitudinal direction others to a lesser degree. Attempts were made to quiet vibrations, first by working at the abutments, take dummy bellows out, replace by heavy springs, replace by stiff rods (arrangement used at present,1972), then by trying bellows of different stiffness, also different make bellows. Changing bellows location from one end to the other affected but did not suppress vibrations on some pipes; various types of clamps were tried. None of these changes proved effective. The manifold could be kept in operation (two addional manifolds were there to cannibulize) to establish performance characteristics

Finally two tees for 2L & 6L, the most violently vibrating pipe, were made of standard weight 1/4" thick 3 1/2" stainless steel pipe and fittings, and welded on flanges at the head flanges. Vibrations were eliminated when these rigid tees were used. It was then concluded that a manifold made of rigid tees, presumably made of stainless steel castings could be made satisfactorily; but that the cost would be very high; the combination of heavy rigid castings and light wall tubing is not an attractive design solution. Just about that time the streamlined single duct manifold became ready for testing. The eight duct performance concerning acceleration smoke for locomotive was definitely not outstanding; all these reasons led to discontinuation of further work on the eight duct manifold at that time.

Three years later in 1968 interest in the eight duct manifold was revived; the emphasis now began to veer away from the locomotive, for which the streamlined single ducter with improved fuel limit controls appeared acceptable, to river boats and diesel-generator sets where rapid acceptance of load became a mandatory requirement.

Patterns for the tees were prepared and two complete sets of castings were procured and machined. These were matched with bellows, pipes, and Vee bands still in the laboratory from the earlier program described above. One of these manifolds was installed on one engine of the river push boat Naye, the other was used for short testing in the laboratory, then stored until the laboratory was dismantled in 1970 when it was transferred to Renewal Parts stock for possible use in the field. This is the manifold which was then applied under "crisis" conditions to the Tuna Boat Quo Vadis. That means that the Quo Vadis did not get a completely new manifold but one where many components had been on and off an engine several times even if they had not seen much running. The original Vee bands did not fit properly in the laboratory and had to be replaced; whether all original Vee were scrapped, as instructed, or whether some found their way on the Quo Vadis manifolds cannot be asserted without reservation.

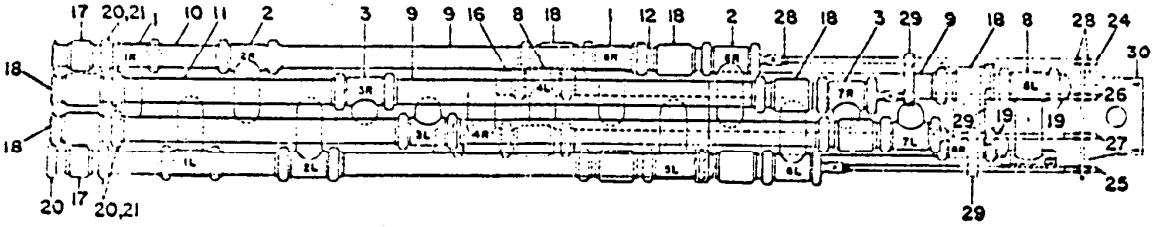
This type manifold is shown as sketch 28-6. Initially there was some complaint about manifold vibrations on the river boat and clamps, item 29 on the sketch, were added and reportedly eliminated all vibrations. Nothing was heard about mechanical performance of this manifold until the recent overhaul (early 1972) after some three years of operation when dissatisfaction due to frequent leakages of the joint was expressed to inquiring Alco personnel.

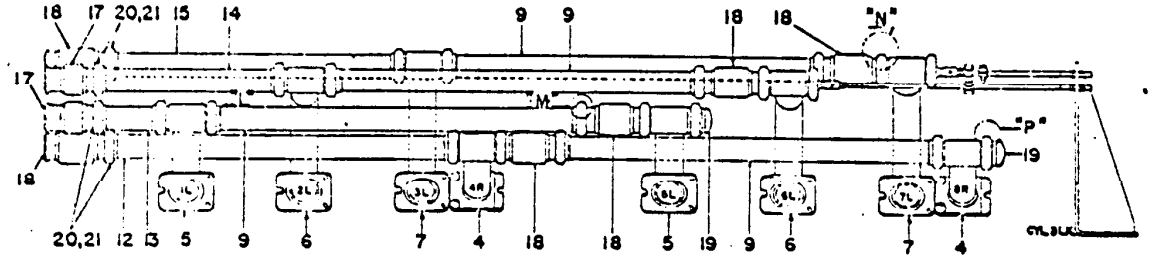
The Quo Vadis manifold apparently showed no leaks at initial installation but bad leaks developed shortly afterwards; Alco had to pay for repainting the engine room which had become sooty all over. New Vee band sophisticated heat resistant material were supplied; they alleviated the leakage problem, but do not seem to have solved it completely. This manifold, in contrast with the river boat manifold, is completely enclosed by Alco in a metal heat shield which was replaced by the customer by wrapped insulation. It is likely that performance of Vee band flanges and gaskets is adversely affected by operating at higher temperatures on the insulated manifolds.

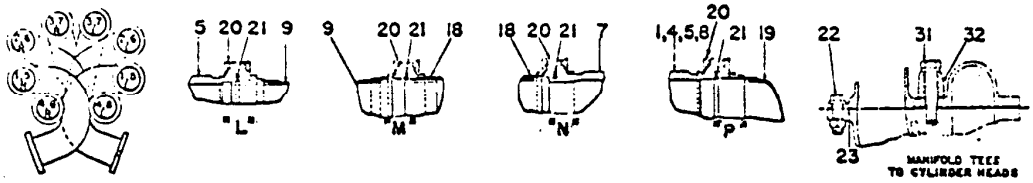
The history of these two manifolds has been told in detail because it is possible that their unsatisfactory behavior is due to their peculiar background and to being prototypes.

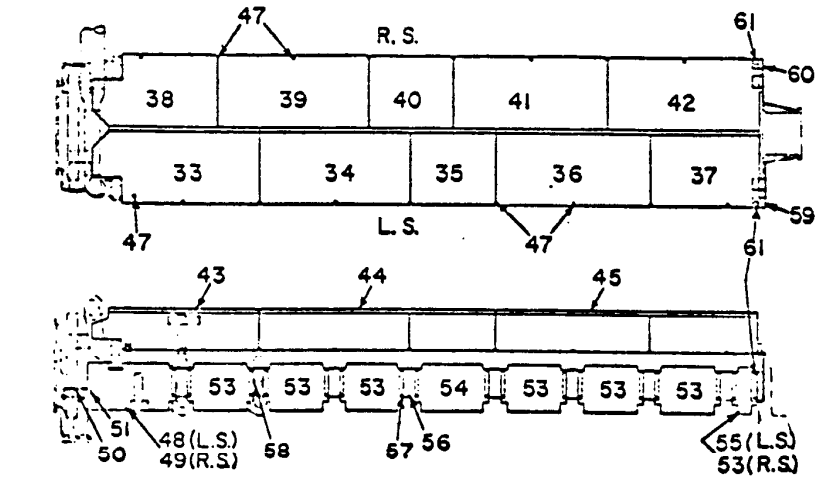
Six duplicate manifolds have been applied since, but none has seen any extended engine operation to date.

EXHAUST MANIFOLD AND SHROUD









SKETCH 28 - 6

EXHAUST MANIFOLD MATERIALS

For earlier conditions see ref. 28-1. One item not usually recognized is repeated here. Contrary to general published data intergrannular corrosion can occur near welds in stabilized materials exposed to temperatures in the 1100 to 1300° range if the atmosphere is slightly carburizing, as diesel exhaust gases are likely to be. Manifold heat affected zone failures could be duplicated in the metallurgical laboratory, when atmosphere was carburizing but not in normal atmosphere, the material then in use was 347.

The cast iron of the original 251 manifolds Aclo spec 31-P-5420 contained about .5 chrome, 1 to 1.5 nickel, .4 Mol; this is also the material always satisfactory for the elbows next to the cylinder head.

This was superseded for tees, pupes, and upper elbows by Niresist #3 Alco 31-P-5421 and later on some parts by ductile Niresist #3 Alco 31-P-5423. These types were chosen for their presumed low coefficient of expansion, emphasized additionally by narrowing nickel content tolerance to the high range. All this was done on advice of International Nickel, who never did point out that the expensively bought low coefficient of expansion became nearly as big as that of the less expensive less nickel containing D2, once operating temperatures of over 1100° were reached.

To date only the 12 cylinder low manifold castings are made of ductile Niresist D3, this is the way the patterns were made originally and since there were no failures no material change was made.

Beginning in 1963 a foundry, Dodge in Philadelphia, was found who could and did make very satisfactory castings, of reasonable wall thickness of stainless steel. The first castings were turbo gas inlet castings made of CF20.

This was later changed to CH20 presumably for slightly improved resistance to cracking and improved weldability for salvage welding. Since that time all exhaust manifold and similar purpose castings were made of these steels. Once Dodge proved that such castings could be made Abex retracted their earlier statement that parts could not be made in stainless steel and learned to make them also.

All fabricated manifolds since the early Sixties were made of 321 stainless steel, including most bellows.

Efforts were made to make the 12 cylinder short bellows of Inconel 600 to overcome failures, but availability is questionable and substitutions of 321 had to be allowed several times.

Flanges of the fabricated manifolds bolted against the cylinder heads are made of low carbon steel plate, they are fully satisfactory.

Flanges for the 12 & 8 cylinder bellows from American Boa are made of forgings of 5 chrome, .5 moly steel AISI 501. They should be inexpensive and perform satisfactorily, but this steel does not seem to be readily available.

Bolting of uncooled manifold parts have been made of Incoloy 901 since the later fifties. It overcame the persistent complaint of joints loosening and leaking. Cost and availability are serious objections, see sketch 28-7.

In many cases hot flanges are clamped by 4 bolts and two pairs of double nuts; the bolts can be locked either by wire or bent plates. The double nuts are low carbon steel forgings copper plated. They are made thick enough so that the bolts

PROPERTIES OF BOLTING MATERIALS

AT 1000°F

AT 1200°F

11-13-57

do not protrude, with their ends below the nut surface. With the Incoloy 901 bolts this practice may have lost its original purpose, but it is still being continued. Under high temperature condition bolt ends protruding from the nut tend to oxidise and will seize when unscrewing. The copper plated mild steel nuts serve their purpose well under all temperature conditions experienced. When single nuts must be used silver plated commercial high temperature lock nuts are used.

ENGINE PERFORMANCE WITH VARIOUS EXHAUST MANIFOLDS

With the original 4 duct exhaust manifolds both on the 12 and 16 cylinder engine acceleration smoke and load acceptance were an area on which hardly any question arose.

It was typical, if unfortunate, that the manifold on the 12 cylinder engine was changed from the high shrouded slip joint manifold to the low bellows manifold without conducting any recorded performance test. Reports from production testing frequently done at MLW and very infrequently at the time in Auburn do show that some unexpected differences in performance did take place which are very difficult to evaluate since changes in turbocharger models occurred more or less at the same time.

Ref. 28-13 contains a review of 1964 of manifold configurations and tests which are the basis for choosing the only 4 duct configuration which can be used satisfactorily. A manifold to figure 2 was tried on the market by a "pirate" but the Alco test results discouraged customers.

The difference between highest and lowest cylinder exhaust temperature was for a long time the prime concern on the 16 cylinder engine.

Much, if completely futile effort was expended to even up these temperatures, among them were divided air manifold between right and left bank, then between cylinders on the first and second set of four cranks; timing changes on some cylinders extensions on some of the manifold pipes to change possible wave patterns. The most effective for reducing cylinder exhaust temperature was the introduction of an orifice reducing the area between air manifold and elbow. It was not pursued.

When the dual fuel engines forced improvement to avoid detonations in some cylinders, the first eight duct manifold was designed; for drawing and performance data including indicator diagrams see ref. 28-3. Compared to the four ducter the eight ducter reduced the difference between maximum and minimum cylinder temperatures from 296° to 113°F at 193 MEP but average turbo inlet temperature rose from 1167 to 1201°F; fuel consumption decreased slightly. As stated before this manifold was mechanically deficient for locomotives.

MEL 3370 Dec. 61 shows the first comparative run between 4 duct and single duct. The "experimental" firing order is the one adopted since that time. At full load 193 MEP the single ducter reduced fuel consumption from .368 to .358; pressure difference between compressor discharge and turbo inlet from 15.8 to 11.4"HG; gain in turbo inlet temp 14°F. It also shows that acceleration without some control device is unacceptable, but can be made acceptable with a control device. Tests on a locomotive an the Santa Fe showed more improvement in compressor discharge pressure and temperatures. This test showed no direct comparison of lower notch performance; at that time with 4 duct manifolds, locomotive demands did not seem to need particular attention to part loads.

MEL 3495, Sept. 65 gives a comparison of performance between the original square entry and the prototype streamline manifold. The steady state conditions show some improvement, turbo inlet temperature at part load drops about 70°, air flow increases 5% more at part loads. Duration of heavy acceleration smoke (over #5) drops from 18 to 12 seconds.

Extensive development work on manifolds was conducted in 1965 and 66 results of which cannot be summarized in this write up. Ref. 28-13 contains analyses of some of these test results. MEL 3501 compares performance of 8 duct, 4 duct, streamlined and original single duct at 222 MEP 1100 RPM.

About this time the first "Power Paks" were tested in the shop. Load acceptance capability with the single duct manifold did not meet promises of sales publications, whether or not the installations actually needed this capability was not known.

Since the units then on order for Latin American were supposed to operate only as standby unit it was decided to equip them with the, by then, absolete 4 duct manifolds. Load acceptance was as desired. Predictably the units operated more than just as emergency standbys and just as predictably the manifolds failed and had to be replaced by single ducters. Particular load acceptance capability was not needed. A similar instance occurred in a Virgin Island installation.

Since early 1969 one river push boat was in operation with one engine equipped with streamlined single ducter the other with an eight ducter. Actual reports as well as a film taken during acceleration shows both engines smoking badly, duration of heavy smoke on 8 duct 21 seconds on single duct 34 seconds. A recent visit in 1972 indicated improper nozzle ring on the turbo and smoke apparently could be cleared up with 8 ducter, with additional simultaneous change in injection nozzle size.

Later tests at higher ratings 250 MEP range were made with 8 ducter and streamlined manifolds, they are recorded in MEL 3524 and 3524A, which also contains manifold pressure diagrams.

Additional development work was done in connection with AEC quick start and load acceptance requirements. Results have not yet been recorded in a readily accessible manner.

A subjective summary of manifold characteristics is:

1. 4 duct manifold, optimum load acceptance and acceleration smoke. Intractable difference in cylinder exhaust temperatures. Mechanically not impossible (consider Hanslon Wilson approach).

2. 8 duct manifold, load acceptance good, acceleration smoke needs control. Mechanically difficult and therefore of questionable reliability.

3. Streamlined single duct load acceptance and acceleration smoke bad; can only be applied if no fast acceleration and load acceptance is required. If this is the case manifold is most reliable and cheapest unless fabricated in Canada.

EXHAUST MANIFOLD SHROUDS, SHIELDS, BLANKETS

Like all parts of the 251 engine, exhaust manifolds and coverings were designed for locomotive application. In the mid-fifties locomotives were still built in two basic versions; the stream-liner and the switcher. In the stream-liner personnel could be in the engine compartment with an operating engine; in the same compartment was also much electrical equipment. This condition postulated reasonable heat insulation of the manifold and minimum leakage of exhaust gases and soot; soot deposit on electrical equipment originally was a serious source of electrical failures.

In the switcher locomotives and the later road switchers, the engine is under its own hood with louvered doors; the electrical equipment is also separated more or less effectively from the engine area by bulk heads. These conditions minimize the importance of thermal insulation. Adoption of bellows joints had overcome earlier leakage problems.

On the 244 engine blanket insulations had been used which were very unsatisfactory, especially since they had to be removed frequently for manifold repair. The shrouded manifolds developed for the 244 and adopted for the 251, see sketches 28-1 and 2, served their purpose quite well.

Sketch 28-8, figure 1 shows the typical shroud construction; a mild steel outer skin 14 gage, the minimum thickness for ready welding; welded to it are stainless steel cotter pins, an inner skin of the least expensive 410 stainless steel, non-weldable, held by bending the cotter pin over a washer. Between the two sheet metal is glass wool, somewhat compressed. Weld studs were tried but found unsatisfactory, details not recalled.

Figures 2, 3, and 4 show the evolution of a water blanket. At first, Figure 2, there was a tubular inner member to which spacers were welded and an outer member was wrapped around the spacers. The outer member was damaged in short time by a failure mechanism similar to that of a magnetostrictive device. Repair was made by drilling holes and plug welding, always a source of ugly warping. The next step, figure 3, was putting spac bars thru both skin sheets and fillet welding sheets to the spacer bars; leaks and cracks developed and reinforcement patches had to be applied with intermittent welding to both skin sheets.

Figure 4 shows a section of a water blanket which performed well and could serve as model if water blankets were again needed. It consists of an inner tubular member to which an outer member of corrugated sheet is applied by spot welding. The corrugations serve as passages for cooling water. Header sections fitting the corrugations are fillet welded at the ends.

Figure 5 is a typical section of the shields designed for the LST engines. The usual difficulty of welding a spacer between two skins is overcomed by spot welding with minimum distortion. First the Vee shaped reinforcement or spacer is spot welded conventionally to the outer skin, both parts are of equal wall thickness; then the thin inner skin is spot welded to the spacer. One of the electrodes is conventional, the other a plate. This was developed in the Schenectady locomotive Sheet Metal shops and resulted in very neat appearance. The space between the sheets contains glass wool batts.

In reviewing stainless steel shrouds, the 13 ga wall thickness is surprising. This was chosen to use up a large amount of surplus stock from the time when stainless steel locomotive cabs were made. It is possible that this overly thick sheet was specified even later on simply by copying earlier drawings inadvertedly.

Whenever a shield was designed in a way that it would touch, intentionally or not, a part of the manifold, one or both of the pieces were destructively. A determined effort was therefore made to support any of the later metal shields from some other part of the engine structure, rather than from the manifold.

SUBJECT _____ GROUP _____

TYPE _____ DRG. NO. _____ SHEET ____ OF ____ SHTS.

CAL. BY _____ APP. _____ DATE _____

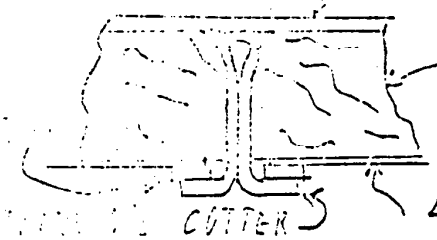


FIG 1
FULL SIZE

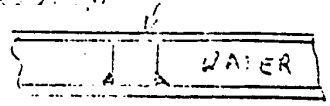


FIG 2

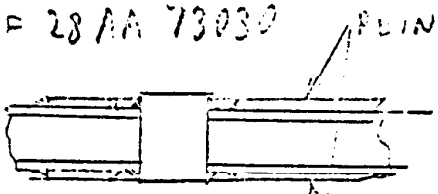


FIG 3

13 GA
HALF SCALE

REF 28 EE 73235

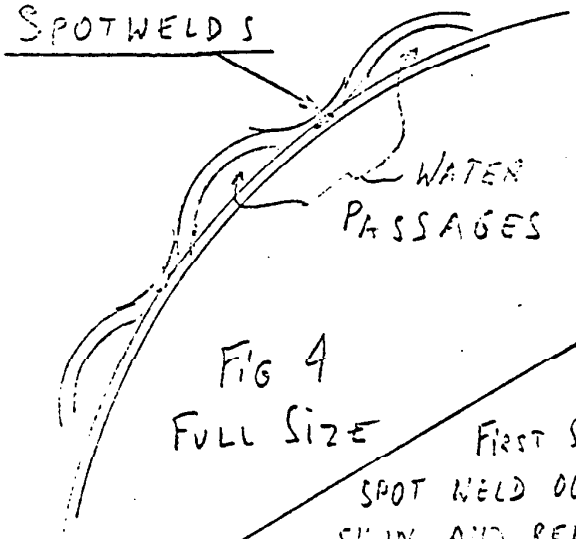
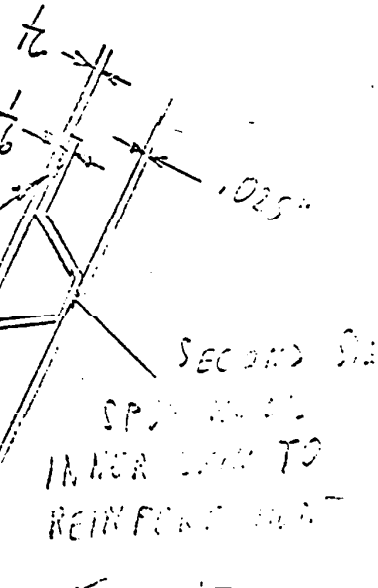


FIG 4
FULL SIZE

REF 28A 71661



FIRST STEP
SPOT WELD OUTER
SKIN AND REINFORCEMENT
TOGETHER

FIG 5
FULL SIZE

SKETCH 28 - 8

It was mentioned that current locomotive practice, road switchers, either do not need shields or can be satisfied with simple arrangement. On the 12-cylinder no shield is provided for locomotive application but a design is available. For the 16-cylinder a simple sheet of aluminized mild steel fastened to the water headers has been used since adoption of the single duct manifolds and, surprisingly, complaints or comments have not reached engineering.

As long as no effort is made to enclose the manifold completely, the shields are relatively easily made; if specifications are strict so that all manifold leg portions must be covered, the shield becomes awkward and expensive. No effective simple design is in existence.

Blankets have been frowned on since the time of the early 244 engines. If leaks do occur blankets are liable to get oil soaked and must be considered a fire hazard. There is also the apprehension that blankets will increase metal temperatures to the point where manifold life will be shortend.

The blankets recently applied to Marine installations should be watched before adopting this insulation across the board.

AFTERCOOLER GENERAL

All but the first few 6 cylinder engines, probably the ten which went to Cuba, have been equipped with aftercoolers. When the 251 design was started a little experience with aftercoolers had been gained on the latest 244 engines which was used as basis. Most of the aftercoolers used are air to water coolers, except for the 6 D production engine which uses an air to air cooler, and one experimental 12 cylinder engine which had been in several years' field operation. In the following the two types will be described in separate chapters.

For most aftercooler calculations Alco has relied on the manufacturers although for some an outside consultant was used occasionally for guidance. Few clear performance specifications for the coolers seem to exist. A few are stated on the drawings, for most they must be reconstructed from calculated results submitted by the vendors and contained in engineering file folders (Aftercooler). Many engine test reports also contain fairly complete data.

AIR TO WATER COOLERS

The earliest trials were made with a side mounted flat tube cooler on a 244 engine; this cooler failed mechanically in short order. This led to strong emphasis on mechanical reliability of all coolers and particularly to avoidance of anything but round tubing, although possible performance gains with flat tubes were always tempting (G & O comparison of effectiveness of 16C cooler claims possible improvement from 73% to 82% by accepting flat tubes instead of round tubes). The first coolers per print 2S-A-71149 used in production and still specified for 6 B engines are made by Rome-Turney, of 1/2" OD copper tubing with [illegible] soldered on continuous spiral fin. The fin OD is 1 1/4" with 12 fins per inch spacing. They are two pass coolers flowing the full 300 GPM jacket water of the 6 cylinder engine through the cooler. The cooler is not of an efficient shape but utilizes otherwise completely wasted space in the air manifold. Some guide vanes in the manifold attempt to counteract the uneven cooling of the air to the different cylinder but typical air temperatures to the cylinders at 207 MEP do vary from a minimum of 174°F to a maximum of 215°F, this with an air temperature before the cooler of 292°F.

At the beginning, it was believed that the short "U" shaped tubes of the cooler would be held firmly enough by being rolled in the tube sheet, note the internal reinforcing ferrules, but even there, failures forced the adoption of a substantial sheet clamping the bends of all the tubes together to keep them from vibrating.

The absence of failure reports on this cooler and generally good performance of the 6 cylinder engine confirms the choice of a rugged if inefficient cooler for this comparatively low rating. No alternate source for this cooler was ever developed.

The coolers for the Vee engines were developed together with Modine again with great emphasis on ruggedness. The tubes are placed vertical to avoid any bending at the joints in the tube sheet; the lower header can freely expand relative to the housing but a spud sliding in a gland resists horizontal forces when the locomotive is bumping cars or the LST engine is getting bombed. This design requires removal of the turbocharger before the cooler can be pulled out for cleaning, a recognized disadvantage which was not too objectionable in locomotive service, because a small hatch over the turbo charger could be removed relatively easily and make the area quite accessible. Some of the railroads insisted in having a large opening in surrounding casing with a flanged cover to inspect aftercoolers for dirt; the standard design has only a fairly large pipe plug for this purpose. Actually dirt becomes only a serious question if oil bath filters are used and oil carry over from the filters exists.

This type aftercooler is being built to print 2S-A-71314 in two different lengths. The 16 cylinder core is 6 3/8" longer than the 12 cylinder core. One of the more frequent uses of the 12 cylinder engine was in the low switcher locomotive hoods while the 16 cylinder was then used only in the higher profile road locomotive cabs ("covered wagon", "streamliner").

This difference in turbocharger height and space available for the aftercooler was carried through until very recently when the 12F was equipped with the higher turbo support and an exhaust manifold designed to suit.

The early coolers by Modine were somewhat overdesigned, e.g., the holes in the tube sheets had two circumferential grooves cut in to anchor the rolled-in tube more securely. Later G & O competed and from then on furnished most coolers of this type at lower cost. The coolers were made of 3/8" OD X .049 thick copper tubes and copper fins, 9 fins per inch. For manufacturing reasons the fins are made in two pairs of two types of sheets, one containing two of the three rows of tubes. The manufacturing process was to use tinned tubes, assemble the tinned fins on the tubes, then bake the core in an oven which presumably would solder the fins to the tubes. The fins had holes stamped in them in a way that the edges were not clean but had triangular fins bent up to make more contact for soldering between tube and fins and to act as spacers.

One frequent defect in the early sixties was broken off parts of fins, about a couple of inches long and of a width from the edge of the fin to about the center of the first tube row. It seemed to occur only on the G & O built coolers and was caused partly by inadequate solder joints and partly by the unsupported edge of the fins vibrating on the air inlet side. Fairly large patches of the fins were thus broken off. Normally they fell harmlessly to the bottom and caused no damage to the engine. The cooler could be turned around and run for another year or more. The breaking of fins was found in less than a year's service if one did look. It was overcome about 1965 by heavier embossing thereby stiffening the fins. For the high shock LST coolers the two end faces, either might become air-in side, were solder-dipped to insure a better joint.

Later in 1969 as a cost reduction cooler construction was changed to copper tubes expanded into aluminum fins, 16 fins per inch. The change was made after first making a performance comparison on the engine and then letting ten such coolers operate in the field for a year. These coolers are built for about 80 to 100 GPM two pass waterflow. The bottom header has a drain connection which is supposed to be connected into the water piping. Several cooler headers cracked by freezing on stored engines, when connecting the drain had once been forgotten. The upper header must be vented not only to to make sure the cooler fills, but also to avoid steam pockets.

As ratings were increased above 220 MEP a new cooler design was developed jointly with Young; their prices were more attractive than G & O's, also the rash of broken fins had soured G & O's reputation. Their cooler was, however, approved later on. The cooler has over 20% greater frontal area than the former 16 cylinder cooler; it is also substantially deeper. The increase was achieved by giving up the stipulation of vertical tubes and acceptance of a horizontal tube with tube sheets and headers vertical; the back header again has the spud extension still allowing the tubes to expand freely, but it now also serves to support a substantial part of the core weight. The tubes are now made of 3/8 OD, .0375 thick, expanded into aluminum fins, 16 fins per inch; expanding of the tube limits permissible wall thickness to not much more than this number. This core is shown on 28-A-7153S. .035

This print shows the cooler for the 16 cylinder engine, while 28-A-73519 shows the lower height cooler for the 12 cylinder engine of this type. The size established to meet the limitations of the 12 is also used on the 8 cylinder engine and the 6 F engine. These coolers are built for a 4 pass water flow limited to 130 and 90 GPM respectively to avoid erosion damage at the tube entries. 71538 73519

These cores had a good performance record on 16 E engines. When used on 16 F engines they developed in 1971 frequent failures of the lower front tubes at the main tube sheet joint. These failures seem concentrated in Canada and Australia. A believable explanation has not yet been found; tube material is being uprated to 90-10% copper nickel; but unclear reports of strange air ducting in Australia resulting in unbelievably high air temperatures may well be the primary cause of the failures.

A still larger cooler of generally the same design discussed in the preceding paragraph was designed for the 18 cylinder engine. It is shown on 28-A-71651 as based on 3/8 OD .035 thick copper tubes, 16 plate fins per inch. Although originally specified as aluminum fins, they were delivered as copper fins and the drawing changed accordingly. It has about 48% greater frontal area than the one for the 16 cylinder, it has a two pass water flow. One failure has been reported to date (summer, 1972). This failure occurred on the 640 locomotive on the CPR (first 18-cylinder loco).

WATER PIPING ARRANGEMENTS

The water for the coolers on locomotive engines was taken originally by putting the cooler in parallel with the supply to the right cylinder bank; in this supply line an orifice was placed to divert about 100 GPM through the aftercooler. This caused the temperature in the engine outlet header from the right bank to be about 5°F higher than that of the left bank, which difference was accepted. This piping had the advantage of being completely part of the engine. For stationary and marine applications super cooled water was usually diverted in and out of the aftercooler. When the side mounted cooler and more cooling of air was wanted, super cooled or separately cooled water was piped into the aftercoolers. This is also the current practice. On some engines, notably the 8 and the 18, the coolers are placed in parallel with the whole engine which, due to the flat pump characteristic, increases the water flow through the whole system.

AFTERCOOLER CASING AND TURBO SUPPORT

The air manifold in the Vee of the 251 cylinder block with exhaust manifold above it and a single turbocharger mounted at the free end made for a natural design of a casing combining the functions of enclosing vibration damper and gearing, forming housing and mounting for aftercooler cores and their headers and for supporting the turbochargers. It also incorporates eyes for lifting the engine front end. Except for experimental installations of additional aftercoolers, no alternate designs for turbo support and aftercooler casings were seriously tried except when the installation of one smaller turbo at each end was considered.

The location of the turbo support (the word is meant to include the aftercooler) in this area implies certain limitations. The locomotive profile determines the upper limit of height, the extension shaft and its coupling the lower limit. The width is limited by the need to be able to pull the two camshafts out of the block without removing the turbo support. Somehow it must be possible to bring the cooling water to the ends of the water compartments in the block. A further limitation now somewhat neglected was the need to lift the brake air compressor vertically between the coupling members on either side of the compressor until the OD of the coupling flanges clear. This gave the strange, aerodynamical wrong shape of the front plate of the 16C turbo support. It is understood that now the compressors may be slid out sideways. Some way must be found to connect the levers at the fuel pump control shaft at two sides of the engine. Add to these requirements the need to use different turbochargers and different aftercoolers and the deplorably large number of different turbo supports going through the shop becomes understandable.

One feature is common to all supports. To fit cooler cores the casing must have large flat walls. Considering that current boost pressures, hover around 30 PSI. The thickness of such plates must be substantial or ribbing must be added judiciously. The turbochargers themselves may occasionally go more or less out of balance which can impose severe loads on the support. For this reason above, it is essential that all piping for water and oil be made of hose.

The turbo supports for the side mounted aftercoolers had an insufficiently rigid mounting for the turbo on the top plate. Field repairs, the first was not good enough, had to be made to overcome frequent failures. The production parts have been redesigned and are now presumed to be satisfactory.

On the top mounted cooler the core does not fit tightly into its casing, clearances are those maintainable with a weldment. This does cause some bypass of air. A test with sealing strips was once made and air temperature after the cooler was decreased slightly at the cost of a small increase in pressure drop, which caused the temperature into the turbine to go up a shade. Accordingly sealing strips were not introduced, in later designs they were provided. Looking at the core development from top mounted to the side mounted and again to side mounted for the 18 cylinder, one can see how the increases in size gradually required some forced designs to stay within

the limitations enumerated. The clearance to the extension shaft gets less, an insert must be welded into the front plate for coupling clearance. On the 18 the back wall of the aftercooler compartment has become a frame, whose surface just clears the camshaft during removal, to support a cover which just makes a little longer core possible.

When designing around turbo chargers and aftercoolers radiated heat from some areas of the turbo and heat of the air discharged by the compressor must be kept in mind. This refers to the hoses required to avoid vibration failures of steel tubing. Air temperatures in excess of 300°F are beyond the capacity of general materials for gaskets or diaphragms. Note that both the manifold pressure relief valve (Fisher valve) and the acceleration control device (with a Bellofram) must be installed on the down stream side to prevent deterioration due to heat. (The 18 cylinder turbo support originally had the valve on the "hot" side, it may or may not have been moved.) The Fisher pilot valve has temperature resistant diaphragms, all of them doubled up for redundancy and with a thin teflon sheet sandwiched between them to protect against having the main valve stay open due to heat cracked diaphragms.

While the preceeding discussion dealt with the Vee engines, many of the remarks apply also to the recently designed 6 F engine in which the original air to air aftercooler of the 6 D engine has been replaced in approximately the same location by an air to water cooler. The camshaft can be withdrawn through an opening in the turbo support..

AIR TO AIR AFTERCOOLER

In 1960 the thought came up to make use of the possible decrease in charging air temperature available if air is directly cooled by air rather than by the detour of air cooling the water and the water cooling the charging air. An additional justification was that these engines were destined primarily for overseas into desert like, water poor areas, India, Chile.

A core to drawing 28-B-71415 was developed by Trane, later many cores were purchased from Modine and Marsten in England. It is made completely of aluminum, with corrugated thin aluminum sheets being sandwiches between flat plates. The corrugations of adjacent sheets running alternatively horizontally and vertically. The whole assembly is brazed together. On the engine the boost air passes the vertical channels of the corrugations, the cooling air the horizontal channels.

The cores look flimsy and caused many complaints in the beginning; the ends were crushed and Devcon was used in large quantities to fill up damaged passages. Actually there was and is nothing much wrong with the cores but the original design of mounting the core in the steel casing was completely unsuitable. The mistakes were neglect to allow for differential expansion of aluminum core and steel casing, use of thick felt, disregarding the limited ability of felt to withstand elevated temperatures. "Fixes" were made in the field and more heat resistant armored gaskets were installed. The housing was redesigned and was made more flexible and the core is now not exposed to crushing by overtightening of bolts. The assembly seems to be living but should not be taken as a model for a new design without careful analysis.

The cooling air is supplied by an engine mounted Westinghouse fan. It is driven from the extension shaft gear through a quill shaft, having all the costly attributes of such a design. A trial with a solid shaft broke the fan impeller dramatically. The fan requires 40 HP at full engine speed. It was a controversial subject if the engine output as measured at the generator +40 HP could be regarded as Gross Horse Power. This is the way the GHP was interpreted for the Indian contract. As justification it was cited that the SAE defined basic engine does not include the fan. When comparing fuel consumption data this uncertainty in reporting should be watched.

The turbo support was originally designed as a weldment to permit installation of a turbo gear drive. To reduce cost it became a large complicated iron casting still including the pads for the never used gear drive.

In the early sixties a 12 D cylinder engine was built utilizing air to air aftercoolers, it operated quite satisfactorily for a few years in heavy field service. The space requirements for ducting the large amount of cooling air for the fans caused the 12 cylinder engine to become practically as long as a 16 cylinder engine. All efforts to proceed with further development of air to air coolers were dropped.

REFERENCE - SECTION EXHAUST MANIFOLD

28-1 Exhaust manifold history for the 244 engines
October 1953 see Miscellaneous Reports.

28-2 Typical Shrouded Cast Manifolds

16-cyl. 28-AA-72965
12-Cyl. 28-AA-72764
6-Cyl. 28-AA-73305

Description 16-cylinder MI-1002A
12-Cylinder MI-11077
6-Cylinder MI-11053

U.S. Patent #2660031 of 1953

28-3 8-Duct Cast Manifold - Assembly drawing 28-AA-73168
- Detail of bypass opening 28-A-73141
Performance data including Cox Indicator diagram -
MEL-3057, 3057-A, 3057-B.

28-4 12-F Manifold 28-AA-73853

28-5 6-D Manifold 28-EE-73760

28-6 18-Cylinder Manifold 28-EE-73700 Also support 28-EE-73860.

28-7 8-Cylinder Manifold 28-A-73634

28-8 Single Duct Production Manifold 28-EE-73457

28-9 Tests with reduced and increased manifold sizes.
Project 863461. larger manifold per 19-LYA-1363

28-10 Dual Fuel Manifold Alco code 72-08120-534
Alco Cat. No. 22816071, Flexonics drawing D-03408, 03407.

28-11 Ontario Research Foundation Report ENG. R-71-59 of May 20, 1971.
Also detailed description of failure in Correspondence File
Streamlined Exhaust Manifold 1966 - 67.

28-12 8-Duct Exhaust Manifold - Flexonics print J-02173
Layouts 19-LYA- 1365 and 1376.
Photograph in Correspondence Folder 8-Duct Manifold 1965.

28-13 Correspondence File Folder Exhaust Manifold test survey.

ENGINE MODELS, DISTINCTIVE FEATURES

The following will outline the principal differences between the various 251 engines with Model designations "A" through "F" Basically, the letters represent improvements in the engines accompanied by higher ratings.

New Models of engines with different numbers of cylinders were not necessarily introduced at the same time. The discussion will therefore, describe the changes for engines of the various cylinder numbers independently from each other.

This outline will be based on locomotive engines because there the ratings and different executions are more clearly defined. On engines for other purposes, the differences between the various models are less distinct because special requirements for speeds, loads, types of operation tend to force adoption of certain features of one model onto a basically different model.

As the years go by, many improved components, developed to withstand the greater stresses imposed by the higher ratings of the newer models, are also offered as replacements and improvements for older models.

The following will describe the models as they left the factory originally and will disregard available items of upgrading.

6 CYLINDER ENGINE

6-251-A, 850 GHP, 1000 RPM, 168 MEP
This is the first 251 production engine, not more than about one dozen were built about 1953. Its main difference from all later 251 is that it does not have an aftercooler. Interchangeability with parts of later engines is limited. Cylinder block has camshaft bearings in different longitudinal location than later engines, shape and location of water connection to head is different, airmanifold has no flange for aftercooler. Injection equipment is an early version, port closing is at a different cam height. Most original "A" engines were either reworkedor scrapped.

6-251-B double rating 975/1050 GHP, 1000/1025 RPM, 192/202 MEP
There is no difference in the engine between the two ratings, racksand speeds are set differently. Turbocharger is 320, crankshaft is located with half thrust washers at center bearing #4, governor was Woodward without head control, injection pumps have 15 mm plungers, overspeed trip turn fuel pump control shaft to no fuel. In these respects it is the same as the 6A model. But it does have an aftercooler, and the governor was soon changed to "with" load control.

6-251-D 1365 GHP 1100 RPM 245 MEP Production started in 1961.
The 6D is equipped with an air to air aftercooler. The cooling air being furnished to the aftercooler by a blower mounted on the engine and driven from the front crankshaft gear train. Valve timing has been modified from the original B rating. Fuel pump diameter has been increased from the original 15 millimeter to 17 millimeter. This, and the increase in speed, required modifications in the fuel pump drive mechanism to absorb the higher hydraulic and dynamic loads. Crankshaft

thrust is absorbed by a flanged thrust bearing at the generator end. The overspeed stop functions by cutting off the air supply after the aftercooler. This engine is equipped with a 350 turbocharger and a governor influenced also by air manifold and barometric pressure. The 6D as also the 6A, 6B have 12.5:1 compression ratio pistons and 123 3/4 degree overlap camshafts. The cylinder block is not interchangeable with that of the B engine.

One feature common to the 6, A, B & D engines is that lub oil cooler, filters and strainers are engine mounted, so that the engines are self-contained as far as lub oil is concerned. This feature is not duplicated on any other Alco engine.

6-251-F 1500 GHP 1100 RPM 269 MEP
This engine has an air to water aftercooler same core as the 8 cylinder, 131 turbocharger, 140° overlap camshaft, 11.5:1 compression ratio piston. overspeed stop again operates on the fuel pump control shaft. While it has been laboratory tested it has not yet gotten into the field.

No 6C or E engines were developed.

8 CYLINDER ENGINE

8-251-E 1700 GHP 1000 RPM 252 MEP
Introduced in 1966 as a locomotive engine. Side mounted aftercooler, 131 turbo, 12.5:1 compression ratio piston; 123 3/4 degree overlap camshaft; the distinguishing feature is that the inherent unbalance of a 45° 8 cylinder Vee engine is partically counteracted by two balance shafts located in the base below the crankshaft. The balance shafts are not equal in unbalance nor are they located at equal distance, from the engine center line. They are driven at double engine speed. The first series drove the balance shafts thru an idler from the split crankshaft gear, the later ones from the extension shaft gear. Gear drive at front end is not interchangeable with that of other engines. Due to unbalance and vibrations speed is limited to 1000 RPM. No model change has been made to reflect change in balance shaft drive; nor the recent changes for locomotive engines for MLW which introduced a new base with longer balance shaft bearings and 140° over lap camshaft. Nitrided crankshafts are mandatory.

12 CYLINDER ENGINE

12-251-B 1950 GHP, 1000 RPM, 193 MEP
While a few type A engines were built as prototypes all production engines started in the early fifties as B engines. Vertically mounted air to water aftercooler, type 500 turbocharger, 12.5:1 compression ratio piston, 123 3/4 overlap camshaft. Crankshaft thrust is at center main bearing, governor mostly GE; overspeed trip acts on fuel pump control shaft, retained in all but the 12D; injection pump plunger i.e. 15 mm.

12-251-C 2150 GHP 1025 RPM 207 MEP
Only difference is change to 600 turbo which was later replaced by an up graded 500 style turbo. While not strictly a model change, change from GE to Woodward governor more or less coincides, same is true of change in thrust bearing location from #4 to #7.

12-251-D 2600 GHP 1100 RPM 235 MEP
Air to air aftercooler one prototype operated for several years in field; did not reach production.

12-251-E 2600 GHP 1100 RPM 235 MEP
Not built by Alco as a locomotive engine; small air to water side mounted aftercooler 131 turbo. 8 counterweight nitrided crankshaft is mandatory all preceding production models have 4 weight with nitriding being optional, 11.5:1 compression ratio pistons, 17 mm injection pumps, 140° overlap chamshafts.

12-251-F 3000 GHP 1100 RPM 269 MEP
Not built to date by Alco as a locomotive engine, Large side mounted aftercooler, 165 turbo larger diameter exhaust manifold, 11.5:1 compression ratio pistons, 8 counterweight, nitrided crankshaft, 140° overlap camshaft. Air valve seat angle has become 30°, on all others it was 45°.

16 CYLINDER ENGINE

16-251-B 2600 GHP 1000 RPM 193 MEP
Production started in mid fifties. Vertically mounted air to water aftercooler (longer than on the 12), 700 series turbo, originally 4 duct later single duct exhaust manifold, 123 3/4° overlap camshaft, 12.5:1 compression ratio pistons, 15 mm injection pumps. 4 counterweight crankshaft, thrust at #9 bearing except on first series. Governor predominantly G.E. A change in firing order was made within this model.

16-251-C 3050 GHP 1050 RPM 216 MEP
Introduced in field 1965; same as 16-B except turbocharger starts as 800 but is soon replaced by up graded 720 series turbo, crankshaft becomes 8 counterweight to permit speed increase to 1050 ; injection pumps are increased to 16 mm.

No D model developed.

16-251-E 3300 GHP 1100 RPM 223 MEP
Introduced 1966 or 67. Large side mounted air to water aftercooler 11.5:1 compression ratio pistons to limit firing pressure. "Streamlined" single duct exhaust manifold, 16 mm injection pump, Starts with overspeed trip cutting off air later changed back to turningcontrol shaft to off." Original series goes out with 123 3/4° overlap camshaft and 720 turbos, becomes 140° overlap and 165 turbo after these parts become available. Crankshafts for this and F engine must be nitrided.

16-251-F 3900 GHP 1100 RPM 263 MEP
Generally the same as the E engine but starts with 165 turbo, and 140° overlap camshafts, uses 17 mm injection pumps. Has larger oil capacity base, cylinder heads have hardened guides and 30° air valve seats; governor has a fuel limiter incorporated; went into production about 1968.

18 CYLINDER ENGINE

18-251-F 4500 GHP 1100 RPM 269 MEP
First locomotive engine delivered in 1970. Large newly developed air to

water aftercooler, 165 turbo but with modifications to suit larger engine. Nitrided crankshaft with 18 counterweight, 11.5:1 compression ratio pistons, 140° overlap camshafts, 30° air valve seat, hardened valve guides, 18 mm injection pumps, new larger capacity lub oil pump, front end gear train speeded up by 10% compared to 12 & 16 cylinder engines.

No earlier version were built.

It should be noted that great effort was always made to retain interchangeability between the various models, at least in the ability to replace older parts by newer ones. In these categories fall the power assemblies (head, liner, piston, connecting rods), main, connecting rod camshaft bearings; most gears, except for the 8 and 18-cylinders which required redesign. Fuel pump drive mechanisms of the later models were designed for greater load carrying capacities, newer style parts may be installed in older models, e.g. current style exhaust manifolds and crankshafts will fit older models.

In rebuilding older models, advantage may be taken to use upgraded parts for replacements.

Cylinder Block 251

General Considerations

The design of this block was made with full knowledge of the painful development and need for at least three major redesigns to overcome failures of the cylinder block for the preceding, model 244 engine. The design intent was to eliminate failures and reduce cost. This led to a tendency to over design for the ratings then current but provided the margin for the gradual increase in ratings since that time. Whatever stress and stiffness calculations were made were based on comparison with the earlier block rather than on absolute presumably acceptable stress values.

Reduced cost was achieved by accepting a wet block, and be designing for fixed, accurate center distances between crankshaft and camshaft bores; with elimination of idler gears, anticipating the procurement of a single purpose boring mill.

Making the block of cast iron or ductile iron was suggested several times as a cost reduction. The Cooper Besemer - GE competitive engine does use a cast block. Preliminary estimates indicated possible savings perhaps up to 20%.

Alco engineering recommended strongly against attempting such redesigns. It was one of the features of the welded blocks, first the 244 then the 251 that failures did not begin to occur until many months, up to two and three years after they went into service. Corrective actions became extremely expensive due to the large number of units in field operation when such failures became noticeable. There is no reason to expect that a new cylinder block made of cast iron would behave more predictably, especially considering the likelihood of undetected casting defects, lack of Alco's control of suitable foundries and the loads imposed on this structure which due to increased speeds and ratings are now more severe than ever.

An important advantage to users is the ability to restore welded cylinder blocks virtually regardless of the degree of damage, say from a broken connecting rod or from a collision. Such repair work, controlled and described later, forms a continuing source of income to Alco. The G.M. engine, for years Alco's principal competitor, is also based on fabrications rather than on castings.

Materials

The blocks are fabricated from plate and forgings. most recently an extrusion has replaced a part previously machined out of plate. Except for a very early unsuccessful attempt, Alco did not use steel castings for highly stressed components in fabrications. When India first started to build cylinder blocks, the question of substituting steel castings for the saddle forgings was examined; there would be a noticeable decrease in fatigue strength, and of course the casting integrity is inferior to that of forgings.

In the following listing properties of the main bearing cap forgings are included, although they are not part of the weldment. But strength investigations of

Dimensional Limitations and Assumptions

The basic dimensions were established as follows:

1. The base rail is 2-3/4 below the crankshaft, a dimension determined by the largest cutter suitable to produce the serrations.

2. The side plates are about as close to the liners as possible. Recent studies about increasing the cylinder bore, indicate a maximum increase from 9 to 9-7/16 bore is possible providing thinner liners, presumably made of steel, are used. The camshafts based on a 4-1/2 bearing diameter were brought as close to these side sheets as possible, considering the possibility to restore damaged bores by inserting a 1/4" thick bushing.

3. The camshaft and base rail location determined the size of the camshaft gear which in turn established the minimum height of the block. Note that the base rail of the block is locally reduced in thickness to give clearance from the camshaft gear.

4. Longitudinal dimensions were dictated by the crankshaft design which was considered of paramount importance not admitting of any compromise whatever. This resulted in increasing the cylinder center distance from 16-1/2" on the earlier 244 engine to 18-1/8 between intermediate cylinders of the 251 engine and 20-3/8 in the center, to accomodate the wide center bearing.

   Performance records of crankshaft and bearings of the 251 engine support this design decision although it made the engine long and consequently more costly.

serration in saddles and matching caps were made simultaneously. The caps, which are not subjected to stress relief, are of 1045 steel quenched and tempered to 216 BHN.

The steel in the block itself, are low carbon types, plate and bar stock, the saddles are 1021 material; all these parts are stress relieved in the completed weldment at 1150°F.

The resulting mechanical properties are: (ref CY 1)

Location	Bearing Cap	Saddle Forging	Plate Typical	Weld Typical
Steel	1045 q & t (Not part of weldment)	1021 Str. Reliev	ASTM A283-Gr. C Str. Reliev.	Lincoln L61 Oxweld 36 Str. Reliev.
Tensile PSI	99500	72500	55000 to 65000	58000 to 61000
Yield PSI	57000	44500	30000 Min.	31400 to 33800
Elong %			27 Min.	28
Endurance on notched bar .010 radius same as original serration root radius		17500		
Endurance on notch bar 1-1/32 radius same as serration radius PSI	29000	20000		
Shot peened same radius	37000	25000		

Serrations were actually shot peened in the early sixties on 16 cyl. blocks but procedure was discontinued when shot peened serrations did not show any noticeable improvement in resistance to serration cracking in the field.

The cylinder block weldments must be stress relieved to avoid existence of residual stresses of approximately the yield point. A not stress relieved weldment of the complexity of a cylinder block where residual stresses are not controllable cannot be expected to support service stresses without cracking. The only acceptable stress relief is a true thermal relief in a furnace. "Vibratory" stress relief, economically very attractive, was investigated by Alco and found completely without value ref (C 2). The fact that thermal stress relief is required, precludes the adoption of low alloy or other higher strength steels, since such steels virtually loose any properties better than those of mild steel in the thermal stress relief.

Weldment Design

The welding design is based on placing all higher stressed welds longitudinally to make machine welding (submerged arc) possible in many areas. It also makes possible the intermediate machining of weld preparations which serve to ensure relative dimensional accuracy of the design. Unless otherwise indicated, all welded dimensions are expected to be within ± 1/8".

To maintain the overall length and aid to avoid excessive variations in the thickness of the end plates, also to insure a tight fit up to the end plates the block weldment is milled at both ends to length prior to attaching the end plates.

It is the design intention to place all highly stressed welds (primarily the saddle joints) so as to get them away from stress raisers due to changes in contours. The most typical example is the design of the "spline" in which member the loads from the firing forces are transferred from the inner side plates of both banks to the single vertical projection on the bearing saddle. All the welds joining side plates and spline, outer side plate, and saddle tangs, spline and saddle are carefully kept away from any abrupt changes in cross section.

The spline is now and was for the past few years being made from extrusions, but prior to that it was most economical to mill the spline out of a heavy plate. This places the high stresses across the grain of the plates. No case of failure due to laminations in the spline is known. Laminations in top and middle decks occurred occasionally, they were chipped in deeply, welded up, and have not caused known failures.

The welds between top deck and side sheets, full penetration by submerged arc expected, but not back chipped, violate the rule of not placing welds into points of major cross-sectional change. The earlier 244 block had tangs machined out of the top deck, a very costly operation, this was intentionally not done on the 251 block for cost reasons. Only very few failures at this point have occurred and they were triggered by faulty welding. The lower contour of the top decks is machined giving an accurate and straight weld preparation, but excessive stock removal to make tangs is avoided.

All but one weld is made from both sides. The main strength welds between saddle tangs and adjacent members are back chipped to give an expected 100% penetration. All saddle tangs are made excess length, with full cognizance that this makes for more expensive forgings. They are welded full length against spline, base rails, and side sheets and then torch cut to design length. Print 31-AA-71394 is a typical illustration of this method. This is one of the important features of the Alco Block design. The 244 cylinder blocks had many failures in the corresponding joints because the end of the welds were "capped"; capping, even if carefully done, is likely to have some slag inclusion near the ends which act as nuclei for failures.

False run outs, little plates with weld grooves which permit starting and ending the welds and which are knocked or burned off afterwards, were considered in lieu of the wide saddle tangs, but were judged inferior and risky. G.M. used them at one time.

31-5

Weldment Design - cont'd

Similar considerations apply to the joint between the end plate item 25 and the end bearing blocks item 17; the difference is that the leakage would be lube oil. Originally the bearing blocks were attached to main structure prior to milling off for the end plates. The blocks were then fillet welded to the end plate. Oil from the camshaft bearing entered the gap between cam block and end plate and leaked out of the threaded bolt holes holding the various casings to the cylinder block. To avoid this, an extension was added to the bearing block protruding thru the end plate and the expensive weld per detail section L-L was adopted.

Weldment Design - cont'd

In occasional xray examinations of the saddle tang welds, any indications of lack of penetration closer to the ends than 1-1/2" were burned out and rewelded. Only very light indications in the center were considered acceptable. Xray examination can reasonably be made only on the preassembly of base rails, spline, and side sheet.

The only one-sided weld is between the top decks and the center deck over the air manifold. No feasible way for double-sided welds was found, but the edge preparation makes a nearly full depth weld possible. A few failures of this weld occurred some years ago. Instructions for a field fix were worked out, a copy is attached as supplement CY 1 because it could be used with slight alterations for many other defects. The defect in the center top desk has not been reported in recent years.

Aside from the known weakness of one-sided welds, another strong reason for avoiding them in compartment exposed to lube oil is this: a joint that is not closed by welding presents a small gap in which sand from a sand blast operation will invariably get lodged, in operation such sand will gradually enter the oil with potentially serious consequences. Similar considerations apply to intermittent welds in oil spaces.

To facilitate policing practices regarding one-sided end intermittent welds in oil spaces, they have been virtually avoided on all engine work. There exists some considered exceptions.

Special precautions against water leakage; print 31-AA-71632 is used as illustration but the same conditions exist on all blocks.

Consider the water compartments in cross section AA in particular the upper welds between the middle decks items 8 and 9 and spline 5 or side sheet 6 and 7. It is conceivable and it did actually happen that one of these welds at some point is porous, or has a blow hole which permits some water to pass into the joints at the sides of the middle deck between the upper and lower welds. While these joints are presumed not to have a gap, they are not watertight. Wherever in the whole length of the block this porosity may exist, the water will travel all the way forward. If the end plate were simply attached to block and connected by fillet welds any water leakage could and did travel to any part of the joint between block components and the end plate and come out after machining thru a hole drilled thru the end plate into the joint with the other parts of the block. Remembering that the welds inside the water compartment are not accessible for close inspection, it is nearly impossible to find and correct the primary deficiency, especially not if found only after machining.

To prevent the occurrance described, "false" or double end plates item 24 have been adopted. They close off the water compartments. Note also the capping of all pertinent longitudinal welds specified, e.g. in detail section C-C. After these small end plates have been applied, the water compartments are pressure tested.

This is a point in fabrication where any corrections can be readily done. Only after this pressure test has been passed will the end of the block be milled off square to receive the end plate.

Failure Prone Areas.

In the early Sixties a substantial number of cylinder block failures occurred in two interrelated areas; they were concentrated on 16 cylinder engines, although a few odd failures on 12 cylinder were found.

One phase of the failure was cracking in serration roots, the other cracking in cam bearing blocks. While the failures did occur on any saddle, there was a preponderance on the more heavily loaded saddles, the cam bearing failures followed the same pattern. That is, failures predominately occurred on saddles and cam 3, 5, 7. Reviewing the record in 1961 such failures on serration and cam block, were found on about 50% of engine, then on the Sante Fe after about 300,000 miles. Several changes were made simultaneous, those affecting the serration will be discussed when dealing with machined blocks.

The cam bearing block failures were over come, or at least reduced to acceptable frequencies by weldment changes. Refer to sketch CY1 showing the various steps in reaching the current design. The crack shown in the original design originates at the sharp end of the fillet which forms a real stress raiser. The stress causing the failure is believed to be due to firing pressures which put the side sheet in cyclic tensions and attempt to stretch this rather elastic member. The bearing blocks attached to the firing bulkhead are rigid and attempt to carry the firing forces for which they are not designed.

Original attempts to chip out the crack and reweld it in place were unsuccessful, but boring out the bore over size, inserting a bushing, welding the bushing with a 1-1/2" long weld and leaving the crack was completely successful; refer to print 50 B 73705.

The "intermediate" design which stiffened the bearing block was a typically wrong approach. It was made "livable" by making it more flexible by milling and grinding at least the most highly stress bearings. This showing improvement, a design was adopted in 1961 in which the bearing block as well as the bulkhead were shaped not to resist forces transmitted by the side sheet, while a gusset was added between bearing block and outer side sheet to aid in resisting cam and fuel pump loadings. This design, originally considered in the light of a stop gap has proven quite satisfactory.

A year or two later a rush of failures occurred in the weld between cam bearing block and side sheet on blocks fabricated by M.L.W. Such failures were not completely unknown on Auburn made block but were very rare. The Canadian failures were very serious because the cracks propagated into the side sheet, with little chance of repairing them except by following the repair by full stress relief. The type of crack is shown on sketch CY2.

These failures were primarily due to bad fit ups between bearing blocks and side sheets; it is necessary to get good contact between the two members so that there are no gaps for the fillet welds to bridge.

ALCO ENGINE INC.

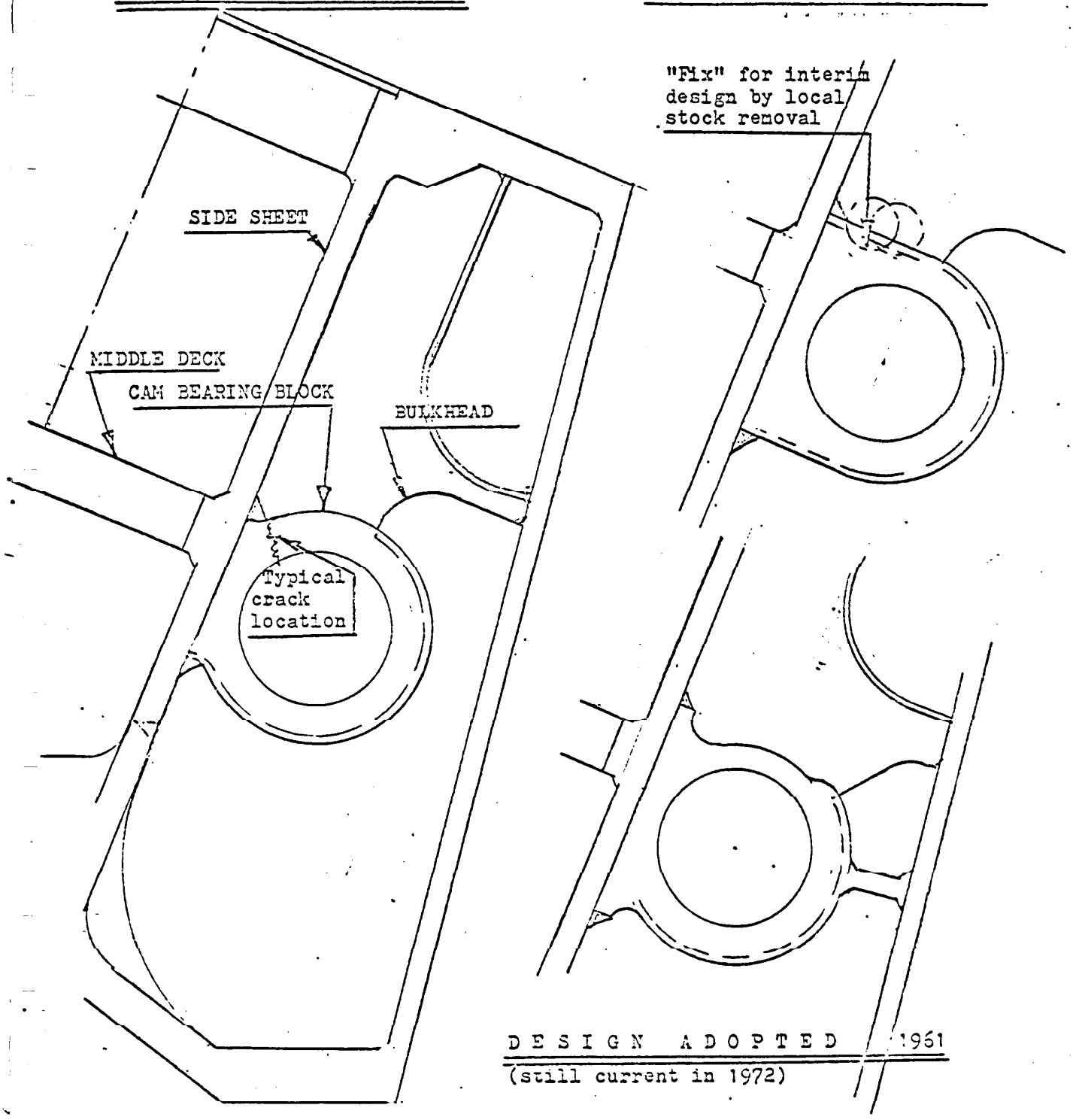
SKETCH CY I

BY F.C.T. DATE 3/13/72 SUBJECT CYLINDERBLOCK REVIEW SHEET NO. OF

CHKD. BY DATE Illustration of cam bearing block dovelopment JOB NO.

ORIGINAL DESIGN

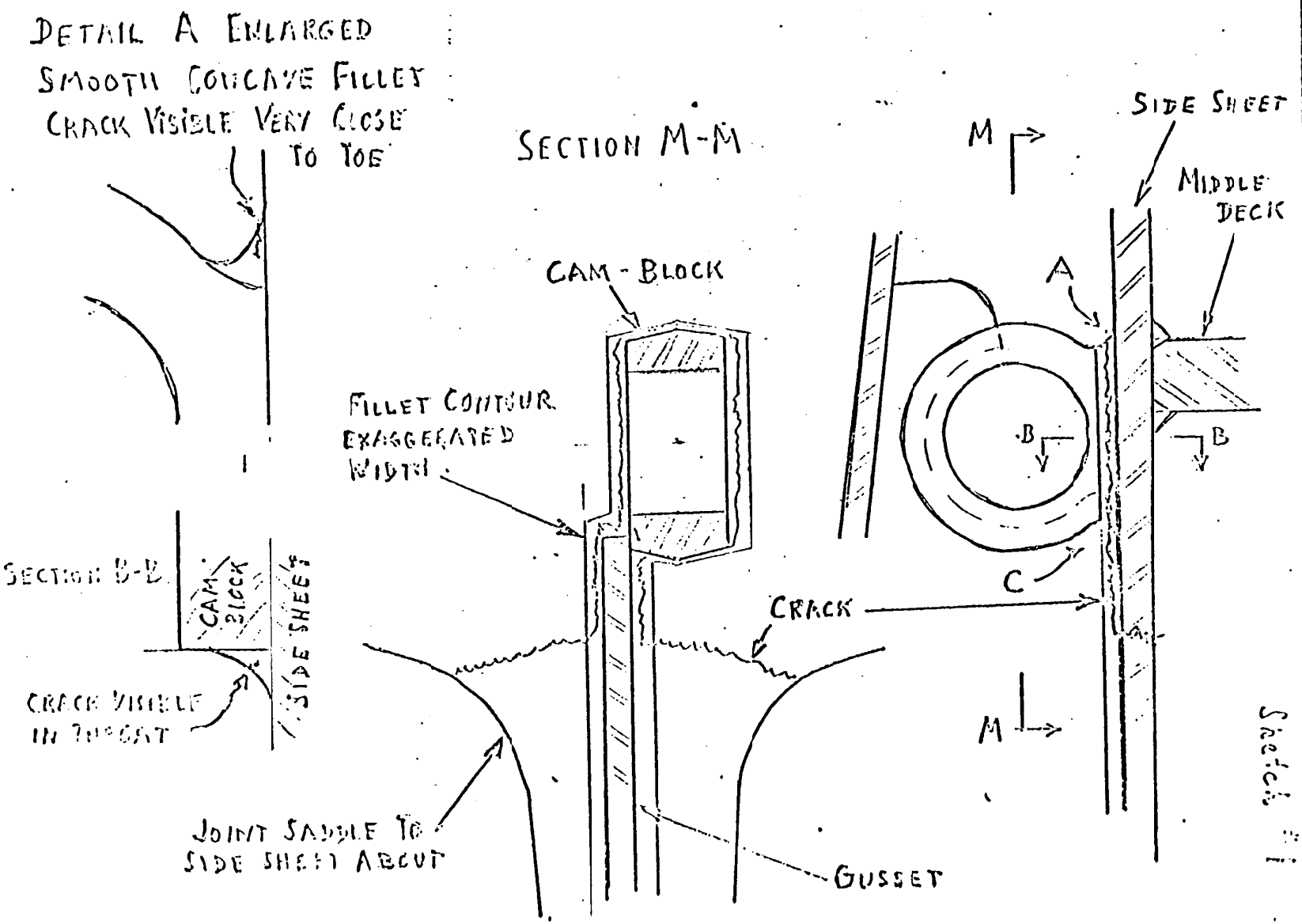
INTERIM DESIGN



DESIGN ADOPTED 1961

(still current in 1972)

SP 10935A



TRIP REPORT 10-30&31-62 M.L.H.

12 -251 CYLINDER-BLOCK

DRG. NO.

SKETCH CY-2

Sketch #1

Cost Reductions in the Weldment

Originally a stiffening plate, with large "rat" holes at each corner, was inserted in the water compartment above the center saddle. It was conceived as preventing diagonal distortions of these compartment. They were eliminated after having a few blocks without the stiffeners several years in service. The middle deck thickness was changed from 1 3/4 to 1 1/4 also after trials on several engines for several years.

Product engineering never controlled welding methods, electrodes or even weld preparations. Over the years many changes for "cost reduction" were attempted in these areas and some were carried thru but only after consultation between product and industrial engineering.

One cost related set of weldment changes relates to the drain channels inside near the bottom of the outer side sheets. The original design provided a channel to collect fuel leakage from fuel pumps and nozzles. This was led thru tubes with clearance holes in the side sheet into the channels, from where it was piped to the fuel tank. Realizing the danger of contamination of the lube oil by fuel these channels are welded to the side sheets and pressure tested before application to the block.

On these earlier engines the grommets on top of the covers over the fuel pump drives were deficient; water from heavy rains and from cleaning engines with steam and hot water jet (a practice on some railroads) went past these grommets and entered the drain channels and from there into the fuel tanks.

The simple channels were then replaced by pairs of angles one of them collecting thru tubing with sealed fittings the leak off fuel and returning it into the tank, the other collecting water runoff, lub or fuel leakage past gaskets which would be dumped. In the meantime the upper grommets were improved considerably and will not admit water, if correctly installed (not a certainty). In many installations the leak off fuel is not returned in any case, but everything is dumped, so that single channel was readopted some time ago.

The LST engines show the absurdity of having the double angle passages on the inside and outside a harness of tubing connecting the four ends of the angles and leading to the bilge thru a center connection.

These channels have caused additional mishaps in a few instances on locomotives. Fuel tanks on railroads maybe filled with pressure thru tight connections. If filling is done carelessly the fuel may rise thru the drain channels and enter the crankcase thru fuel pump crossheads. The fuel pump covers now have holes thru which the fuel will run out as fast as it is pushed in; on earlier pipings various means were used to prevent such occurrences.

Cylinder Block Machining

Since the first 251 block was built, the only significant later changes in machining are:

1. Relocation of crankshaft thrust from the center saddle to the saddle at the generator end; for details see discussion in section crankshaft.

2. Introduction of a cast iron (now ductile iron) insert at the lower liner seat. This area was a source of constant complaint, no seal ring lived and the bore in the middle deck fretted badly. The pressed in Cl sleeve together with a Viton "O" ring at the water side, have completely eliminated the complaint.

3. Originally the bearing serrations had flat bottom with .010"R fillets; they were changed to serrations having as bottom a full 1/32 radius. This required reducing the height of the serration teeth.

The serration change was one of the several changes made in 1961 to overcome a serious epidemic of serration cracking, mentioned earlier. Such cracks varied from a harmless breaking out of one serration for the whole length to cracks originating in the bottom of a middle serration propogating under about 45° until it reached the bore. A number of changes were made simultaneously to overcome these failures; they are listed below in the order of probable effectiveness:

1. Increase main bearing stud stretch from .021" to .030". Both during machining and in service. When stretch was increased on blocks bored with lower stretch changes in bore dimensions were mostly less than .001". The increased stretch required addition of hardened washers to prevent the nuts from galling.

2. Changed number of counterweights from 4 to 8 to reduce the horizontal component of the bearing loads. This change was very effective but was adopted somewhat later. It made increases in speed and power possible, rather than merely correcting the earlier deficiency.

3. Changed the serration root radius from .010 to 1/32.

4. Changed firing order from 1,5,7,3,8,4,2,6, consecutive, to 1,4,7,6,8,5,2,3 consecutive. This cut the torsional vibrational amplitude of 2-1/2 order at full speed in half. Some strain gage measurements at saddles also linear vibration amplitudes show some gain; torsional vibration amplitudes on free end of camshafts are much worse. Ref CY3.

5. Serrations were shot peened for a short time. This was soon discontinued as being costly and apparently not effective.

6. Tolerances on serration spacing was tightened. This probably had little effect in a material with relatively low yield point. Also machined serrations, always broached in the cap and milled in the block had even then respectable accuracy. A later epidemic of bearing failures on MLW built 16 cylinder blocks was traced back to improperly machined serrations. MLW machined these serrations in the blocks with short broaches, not well supported.

Cylinder Block Machining - cont'd

7. Normalizing of saddles which had been discontinued as cost reduction was reintroduced, more in the thought of not accepting any unnecessary chances than in expecting significant gain in properties.

Now over eight years after these changes were adopted and ratings were increased, no significant number of such failures are reported. On the other hand, the block of a 16 cylinder engine rebuilt by a railroad was recently received for repair. This block showed all the failures found in the early Sixties. But the railroad had the block rebuilt with a four counterweight crankshaft and operated the engine as a C engine at 1050 RPM.

Cavitation Damage

The Alco cylinder block if operated with water treated according to Alco recommendations is not subject to cavitation damage. If no or inadequate water treatment is used, serious damage will occur.

An early specific instance of cavitation damage triggered by local stress concentration is mentioned because it highlights points to be watched. The original practice in the welding shop was to put temporary braces inside the Vee of the air compartment at the center of every other cylinder and tack weld them to the side sheet. After the top decks were welded to the side sheets, the braces were knocked off, leaving part, of the tack welds on the side sheets. Cylinder block, returned from the Pennsylvania R.R., which always had a record of inadequate water treatment, showed severe cavitation damage inside the water compartment originating opposite the spots where tack weld on the air side had been carelessly removed. This occurrance was prevented from reoccurring by making sure that the tack welds were ground off smooth.

Usually the most pronounced area of cavitation damage is in the bore of the upper liner seat where water velocity is high. Also reported are cavitated areas in the side sheets opposite the liners.

Repair work will be discussed in a succeding chapter. Principally it is preferred to smoothen cavitated areas rather than build them up with weld unless the depth of cavitation makes build up mandatory.

Attempts to find a way to prevent cavitation is described in ref. CY4; neither of the two approaches, one a thicker liner, the other interposition of a stainless sheet metal sleeve between liner and block (to dissipate presumed pressure waves) helped at all; standard liners and standard block showed very little cavitation damage in 575 Hrs. with untreated Schenectady city water at 1050 HP, 1025 RPM in the 6 B engine.

Block Salvage

Standard practice sheets 31P7501, pages A thru G together with drawing 50-EE-72601 are the basic guide lines for block repair. It is not expected that these instructions cover in detail the repair of any conceivable damage, nor is it expected that any block salvaged by Alco was treated in every particular instance as instructed. Drawings and standard sheet were reviewed jointly by product, industrial and welding engineering in 1968 and represent a quite reliable document.

Repair welding, especially building up some surfaces by laying weld beads close to each other should be avoided where possible both to keep labour costs and residual stresses to a minimum. Stress relieving will reduce residual stresses but introduces the need for extensive machining to offset distortions. Paints which prevent, or at least reduce, scaling are being used on machined surfaces. Scaling is further reduced by stress relieving salvage blocks at about 100° lower temperatures than new blocks.

Welding of serrations especially should be avoided if at all possible; it is much preferred to recut serrations into the parent metal even if this brings the pitch line of the serrations above the bearing bore (.100" max per drawing). Residual stresses by salvage welding were investigated (reference CY5), furnace stress relief is effective, welding of small beads with hammer peening each bead is quite effective if done with care. Preheating, postheating local stress relief is useless.

Machining to undersizes and restoring dimensions with shims in lieu of welding up large contact areas has become the preferred practice in recent years. If the sides of the block under the fuel pump supports are remachined, injection and cam timing are affected (1/32" stock removal corresponds to about 2 crank degrees). Timing discrepancies between right and left banks are accentuated. Performance of lower liner bores has been so much improved by using the cast iron inserts, that any of the early unsleeved blocks returned for repair should be sleeved at this location.

The design conditions at the upper liner bore are such that attempts at restoring cavitation damage by sleeving was not successful, mainly because the sleeve is cut nearly in two by the water outlet hole and therefore could not be installed with a tight fit. Sleeving of the upper liner fit was done by the shop with great frequency in recent years resulting in many complaints from customers about rapid cavitation damage. The practice of machining blocks for the upper sleeve was stopped late in 1971.

One frequent consequence of a main bearing failure is that heat generated by the failure has caused one saddle and usually also the cap to "close in", that is, the horizontal bore dimension is reduced. This is the typical case of requiring build up of bore by weld cutting the serration in the block slightly deeper into the parent metal, recutting the serrations in the cap and boring the damaged bore in line with the adjacent bores. No other bores need touching and no stress relief is required; refer to 31P7501, item 1.4C, 3,1,4 and 3,3,2.

Block Salvage - cont'd

A common occurrence on returned blocks is that the vertical bore dimension has become smaller. Presumed reason a slight yielding in the serrations. Correction at Alco is to establish degree of undersize and possible misalignment of bores to each other by mandrel, then correct by hand held hone; by placing the block on horses and taking advantage of the fact that the weight of the hone causes it to cut slightly more downward, alignment correction are possible.

Tools to bore out cylinder blocks at upper and lower liner fit, also of cam bearing blocks, without removing the cylinder block from the engine were developed. They are portable tools operated by air motors. One camshaft boring fixture and one or two liner seat boring fixtures are owned by Alco and rented to customers; these tools are under the jurisdiction of the service department. Complete description of the liner seat bar is given in reference CY 6 while only limited data on the cam block bar are in reference CY 7.

6 Cylinder Engine Block

Preceding discussion is based on the cylinder blocks for the Vee engines. The cylinder blocks for the 6 cylinder engines follow the same principles. The saddles, also forgings connect directly with the side sheets without requiring the "spline" member. One significant difference is that the blocks contain as an integral member the lube oil header which is connected to the saddles, i.e. to the upper part of the bearings by rigid tubing which must be regarded as completely inaccessible on an assembled engine. Once the need for keeping this piping clean was appreciated by shop personnel, no difficulties were experienced.

In contrast with the Vee engine blocks no failures, except cavitation damages, seem to have occurred on any 6 cylinder block regardless of operation at 203 MEP at 1025 RPM or 243 MEP at 1100 RPM. This fact suggests that the block has substantial margin for uprating or potential for cost reduction

31-17

SUPPLEMENT

CY 1

INSTRUCTION - For Emergency Repair of Cylinder Block Manifold Leaks with Engine in Locomotive. Page

MATERIAL AND EQUIPMENT REQUIRED:

1. Cleaning Solvent - Kerosene or Fuel Oil, also Carbon Tetrachloride or equivalent.
2. Clean Cloths.
3. Air Chipping Hammer and Chisels, also round nose chisels for peening.
4. Portable Air or Electric Grinder.
5. Welding Machine DC Generator or Rectifier.
6. Welding Electrodes 50 pounds of size 1/8. E 6010
American Welding Society designations ~~- E-6018 or E-7018 preferred: E-6016 or E-7016 second choice.~~

PREWELDING PREPARATION:

Remove hatch, exhaust manifold, air elbows, and water headers.

1. Clean top deck with kerosene or fuel oil. Then wipe with rags soaked in Carbon Tetrachloride or equivalent.
2. Examine area near the joint of the horizontal manifold cover and the top decks, to determine the extent of cracks or defective weld areas.
3. Depending on the location of such areas it may be necessary to remove some of the cylinder heads and some of the inner-most cylinder head studs to give adequate room for chipping and welding.
4. Cover all openings to cylinder heads, or liners if heads are removed, and air manifold to prevent entry of foreign material during chipping and welding.
5. It is expected that the defective areas are not continuous but individually less not much over one foot long and separated from each other by several inches length of sound metal. Chip out the defective material with grooves about 3/8" deep. Keep the groove as narrow as possible with somewhat sloping sides and slopes at the end.

INSTRUCTIONS (Cont'd.) 2.

6. Should the defects appear to be continuous for a length much greater than one foot length, chip out grooves about one foot long then skip about one foot and chip another foot long groove. Continue this pattern of grooves and unchipped sections as required. Complete welding according to the following instructions. Then go back and chip out the remaining defective areas and weld them in the same manner.

WELDING PROCEDURE

1. Note location of chipped out areas in relation to the cylinder heads. The nearer to a cylinder centerline welding must be done, the more care should be used to avoid distortion.

To minimize distortion, start the bead opposite the nearest cylinder centerline and work away from it toward one end of the chipped groove. Then work in another area. Resume welding in the first groove only, after the surrounding metal has cooled down so it can to touched by hand.

At this time start the bead again at the cylinder centerline, and work in the opposite direction towards the other end of the groove, slightly overlapping the beginning of the first bead.

This applies to the root pass as well as to the succeeding passes.

2. If there are small cavities or blow holes in the bottom or the ends of the grooves, peen them shut before starting to weld.

3. Place a root pass in the bottom of one groove. Immediately clean weld, remove slag and peen lightly.

4. Repeat procedure 3 in other areas until all grooves have the root passes. Examine for defects, chip out and repair if necessary.

5. Deposit, succeeding passes one at a time, using the instructions of Item 1. Clean, remove slag immediately after completing the individual beads and peen somewhat heavier than on the root passes.

6. If the defects are so extensive as described in "Prewelding Preparation" item 6, chip out the remaining areas at this time and weld.

7. Smoothen top deck with grinder to provide sealing surface for air elbows and water vents.

CY-1

WELDING PROCEDURE (Cont'd.) 3.

8. Remove the covers from the various openings, examine all ducts especially, the air manifold for chips and weld spatter.

~~The following precaution will improve the quality of the repair. The recommended low hydrogen electrodes have a coating which is hygroscopic. Therefore, to preclude possible moisture pick-up, all electrodes from opened packages should be baked at 550°F for one hour, and then kept warm during the entire repair.~~

P. S. Vaughan /T

P. S. Vaughan,
Chief Engineer
Diesel Engine Design

FCT:pt
1/8/60

SUPPLEMENT CY 2

Blocks for the LST engines are virtually the same as standard blocks; the large round opening in the end plate at the drive end is normally burned out, on the LST it is bored to receive the bronze crankshaft seal retaining and shock deflection restricting member. A few extra holes to hold the fuel header, some pipe flanges, water header supports take care of the high shock requirements. On the standard blocks, base and block location is established by two dowels near the ends at the right engine side, all shear forces are resisted by friction from the cap screws between the rails on block and base. On the LST ream bolts were added so as not to rely on friction during shock.

References Section Cylinder Block

CY 1 Met Lab Reports 3092 & 3092A, 3378

CY 2 "Vibratory Stress Relief" Correspondence File 1968

CY 3 Mechanical Lab Report 3378-C

CY 4 Chem Lab Report 1117

CY 5 Residual Stresses Mechanical Lab Reports 3092, 3092A, 3167, 3235

CY 6 Liner Seat Boring Bar Mech. Lab Report 3175, 3175A, 3131, 3131A, 3131B

CY 7 Camshaft Boring Bar, Terminated Engineering Project Folder 863921

FUEL PUMP DRIVE AND VALVE LIFTER MOUNTING

General and Initial Development

This discussion uses assembly drawing 32-EE-72248 as basic reference. The arrangement of the push rod lifters was taken over from the old Alco 12½ x 13 model; it is simple and causes few complaints, and is virtually unchanged from the first design; it has operated satisfactorily without pressure oil lubrication, depending on oil catching depressions to lubricate both the fulcrum bushing and the roller pin.

The fuel pump lifter design was chosen from many layouts of different approaches as being the least objectionable. Attempts were made to design crossheads but the pumps protruded and made the engine too wide for locomotive use. The later redesign proposals (1968) to mount the pumps on top of the block could disregard the fact that an In-line engine would not leave any room between the heads for pump installation. When the 251 was first designed, In-line engines (including an 8 In-line) were considered fully as important as Vee engines, and it was mandatory that any basic design should apply to In-line and Vee engines equally. After original painful debugging, the fuel pump drive design has performed reasonably in actual use, but it did and does have enough shortcomings to discourage adopting it for any new engine design.

In the first prototypes serious jumping of the rollers was experienced. It was not due to cam deceleration because it did not jump if the pump was cut out. The jumping was due to deflection of many components during the time that the pump operated against hydraulic pressure and the sudden release of this pressure when the pump ports opened, and all components resumed their undeflected condition.

The jumping was overcome in two ways; an auxiliary spring was added, originally the spring in the pump had been dimensioned to handle cam dynamics alone. First two rat trap springs were added at the lifter fulcrum, but then the casting was redesigned to accept one auxiliary spring under the pump.

These springs were rather highly stressed and some failures (not excessively) were experienced. They underwent several redesigns, guided also by high speed photography. Originally, a spring retainer was sandwiched between pump and support casting. The sandwich fastened by countersunk screws to the casting so that the pump could be removed without disturbing the spring, and originally, when the free length of the spring was quite long, without having the spring jump out into one's face. When the spring was later shortened, this sandwiched spring retainer was eliminated and the pump mounting face of the casting raised accordingly.

The second phase was to minimize elastic deflections. Deflections under static hydraulic pump load of all parts concerned were measured and those showing substantial deflections were stiffened. These were primarily the lifter forging and the support casting. The lifter became much larger and its mass was so increased that part of the auxiliary spring force had to overcome the cam deceleration forces. On the casting the ribs and high bolt bosses were added to stiffen it. At that time it was made of cast iron. In the cylinder block the camshaft bearings were widened.

These changes together overcame observable roller jumping at speeds 1000 RPM and fuel pump plunger size 15 mm, then in use.

The clearest way of observing roller jumping, not necessarily employed in the early development, was to apply a strain gage to the upper side of the lifter eye above the roller pin; this area is fairly high stressed and the oscillograph trace showed jumping quite clearly. Ref. 32-1.

In the beginning, it was impossible to operate the fuel pump drive without scoring. The alignment of casting to cylinder block and camshaft with the two dowels had been

watched from the beginning and had given no complaint, but other misalignments, some of it due to camshaft deflection, some probably due to deflections and minor out of parallelism of the various small parts, caused corner loading and metal pickup.

This was overcome by increasing all running clearances between pins, rollers, bushings, beyond normally accepted values. A typical example is the running clearance of .004/.005 between the pin of 1-1/8" diameter and the roller bushing. While these clearances may require excess oil supply, they are absolutely essential to permit the rollers to align themselves to the cams.

Also to overcome scoring and pickup, lubriting of various parts in various combinations was tried. It was then established, and proven again later on, that operation of lubrited hardened steel parts against lubrited hardened parts gave best results, contrary to general recommendation. Lubriting, however, must be closely controlled; "sugary" surfaces cause rollers to slide on cams and cause irreparable damage in the first hours of operation. Ref. 32-2.

The original design included a thin gasket between casting and block. This had to be eliminated because it caused the bolts and the assembly to loosen.

LATER DEVELOPMENTS

In 1960 the 6-D engine was developed for higher power requiring 17 mm pumps and to operate at 1100 RPM, it was felt to be necessary to widen the cam roller from 1" to 1-3/8" which increased the mass; this and the higher speed required higher spring forces which could not be attained by a single spring in the available space;this brought about the current double spring design. The casting was changed from flake iron to ductile iron. Unfortunately, at that time it was not recognized that the capacity of the floating bushing in the lifter fulcrum also needed increase. Therefore all the early 6-D perhaps 100 engines have a hybrid arrangement with wide roller, double springs and original narrow floating bushing, involving non-interchangeable castings and lifter forgings. When higher speed and larger pumps became standard on the Vee engines, the hybrid arrangement was superseded by a forging with the hub length made equal to the width of fork for the longer roller and the eyes on the support casting were spread accordingly. This then also allowed for the use of a longer floating bushing at the lifter fulcrum. This is the design adopted for all new production at the end of 1971. Before that time B and C engines retained the single spring arrangement purely for cost reasons; when its usage dropped its cost increased over that of the double spring design.

Also at the time of the 6-D it was felt to be necessary to add two piston rings to the fuel pump tappet to reduce the loss of lube oil into the fuel which does occur by passing past the tappet. Actually there was only one complainant, Service Pipe Line, which operated 251 Dual Fuel engines where the contamination of the small tank for pilot fuel may have been of consequence. Service Pipe Line did therefore get tappets with piston rings for their Vee engine.

In all other applications, the quantity of lube oil lost was considered insignificant and the piston ring on the tappets were omitted from production engines soon afterwards. The assembly was tricky and rings could be broken without knowing it. Several MEL reports have quantitative data on lube oil lost. More important is that it has been confirmed that fuel does not get into the lube oil. Ref. 32-3.

All of the castings have two 3/4 cored holes connecting the outside with the spring compartment. Their purpose is to break any suction or pressure due to the pumping action of the fuel pump cup which might affect the existing reasonably effective separation of lube and fuel oil. The holes do let some crankcase vapor escape into the fuel pump compartment; but only laboratory personnel object to this because nobody but lab and shop run the engine with fuel pump covers removed. Trial runs were made with the holes plugged and leak off fuel consistency was not affected. The holes therefore could be plugged if there were any good reason to do so.

DEFICIENCIES, CORRECTIONS, ATTEMPTS TO CORRECT

Push Rod Lifters:

The heel of the lifter forgings at the fulcrum end keeps the lifter from falling down when applying the assembly to the cylinder block. It has happened that die wear let the heel become so large that it kept the roller from contacting the cam on the base circle, creating noise at contact and damaging cams and roller. A check in 1971 indicated that the shop has been machining the heel for several years rather than relying on checking forging dimensions.

No complaints concerning excess wear on roller and pin are known on Alco made engines. MLW had some complaints and is believed to make a differently grooved pin. Snap rings have broken with serious consequences on MLW built engines not on Alco built ones. MLW is believed to have changed the grooves in the pins from round to square; and from round snap rings to Truarc rings. Alco had no reason to change and did not want to risk hardening cracks in square cornered grooves.

In recent weeks, March 1972, the reason for different performance of snap rings has come to light. Alco received from the common source of Alco & MLW (National Lockwasher) a shipment of snap rings made of Music wire, duplicating those in use at MLW instead of the intended snap rings made of spring steel and heat treated after forming. The former yield permanently in application, the latter deform elastically.

Roller:

Some deficiencies on the fuel pump drive have occurred for which no clear solutions were found, yet the complaints have disappeared. Examples are:

Roller sliding instead of turning; from recollection this was a serious problem which seemed to have gone away without design changes; although a number of modifications had been tried. Among such modification were, adding circuferential grooves to the floating roller bushing, crowning the roller, an antifriction bearing in the roller, bronze bushings. Ref. 32-4, also 32-6.

Improvement of manufacturing practices probably accounted for their disappearance. The roller OD showed grinding chatter, temperature and chemistry of lubriting both was better controlled, corners of the roller are rounded on a belt grinder to preventedge contac between a chamfer and the cams. Application of Molycote powder to cams was beneficia..

When the roller was widened and the pin diameter increased to compensate for the greater span, the ends of the pin would hit the air and exhaust cams. Flats were added to the pin to give adequate clearance. But the flats together with the need to match oil holes in pin and lifter eye demand a pin lock insuring angular orientation. The pin lock plate made of case hardened sheet metal and fastened by two bolts serves this purpose. In the late sixties, several cases occurred within a short time where the pin lock plate showed indications of a heavy axial force on the pin, the plate had segmental depressions where the slot in the pin pushed against the plate. In one case the segment was actually punched out, in a few other cases the 5/16 cap screw heads had popped off. This type of failure was never explained (what could cause a substantial longitudinal force on the pin?) nor was a remedy found. A few of these failures had serious consequences but they did not recur

Some cases occurred in at two different periods; the spherical seats over the adjusting screw in the lifter cracked; they were due to the spherical radius of the seat being too small (outside drawing tolerance).

Floating Bushing

One of the most persistent shortcomings is excess wear on the lifter floating bushing. With loads and oils then in use, performance on locomotive engines was considered adequate until the early sixties. The ease with which wear was detected, .010" radial bushing wear would retard injection about 10 crank degrees, and low cost for replacement bushings counteracted loud complaints. Typical wear figures of that time 1960, see MEL 3284, for wear after 40 months in freight service are: bushing O.D. and pin insignificant, bushing I.D. .023" max diametral wear, average .011". All bushings did rotate as intended. At the same time, however, complaints about excess bushing wear were received from stationary applications which operated with sustained (not high) loads and speeds.

A large number of different materials and some design combinations were tried. But laboratory testing for such parts is of limited value. Differences in wear after several hundred hours are too small for clear evaluations, except if results are very bad. Any laboratory wear result needs field tests in several applications. The recently adopted Hi-Silicon bronze bushing, did show up good on laboratory test, then on river boat and locomotive field test. It took a very long time to come to the decision to try it, it is so difficult to machine that the bushing becomes very expensive; it is definitely not considered a bearing alloy. It has however been used very successfully as the fuel pump roller pin of the 539 engine. Ref. 32-5. In the lab testing the High Silicon bushing, Manganese bronze, also showed substantial improvement over the original SAE-64 bronze but not nearly as much as the High Silicon bronze.

Current trials include babbit lined non-floating steel bushings; in Canada also cast iron bushings and nitrided steel pins. Pins made of 52100: not much improvement should be expected from the choice of 52100 because this pin has always been of carburized steel which has at least as much carbon content and as hard a surface as 52100. Surface improvement however should help. Pins without flat, Ref. 32-6, may help by improving trueness of contour by centerless grinding.

Trials with negative or inconclusive results made in the past include: Ref. 32-6, bushings of SAE-68 (aluminum bronze) SAE-63 leaded gun metal gear bronze), Standard bushing of SAE-64 but with oil groove soldered up at ends, This was virtually not producible and trials were made, the bushing shortened and oil grooves stopped off by adding narrow ungrooved rings. Ref. 32-7. Lead thin copper thinly plated bushing and honey chrome plated pins, Ref. 32-8. There probably were some more different trials.

Oil additives have a great influence on bushing wear per ref. 32-9, copper content increases 6 times if high dispersant oils are used.

One way to eliminate bushing wear would be to unload the bushing during part of the cycle. Experimental drawings for such an arrangement exist, it was never tried being an example of desperate gadgetry. See assembly drawing 98-C-32818.

Fuel Pump Support Casting

The casting was originally made of cast iron, Alco spec 8024, which is approximately a 40,000 tensile iron, the original foundry CWC, met mechanical property specifications. For the "hybrid" 251-D ductile iron was specified. When the design for the widened lifter hub was adopted, it was felt that stresses in the casting could be carried by the original flake iron. Unfortumately, the foundry had been changed to the Abex plant in Mahwah, who never did meet property specifications but had failed to advise Alco of this fact. It was only after several failures with heavy contingent damage occurred in the field about 1966 that this point was recognized. The failures all were breaks of one side of the fulcrum pin supporting lugs, letting the lifter assembly drop into the path of the connecting rod. The break was always more or less parallel with the mounting face on the block. After intensive investigations including strain gaging, Ref. 32-10, ductile iron was adopted for all fuel pump support castings. Limited changeout of castings was made in the field.

One pecularity was, that most, if not all, failures occurred on fuel pump supports on the Right Bank of the engine, (considering only locomotive engines with normal direction of rotation). Vector analysis of the forces does indicate less favorable loading on the right bank than on the left bank but the demonstrable difference never seemed to explain the very obvious difference in failure frequency. It is an Alco recommendation, not necessarily in print, but made to many customers, to install flake iron castings on the left bank only and apply ductile iron castings on the right bank (clearly visible pattern numbers are different for the two materials.)

When changing castings from the style for short lifter hub to that with longer lifter hub, the drilling for oil supply to the fulcrum pin was mistakenly not changed, resulting in some cases in serious restriction of oil where holes in pin and casting match. This point was checked in the field by removing oil line and probing with a wooden stick (sharpened pencil but lead broken off). In some forgotten cases, it could still lead to still more bushing wear.

The assembly drawing shows as an alternate the nozzle spraying oil at the fuel cam roller. The nozzle is screwed into the fulcrum pin for the push rod lifters, which are drilled for the purpose and are fed by additional drilling in the casting. As mentioned in section cams such nozzles were installed on several engines in riverboats. It is not established if it reduces cam wear or not.

Fuel Pump Drive

Salvage:

Hardly any salvage instructions except replacement of parts are in use. At one time it appeared that the holes for the roller pin in the air end exhaust lifters would wear out; salvage of lifter 24B73128 with Clevite bushings their #130307 were run in an engine satisfactorily, but wear in this area did not become a serious problem. Bushings and salvage sketch were therefore not introduced.

Bushings to restore worn fuel tappet bores in the pump support castings were occassionally requested. Wall thicknesses of casting are too thin to make bushings feasible, wear does not seem to be serious enough to cause complaints.

While the fuel pump lifter bores showed hardly any wear as long as SAE 64 bushings were used, the current high silicon bushings may cause wear to a degree to make a salvage method desirable, insertion of a heat treated steel bushing is shown on 50C74226.

Cost Reduction

Pushrod lifters of pearlitic malleable iron 24B73129 instead of standard forgings made of 1045 steel,were procured and installed in three locomotive engines in 1963 on the Erie R.R. Nothing adverse was ever heard from them, but the project was not followed thru. The casting was made of elliptical cross section rather than "H" beam to avoid coring. The stresses are low; the use of a casting is entirely feasible but present availability would favor 80-55-06 ductile iron. Review of costs be worthwhile.

An ambitious study of a redesign of the fuel pump drive,both to reduce cost and also to overcome the short comings of the lifter design,was initiated in 1968, the principle is shown on layout 19LYA1587 (Ref. 32-11).

While the basic aim of replacing the lifter by a crosshead is desirable, the layout embodies features that are not proven, may cause difficulties in the shop, may trigger serious failures; they also disregard interchangeability with former engines and components. Examples for above statement are:

The important crosshead guide is located in bores created during machining of the block. The fastening of the guides is by bolts in holes which must be drilled and tapped in a plate prior to fabrication. It is not realistic to expect sufficiently close dimensional control of welding shrinkage and distortion to keep 32 bolt holes and 16 bores within acceptable dimensional relation.

The plate carrying the crosshead coring is welded to the side sheet in a way to create stress raisers ,which are likely to cause cracking of the side sheet and/or cavitation damage on the water side of the side sheet. A completely new fuel pump design is required. Am. Bosch did present quotations. The proposed design is not applicable to the 6 cylinder engines, nor is there any possibility of adapting an existing cylinder block to the new design. Estimates were prepared in 1968 claiming savings of $1390 - per 16 cylinder engine with an investment of $226,120. This figure assumes that one half that is $100,000 - of the cost for a new boring machine could be charged to C.R. of current production. The figure provides for lab testing but not for any expenses for field testing. While many detail drawings had been made,both originals and prints were inadvertently destroyed when clearing out the Schenectady vault. Enough information exists to recreate the drawings if necessary. MLW at one time showed much interest in this project.

FUEL INJECTION EQUIPMENT

Injection Pump

The American Bosch CQ size pumps has been used exclusively on 251 production engines in the past.

To provide adequate fuel delivery for the various size engines Alco uses pumps with 15, 16, 17, and 18 mm plunger diameters. Except for plunger, barrel and delivery valves all other components are the same and interchangeable.

The pumps have been subjected to no major changes except for the rather disastrous cost reduction change of the baffle ring and the harmless change of shortening the guide cap also for cost reduction. The baffle ring originally and now again is made of hardened steel tubing, and was changed to a rolled up piece of spring steel. The purpose of this ring is to absorb the hydraulic shock wave emanating from the port hole in the barrel when the pressure in the chamber above the plunger is suddenly released on port opening. At the higher ratings and speeds the pressures above the plunger will fall in the range from 14000 to 16000 psi Ref. 32-14. The cheaper rolled up baffle rings would desintegrate locally and the debris might damage the plunger or it might be carried back into the header to damage other pumps. This occurance is not strange, the shock waves coming back into the header could not be handled on the 244 engine as long as rigid tubing was used between header and pump. The failures disappeared completely when Aeroquip hose was introduced at this point. When considering the fuel headers one must forget the 40 psi read on a gage and controlled by a relief valve. One must think of sharp pressure peaks not less than 1600 psi, this figure was once determined on the 244 and may well be higher now. A few years ago it was attempted to protect pump and especially nozzles by a stainless steel wire screen in the fitting between header and pump. After a few hours of operation the screen had completely disappeared. Ref. 32-12.

Considering prior experience on the 244 it appears strange that rigid forgings are used between the 251 header and the pumps. The 251 started out with Aeroquip hoses, but deflections of pump and support during operation caused the injection tubes to break.

Joining the pumps rigidly to the header corrected the tube breakage. Note, however, that on the standard engine the header is not rigidly fastened to cylinder block, in other words the pumps are harnessed together but not to the main engine structure. On the LST engines, being afraid of high shock conditions, the headers were fastened more or less rigidly (not absolutely rigid) to the block, with no ill effects to date. Note that the bolts between header and the banjo forgings to the pumps are top grade, reduced body, of high tensile steel. Anything less good did not live even at lower ratings. The spiral wound gasket with Neoprene filler has never been reported to have leaked.

Between the delivery valve body of the pump and the injection tubing the Alco designed and pretented snubber valve Ref. 32-13 is interposed, it is actually contain within the extra long nut on the injection tube. The snubber valve was originally developed to correct cavitation erosion failures on the 244 injection tubing which it did most effectively. It was obviously carried over into the new design. It permits virtually unrestricted flow towards the nozzle but only very small flow in opposite direction when the delivery valve with its retracting relief piston seats. It thereby prevents or at least reduces the intensity of cavitation caused by negative pressure peaks.

As long as ratings required only 15 mm plunger diameter pumps alloy steel annealed tubing 3/8 OD .093 ID was satisfactory. When higher line pressures and larger fuel quantities were introduced with the 6D engine, (17mm pumps), tubes began to fail systemically, also required a larger ID. At first all tubes for the higher rating were individually heat treated after bending and swaging, but tubing made of heat treated alloy steel has proven satisfactory Ref. 32-15. Use of the larger .107 ID tube for smaller engine output is only insignificantly less desirable than the original diameter. Since proportionately large number of engines does require the higher strength tubes, and the danger of mix ups is of course, always present Alconow specifies only the larger ID higher strength tubing. This change initiated years ago became only recently effective due to large stock of original tubes.

The studs clamping the two bolt flange of the injection tube to the head originally were of standard stock, at the higher rating of the D engine, these studs broke several times letting the nuts fly off like projectiles at about eye level of attendants. The studs, Ref. 32-16 were upgraded accordingly, and no failures seem to occur.

For long time Bosch had their pump springs cadmium plated with rather frequent resultant fatigue failures. Several design or manufacturing changes were made on the springs.

Nozzles and Holders

In contrast to the pumps, nozzles and holders were changed significantly several times. The changes are most easily discussed by reference to sketches 32-1, 2 and 3.

The original type sketch 32-1, Nozzle ADB, shows very pious intentions; get the lapped fit away from the heat, allow reasonable space for a safe not too highly stressed spring, provide adequate lubrication for stem, spring, seats, and contact points. It did all that but the masses were too big and engine performance was not good; when it was changed to the style in sketch 32-2 the rack position in all engine test instructions had to be reduced by one mm to compensate for reduced fuel consumption. This nozzle did , however, establish the shape of the holder which had to be retained for all subsequent nozzles. These early nozzles had 60° seat angle; there were already 9 holes .350 dia with 157° spray angle, although a spray angle of 165° was tried.

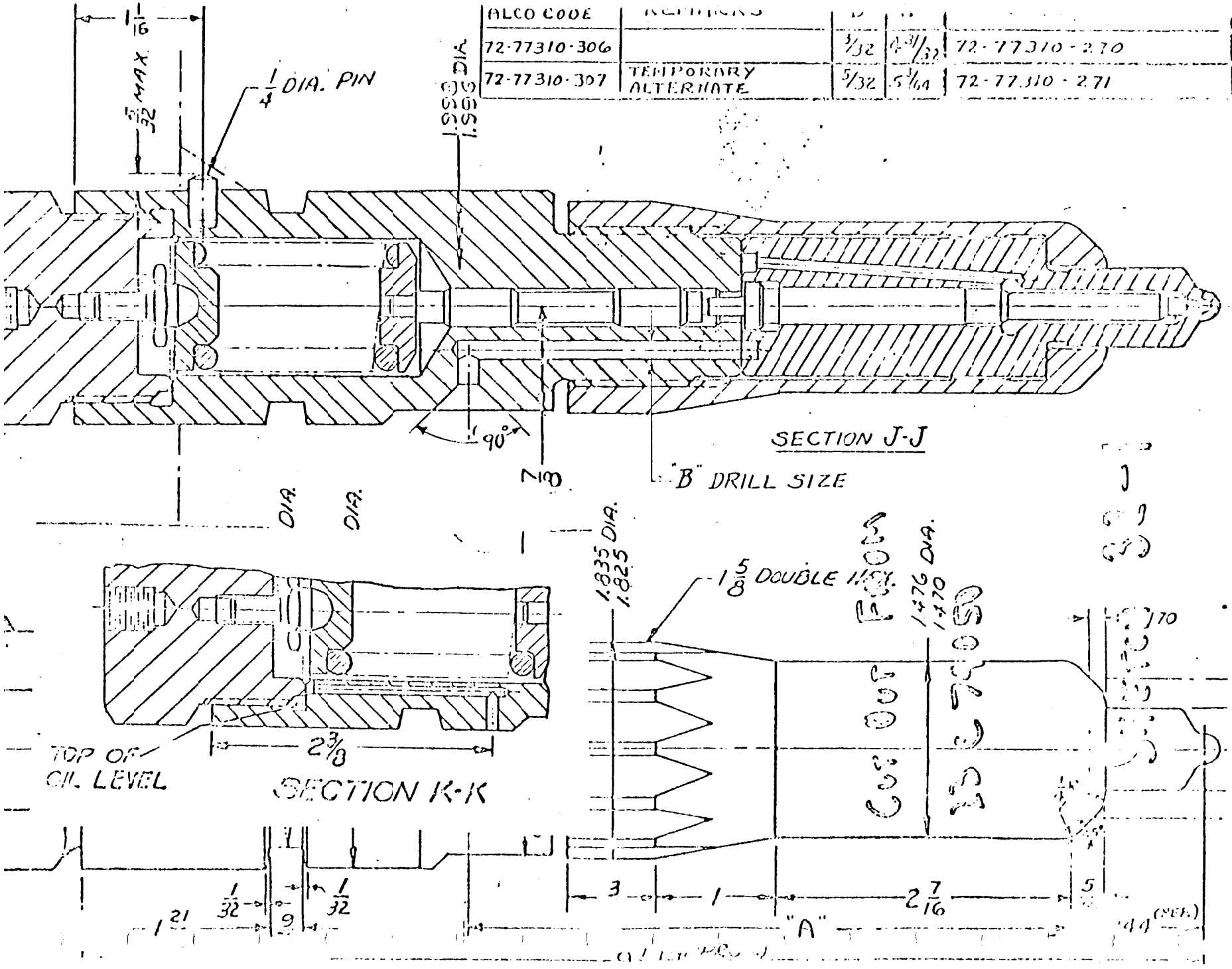
This nozzle holder was superseded in 56 or 57 by one to sketch 32-2, it shows already the currently used ADL type nozzle, 90° seat, 9 hole .350 dia and 157° spray angle. This configuration except for hole size has been used in all 251 engines built to date. The hole size varies from .325 for low power and low speed, by steps of .350 (most C rated engines), .375 (most E and F engines) to .400 mm for maximum ratings at high speed to match 18 mm pumps.

Relatively frequent failures occured in the spacer which forms the spring compartmen Cracks thru the long drilled holes, subjected to high fuel pressure fluctuations were the normal mode.

The construction was, therefore, changed again in 1959 (?) to that shown in sketch 32-3. This also showed one weakness in that cracks appeared between the 90° tube seat and the longitudinally drilled supply hole to the nozzle as shown on sketch 32-2 it was corrected by moving the long hole to the opposite side of the body, as in sketch 32-3.

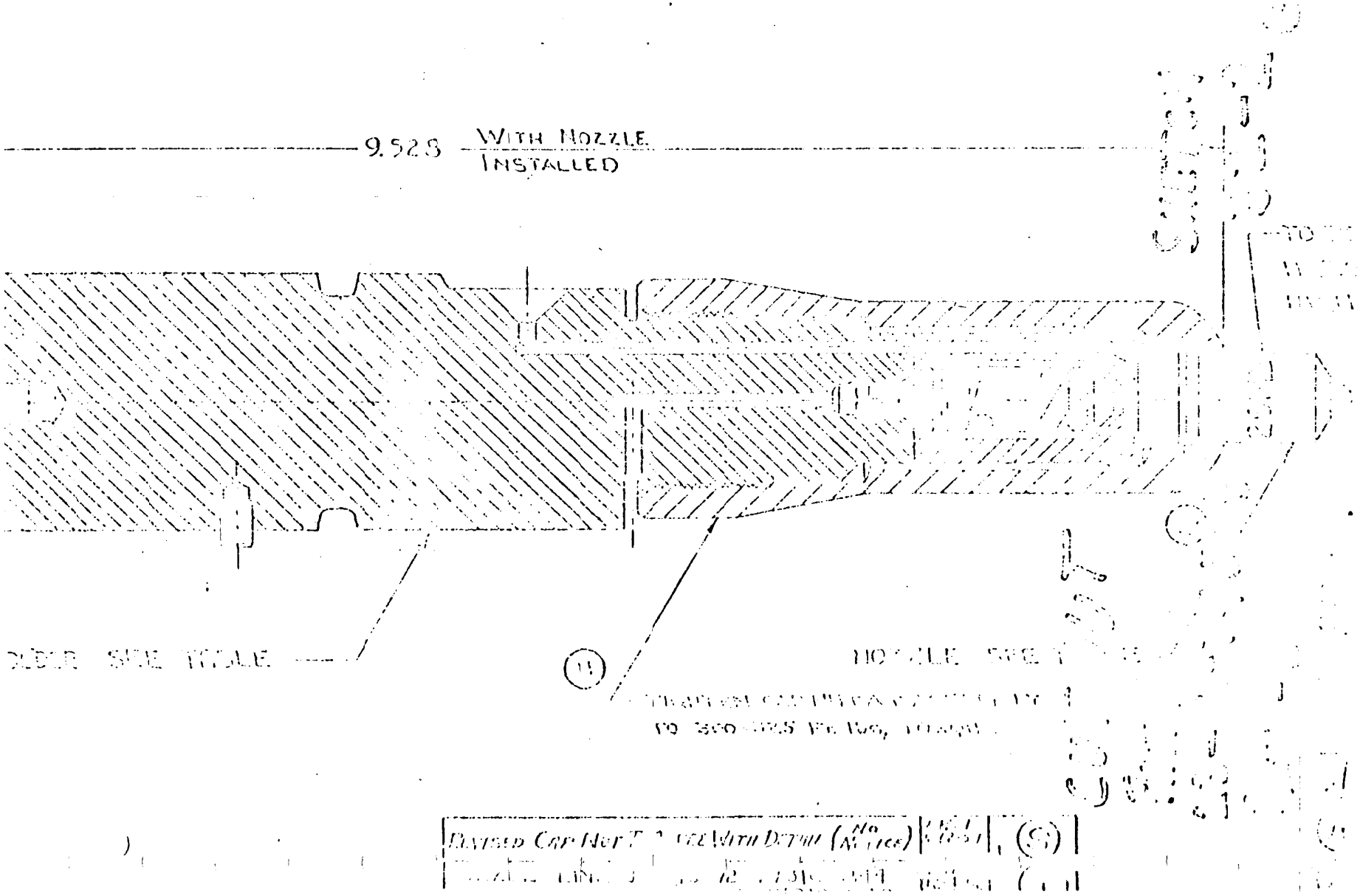
Over the years various changes were made to the nozzle body itself to make it more

ALCO CODE	REMARKS	D	A	
72-77310-306		3/32	4-31/32	72-77310-270
72-77310-307	TEMPORARY ALTERNATE	5/32	5 1/64	72-77310-271



			23-C-14073-1	
	12-77310-313	12-77310-255	23-C-14073-2	

9.528 WITH NOZZLE INSTALLED



REMARKS	ASSEMBLE TO CODE	NOZZLE HOLDER			DESCRIPTION	
[illegible]	[illegible]	[illegible]	[illegible]			
[illegible] PLANT	72-77310-317	72-77310-285	23-C-74073-1		ADL-157TM-[illegible]	
[illegible] MATERIALS DIESEL FUEL	72-77310-318	72-77310-285	23-C-74073-2		ADL-157TM-6[illegible]	
ULAR MATERIALS		72-77310-285	23-C-74073-4			
RALLOY NOZZLE	72-77310-329	72-77310-287	23-C-74073-7		ADL-157TM-822-6	
[illegible]	72-77310-325	72-77310-287	23-C-74073-[illegible]		[illegible]	
RALLOY NOZZLE Y CRUDE OIL	72-77310-331	72-77310-287	23-C-74073-1		ADL-157TM-823-6[illegible]	
ULAR MATERIALS	72-77310-327	72-77310-287	23-C-74073-4		ADL-157TM-692-C	
RALLOY NOZZLE	72-77310-330	72-77310-287	23-C-74073-6		ADL-157TM-821-6	
RALLOY NOZZLE EC 2A4100 APPLIES	72-77310-331	72-77310-287	23-C-74073-8		ADL-157TM-825-6	

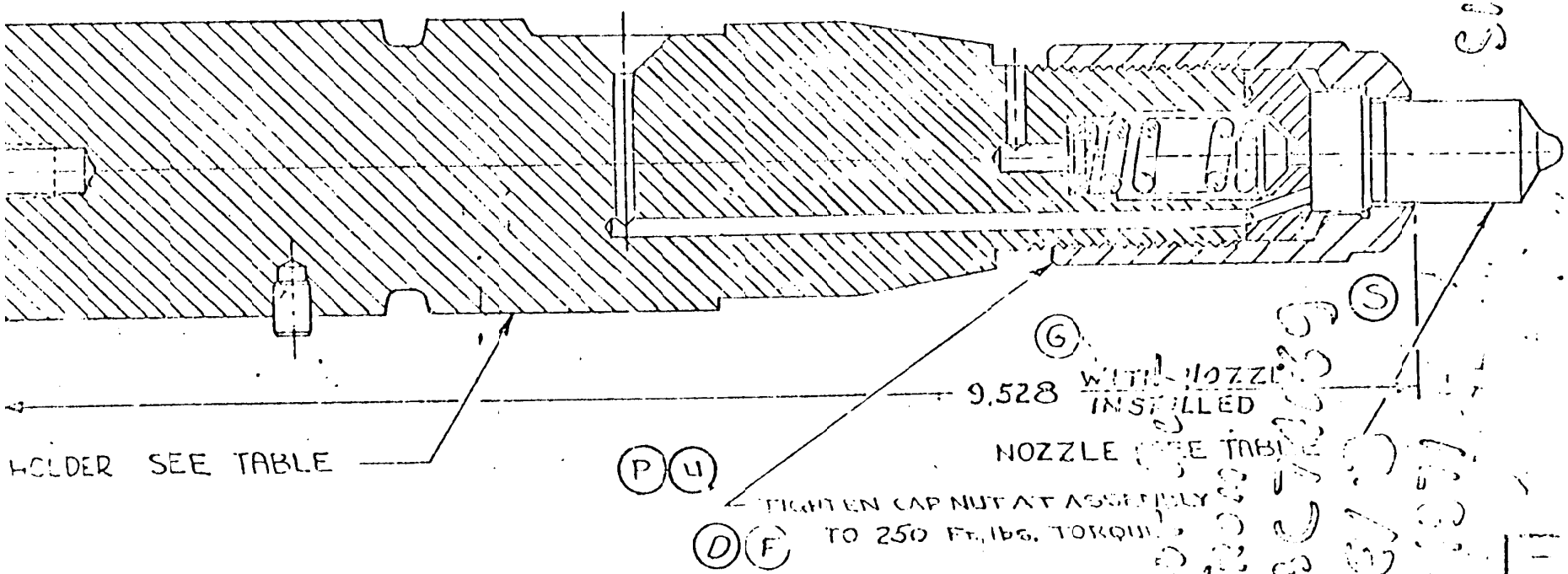


TABLE WAS "REGULAR DIESEL FUEL" (No Notice)	W.B. 2-24-60	(E)	ADDED LINES 4 & 5	R.C. 12-59	(S)
[illegible] 250 FT. LBS.	R.C. 12-59	(F)	REVISED CAP NUT [illegible] WITH DETAIL (No Notice)	A.A.I. 6-12-59	
[illegible]		(G)	[illegible]		

resistant to cracking at one point or another. For long time the excuse for broken nozzles proffered by Bosch and accepted by Alco was excess load due to plugged holes. While this is a very important contributory reason it is now contended that a nozzle must continue to live even if at least two holes are plugged.

One of the most important changes of recent years was the adoption of Nitralloy for the bodies; this change also had a learning curve and its effectiveness seems to depend on rigid quality control of the material and control of the heat treatment. Other recent changes include a slight moving of the three supply holes to give enough wall thickness between these holes and the bore to avoid hardening thru also making the circumferential groove at the lapped joint round rather than square.

Nozzles with the latest improvements were put into production in June 68 and the last digit of the Bosch designation is an underlined 6.

Fuel Pump Test Stand

At one time it was felt that engine failures occured due to unequal fuel delivery and overloading of some cylinders.

Maintainance of injection equipment for the railroads by independent if Bosch authorized, injection equipment service outfits gave Alco engines a bad name, because such dealers wanted to replace and sell parts. Alco, therefore, designed a relatively inexpensive (in comparison to the Bosch product) pump test stand also various service tools, and instructions. The description of the test stand is included in MI 11130 it has three operating stations for the D, CC, and CQ pumps used on Alco engines. It operates at constant speed, but can run in both directions of rotation. In the forward direction the cam flank corresponds to that in the engine, in reverse rotation the cam shape is so modified that the plunger motion at test stand speed simulates that of the engine at idle speed.

To have some absolute means of establishing correct calibration for pump inspection or rebuilding Alco furnished "Master Pumps" and also calibrating nozzles.

While the principle is correct one decision was made years ago which caused Alco customers to be charged for an unjustified amount of labour expended on the master pumps. The time never did seem right to correct this condition.

The principle was: The master pump must give absolutely correct delivery quantity. That could not be done at the two points of near full fuel and near idle fuel. Rather than setting say the full fuel point correctly and stamping the pump with a reasonable deviation found on this particular pump at idle, two master pumps are furnished, one with the rack pinned to give "absolutely" correct full fuel and a second pump pinned to give "absolutely" correct idle fuel. It is actually not the true full fuel but some arbitrarily chosen rack position near full fuel.

To make matters still more inconsistent, one Alco invented calibrating nozzle is used for both full and idle fuel quantities. The nozzle incorporates two orifices and check valves to make either of the orifices effective.

The Alco stand does not give the same amount of fuel as a Bosch calibration stand. This necessitates getting a correctly calibrated pump with carefully recorded quantities from Bosch and running it on the Alco test stand, and using the valves obtained there for any publications for Alco customers.

Injection Equipment Not Made By American Bosch

Injection equipment made by Bendix was tried on 251 engines Ref.-17, but not adopted, this was done for not technical reasons. At a later date Bendix equipment used on C.B. - GE engines was obtained and examined. MEL report 3478 gives a comparison and records useful bench test data also for the Bosch 15 mm pumps.

An accumulator injection equipment to an Alco design was tried, but did not become operational Ref. 32-18.

In the late Fifties Alco duplicated American Bosch injection equipment, both for the 244 and the 251 engine. Bosch parts were measured and drawings made, parts were metullurgically examined. Machine tools were purchased and some 150 prototype sets were made and put in the field. Ref. 32-19.

At this point American Bosch realized that Alco was serious in demanding lower cost injection equipment. Some agreement was reached between the two companies, Bosch took over the special machine tools and reduced price substantially; Ref 32-19 gives information on dimension and materials then established by Alco's own examination.

The injection equipment now offered by Fipco also attempts to duplicate American Bosch equipment, of current vintage in every detail.

References - Fuel Pump Drive Group

32-1 Roller Jumping: Fragmentary(useless) records in correspondence file.
Injection Equipment 1952/53.

32-2 Lubriting: Alco Practice Sheet 99999va
Effect of Inadequate Lubriting MEL 2941
Comparison of Lubrited and not Lubrited Pin, Weekly Lab 1-2 thru 1-21-1968
Lab 2.

32-3 Lub Oil Loss in Fuel Leak Off
MEL 3075,4.5 gal lub per cyl per 1000 hrs.
MEL 3419H,8 gal lub per cyl per 1000 hrs.

32-4 Various Roller Arrangement
MEL 3076,1000 HR lab run, includes crowned roller, needle bearings.
Also Weekly Lab 1-2 thru 1-21-1968, Lab 1, 2, 3 and 4-22 thru 5-19-1968
Tab 1, 2, 3

32-5 High Silicon Bronze Weekly Lab 1-21-68, also 4-22 thru 5-19-68 tab. 4

32-6 Various wear data on floating bushing and related parts
MEL-3312-C, 1158 hrs. lab 6-cyl std part
MEL-3470-A,3 year switcher operation std part 6 cyl
MEL-3390-D,1030 hrs 12-cyl D-engine up to 1200 RPM 243 MEP mostly
std parts

32-7 Special floating bushings MEL-3346,see also Project 863165

32-8 Plated bushing 3375E, Honey Chrome Pin 3375 catastrophic wear

32-9 Lube oil conventional vs. high dispersant, memo McGowan to Tromel
12-17-68 Lub Oil File.

32-10 Support Casting MEL-3510, 3510A, 3510B

32-11 Fuel pumps on top of block; G.O. folder 801013; large package of prints
in F.C. Tromel file cabinet.

32-12 Screen in, inlet filtering Weekly Report June 4, 67 Dec. 8-67

32-13 Snubber Valve MEL 3032, 3114, 3489, 3218

32-14 For Line Pressure see Bosch Test Reports Scattered thru Correspondence Folder
Fuel Injection Equipment 1966, 67, 68.

32-15 Injection Tubing MET 3246 ID Compound MEL 3401

32-16 Stud Up Grading MET 3116

32-17 Early Tests of Bendix Equipment MEL 2998, 2998A,B,C
Late Comparison of GE-Bendix and Alco Bosch MEL 3478.

32-18 Alco Accumulator Injection System MEL 83019

32-19 Alco Duplication of American Bosch Equipment
Pump Assembly Drawing 32A71064
Nozzle Drawings 23C74073 thru about 23D74082
Metallurgical Examinations MET 3026
Bosch and Engine Test MEL 3314

GEARING GENERAL

All spur gears used on the 251 engine, have 20 degree involute, full dedendum, full radius root teeth. All are hobbed and most are shaved after hobbing. On all shaved gears it is the practice to shave one gear in a pair straight, the other eliptoid. Gear dimensions are calculated per applicable Alco standards ref. 35-1. They are measured with Illinois Tool gage blocks rather than with pins. All gears are either machined all over or statically balanced.

CAMSHAFT GEAR DRIVE

The camshaft gear drive was originally based on 6DP helical gears with a 13° helix angle. This arrangement operated fully adequately on the 6 and 12B engines. Even the governor drive, spur and bevel gears performed unexceptionally.

The basic design features were established then and never changed; they include:

A split crankshaft gear, the split follows a tooth root, then at the helix angle, now straight. Four (4) hollow hex capscrews, 2 each threaded into each half, hold the gear together.

The camshaft gear with a taper fit on the camshaft and anut with a special locking-off feature. The right and left side camshaft gears were then identical; now on some engines which have neither flywheel or generator fan to carry timing marks, the right side camshaft gear sometimes has to have stamped engine timing marks.

The ends of the teeth in mesh are not visible; meshing the gears in correct position is done by setting the crankshaft at TDC #1R and lining up markings on the camshaft gears (same on both gears) with the sides of the cylinder block.

These gears were made 86B45 at 302 to 340 BHN crown shaved crankshaft and straight shaved camshaft gear.

The center distance between camshaft and crankshaft gear is fixed so that there is no adjustment of backlash possible. Original backlash figures were calculated on the high side to permit raising the crankshaft bore for salvaging by .010" (by recollection) and still retain workable backlash. Backlash depends also on whether the camshaft lies on the bottom of the bearing or whether the fuel pump lifter of the last cylinder has raised the shaft to contact the top of the bearing.

The lately introduced nitriding causes a slight growth, which is quite noticeable on the large camshaft gear. To counteract this the web is painted with a suitable compound to prevent the web from getting nitrided; tooth thickness also was reduced by .0015" to counteract such growth.

When 16 cylinder engines got into more severe service on Santa Fe R.R. locomotives serious failures began to occur epidemically. These failures were clearly related to [illegible] torsional vibrations, on each cam gear eight equally spaced groups of teeth showed substantial wear, on both flanks. On the crankshaft gear wear occurrence was less clearly marked but breaking of teeth near the root was more frequent.

Outside consultants Buckingham of MIT and Bjoernberg of Illinois Tool were consulted in; they disagreed violently on proper approach. Buckingham strongly recommended severely shortened addendum on the cam gear. It turned out that

space limitations of adjacent parts made straight 4 pitch gears the only feasible choice. Even these gears had to have their addendum shortened by .020" to permit the crankshaft-generator coupling bolt heads to pass by the OD of the crankshaft gear. Both expertsagreeed on lack of particular merit of shaving the gears but did recommend shot peening and lubricing to improve break-in conditions. Dynamic gear loads were concurrently reduced by development and introduction of the camshaft vibration dampers. With these changes gear wear and breakage problems disappeared for about five or six years, so much so that shot peening of the camshaft gears was discontinued although crankshaft gears shot peening was kept. There is one hidden benefit in shot peening, in order to peen all sharp corners e.g. intersection of tooth root with the sides of the gear must be carefully rounded, Alco calls for this as a special operation.

As previously mentioned the camshaft gears are identical. The key location in the camshafts was so arranged that the camshaft gears with their marking lined up with the sides of they cylinder block would place the cams in correct angular relation with the crankshaft. This had been worked out on a layout matching angles between the meshes with certain number of tooth speces. When changing from 76 helical 6DP teeth on the crankshaft gear to 52 straight 4DP teeth, this relation became upset. Key location was compromised so that the right bank camshaft was slightly retarded and the left camshaft slightly advanced relative to the crankshaft. Total difference between the camshafts about one crank degree.

Toward the later Sixties gear wear and failure complaints re-awakened now strongly on river boats with high load factor and also on speeded up, uprated locomotive engines. Nitriding the gears proved a very effective correction. Changing the material from 86B45 to the higher chrome containing 4150H enhanced the gain in strength by nitriding.

In Canada where suitable nitriding facilities do not exist, TUFFTRIDING was substituted. It appears to perform as satisfactorily as nitriding, but some uncertainty exists because MLW added in an unknown number of cases outboard bearings beyond the camshaft gear, which also reduce gear dynamic loading substantially.

On the 18-251 engine the face width of the gears has been increased from 2½ effective to 3" to enhance resistance to failures. All other features have been retained.

The usual way of lubricating this gear train has been by spitters spraying oil on top of the camshaft gears disregarding differences in direction of rotation. There is no proof that this method is in any way deficient. During the time of gear failures, just before the nitriding of the gears was found to be effective a rather complicated system of gear sprays into the meshes was intorduced. It consists of o line coming thru a bulk head fitting in the top deck of the cylinder block, the other out of the end of the lube oil header and is clamped to special pads welded on the base, both lines ending in spitters next to the two gear meshes.

This sytem is now specified for both the 16 and 12 cylinder F engines, whether it is justified or not is open to question.

Governor and Tachometer Drive

Tachometer generators were driven by a spur gear meshing with camshaft gear. The same size gear drives the governor thru an additional pair of bevel gears. No difficulties with either drive is recollected with 6 pitch gears, at least not on 6 & 12 cylinder engines. When the 4 pitch gears became operational and the spur gear for governor and tachometer had to be changed to 22 teeth of 4DP, failure of the spur gears on 16 cylinder and 6D engines became frighteningly prevalent. Frighteningly because one tooth broken on the governor drive gear more often than not caused the whole gear train to fail. It frequently elongated or cracked also the last camshaft bearing block in the cylinder block.

The usual first attempts to improve mechanical properties of the spur gear steel were predictably useless. Flexible drives were suggested, several were tried on paper, one was made and applied on a lab engine. In the meantime, relief was found by using gears made of Textolite. Some of them lived for well over a year some lasted only several months. (It was then suggested that phenolic laminates of this kind have much better properties if they had been for half a year or longer in storage than fresh stock). In any case engines could be operated with such gears without fear of major casualties to the gear train. Gradually these gears were replaced by nylon gears. If certain peculiarities are recognized such gears give respectable performance with acceptable life. They have a high coefficient of thermal expansion, about ten times that of steel and they absorb moisture. Current, apparently satisfactory practice, is to boil the gears in water for 10 minutes, just before shrinking them on a shaft. Before introducing the shrink fit and boiling in water, loosening of the gear on the shaft and hammeri out of the keyway was a frequent occurrance. Contact pressures must be kept low. To reduce frictional wear the addendum is made only 0.4 of the standard addendum. It is not recollected how this figure was arrived at, but it works. To compensate for the high thermal expansion cold back lash must be set high .025" to .030".

While Alco can and does cut all spur gears, bevel gears must be obtained from outside vendors. The governor drive bevel gears are based on the Gleason Zerol System. If the backlash is set correctly and they are driven by the nonmetallic spur gear one never hears about them.

Currently we are experimenting with nylon gears bolted to a hub to overcome loosening.

FREE END GEAR TRAIN

When the 251 engine was first designed reasonable sized openings were provided in the side sheets of the base in the free end compartment to permit inspection of the gears without excessive removal of parts. On all locomotive engines up to recent complaints by Canadian railroads on 16E and F engines, which are not duplicated anywhere else, the free end gear train was so free of any deficencies that the inspection openings have long since been eliminated. These free end gears were always 5 pitch, straight spur, shaved, the extension shaft gear crowned the others straight. Material 86B45 in the 302 to 340 BHN range.

On marine engines with additional raw water pump above statement does not apply with equal force, some serious difficulties with the idler have been experienced at one time. In this case an idler gear is meshed with the lube oil pump gear. The idler is rotating on antifriction bearings on a stationary pin. The pin is clamped in a bracket fastened to a special flange on the side of the base; the bracket also supports the raw water pump at a controlled center distance from the idler pin. When first developed it was realized that the lube oil pump drive gear would get about double the tangential load at the mesh with the extension shaft gear and in addition would get the teeth loaded in reverse bending at the idler mesh. To compensate for this load increase shot peening was specified. This gear never showed any primary distress. The shot peening was recently changed to nitriding, this is considered more effective, costs about the same and is the method now adopted by Alco as standard gear enhancement.

The failures which did occur, probably on all installations of the original design, were due to lack of rigidity of mounting the idler pinion, and lack of proper alignment, between idler and lube oil pump gear. Since the mounting of the idler was improved no significant number of deficiencies have been reported.

The gear train is lubricated by feeding oil from the lube pump into the dammed up rim of the pump drive gear which has holes drilled in six equidestant tooth spaces, so that oil is thrown out by centrifugal force all around. The mesh between raw water pump gear and idler has its own spitter since this area is shielded from the general spray.

Free End Gear Train With Increased Pump Speeds

During the developmnet stage of the steel cap pistons the need for high lube oil header pressures, 80 psi minimum was over estimated. There also were some engine on order with 750 RPM; in addition a demand for substantial increase in lube oil capacity for the 18 cylinder was anticipated.

Consequently a gear train was developed which gave approximately 12% increase in pump speeds but which as a set is fully interchangeable with the original gears. These gears are currently specified on the 18 cylinder engine, and thus have become a production item. Their use should be considered on any 16 cylinder engine operating at speeds 900 or lower. At 750 RPM installations it should be used also on 12 cylinder engines. Service experience on this gear train is limited at present to one severely pitted lube oil pump drive gear which was examined at Auburn and led to the decision to call for nitriding all gears in this train. It should, however, be remembered that the gear came from a pump on which misplacement of the oil pressure relief of the pump impellers had created excess loading on the drive.

8 CYLINDER ENGINE GEARS

The eight cylinder engine requires a drive of the two balance shafts at double crankshaft speed. The original design which was abandoned for the first order of locomotive engines but readopted before executing the 51 engine order for the LST drove everything from the extension shaft gear. Number of teeth and center locations differ completely from the standard drive. The gears do remain 5 pitch straight spur with shaved teeth. The design is arranged so that all gears transmit power, so that there is at least a tendency to keep the teeth in contact. The extension shaft gear drives the lube pump gear, which drives one balance shaft gear, which drives the second balance shaft gear which drives the water pump.

Gear noise and vibrations on the laboratory engine led to a redesign which transferred the drive to the generator end. An idler is mounted in a bracket in the base which meshes with the split crankshaft gear and a gear on the right side balance shaft. This shaft carries a gear also at the free end which drives the lube pump gear and the other balance shaft gear. This again drives the water pump. The extension shaft gear is omitted. The first locomotive orders, about 15 engines, were delivered this way. One idler gear failure was reported otherwise these engines have now been in satisfactory service for five or six years. The really serious draw back of this design is the virtual inaccessibility to the idler, its bracket and bearings.

It is possible to replace these parts with the engine in the locomotive, but it is a typically desperate design. On realization of this draw back the design was reconsidered and about four locomotives were delivered with drive from the front end. No reports on them have been received. The LST engines are equipped with the front end drive and also have the drive for the raw water pump. Examination of two such drives after one year's operation is reported to have shown the need for replacement of all gears except the lube pump gears. The LST engines operate at 900 RPM constant speed while the locomotive engines operate at various speeds with 1000 RPM maximum. At 900 RPM rather high torsional vibrations are said to occur, which together with cumulative backlash in the sucessive meshes appears to impose severe loads on the teeth. To correct this wear conditions, nitriding of the gears has now been specified, it will be about the end of 1973 before the influence of nitriding can be properly evaluated.

On this drive there are two featues which may accentuate or possibly cause excess gear wear. The small gears on the balance shafts are identical and both have crowned teeth. The other feature is that these two gears are protected from dipping into lube oil in the base by a small trough, which is supposed to be kept empty by the gears sweeping out all oil. Looking at this design on paper raises the suspicion that the trough effectively prevents the gears from getting any lubricant on the teeth.

SPECIAL GEARS

Also does machine starter ring gears, they are based on SAE standard, 6/8 pitch. The ring gears are cut into three equal interchangeable sections and can be replaced easily. Early failures were due mainly to insufficient rigidity of the starter motor mounting. Since this has been corrected no failures have been reported.

On some engine-generator sets with electric starting and no fan disk a barring device is used in which a small pinion with its own bearing support is temporarily mounted on the right side camshaft gear cover. The pinion engages the camshaft gear. Although only manually operated the tooth load can be very high and may damage the camshaft gear. Use of this device should be discouraged.

Mentioned under section lube oil pump are the impeller and idler gears cut for 30° pressure angle teeth, straight teeth for the new 18 cylinder pump, helical teeth for all previous pumps. Neither manufacturing nor service departments have questions or report deficiencies on these gears.

Governing References

35-1 Alco Standards Sheets

Gear Calculation 15C8350A & E

Full Radius Root Tooth Form 01P4570

Measuring Over Gage Blocks 02P6380

Shot Peening Gears 01P6200

Nitriding 31P5302

Tufftriding 99999XC

GOVERNOR AND CONTROLS

G.E. Governor Features

Virtually all 244 and all early locomotive 251 Vee engines were equipped with the G.E. Electro-hydraulic governor. The Woodward governor was accepted as Vee engine locomotive standard in the early sixties but certain railroads, including SP & S as well as the Indian railroads, have continued to use the G.E. governors. Design and operation of these governors was controlled by G.E. and the Alco electrical engineerir group. The engine group's concern was mainly limited to application; in consequence of which there are virtually no data on notch loads and other design features available.

This governor consists of three units. The governor proper, a permanent magnet tachometer generator and an independent electric motor driven lub oil pump with suitable piping.

The governor itself houses a mechanical portion with linkages, cams, even a collapsib link; a hydraulic section including a servo motor, a small oil reservoir with regulating valve, and also an electrical section including solenoids and three rheostats. The governor does not require any drive. Originally, the governor was mounted on the side of the turbo support similar to the mounting on the 244 engines but locating it on top of the block above the left camshaft gear reduced control shaft vibrations and permitted to have similar installation conditions as the Woodward governor, which originally was used on marine engines only. The G.E. governor has gone through several model changes, latest model believed to be MG8. The independent lub oil system consisted of a double ended electric motor, with one Tuthill pump mounted on each shaft end. One of these pumps served as fuel transfer pump, the other to provide pressure oil to the governor. Like all other locomotive motors, it operated on 72 V.D.C. and controls were so that the motor had to run before the engine could be started. Fuel tank and filter arrangement in the locomotive demanded location of this pump near the free end which forced the acceptance of long oil pipes along the side of the engine, which obviously had to be of large diameter (3/4 and 1" by recollection) to limit fluid friction, requiring careful purging at first starting but apparently no later difficulties arose.

The tachometer generator, which is also used with Woodward governors depending on which locomotive excitation system is installed, is mounted on the right cam gear cover and driven by a Nylon gear meshing with the camshaft gear. There were at one time various defects caused by torsional vibrations, shaft breakage, loosening of the rotor on the shaft also damage to the rotor itself. It was overcome by making the parts more rigid and by adoption of the Nylon gear. Torsiograph tests were made but no meaningful data were ever developed. No failures have been reported in recen years.

The output torque of the governor shaft is 130 lb. inch. In looking at the fuel pump racks when a G.E. governor is used one notices that the racks seem very steady, control appears to be done mostly by changing excitation. All actions of the govern are rather fast although the latest version has some delays built into the hydraulic circuit. During development of overspeed mechanisms, it was noted that under certai circumstances (transition?) the governor output shaft and with it the whole linkage will go to zero fuel and back to some fuel position so fast that the motion is hardl visible to the eye.

On railroads who became used to it, the G.E. governor performed well; but on other roads the need for cooperation between plumber, electrician and mechanic, all of them involved in the governor, led to operating and maintenance complications.

Alco replaced the G.E. by the Woodward governor primarily due to preference by many domestic railroads. EMD used Woodwards and test stands as well as people trained to handle Woodward governors were widely available. Due to costing of the whole electrical equipment, including the governor, for one locomotive by G.E., and only minor deductions, if the governor was omitted it was never clear whether the Woodward governor represented a cost advantage or not. Alco was not able to present sufficient technical reasons to convince India (DLW) to adopt Woodward. It is interesting that G.E. on their own locomotive have never used a G.E. governor. It is rumored that the fast action of this governor caused intolerable stresses on their original articulated connecting rod.

At one time it was tried to reduce the speed of governor action by dashpot, but dashpots have the feature that the resistance is zero at the beginning of a motion and gradually increases. This is just opposite of the motion wanted for engine control. The acceleration control device, to be discussed separately, is applicable and effective with a G.E. as well as a Woodward governor.

For details of G.E. governor, see Ref. 36-1.

Woodward Governor Mounting

There are two forms of locomotive governor bases available; one with a spline drive and one dubbed Alco base originally developed on the old SI governor for the 539 Alco engine with a downward extension, and a keyed and threaded shaft end to mount a gear directly on the governor. This type builds lower and therefore was and is used on the 6-cylinder engine to fit into low profile locomotive hoods. It also makes for a cheaper drive but does require checking of gear backlash (shimming of the governor) at original installation and at governor replacements.

This type was also used on the early Vee engines, both for locomotive and marine application. On Vee engines it was replaced by spline drive governors which require an additional housing with a vertical shaft in two ball bearings to carry the bevel gear on the bottom, and a splined sleeve at the upper end. The locomotive standard uses ball bearings but there exists also an earlier design with bronze bushings which fits some UG-8 or UG-32 governor bases for generator or drives. The spline drive permits setting backlash once and for all between the bevel gear pinion lower and upper housing; at governor changeout no attention need to be paid to gears. The spline and flange connection is the same as used on governors for EMD engines.

The lower support casing, which carries the horizontal lay shaft with the bevel pinion and the spur gear which meshes with the camshaft gear, is the same whether a governor with "Alco" base is applied or the intermediate casing used. The horizontal shaft and all gears are the same for Vee and In-line drives. There are two sets of bevel gears available to compensate for differences in rated engine speed. The Nylon gear, which occasionally wears more than desired, overcame epidemical gear failures (one broken tooth was likely to ruin the whole cam gear train) on 16-cylinder and 6-D engines.

For the high shock LST engines special "Alco base" governors were used because the parts mounted in this location are subject to very high shock loading, and the reduction in height became very beneficial.

Difficulties in the Governor Drive

Drive complaints include loosening of the Nylon gear on the shaft; this was largely corrected by recognizing that Nylon has a very high coefficient of thermal expansion, so that the originally normal tight fit between shaft and gear became very loose at operating temperatures. Originally there was no good way to hold the pinion shaft from turning when tightening the shaft nut. Now there is a 1/2" square recess in the pinion to hold the shaft. For the old shafts a fixture was made which grabs the teeth of the pinion for this purpose.

Occasional complaints are received on wear of the spline or spline bushings. Experimental Nylon bushings were tried in the field; they seemed to have worked well but no reliable feed back was received. A method of introducing an oil spray appeared satisfactory on engine test, is now due for field test. Oil spray were applied to the bevel gears when they seemed to wear excessively. This was overcome by the use of the Nylon spur gear. For normal operation oil from the cam gear sprayer appears to be sufficient.

A number of design studies of flexible drives were made, The only one seemingly feasible using synthetic rubber balls between two disks with conical recesses was actually tried, it operated as intended but fretting wear between pilot and gear condemned it.

In 1960 the first non-metallic gears were made of Textolite (Phenolic laminates). The drive became smooth, wear of the textolite gear was excessive in many cases. Introduction of the Nylon gear resulted in acceptable if not perfect life. The Nylon gears are cut, after original attempts to use cast Nylon gears with the teeth formed in the mold, proved unsatisfactory.

All governor supports include a collapsible link mechanism. On the Vee engines this is based on torsion springs, both for G.E. and Woodward governors. The G.E. governor actually has a collapsible link feature built into the governor but this had not been realized at the time. On the 6 In-line engine the mechanism is based on a compression spring.

Woodward Governors; Basic P.G. Governor

This discussion deals primarily with locomotive governors and will avoid duplicating any information given in the various Woodward bulletins, and also Alco MI's. The Woodward bulletins give complete basic information but they are sadly incomplete when special features like fuel limiters and altimatic controls were applied. For this reason the locomotive builders, including Alco, had to develop manuals pertaining to the governors used on the particular engines. Ref. 36-2 lists basic Woodward and Alco bulletins.

The 6-cylinder 251 engine was from the beginning controlled by Woodward governors. Originally they were PG governors without load control. At first there was some kind of fuel limit lever used which could never be kept in adjustment. It was soon abandoned and a time delay relay (AGASTAT) was incorporated in the excitation circuit. Some turbo surging was experienced in notching down and the games of changing the hole sizes in the speed setting pilot valve began. The smallest hole ever used on production engines was .039", the largest 1/8" which is suitable only if speed or load increase is controlled by something else like an acceleration control device. .046" diameter is about the most useful size if no other delaying features are used.

Speed Setting

An early order for 6-cylinder locomotives on the Boston and Maine and one other installation used pneumatic speed control, all other locomotive governors have the speed controlled by four solenoids. The arrangement of the solenoids is such that energizing them in rather well established combination gives 7 equal speed increments between idle and top speed at the eight standard locomotive notch positions. In this connection it must be kept in mind that all domestic locomotives must be able to operate in MU (multiple unit) with locomotive units of the various builders indiscriminately hitched together. If the notch control lever on the leading locomotive unit connects a certain set of wires to the solenoids, all locomotive units must respond in the same way. It is possible to modify the controls on one unit by electrical gadgetry to jump a particular notch speed. The original 18-cylinder locomotive governor and controls were so built, to avoid a certain speed at which excessive chassis vibrational response was expected. Fortunately, the vibrations did not materialize and the standard speed sequence could be used. It was understood that one G.E. locomotive had to have such an arrangement. There still would have been a response to the notch position by difference in excitation in the two notches even if the speed was kept the same.

In attempts to reduce acceleration smoke, a 16 notch control was tried on the UP. The normal eight notch speeds were retained but for each notch speed two excitation levels were provided. The gains did not justify adoption of such complications.

Load Control

The original PG governor provided the fuel rack position to maintain the set notch speed regardless of load. Note that in these discussions speed is always understood to be engine speed, which is virtually unrelated to locomotive speed.

To improve operation, primarily smoke reduction during acceleration, also protection against engine bogging, the load control governor was adopted for the 6-Cylinder engines in 1957; at about the same time various trials with Woodward load control governors on Vee engines were started.

The basic feature of the load control governor is that a definite rack position is assigned to each notch speed. If the engine cannot operate at this speed with the specified rack, a hydraulic servomotor will turn a rheostat which will increase or decrease generator excitation; that is, if the engine at the given rack wants to run faster than the specified notch speed the generator load on the engine will be increased.

In all Alco installations the rheostat is part of the governor, EMD uses the same governor mechanism but servomotor and rheostat are separate units not furnished by Woodward.

Alco used two of several available rheostats. On the 6-cylinder and now also on Vee engines, the 30.5 ohm rheostat with tapered characteristic is used. The Vee engine governors up to 1966 used the 10.7 ohm rheostat with straight line characteristic. The different values 30.5 and 10.7 ohm are insignificant because the excitation system can be arranged to respond equally to either. Significant is the difference between taper and straight line characteristic. On the straight line each degree of rheostat movement corresponds to a constant increment in resistence. On the tapered the resistance increment is small at the beginning of the rheostat travel and gradually increases to a maximum at the end of the travel. During operation the movement of the rheostat can be observed by watching the indicating dot in the round rheostat window.

Under steady state condition, when the load is controlled by the governor, the dot may be in any position in the window; if one cylinder were cut out, or the cooling fan is switched on, the dot moves towards reduction of excitation. Note a frequently confusing wording: one talks of minimum and maximum field excitation corresponding to minimum and maximum load on the engine, but this always opposite to the maximum and minimum resistance position of the rheostat.

In idle position of the engine the load control may be arranged for maximum or minimum field start, which affects locomotive handling. At this time only the governors on the 6-B have minimum field. All others have a maximum field but the maximum field is actually negated by energizing the over riding solenoid (ORS). The effect is when the engine is idling the load control is at minimum field, due to the ORS. As soon as the control lever is moved out of idle to any notch, the ORS is de-energized and load control goes toward maximum field; but this motion is delayed by hydraulic orifices and especially with tapered rheostat load is applied gradually. Depending on locomotive control system, the ORS can be touched at any load increases which quickly reduces excitation and rebuilds more gradually. Load control can decrease field current by a certain percentage. Originally this was in range of 20% of full field current; this percentage was increased substantially, limited by occurance of instability.

Degree of load control, adoption of the tapered rheostat, using of ORS and establishing load control timing is based on experimentation in the middle sixties to reduce acceleration smoke on 16-cylinder engines with single duct manifold. At this time only Alco became aware of the possibility of operational improvements by integrating the work of the engineers responsible for electrical controls and engine and governor operation.

Acceleration Control Device

Discussion of Woodward governor development is interrupted at this point by description of the Alco-MLW acceleration control device. This device pointed the way and acted as a goad in the development of the later Woodward fuel limit governors.

It was first developed to help in switching operations of Canadian built 6-B engines, which were substantially more difficult to handle because they were equipped with a Canadian made modification of the Vee engine generators instead of the much smaller (lower inertia) generators used by Alco. The device consists of a spring loaded piston which actuates a cam. The piston is exposed to air manifold pressure, the cam restrains the motion of the fuel pump control shaft, thereby, overriding the governor by overcoming the springs in the collapsible link. The shape of the cam is strange and developed strictly by cut and try methods. In addition to responding to the difference between air pressure and spring load, it also is influenced by the forces which the collapsible link spring exerts on the control shaft. It must not only impose limits on the control shaft travel, but these limits must be set so that at each individual steady state notch position, the cam is far enough away from the control shaft to allow some rack increase to permit speeding up the engine to the next notch.

Alco started out with the MLW device when first testing the single duct manifold on the 16-cylinder engine in 1961. After some completely unsuccessful attempts to design the device rationally, cut and try methods were again used on an engine equipped with a G.E. governor. The device was later also checked out with Woodward governors. The principal requirements, which are met by the basic device shown on Ref. 36-3, are the need to have enough fuel for starting and for accelerating from idle speed. This initial position is 15 mm nominal. A very small increase in manifold pressure must allow more fuel. To make the piston respond with a minimum of friction, a Bellofram diaphragm is used and a very light spring. After the engine speed and load has slightly increased, the turbocharger quite suddenly begins to build up pressure more rapidly. In the control device the piston has travelled a certain amount against the resistance of the light spring. To match the sudden rise in manifold pressure a second much heavier spring is contacted. The device still restrains the control shaft until manifold pressure reaches about 5 psi, when the cam gets out of the way completely. While single springs were tried only the double slope of the force of two springs could be made to work.

On all later executions, the contact between roller and cam is so arranged that the line of contact passes close to the cam center at the initial position. Forces exerted by collapsible link on the control shaft can influence the cam position only very little while air pressure and piston spring force is low.

The cam is shaped so that the increment in fuel rack increase is very small per degree of cam rotation in the beginning of the travel, but increases rapidly as the cam moves. This allows another adjustment of the mechanism by making the roller contact higher or lower on the cam, thereby, changing the rate of fuel increase.

This device is admittedly a crude affair; one clear shortcoming is that the piston rod exits the housing on top. Dirt, water leaks from the headers above have caused the rod to stick. Any redesign should reverse the position. The clevis on top of the rod must be able to swivel, otherwise the diaphragms are easily torn when making adjustments. After the Woodward locomotive governor became equipped with an effective fuel limiter, the incentive to optimize the acceleration control device was reduced, although it is still specified in many installations. Designs and kits for application to virtually all engines, including special 6-B and 6-D, also 16-244, have been worked and applied in the field.

Woodward Absolute Pressure Bias and Fuel Limit Governor

In 1960 development was started of the 6-D engine to operate at over 240 MEP and 1100 RPM instead of the then current ratings of 200 MEP at 1000 RPM. In order to reduce temperature of manifold air, an air to air aftercooler was adopted, other changes were nearly insignificant insofar as they would have increased load carrying ability of the various components. The intended approach was that the engine if in proper working order, had enough margin to accept higher loadings and that control devices would be introduced to protect the engine from operating at high loads if some equipment should become defective. With this in mind the absolute manifold pressure bias governor with fuel limiter was adopted. Ref. 36-4.

If the manifold pressure should drop for any reason fuel would be reduced. (Since the early gasketing of the aftercooler was deficient and substantial air leaks occurred frequently this feature of the governor was put to good use). Also incorporated was a temperature sensing "SOLRAC" valve which was so connected that it simulated a low air manifold pressure signal to the governor calling for reduced fuel. About 50 engines went out with the Solrac valves in each of the exhaust manifold pipes but could never be made to work reliably. Even if they should have worked reliably, they could not perform as intended because the Alco engines have a very flat curve of turbo inlet temperature in relation to load. These valves were scrapped and no other device to protect engines from high exhaust temperatures became operational on any Alco engine. (The tunnel operation protection used on the S.P. uses ambient temperature as a signal).

This governor does include a fuel limiter during acceleration, in which a piston location controlled by absolute manifold pressure thru a cam over rides normal governor function and restricts increase in fuel supply. It is therefore intended to act like the acceleration control device described in preceeding section but the cam is small, difficult to adjust and the linkage not sufficiently rigid to perform its function without uncontrollable deflection. The limiter does function but not too reliably or effectively.

The basic feature of this governor, the fuel limiter and fuel reduction when manifold pressure drops should be considered side issues in that fuel rack position in the various notches does not follow a straight line relation between no fuel and full as on the basic load control governor but instead is influenced by air manifold pressure.

Actually, if the tube to the air manifold is disconnected or if the turbo were blocked, there is a straight line increase of fuel rack from no fuel to a very much reduced max fuel position. But if turbo charger is operating the air pressure thru a special device causes changes in the internal governor linkage so that the rack position is substantially increased.

The resulting notch load curve is therefore not any more a straight line but resembles closely a 4th power of the speed curve. What the actual load in the various notches will be depends also on the locomotive excitation system; whichever excitation or governor setting gives the lower power, is the ruling influence.

An inherent consequence of giving fuel in normal operation in some dependence on air manifold pressure accentuates the normal variation, say up to 2.5"Hg between performance of different turbos. If a turbo is especially good, i.e. gives high discharge pressure, it will be penalized by getting more fuel, while the less good turbo gets less than normal fuel and the engine performance falls further below par.

The 6D engine had been developed and tested with a 10 or 11 sq. inch nozzle ring in the turbo and at full load the manifold pressure was 54" Hg, and the governor

was set accordingly. When engine was finally tested in the locomotive it took up to 10 minutes to develop full power because the engine needed full fuel to develop full power but could not get full fuel until full 54" HG manifold pressure was reached. At that point the nozzle ring was reduced to 9 sq. inch and the Fisher manifold relief valve was permitted to blow continuously to keep manifold pressure at 54"Hg. In this way the full load was quickly reached and the first order of Indian locomotives with 6D engines was delivered.

This governor appears to have been designed, although disclaimed by Woodward, for the Roots blown EMD engine. The idea to modulate fuel to absolute manifold pressure was valid for this engine and gave a certain amount of altitude derating. It was worse than useless as an altitude derating device on a turbo charged engine where the turbo charger, if left to itself, attempts produce nearly constant absolute manifold pressure by increasing its speed to the point of self destruction. But in the case of low altitude operation in India this consideration did not apply.

Mentioned, without further discussion is a change between earlier and later governors of this style, the adoption of the two slope linkage. It was introduced to increase the very low power in the lower notches. When correctly adjusted it performed well, but the additional gadgetry did create an additional potential (and actual) cause of defects.

Woodward Pressure Bias and Altitude Derating Governor with Fuel Limiter

The "absolute" manifold pressure sensor used on the governor described in the preceeding section consisted of a stack of two equal diameter bellows. The inside of the upper bellows is connected to the air manifold, the lower bellows is fully evacuated, that is an aneroid bellows. The upper and lower ends of the bellows stack are fixed (upper end adjustable) the location of the joint between the two bellows varies according to the "absolute" manifold pressure and to this point the mechanism influencing governor linkage is attached.

When Alco insisted that controlling fuel delivery by absolute manifold pressure was not satisfactory and that the reduction of fuel must be made much more dependent on altitude to limit turbo speeds the areas of the two bellows were made unequal, the aneroid bellows retained the original effective area of 0.3 sq. inch while the upper manifold pressure bellows affective area was reduced to 0.16 sq. inch; the spring under the sensor piston and certain settings were changed to compensate for changes in the bellows stack. In all other respects the governor remained the same as the pressure bias but influence of barometric pressure, that is of altitude was increased roughly in the ratio of the areas.

The first governors of this type were installed on 12 cylinder DL543 locomotives on the Southern Railway ofPeru. This road had been using for several years successfully 12 cyl. locomotives with G.E. governors permanently derated to 1320 HP. Performance of the engines with the Woodward derating governor was unsatisfactory from the beginning. One of the most serious objections was that low notch loads varied with altitude, and while the governor was set for maximum field start at sea level, the mechanism was such that this changed to minimum field start at high altitudes. Since the trains had to be able to stop and start at heavy grades minimum field start and reduced loads in the lower notches caused intolerable train handling difficulties. The location of the fuel limiter cam changed with altitude so that the rate of fuel limiting differed at different altitudes, meaning that it did not prevent belching of black smoke when accelerating at altitudes.

Conditions were aggravated by lack of governor test stands at Alco and in Pern and lack of clear instructions. Woodward leaned toward their known policy according to which governors are perfect and must only be touched on test stands at Woodward plant.

In time instructions were worked out to make a number of adjustments statically or on engine at idle quite satisfactorily. A Woodward engineer was dispatched to Peru and made the governors work after a fashion; to counter act the variation in air pressure between different turbos, the Fisher valve was set to blow off at the pressure the least good turbo was expected to deliver.

About concurrently 6D engines were to be delivered to the Peru Central and the Southern as well as to Chile which also had to contend with substantial if less seve altitude variation than Peru. Similar difficulties as on the 12 cylinder engines arose when the DL535 locomotives with 6D engines became operational.

By recollection no MI covering these governors specifically is in existence, the absolute pressure bias 36-4 governor and Woodward test instructions are probably the best references available.

Woodward Altitude Derating Governor

Soon after the previously discussed 12 C engines went to the Southern of Peru an order for several 16 cylinder engine locomotives for the Central of Peru was received. This has the most severe conditions of all; the track rises from sea level with a practically uninterrupted 4.5% grade to 15600 ft. and at the top is a 3000 ft. long tunnel. At the time the order was received some of the experiences with the 12 C engines were available.

It was, therefore, decided to use the governor described in the two preceeding sections but to remove the fuel limiter end to make ineffective the influence of air manifold pressure. See reference 36-5. The regular Alco acceleration control device was applied.

The locomotive excitation system was relied on to limit loads within reason at all lower notches.

This provided a satisfactory solution and was ultimately retrofitted on the 6D engines in Peru and Chile.

A pertinent experience on the 16 cylinder engines was a succession of turbo disk failures (720) virtually on the same spot in the tunnel at the extreme altitude. The turbo failures were corrected by increasing the nozzle ring, but the condition triggering the failure at this particular spot was later found in a peculiarity of the electrical controls, which is probably quite universal. Depending on locomotive speeds and load the connection of the various traction motors varies from series to parallel and mixtures of the two. During transition, that is during the automatic change from one connection to another the engine is instantaneously disconnected from all load accompanied by an erratic movement of the racks. The time for this process is so short that the rack movement is overlooked unless one observes carefully and waits for something to happen. The combination of load, train speed, altitude, turbo temperature increase due to the tunnel and momentary loss of control during transition caused the turbo failures at the particular instance.

On the basis of the experience gained in Peru, Woodward was induced to develop in cooperation with Alco a governor with the following characteristics:

1. An altitude derating feature completely independent from manifold pressure. Altitude changes must not affect the setting of maximum or minimum field start.

2. A fuel limiter controlled by manifold gage pressure. This to be equally effective at any altitude.

3. It must be possible to include both independent features in the governor.

Ref. 36-6 lists the MIs describing the resulting governors. The desired characteristics have been met.

On MI 11156A Page 3 shows the cam which insures the constant field start condition over the altitude range.

On MI 11204 the schematic of the fuel limiter is shown; to overcome inaccuracies due to linkage deflections in the earlier type fuel limiter the sensor piston and cam are subjected only to small forces which then control a hydraulic amplifier which then operates the ORS and the shut down nuts. The bellows stack still includes an aneroid bellows below for engine builders who want it. For Alco the aneroid is punctured and ineffective. If no altitude derating device is specified the Woodward standard load control adjustment is used.

If both devices are specified load control adjustment with the cam and as required for the altitude derating device must be employed.

A prototype of this governor was in regular operation for about one year on the 12D engine on the Santa Fe. On removal all adjustments were as originally set.

This type of governor has been used for the last few years. Although feed back from the field is uncertain it is believed to perform without complaints as intended.

Woodward Governor Secondary Devices

All Woodward locomotive governors used by Alco include the low lube oil shut down device, (protection against loss of oil suction as on EMD and against loss of water pressure as on GE were never used). The original oil pressure shut down settings were 8-14 psi at idle and 26-32 at full speed. Unfortunately this setting is partly determined by the main speed setting piston spring. When Alco wanted to increase the shut down pressures to compensate for higher oil pressure with steel cap pistons the increase in pressure required the addition of a pressure multiplier at the shut down device to obtain the desired setting 15-20 psi at idle and 32-62 at full speed.

On the first Vee engines with Woodward governors a 3/8 turbing was used to connect the lube oil header at the free end of the engine to the governor low lube oil pressure shut down. The engine would start and stop quickly, because oil pressure did not build up at the governor fast enough. On all later engines the connection is made at the generator end of the lube oil header and led to the governor with larger tubes of minimum length.

When air starting of locomotive engines was first introduced a Woodward rack booster was applied. One booster alone, with the rather low air pressures available on locomotives, did not push the racks out reliably, so two boosters were installed. Finally Alco followed EMD's lead, took the starter button out of the cab and placed it so that one man could push the starter button with one hand and pull extension of the control shaft lever with the other hand and thereby increase rack manually. The boosters could be eliminated.

Much trouble with oil leaks both in operation and filling was experienced. A pan with oil drain was incorporated in the governor support casing, the vent from the sight glass, which used to overflow, is now piped back into the governor cavity.

OVERSPEED TRIP

The overspeed trip mechanism was designed for the earliest 6 cylinder engine and has been retained with only few changes, although it has not proven itself to be wholly free of complaints especially on higher speed engines.

The mechanism serves two purposes, it trips the engine on overspeed and shuts it down; the same shut down mechanism can also be actuated manually at will. There have been frequent cases on marine engine where no overspeed shutdown was wanted, but a manual shutdown was required; in these cases the trip weight and carrier were omitted but the rest of the mechanism is used. The outside appearance between the two conditions is not distinguishable. If the mechanism has shut the engine down it must be reset before restarting, which requires considerable force. Assembly print 36AA76287 is used as basis for discussion.

The overspeed trip consists of the trip weight, a motor spring, cocked manually, and a trigger latch mechanism.

One basic short coming of the mechanism is that it is driven at the low camshaft speed and rotates around a horizontal axis. Normally one fly weight moving radially outward is used; at such a low speed centrifugal force and weight are of insufficiently different orders of magnitude to give a clearly defined trip speed. By calculation it can be shown that the spring force keeping the flyweight against the inner stop balances the forces trying to let it fly out (centrigugal forc plus or minus weight) at speeds approximately 5% above or below nominal depending on whether the weight is acting upward or downward.

The first design was based on 1000 RPM rated speed, spring, spring rate and weight were matched reasonably well to give the desired unstable governor characteristics. Other speeds were thennot taken in consideration. When 1100 and 900 RPM rated speeds came into use there was not enough space left to design really satisfactory springs. Another short coming is that moving the spring seat nut inwards to increase spring load for resisting higher speeds, also changes the location of the center of gravity of the flyweight so that centrifugal force increases. The combined effect makes for unsatisfactory speed adjustment. The result is that overspeed settings are subject to substantial variations and must be set about 10% above normal to be sure no unnecessary shutdowns occur.

Two approaches at correction are available. One design relocates spring setting from the nut on the flyweight to the spring seat in the shaft thereby eliminating the change of center of gravity location when adjusting the spring. Two experimental executions have been in operation for sufficient length of time to make a decision concerning adoption. (One is/on one of the experimental engines in Auburn, one on the A engine).

The other approach may even be mechanically more correct, and service proven although no unsolicited feed back has been received either from shop or field. It has to do with the ship's service sets on the LST's. Every part on the LST had to be designed for maximum shock resistance and it was considered likely that an unbalanced over-speed trip weight would fly out under shock and shut the engine down. Therefore, an old, probably somewhat more expensive design from the 539 engine was adapted to the LST 8 Cyl 251, specified and never heard of since. It did not shut the engine down during the shock test. It consists of two weights moving in opposite direction and tied together by gear segments, and therefore would overcome the influence of gravity on speed setting. It is suggested that field reports be requested.

On the regular trip weights incorrect speed setting, possibly due to stetched springs, springs not properly "heat set" or just due to difficulties in setting

WOODWARD OVERSPEED GOVERNOR

On marine engines and other installations where unexpected engine shutdowns may cause hazards the overspeed trips device is replaced, by a Woodward furnished overspeed governor which on overspeed directs engine lube oil at an increased pressure to a hydraulic cylinder. This operates on the lay shaft in the governor support housing but after the collapsible link. It does not shut the engine down but keeps the speed within preset limits.

When used this governor is driven from the left camshaft gear. It is mounted on a small housing, flanged to the gear cover, which contains bearings and drive shaft with Nylon gear similar to the tachometer generator drives used on locomotive engines.

Overspeed Protection By Air Shut Off

Afew cases of run away engines were reported at one time in spite of the normal overspeed trip.

Recalled is one case where a small bolt from an exhaust manifold shroud wedged between the cylinder block and the control shaft lever (a shield was added to prevent repetition as long as manifold shrouds with many small bolts were used) the other cases were two run aways after some repair work during which the air manifold was left half full of fuel for some reason or another.

To avoid such experiences but probably also because the design of the air to air aftercooler on the 6D did not leave room for the normal overspeed trip an air shut off valve was devised which would close off the opening thru which air passed between turbo support and cylinder block, Reference print 36EE76406. The standard flyweight is used but is contained in a thin housing; linkage to the air valve is replaced by a push-pull cable. In the few (3) applications where Alco used a push-pull cables unreliable operation resulted. Partly this is due to the fittings at the ends are not being rugged enough to match other parts of the engine and the mechanics use too heavy a hand on the wrenches; partly the cables are not built to operate at temperatures near exhaust manifolds. The cables seem to have a tendency t stretch or to bind making operation of the overspeed trip not reliable. All 6D engines are equipped with this device and no field reports concerning it are received. The 6 cylinder laboratory engine, however, was recently found to have this device inoperative and it probably had been that way for a long time.

On the 6D the mechanism also included as a standard a micro switch, although also dependent on the push-pull cable. This microswitch actuates the shutdown solenoid in the governor, that is: it will shut the engine down regardless of the air valve as long as the control linkage is free to move.

The initially apparently satisfactory operation of this shutdown device caused the design of a similar arrangement to be incorporated in the turbo support with horizontally mounted aftercooler for the 16E engine. Reference print 36EE76500. Unfortunately the push-pull cable was retained. The micro switch again was depending on functioning of the cable.

In two reported cases the overspeed tripped, the micro switch did not function, either it was not connected or forgotten, but the air valve closed the passage sufficiently to bring the engine to low speed but not to shutdown. The governor sensing low speed gave full fuel and the engine kept running on full fuel and little air for hours in a trailing unit of a locomotive consist. The engines became coked up throughout and had to be completely disassembled for cleaning.

The more frequent complaints were that the air valve did not trip, parts cocked and would bend. In production the standard overspeed trip was readopted.

After long deliberation no field changeout of existing air valve trips was made but according to recollection a kit was offered to the field which contained a simple mechanism to actuate a micro switch independently from push-pull cable and air valve. It was to be mounted directly on the cover over the flyweight mechanism.

Records indicate that the early 8 cylinder engines also went out with the air shutdown valve.

FUEL PUMP CONTROL SHAFTS"

The fuel pump control shafts were originally made of 1045 ground stock, supported on needle bearings and induction hardened at the bearing location. This was very unsatisfactory, the bearings brinelled the shafts and excessive forces to move the control shafts, especially for the overspeed trip, were required.

The bearings were changed to Bound-Brook sintered bronze bushings, of a grade containing a substantial amount of graphite. (During at long strike a Bound Brook Ollite bushings without graphite were substituted without causing reports of excessive wear). It took several trials to establish clearances now specified on drawings. Hardening could be eliminated long ago.

One question frequently asked is why do you continue having the oil holes drilled in the brackets, while you apparently forgot to drill the bushing so oil would get to the bearing surface? The blind holes in the brackets should be filled occasionally with lube oil. This will be absorbed slowly into the porous bushings.

Each fuel pump rack is connected to the control shaft by a lever including the functions of a double collapsible link. This lever permits the control shaft to move against only small resistance if some pumps have their racks latched out to prevent them from pumping fuel, as one might do if a fuel line were broken or a head damaged, but also if some pumps have their racks frozen or stuck at "full". No deficiencies are ever reported on this lever.

The lever includes an excentric adjustment to position each individual pump accurately. (Contrary to some misconception the lever ratios are not affected by the excentric adjustment). The reason for the excentric adjustment is to be able to work on it easily from the outside without getting one's hands thru an obstructed opening into the control shaft compartment.

In this area are the only powdered metal parts used on the engine, the dog on the control shaft and the excentric. Both parts were expensive to machine, very inexpensive as powdered parts and one part per cylinder made for a large enough quantity to pay for tooling.

Note that the control shaft passes a seal which seperates the part of the control compartment which is open to make the connection to the governor linkage and that part which is enclosed by the covers over the fuel pumps and may contain fuel leakage and small amount of crankcase vapors which may pass the fuel pump tappets.

It is necessary to remove thru the opening in the front end the control shaft with all levers and bearings, before the fuel pump header can be dropped and then pulled out.

REFERENCES - 36 GROUP

36-1 G.E. Governor, see the rather rare American Locomotive Company - General Electric Company Governor Maintenance and Excitation System Maintenance bulletins. One copy in F.C.T. governor bulletin binder. Also Alco MI-24027.

36-2 Woodward Bulletin 36006 (Green cover)
" Basic Load Control 36630A
Alco MI-11048A, MI-11125 and MI-11132A.

36-3 Alco accelaration control basic cross section 32-B-74378
A typical application 32-A-74448 (6D).

36-4 Absolute pressure bias and fuel limit governor. Woodward Bulletin 36601
Alco MI-11046A, also EB-87.

36-5 Altitude derating governor
Alco MI-11172.

36-6 Load control governor with fuel limiter alone. Alco MI-11204.
Altimatic derating alone Alco MI-11205
With combined fuel limiter and altimatic derating Alco MI-11156.

LUBE OIL PUMP

The current design of lub oil pumps on all engines,except the 18-cylinder discussed separately, have the same design features and are viturally free of complaints.

The original design was developed for the 244 engine by Roper; the only significant change when going to the 251 was a cost reduction change to support the drive gear from the pump shaft rather than to end the pump shaft in a spline, which engaged a splined hub on the gear which in turn was supported in a special housing on two ball bearings. This was made to meet the original specifications to permit pump changeout without disturbing gear backlash. By the time of the 251 Alco's control of gear dimensions was sufficient to state that gears were interchangeable, so that any gear installed at the originally set center would mesh with its mate with satisfactory backlash.

Instead of the costly separate gear drive of the 244, the 251 pump have a male pilot centered in the bore of a sandwich plate. At the first assembly the sandwich plate, with a pump and gear, or originally with a master gear on a simulated pump, is adjusted to give correct backlash and doweled to the engine base. As long as base and cylinder block are kept together, this sandwich plate location will give correct backlash with any pump and gear selected at random. The same sandwich plate construction is used on all 251 fresh water pumps. On the 8-cylinder, due to the need to account also for setting backlash of the balance shaft drives, a variation of the sandwich plates is used. The lube pump sandwich plate is enlarged and the water pump sandwich plate is mounted on this lub oil pump sandwich rather than on the base.

All impellers are 10° helical gears, 30° pressure angle. For the 6 & 8-cylinder pumps they are 5-1/2" long, for the 12 and 16 cylinders 7-1/2" long. The gears are premachined and the teeth cut, then they are carburized and hardened with subsequent grinding of OD, ID and the ends. Distortion of the teeth has never been a critical problem. The impeller gear and shaft are supported by heavy flanged bushings. The long bushing at the drive gear end has an oil groove bringing oil into a system of drilled holes in the gear which lubricates the whole front end gear train. The idler has a stationary shaft which is bronze sprayed to form the bearing surface for the hardened idler bore. Circulation of lub oil is provided by drilled holes in idler shaft and pump cover. Correct installation of the thrust washers is, contrary to Alco practice, not fool proof but depends on matching stamped numbers. No failures due to wrong assembly have been reported. It is probable that lubrication is so ample that reversed installation of the drilled thrust washers does not matter.

Drive impeller and shaft were originally made of one piece, then for C.R. changed to two piece construction using a Woodruff key as driver and a snap ring engaging one groove each in shaft and gear bore for holding it together longitudinally.

About 1958 Alco copied Roper drawings and took over manufacturing. In copying hardening of the shaft was not duplicated. The Roper shaft had a case around the snap ring groove in the shaft, on the Alco shaft the bottom of the snap ring groove remained soft and triggered a rash of failures. The correction was elimination of keyway,groove and use of a plain shrink fit with .001" to .002" interference. Most of the pumps with the questionable shaft were replaced either at or before failure. It is possible that such shafts are still in a few 6-cylinder engines.

Another type of failure occurred on one or two batches of the idler shaft. The sprayed on bronze came off the steel. It was due to insufficient care in keeping shaft surface free of contaminants prior to spraying. It was corrected by moving the spraying operation into a small isolated room.

The Roper designs emphasized a gradual transition of the passages from the round openings at the flanges to the long narrow ports in the gear compartment. They actually had a patent relating to this point which has since expired. The careful transition has been retained also in the latest 18-cylinder pump design.

Various executions of the casings exist; normal rotation for the 6-cylinder engine, a different one for the 8-cylinder; normal and opposite rotation for 16 (same as for 12-cylinder); also normal rotation for 16 including a relief valve, opposite rotation with relief valve does not exist because design studies did not work out although such a design would have been desirable for 68 LST reverse rotation engines. Depending on specifications, casings were made of plain cast iron (normal) ductile iron and in earlier times bronze. Print 39-A-73115 shows all typical design features.

PUMP CAPACITY

Pump flow capacity was originally determined by wanting adequate flow with hot oil at idle speed, expecting excess flow at higher speeds to be discharged thru the relief valve. With the need for increased flow to meet cooling oil requirements of the steel cap pistons, the capacity of the pumps especially for the 16-cylinder became marginal both at low and high speeds. A new set of front end gears was designed to increase pump speed about 12%. It fits the 12, 16 and 18-cylinder engines. It is specified on all 18-cylinder engines.

For the 18-cylinder the capacity, even of the speeded up pump, was not considered adequate and a new pump with about 20% increase in capacity was developed.

A listing of flow capacities is included as Appendix 39-1.

For many years the pumps were operated at 90 psi pressure; this allowed about 30 psi for drop thru coolers and filters. When the steel cap pistons were introduced, it was desired to increase operating pressure. Some pumps were endurance run on the pump test stand and 150 psi was found to be the maximum pressure not affecting the bushings. A limit of 135 psi was therefore established, to give some margin. Operating experience seems to confirm this limit as satisfactory.

Pump For 18-Cylinder Engine

Gear center distance from the old pumps was retained but the helical teeth were changed to straight teeth. Gear I.D. was decreased. There being no end thrust the thrust washers were eliminated and their length used to increase gear length. The thick solid bronze bushings have become thin walled split steel backed bronze bushings. The gears are made in one piece with the shafts, the idler also has shaft extenions rotating in bushings rather than on a stationary bronze sprayed pin. Gears with their shaft extensions are fully machined and then nitrided. It was believed that making separate shafts and the long internal grinding of gear bores would be more costly than the present design especially if the gears were made of simple forgings. As far as the design of the 18-cylinder lub oil pump is concerned, it should require less labor and less costly material than the old pump; yet it is cited as an item of excess cost.

One error had been made on the original design in that the pressure relief depression at the gear mesh had been put on the wrong side (suction side) of the centerline causing excess gear loads and noise; but all pumps were since corrected.

One peculiarity, obvious when thought about but misleading until then, was discovered when assembling the first 18-cylinder pump; the gears were made reasonably shorter than the housing to provide end clearance, yet when the pump was assembled and all studs tightened the shafts could not be turned. All the pumps, the older ones as well as the 18 cylinder pump, have covers and housing held together by studs extending the full length of all three parts. The cross-sectional area of the 12 steel studs in relation to the cross-sectional area thru the studs puts the housing in compression and shortens it sufficiently to loose all normal clearance. Increasing the clearance to allow for compressive deflection of the housing rendered the assembly satisfactory.

Auxiliary Vane Type Pump

On the first 16-251-E engines in Century locomotives a separate radiator to serve the aftercooler was installed in the locomotive hood above the generator. This radiator was cooled by hydraulic motor driven fan. The source of oil pressure for this hydraulic motor is a Dennison vane pump mounted on the camshaft gear cover. It operates at pressures 1200 psi controlled by a special regulating valve. Originally its drive caused frequent pump and gear failures; by recollection the pump was originally designed for a coupling drive with pure torque, while the Alco arrangement imposes substantial gear load. After several changes in the pump the apparatus seems to function reasonably. It does however affect the lub oil system of the engine proper. The Dennison pump is not capable to operate unless oil is fed to it under pressure; the suction side of this pump is therefore connected to the engine header and puts additional demand on main pump flow capacity. At low speeds this capacity becomes somewhat marginal and if the seals in the Dennison pump become defective, flow and pressure for oil to the bearings and pistons become inadequate. For reference see print 39-A-76004.

A related condition exists on some stationary installations where the remote radiator fan is driven by a hydraulic motor which is operated by a pump at the end of the crankshaft extension. This too is fed from the engine oil system and in a recent case the customer was advised that the engine for this reason must not be operated at low speeds (by recollection 600 RPM).

SUPERIOR

INTEROFFICE LETTER

Memo for Record

From F. C. Tromel

Office Engineering *Date* December 8, 1970

Subject Lube Oil Pump Flow

N. L. Station Building

The nominal lube oil flow in GPM based on actual flow measurements for the various engines is listed below:

ENG. RPM	6 cyl B & D	8 cyl	12 & 16 std gear ratio	12 & 16 special increased gear ratio	18 cyl.
1100	210		290	320	380
1000	195	200	260	290	350
900	175	180	235	260	330
800	155	160	210	230	300
700	135	140	180	200	260
600	115	120	155	175	230
500	95	100	130	145	190
400	75	80	105	115	150
gear ratio	1:1.18	1:1.21	1:1.18	1:1.32	1:1.32

References: Weekly Reports 4-2 thru 16/67, 4-1 thru 21/68, MEL 3540, 3474-F.

16 cyl sped up 18 cyl 8 cyl

F. C. Tromel

F.C. Tromel

FCT:pel

c.c. Messrs: P. C. Anguish
F. Baker
M. I. McKean
D. Roberts
W. Osterman
E. C. Roberts-Auburn
R. Stigler
W. Webster
L. H. Flesher

SS GROUP APPENDIX Tab 1

PIPING AND RELATED EQUIPMENT

Rigid Piping:

Standard weight piping is satisfactory for welded joints. It should not be used if the ends need threading; any threaded pipe or nipple should be made of extra heavy pipe or still larger thickness. Street "ells" must never be used.

Taper pipe threads should be avoided for all sizes 2" and over except for core and access plugs.

Victaulic couplings, used extensively at one time by Alco, have proved prone to leaking and should not be used on new design, They should be replaced on older design wherever they are found: Marman Flexmaster couplings are satisfactory providing their gaskets include the locking feature. In placing clamps and supports, consider slippage of such couplings under vibration and high pressures (lub oil) still a possibility. (It happened on the 18-cylinder test installation in 1971).

For years Alco used mitered pipe joints frequently and butt joints were welded without special preparations except chamfering. The miter joint did give undesirable high resistance in some instances, but no welded pipe joint failures on engines are remembered. In recent times on some marine and stationary engines, some pipe welding specification had to be met. Alco did set up a pipe welding specification based on the use of backup rings, which represents good practice but does not necessarily meet overly stringent codes.

Alco,Auburn is not equipped to pickle piping routinely and substitutes, unless specifically forbidden, cleaning by wire brush and hand deburring. The men who do clean, e.g. main bearing lube oil headers, do a careful job but too much dependence is placed on reliability of the man.

Alco placed regulating valves,especially Fulflo,in the past in the most convenient location, regardless whether they were installed horizontally or vertically. The bypass valve around the oil filter on the 6-B and D engines,and around the lub oil pump on the LST 16-cylinder engines,are placed horizontally. On the 6-cylinder engines, especially when cotton waste filters and the small hoses to these filters caused the Fulflow valve to be partly open muchof the time,the weight of the plunger wore a step in the housing causing improper functioning of the valve. Current paper filters and large hoses probably leave the valve closed and therefore not wearing out. In any future design such valves must be placed vertical.

TUBING

Never use copper tubing in important locations. Alco adopted steel tubing instead of copper after some serious engine failures on 539 engines,due to copper tubing having become crystallized under vibrations.

Do not accept substitutes for Ermeto type 800 fittings. Steel tubing was adopted when Weatherhead-Ermeto was the only supplier of cinch type fittings usable on steel tubing; these fittings then were of the 800 series. For reasons of interchangeabilit Ermeto enjoys a monopoly position at Alco. Normal railroad practice in overhauling is to strip an engine, recondition the components, then rebuild the engines without any attempt to rematch original components. Consider what would happen if tubing fittings of different style or make were used: A cylinder block from engine A has ti fittings on the main bearing caps of make M. During overhaul this block is matched with a base from engine B containing a header with tube fittings of a different

make D. It would be very awkward to order preformed tubes from Alco with nuts M on one end and nuts D on the other. The probability is great that the discrepancy will not be noted, the fittings forced together and the engine lost.

During the LST program when Navy specifications required straight thread fittings with "O" rings, the mixup with consequently ruined crankshaft did occur at least once, the second instance was not proven.

The proposal to use somebody else's fittings for cost reduction or to establish a second source comes up again and again. It was always rejected for the reasons cited. The new wrinkle,however, is that Ermeto has been trying to supersede their 800 series by the 700 series, which is as dangerous as can be,since some of the fittings of the two types can be screwed together, either with insufficient engagement or lack of bottoming of the tube.

HOSES

When designing V engine piping, keep Aeroquip hoses in mind. On the early 244 engines steel tubing was used on fuel piping and on turbocharger connections. Fuel piping is subject to severe pressure pulsations and clamping of the tubes to the structure was never adequately solved. Clamps frequently wore thru tubes, even with liners of various materials. The early G.E. turbochargers were prone to loose a bucket, got out of balance, vibrated and broke the lube oil supply tube causing a fire. At first hoses with clamps were interposed. Alco was one of the first major users of Aeroquip single wire reinforced hoses to correct the deficiencies mentioned above. Hoses are unsightly, but the gain in reliability by using hoses should outweigh appearances.

Alco uses Aeroquip hose 1503, a BUNA lined hose suitable for oil fuel and hot water. Some upgrading for improved resistance to long time exposure to radiated heat is probably possible with currently offered new hose types and may be overdue.

Also now available are decent looking coverings over hoses including fittings, which would keep slowly developing leaks from spraying against hot metal parts. Slowly developing leaks are the only types of leaks one needs to consider. The Alco laboratory anticipated this development by slipping common water hoses especially over the crossover fuel pipe on top of the cylinder block at the generator end. It looked makeshift but eliminated a hazard. The small additional cost for covering the crossover hose should be accepted as a significant reduction of fire hazard.

The connections at both ends of the fuel headers to the hoses were a source of failures.

Both current designs, the 16-cylinder has a new type, are believed to be failure free. Any changes should be approached with caution. The choice of high tensile alloy steel for some of the fittings was not an arbitrary decision but a correction for real failures.

MLW was confronted with some customer's demand to get rid of the crossover fuel line; a rather forced design of a header with welded on fuel return pipe was developed by MLW. It is not known how frequently this is specified, nor how it works.

OIL MIST SPRAY DEVICE

In order to reduce air valve wear, a device was developed to introduce a mist of oil into the charging air. This was done after numerous combinations of different materials for air valves and valve seats gave no acceptable air valve wear conditions at higher engine ratings.

The first trials to introduce oil mist were made with an automotive fuel injection pump and pintle type nozzle. Since it would have required a special reduced speed drive, this approach was discarded in favor of a lubricator fed by brake or starting air. Bench testing of both devices are described in MEL-3494 and 3494A. The lubricator together with a nozzle is the only device tried on the engine itself.

In view of the questionable operation of the air operated spray nozzle in field service, the correctness of the decision not to pursue the injection pump is questionable. Too much emphasis was placed at the time on the failure of an injection nozzle to disperse the oil finely unless it was hot. It is, of course, possible that the fuel injection equipment also would not have worked reliably with dirty lub oil.

There is one published report (source not remembered) which indicates that any benefit of oil on air valves is due not to the oil itself, but to contaminants and additives. Alco tests have proven that clean fuel oil introduced into the air elbow has no effect on valve wear.

Careful review of records show that there are no clear cut, quantitative, reliable data showing that the mist device does cut down air valve wear. Qualitative reports from the field, mostly verbal, do state that air valve wear has been substantially reduced if the device can be kept in operation.

It is known that distribution of the oil mist in the air manifold is very uneven, but to date, no efforts were made to improve this condition; not even changing the height of the spray nozzle in the manifold has been explored.

The device is contained in a sheet metal box which makes it completely inaccessible. The box however was invented to prevent repetition of a serious fire on a locomotive because one of the joints or pipe failed and sprayed oil towards the exhaust manifold.

This device should be considered needed on engines with sustained load, especially if they have 45° air valves but it is not more than an inadequate crutch.

CRANKCASE EXHAUSTER

The purpose of the crankcase exhauster is to maintain a slight vacuum; 0.5 to 1" water column at full load is desired. Under idle conditions vacuum may rise to a few inches, without causing concern, unless some specifications limit the amount. It has been Alco's contention that crankcase vapor should be taken from the engine at the highest point feasible (in the camshaft gear compartment high in the block) to avoid internal rusting due to condensation. This is experience gained on small engines which may not be applicable.

The crankcase exhauster was originally developed for the 244 engine and was reapplied to the 251 on which it worked acceptably at the B ratings.

Its essential features, which make it different from the normal commercial blowers, is the need to make it oil tight and the need to withstand substantial engine vibrations. The first requires gasketed flanged connections rather than slip joints. The second requires a special shaft end of the motor. Normally such impellers are held only by set screws which did let the impeller come off in early trials.

The motor itself on the locmotive had to be a 75 V DC motor but even for stationary applications with mormally available currents and voltages, the special shaft end made the motors specials with long delivery.

On the 244 the exhauster was mounted on a camshaft gear cover, which contained a passage with cast in baffles to serve as oil separator.

Over the years many different approaches were tried on the 251 to separate the oil out of the crankcase vapor. Many of them such as screens in the elbows caused too high restrictions to air flow. The two really effective features are a deep trough in the volute casing with oil drain. Without the trough oil would form a coating or layer on the volute which would drip on the impeller and restrict the openings between the vanes so it would not function (per deductions not actual observation). The other feature is to let the blower discharge vertically upwards into an elbow (miter) its horizontal leg is welded "tee" like into a vertical pipe, the lower end of which, as a trap, is several inches long and blind with oil drain; the upper end is connected to the weather. On the locomotive the upper end is connected to a hose, which makes a 180° turn and discharges the vapors against the road bed. This letting the pipe first go upward and then down again indicates the importance ascribed to the trap. While the crankcase vapors thus handled are by no means free of oil, they are sufficiently clean for most applications.

When the engines were uprated in power and with it the pressure of the turbocharger sealing air which is led into the crankcase increased, the capacity of the blower became less and less adequate. The first step, taken over from a railroad who invented it to stop leaks, was to insert a resistance parallel to the motor field which speeded up the motor; several checks of speeds were made and the resistors tailored accordingly, There were at least two different resistors used. The motor manufacturer, G.E., did not approve of the procedure; but the resistors were placed in a small box attached to the motor by Alco, later by the blower vendor. The speed up did not help the reliability of the motor but did not seem to make it worse either. The practice is still continued.

But even the speeded up motor proved inadequate for the higher powered engines. Therefore, as a crutch two motors and blowers are being specified. On new engines two openings are made in the cylinder block for connecting the two exhausters. For

retrofitting of two motors and exhausters a design 50-AA-74126 has been developed involving an elbow fitting, split into two ducts.

The development of a larger exhauster is long overdue. Work was impeded by the fact that the original supplier, Bayley Blower, went out of business; they started up recently after a lapse of several years. The difficulty now is that Alco does need special blower housings, oil tightness, but that quantities are now not sufficient to make this development interesting to vendors.

Another point of oil leakage has been criticized in recent years, that is leaks past the entry of the motor shaft into the blower housing. It must be noted that the larger exhauster fan is only effective if the opening in the cylinder block is large enough, that is two 3" diameter openings now existing for the double exhausters, must be joined or an equivalent larger opening must be provided. Development of this larger exhauster is virtually complete; implementation awaits the decision to order parts including pattern.

In an attempt to reduce the task the crankcase exhauster has to perform , it is tempting to avoid discharging the turbocharger sealing air to the crankcase. This has been tried several times by different people in different ways, but it has never worked out.

In one of the earlier attempts, not the earliest, two outlets were provided from the central oil compartment on the 320 turbo; one low presumed to let the oil drain off and one fairly high up on which a breather was mounted to let the sealing air escape in the shortest way. It was not possible, more oil than air seemed to come out of the breather. The 320 therefore has two oil drains, one of them high up. It was tried again with a trap about 1968 on a power pack, recorded in a lab report; this had the exasperating feature of working perfectly for a while (10 minutes by recollection) then again discharging oil with the air.

CRANKCASE EJECTOR

None of the crankcase exhausters discussed is capable of discharging crankcase vapors against the restrictions imposed by long discharge piping, nor was Alco ever able to connect them satisfactorily into the stack from the turbocharger. Especially on start up, smoke is pushed from the stack back into the crankcase if the motor driven exhauster is piped into the stack.

On the other hand, the motor driven exhauster has the one great advantage of performing well with the engine idling. If lack of vacuum at idle is accepted, the easiest way is to connect the crankcase to the suction side of the turbocharger, provided means are found to separate oil out of the crankcase vapors. On early marine engines this was done but no oil separator, then tried, was effective enough to prevent the aftercooler from getting oily and plugged with dirt. Motor driven exhausters were retrofitted on these ships.

The LST engines have the requirement that they must continue to operate if they are submerged in sea water up to about crankshaft centerline. This forced the development of an effective seal between crankshaft flange and engine structure; thereby, eliminating a major point of air entry into the crankcase. This made it much easier to handle idling conditions. It was also mandatory to provide an exhauster with sufficient capacity to work against stack pressure.

For this reason, a combination of a modified commercial ejector (PENBERTY) and a reasonably effective oil separator was developed. The final design is shown on 40-EE-71243. The ejector is activated by air manifold pressure, an orifice between the ejector and the manifold permits some adjustment to meet variable ship's piping.

The oil separator is an Alco design, is quite effective in removing lube oil from air without undue increase of resistance to flow. Note the attempt to get any drops of oil out of the air stream into a separated oil trough. A rolled up screen serves as a flame barrier to protect the crankcase.

This combination was thoroughly tested out on lab engines and is recorded in lab reports. The bleed off from the manifold has only insignificant influence on engine performance. Oil separation is quite effective.

When pollution regulations will prohibit the discharge of crankcase vapors into the atmosphere (regardless whether directly or indirectly thru the turbine stack), this separator may serve as basis for leading the vapors again into turbocharger suction. By increasing the number of baffles or their spacing, oil separation can be enhanced especially since additional resistance can be tolerated by going directly into the suction of the compressor.

In a recent attempt to overcome complaints of crankcase pressure on a river boat, such an ejector was put in series with the motor driven exhauster. Verbal report was that the combination performed as intended. This is the type of report which cannot be verified or formalized.

The ejector with separator was tried on a locomotive with 16-F engine in the Schenectady shop. It could not handle the condition prevailing during dynamic braking, that is engine at 1100 RPM full speed with only some small fan load; otherwise, it was reported to function satisfactorily.

PISTON GENERAL

The piston development for the 9" bore Alco engines was such a continuous effort that it is necessary to start the discussion with the 244 piston and mention at what stage the 251 came in and what the few significant differences are. The crown shape of the 244 is a shallow "Mexican hat", nominal compression ratio 13:1; the 251 started the same way but became early on production engines a shallow dish with nominal compression ratio 12.5:1, with uprating to the E engines in 1965 it became a flat top with 11.5:1 compression ratio.

This review will attempt to mention all the more important different designs with brief explanations. More background will be given concerning the current steel cap design, but also the obsolescent but not obsolete aluminum piston with forged inserted ring carrier. This piston is installed in all (16 and 8-cylinder) LST engines, all engines built to date in India and all 244 pistons sold by Alco; it therefore cannot yet be dismissed from consideration.

Relevant piston development started on the 241, the forerunner of the 244 engine, and proceeded in actual Alco production thru the following stages.

1. Forged aluminum body with forged ring carrier to provide for piston cooling.
2. All aluminum cocktail shaker.
3. Cast iron cocktail shaker.
4. Forged aluminum body with cast, inserted ring carrier.
5. Ductile iron cocktail shaker.
6. Forged aluminum body with forged, inserted ring carrier.
7. Steel cap piston.

Stages 1, 4, 6 and 7 have been placed in operation by Alco in a large number of engines.

In damaging competition with Alco, both Gould and Mahle have replaced a large number of pistons by pistons of their own manufacture. Gould's is a cast iron piston based on a joint design mentioned above as #3. Mahle's piston is of cast aluminum with cast-in cooling coil and insert. Lately they are also offering steel cap pistons of their own designs.

When the 251 engine was under development, #1 piston had been found wanting, #2 had been recalled due to failures, #3 seemed promising.

There is one significant difference between the 241/244 pistons and the 251. The 251 piston locates the connecting rod in longitudinal direction by semi circular machined faces giving a total clearance of .013" to .024", while the earlier models follow the more conventional design with the connectingrods held logitudinally by the crankshaft and having large clearance in the piston. This is the main reason why the two pistons use different body forgings, the distance between the wrist pin bosses is about 1" narrower on the 251 than on the 244. 244 pistons can be made from 251 forgings but require excessive machining.

Since replacement of pistons is a substantial business except for that lost to the competition, any clear improvement developed for one of the two pistons was applied to the other as much as possible.

EARLY PISTONS, ALL ALUMINUM AND CAST IRON

From the very beginning all aluminum piston bodies for the Alco 9" bore engines were made as forging; the basic material is the low expansion 12% Silicon alloy 4032, except for the adoption of 2219 in recent years for improved resistance to elevated temperature on some piston types.

In order to provide cooling of the ring area, the upper part of the piston extending to below the ring band was turned down to about 7-3/4" dia. On this diameter a system of grooves was turned, connected by milled slots to form a labyrinth for cooling oil. A forged aluminum ring was then shrunk on over these oil grooves. The shrink fit was and is .019" mean interference for the joint of all forged ring carriers on forged 9" diameter bodies. To make sure this ring would not come off, the first of these pistons also had matching snap ring grooves turned in body and ring, with a steel snap ring inserted for permanent assembly. The piston ring grooves were then machined in the ring after being shrunk onto the piston body.

Cooling oil delivered thru suitable ducts from main bearing to crankshaft, to connecting rod, to wrist pin, to a partial groove in one of the two bores in the piston body. From there it is led to the lowest labyrinth groove, rises thru the labyrinth, passes thru a hole cross drilled thru the body and meets a long discharge hole drilled longitudinally thru the body from a point below the wrist pin as indicated on sketch 42-2. This detailed description of the oil ducts is given because deviations from letting the oil out below the wrist pin at later times were either clearly detrimental or at least questionable.

The shrunk on ring carrier was not new to Alco at that time, it had been used as a method to salvage 12-1/2" diameter pistons which had the ring grooves worn out. New was then the addition of the cooling oil duct.

The cooling oil labyrinth, however, prooved disastrous. These pistons caused one crank case explosion after other; a passage was somehow melted out which permitted combustion gases to enter the labyrinth and go torch like thru all the passages into the crank case. It is not remembered whether the oil did pass the interference fit upward or the combustion gases went downward. Certain is that on an uncooled (without labyrinth grooves) 12-1/2" diameter piston a ring carrier was shrunk on and operated for over a year. (It had proportionately the same high shrink fit). The carrier was removed and black soot was found to have penetrated the shrink fit downward for 2/3 of its length.

In the 9" pistons the snap ring also contributed sometimes to the failures by hammering out its grooves. This occurrence forced the development of a seal weld between body and ring carrier. The seal weld became a quite deep inert arc tungsten electrode weld, which gave little difficulty, both parts to be joined were forgings with high ductility.

These pistons performed well, did not come apart, but had the serious drawback of rapid ring groove enlargement, wear as well as being hammered out. They lent themselves for easy salvage by shrinking and welding on a new carrier. By that time carriers with premachined ring grooves had been made avialable. Expenditure in time and money for frequent piston replacement caused serious customer complaints. Such pistons were continued in production until in about 1954 when cast carriers with bonded ferrous inserts were introduced. In the meantime many completely futile attempts were made on paper and some on pistons to provide wear resistant ring grooves; weld deposits, mechanically held ferrous rings, also early attempts to forge aluminum around a stainless steel ring.

About that time an all aluminum cocktail shaker piston was developed; the rationale how the cocktail shaker should have improved ring wear is not remembered; but the development is remembered as an outstanding example of developing a part, running it successfully in the laboratory for a long time then readying it for production; in so doing making an apparently harmless change to make a very awkward machining operation more palatable. A limited number of such pistons were made and failed epidemically within two weeks of field service due to a stress concentration caused by the easier machining operation. Sketch 42-1.

Development of a cast iron piston was started in 1948 jointly with a small piston manufacturer Gillette and Eaton in Minnesota; they have since been bought by Gould. One of the earliest prints in Alco files 98-E-42217 shows the design approach, which became the basis of all cast iron pistons made in the early fifties both for the 244 and the 251. It also shows a strong family resemblance to the piston still marketed to date by Gould and used widely for 244 more rarely on lower rated 251 engines. In later controversies Gillette and Eaton made the claim that they had furnished the whole design concept to Alco, this claim at the very least is grossly exaggerated.

A good number of such cast iron pistons were made and run in the laboratory and also in heavy service in the field. In the beginning many cracks after short operation were found, initiated by sharp male corners, not only in fillets which had been watched anyhow. By rounding off all such corners, particularly remembered are the corners at the internal ends of the wrist pin bores, short time failures we: overcome.

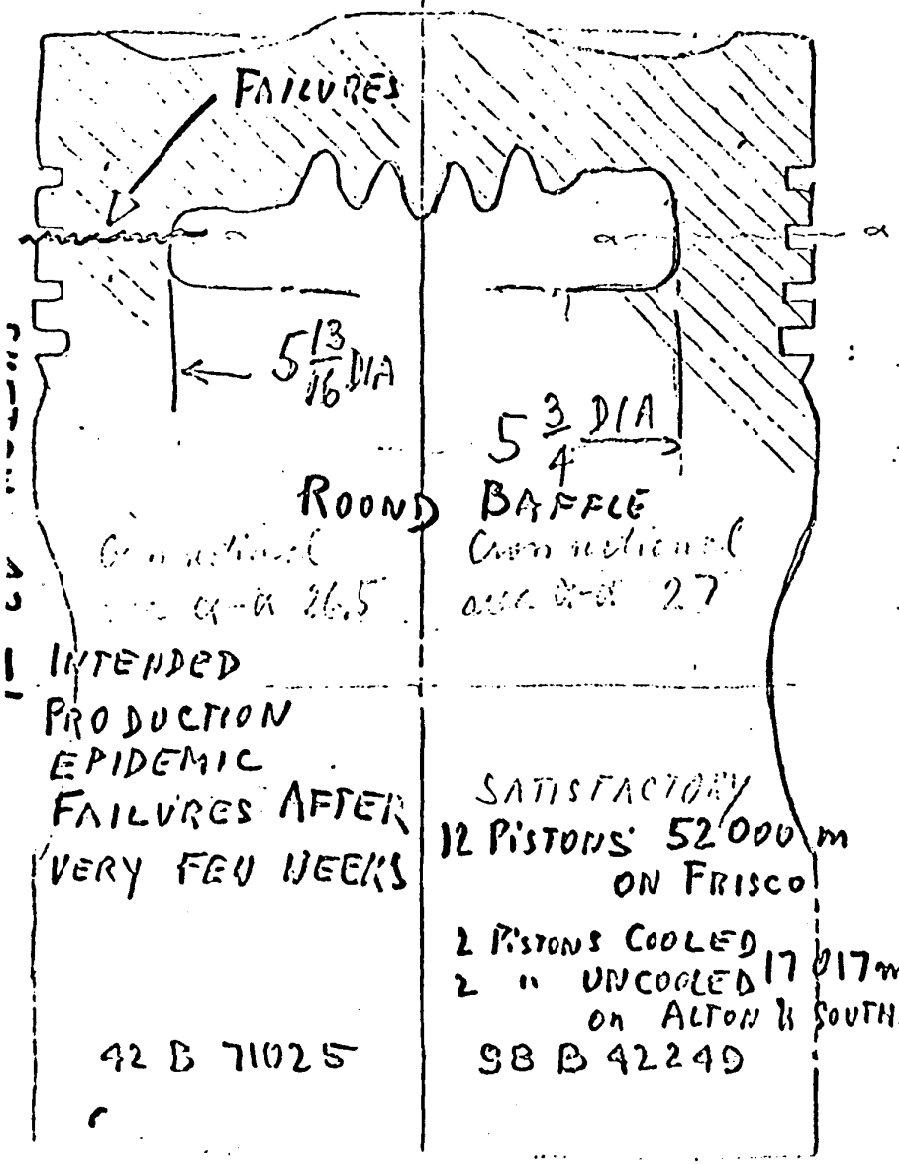
One type of failure which seemed to be basic if only slowly progressing, and which is still usually found on used Gould pistons at least at some ribs for some depth are cracks in the fillets between the oil baffle and the supporting ribs. The material was a high strength flake iron to specification 31-P-5410. On the 244 engine the conclusion was that such iron pistons as then made by Alco from castings produced either by Gillette and Eaton or by CWC, which then also had made a pattern, lived quite reliably in heavy service for about 120,000 miles but would probably have failed before reaching 150,000 miles. The first trials were made with steel backed bronze bushings for the wrist pin, which proved superfluous, the wrist pin operated without galling on the iron. Running clearances against the liners were made less than on aluminum pistons. The engines had the reputation of being much cleaner than with aluminum pistons. All Alco pistons had the baffle closed off with a rolled-in cup. In not a single case did the cup come out. The later Gould pistons have always used rectangular bolted in covers, which permit the use of straight ribs instead of having to angle them to make room for a round opening.

While this piston was still in need of improved reliability it was considered sufficiently far along to equip the first 6-cyl. 251 engines with cast iron piston: to this general design probably to print 42-AA-71018. These engines were installe: in locomotives built by G.E. for CUBA.

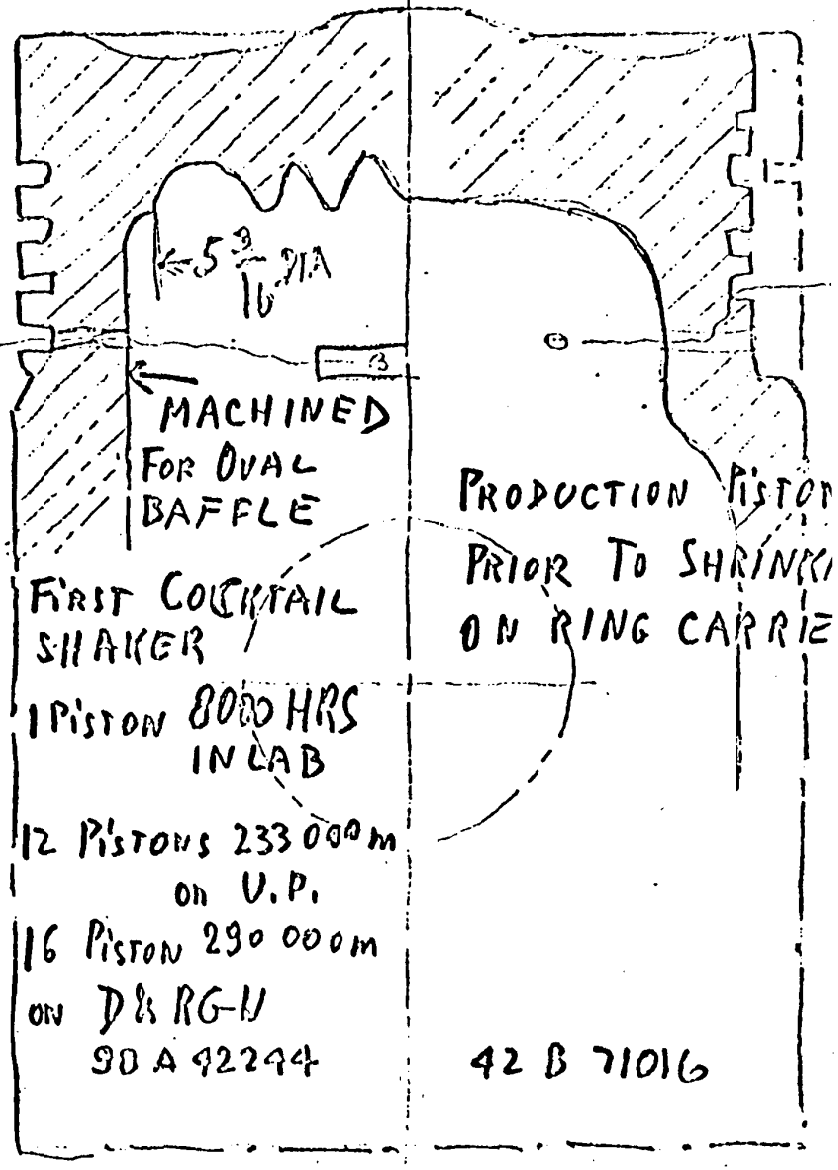
The laboratory 12-251 engine at that time was also equipped with the same type cas: iron pistons; incidentally, these pistons had a shallow Mexican hat type crown.

One day one of these pistons failed in the lab engine, the crack extending from th middle of the crown the whole length of the skirt to the bottom. On hearing about this failure the then company president issued the incontrovertible instruction that no one additional cent be expended in developing a cast iron piston; those that were in the field had to be replaced by aluminum pistons.

MADE FROM SPECIAL FORGING



MADE FROM PRODUCTION FORGIN



This instruction actually killed any further work by Alco on cast iron, that is, flake iron pistons and at the same time induced Gillette and Eaton, i.e. later on Gould, to put somewhat more effort into this development and become before long a successful competitor for Alco pistons in the after market. Ref. 42-1.

PISTONS WITH CAST RING CARRIERS AND ALFIN BONDED INSERTS

The instruction to drop the cast iron piston was given at a time when laboratory tests with a piston with a central recess and pockets for the valves had given improved construction. Such tests were then made with uncooled, solid aluminum pistons which permitted fast procuring and gave valid short time operational results. At the same time the first ring carriers casting, with ALFIN bonded inserts were received, but Alco's knowledge of capability and limitations of these parts was very sketchy, the suppliers on the other hand knew nothing about ring carriers.

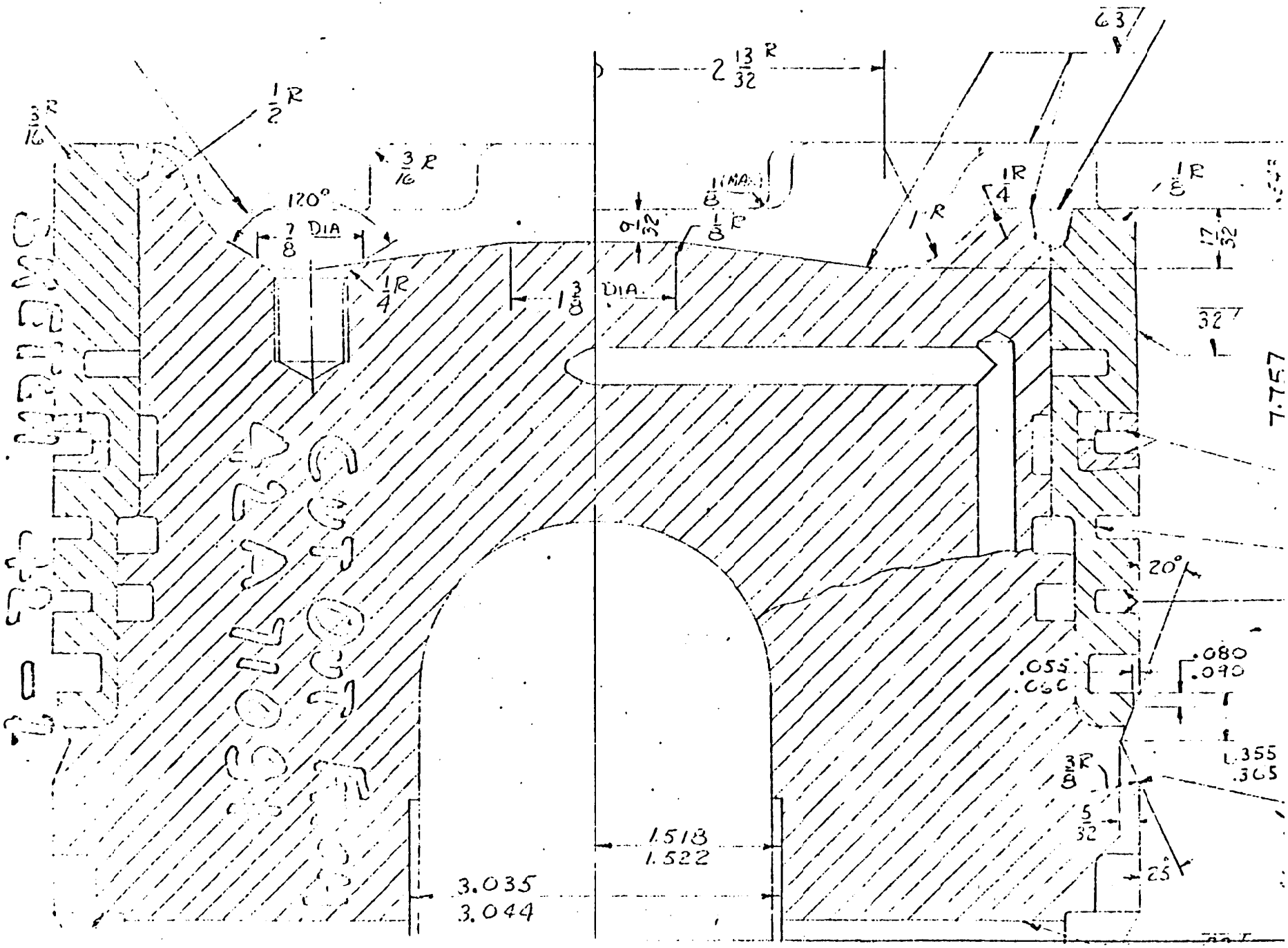
The first piston attempted was a pocketed piston shown on sketch 42-1, it shows a number of features which could not be handled by the cast carrier. First it had to be learned that the shrink fit of .0025"/" used on forged ring carriers had to be reduced to compensate for loss of strength but even more to compensate for the loss of ductibity from 9% to ½%. After many trials the maximum interference without cracking became .0009"/".

While welding the forged carriers presented only small difficulties, welding of the cast carriers became a very tricky proposition which was developed only very slowly over the years to the instructions contained in the earlier issues of 31-P-7600. These practice sheets have since been reworked to fit the forged carriers with inserts and all references to cast carriers have been dropped.

The circumferential welding had been done automatically and this was intended to be continued for the cast carriers. The difficulties of hand welding around the valve pockets were then insurmountable. This is the reason why valve pockets were never again tried or even considered as long as pistons with welded on ring carriers were used. Later it was learned to hand weld after a fashion around the valve pocket of cast carriers for 12½" pistons of the 539 engine; but only a few such pistons are produced each year and scrap rate is frequently intolerable.

Sketch 42-1 indicates another difficulty then encountered. The original forged ring carriers were thin, oil grooving and drilling had been designed to match this thickness. At the cross-section, not shown in the sketch at 90° from the wrist pin the step at the end of the turned down portion of the body and also the bottom of the oil grooves approach the inner contour of the forging enough to make one hesitate to make the ring carrier thicker. On this particular piston the carrier thickness was increased on the top to provide adequate backing of the aluminum under the insert, but original thickness was retained lower down. Matching a stepped bore with some unavoidable if small excentricities further increased assembly stresses in the carrier. While the step was eliminated immediately on the then new 251 pistons, it was continued for reason of interchangeability into the middle Sixties on the 244 pistons when it was decided to carrying the larger thickness of the ring carrier for the whole length and to risk the local thinning of the body. The decision never backfired. It should also be considered that undersized ring carriers were furnished for piston salvage which caused the pistons to be turned .010 smaller at each successive reband; originally 4 undersizes were furnished. When it was realized that aluminum pistons did deterioriate in time, the number of undersizes was cut to two. This did not really insure earlier scrapping of pistons because the customers had learned that careful removal of the old carrier permitted rebanding with carriers of same undersize as that removed.

Sketch 42-1 shows another feature, originally considered important but soon dropped. In the insert small segments were milled out which were intended to anchor the insert firmly and especially prevent loosening while machining the ring groove. It was found that the many sharp corners, even if benched, initiated cracks in the aluminum.


63
2 13/32 R
1/2 R
3/16 R
3/16 R
120°
7/8 DIA
1/4 R
1 3/8
DIA
1/8 R
1/8 R
1/4 R
1 R
1/8 R
17/32
32
7.757
20°
.080
.090
.055
.060
.355
.365
3/8 R
5/32
25°
1.518
1.522
3.035
3.044

All cast carriers were made of A132 which has virtually the same composition as the body forging of 4032. The insert was made of Niresist #1A to minimize the difference in coefficient of expansion between carrier and insert. In all cases the ALFIN process was used to provide a metallurgical bond between insert and aluminum. This bond was never very strong nor could one expect it to extend for 100% of the interface between Niresist and aluminum. A dye check usually would show good bond at the lower side of the insert and anywhere from 10 to 100% lack of bond at the other side. This could also be demonstrated on pistons just removed from the lathe. If one inserted a thickness gage in the ring groove and pushed it up and down many areas were found where cutting fluid could be squeezed out from the interface. This lack of bond wasrecognized and is recorded in the files. It did not cause failures. For specifications see Ref. 42-2.

The ALFIN process calls for a scrupulously clean insert blasted minutes before using it to be dropped into a metal bath chiefly consisting of tin, then placing into a metal coated insert into a mold and pouring aluminum into the mold. The temperature of the mold was controlled by cooling or heating. The process depended apparently on fast transfer,control of temperatures, and proper gating.

The first carriers were obtained pre-machined from Zollner who in turn used castings made by ALCOA. Soon after Alco took over machining of castings made by ALFIN Fairchild, who controlled the ALFIN patent, also Alcoa, later a few from United Engine and Machine in California. Later for reasons of C.R. another vendor Permold was brought in and became chief supplier but his castings caused many failures of a two types not experienced before.

Porosity of the ring carrier made by Permold which happened to occur at the area of the weld greatly accentuated unsound welds, it is believed that castings from the other sources were cast upside down so porosity would be away from the weld. Good castingshave an alloy layer between iron and aluminum in the .0005" thickness range. Permold managed to make this layer frequently .010 thick. This alloy was very brittle,pulverized out and left the insert without support, causing insert failures describes this in detail Ref. 42-3.

The various vendors used inserts of different cross sectional shapes, traperoidal or rectangular with bulbs at the I.D. No difference in performance could ever be detected. Important was that the inserts should have the ring groove premachined. Machining the groove into the solid inserts tended to break the bond. The inserts were held in the mold by three "fingers" centered into the groove. The fingers also kept these points free from aluminum. These empty spots in the groove served to establish all machining in relation to the insert. The cross-section of the insert could be made small because no margin for longitudinal or excentric mislocation had to be provided.

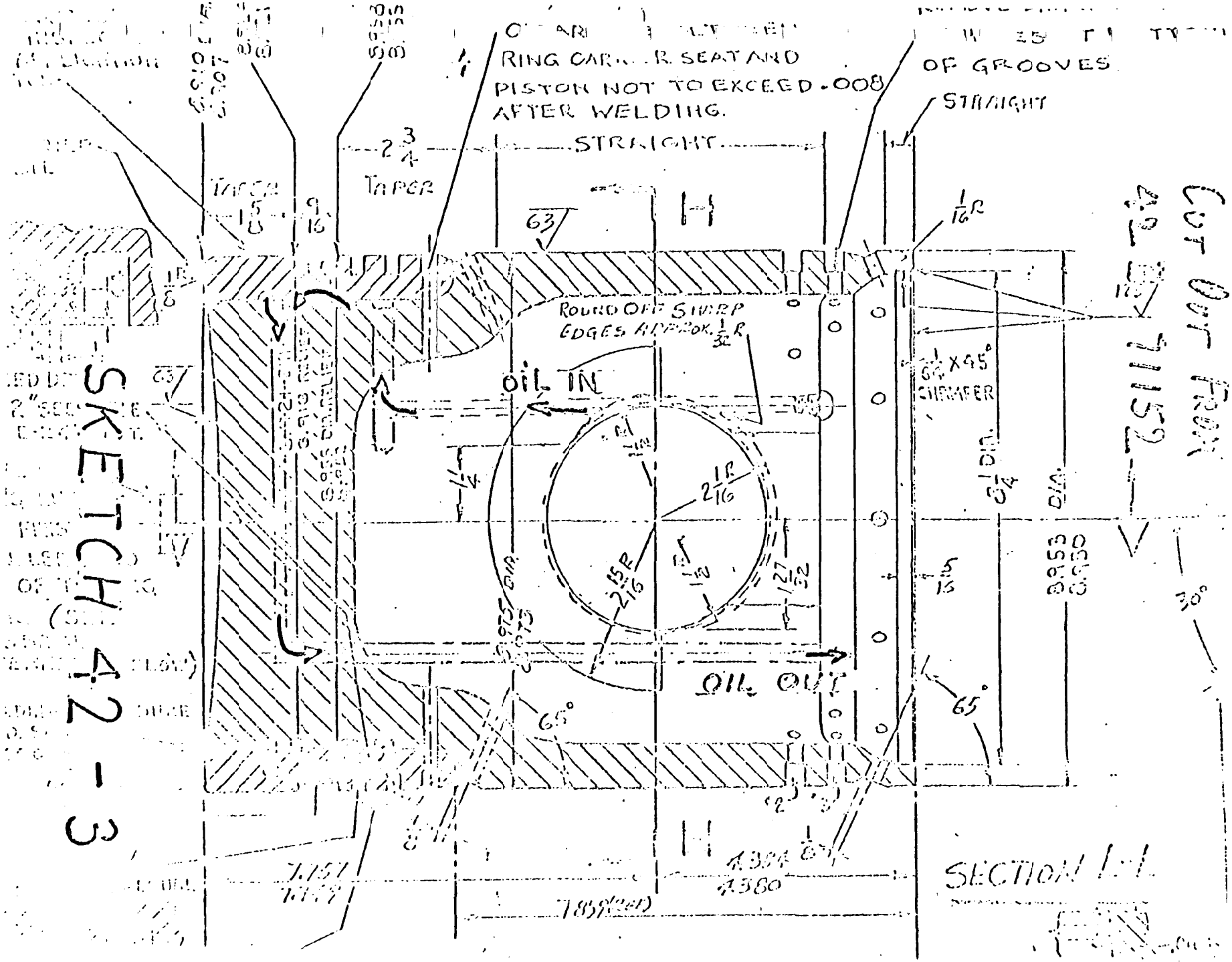
The failures on Permold carriers did not cause the bad reputation of these pistons, they just were an additional aggravation. The real difficulty lay in the low interference tolerated by the cast carrier, which transferred the task of holding th carrier and body together completely to the weld. The interference was .007 with $\pm$ .002 tolerance. With frequent heating and cooling the interference was often lost to the extent that after removing a carrier from a body the lower part of the interface showed evidence of fretting. The difficulties experienced in learning to produce sound welds in Alco shops were compounded many times by the fact that many railroad customers bought fully machined carriers for rebanding mostly from Alco, but also a few from competitors and welded them to the piston with more or less skill under more or less controlled conditions. Under the best of conditions the weld would stop off the interface between ring carrier bore and piston body. that is something akin to a crack meeting the weld. It is a question of time for the "crack" to progress thru the weld. The result was frequent coming off of the carrier from the piston with more or less contingent damage.

While these pistons were subject to the failures described, they had cured excessive ring groove wear. The lower not inserted grooves never wore to any degree which would have made double grooved inserts desirable. These pistons kept thousands of Alco engines running at 190 MEP until early 1963. Some customers operated with very few failures by watching spectrographic oil analyses or by watching crankcase pressures.

One of the latter was the Service Pipe Line with dual fuel engines. Detonations in dual fuel engines made the regular ring carriers unsuitable, they cracked longitudinally. Extra thick ring carriers 1" thick at the upper end, stepped down to 3/4 thickness at the lower end, were made and furnished for dual fuel engines for a short time, Ref. 42-4; later they were withdrawn by Alco and replaced by the original not inserted forged carriers. Service Pipe Line continued to reorder the heavy inserted cast carriers, accepting Alco's refusal of guaranty. Service Pipe Line claimed that crank case pressure increase gave them enough warning of impending failures.

This section started out with the description of a pocketed piston for the 251 engine and the first attempt at using cast ring carriers with bonded inserts. It was continued by discussing cast ring carriers and their subsequent history in general. Now the review returns to the pocketed piston.

Welding the piston with valve pockets could not be done; pistons with a shallow dished crown had given combustion results just a little less favorable than the pocketed piston. Direction was therefore reversed and pistons per sketch 42-3 became the basic long time production piston at first with cast ring carriers later with forged carriers.


RING CARRIER SEAT AND
PISTON NOT TO EXCEED .008
AFTER WELDING.
OF GROOVES
STRAIGHT
STRAIGHT
TAPER
ROUND OFF SHARP
EDGES APPROX. 1/32 R
OIL IN
OIL OUT
1/16 X 45° CHAMFER
65°
SECTION H-H
SKETCH 42-3

DUCTILE IRON PISTONS

As told in a preceding section the performance of the cast aluminum ring carrier with bonded insert was not so that piston development could be discontinued. Another project for the development of a ferrous piston was started. The project however had so many not quite technical difficulties built in, that its lack of success could have been foretold. Priority was low; funding was so inadequate that patterns could be made but no money was available for nearly a year to get castings made from the pattern. The man then in charge of the department did not believe in the feasibility of a ferrous piston nor did he consider the earlier design approach to have any merit "at least not for the 251". Finally flake cast iron was still considered taboo while ductile iron was not proscribed.

The resulting design for the 251 is shown on 42-EE-71140. About the only feature retained from the earlier design is the rolled in cup, only it was made smaller providing insufficient access to cavities very difficult to clean. A pattern was made and much later castings, additional castings were made from the existing 244 pattern of the old design approach. All castings were made of ductile iron to specs 31-P-5411. No technical presentation could budge the prohibition against getting a few flake iron castings.

While the earlier cast iron pistons gave no difficulty of scoring either the piston body or particularly in the wrist pin bores the ductile iron piston were prone to this type of damage, early in field operation, even on the test bed ref. 42-5. Poor thermal conductivity coupled with a high modulus of elasticity led to early failures (7,000 to 30,000 miles) due to thermal stress cracking.

Another metallurgical peculiarity, remembered but not verified by research of records, was that the inside surfaces of these pistons which could hardly be cleaned, much less be blasted and also remained unmachined, acquired either during cooling in the mold or during subsequent heat treatment a thin decarburized layer, of low strength. This in effect lowered fatigue resistance of the high strength ductile iron so that it became lower than that of the flake iron and in the earlier approaches.

Some but not many of these ductile iron pistons were put into the field, it is not remembered when they were recalled or how soon the last of them had failed in locomotive service. The whole development fell within the years of 1956 to 1962. One case is remembered because it was exceptional.

One or a few sets of ductile iron 251 pistons went into dual fuel engines in Canada, probably at Inter Provincial Pipe Line; being for dual fuel engines the compression ratio was lowered by reducing the height above wrist pin and with it the thickness of the crown by .078". The reduced thickness together with the low rating, slightly over 150 MEP was enough to let these pistons perform creditably for a number of years. Some time ago, either to renew these pistons or to repiston other engines the same customer reordered such pistons. Alco attempted to refuse the order, but did fill it in the end without warranty. It is understood that these replacement pistons failed after short service. While the original pattern was then still in existence the castings were made either by personnel not familiar with prior experience or perhaps in a different foundry.

Forged Aluminum Ring Carriers With Inserts

In 1960 while pistons of the general construction shown in sketch 42-3 were production standard, ALCOA suggested that they could make aluminum ring carriers forged around a ferrous insert. While they were not ready at that time to provide a metallurgical bond between the two materials, they were confident that the metellurgical bond could be perfected shortly and incorporated before Alco would be ready to put such carriers into real production. Their confidence was based particularily on one item then going into commercial production a steel sole plate metallurgically pressure bonded to a forged aluminum bottom plate of a household iron.

The carrier ring itself would be made from a DC cast tubing. This tubing would be cut into suitable lengths, a step with excess length would be turned on one end to a diameter slightly smaller than the insert ID. The insert would be dropped onto the step and the thinner excess length would be upset and forged in a die to close tightly over the insert. While the whole ring carrier would get into a die and into a forging press the lower part, not turned down, of the aluminum tube would remain in nearly the original DC cast condition while the upper turned down part would be subjected to a true forging deformation.

Alco first satisfied itself that the claim for DC cast properties being virtually the same as forged properties is essentially correct. DC casting process involves continuous casting of a tube or bar with immediate water quenching. Ref. 42-3.

On the basis of these results and under the pressure of an intolerable percentage of field failures of the cast carrier it was decided to initiate a joint development. The forged carrier with insert was then a venture into completely new territory for both ALCOA and ALCO; in the years following both encountered many detail problems concerning design, forging techniques, metallurgy, welding procedures, setup in machining many of which affected life or performance of the ring carriers in the engine.

To give some idea of the magnitude of the piston failure problem one instance is cited from the period of severe difficulties. On 29 new locomotives with 16C engines 20 piston failures occurred in the period from September 65 to March 66 (14 were clear insert failures, 6 not clearly identified) this corresponded to failure rate of 4.23% of the pistons; but it occurred as 13 locomotive failures (more than one piston had failed in some instances) which gave a 45% locomotive failure rate due to pistons. This happened on the S.P., the railroad with many long tunnels. Five of the twenty failures took place in No.1L the one with least cooling oil flow.

These types of problems continued with little let up during much of the time this type of piston was in full production. It is difficult to judge to what extent the many short-comings have been eliminated. The latest version which has now been in gradually diminishing production since about 1968 and has been limited to the dished top 12.5:1 compression piston,used only up to about 210 MEP seems to experience only very few failures. It is the piston used in all LST's because confidence in the steel cap piston was not yet established when this order was in its early stages.

The currently existing engineering correspondence files as well as metallurgical reports contain sufficient documentation of difficulties encountered and the way they were met to reconstruct the development in detail if the need should arise.

The documentation was sufficient to help in the negotiations which led ALCOA to share, if in relatively small degree, in Alco's losses encountered with this type piston.

The general history of the development was: In 1962, some eight carriers were forged by ALCOA of alloy #2014-T6 with inserts of Niresist #1 and Niresist #3. At least four were operated at high ratings in the 12D laboratory engine for 555 hours. One detrimental defect showed up, the inserts made to about the same dimensions as the one for cast carriers distorted badly in the forgingprocess resulting in unrecognized thin insert section after machining. Otherwise, performance was satisfactory, next trial was made with Niresist #1 inserts of larger cross-section. Ref. 42-7.

Beginning early 1963 some 300 carriers were ordered for field trials. Then and all subsequent carriers were made of alloy 2219.

Then 300 carriers were ordered with full knowledge that no metallurgical bond would be accomplished. Since it was known that the ALFIN bond on cast carriers rarely extended to 100% of the interface and frequently so much less, as mentioned earlier, the decision was made to go into field trials while ALCOA promised to continue development work on bonding.

On this first order considerable delay was caused by metallurgical development of heat treatment. These treatments do include quenching from temperatures in the 1000° range an operation which does influence contact conditions between aluminum and ferrous insert, the extent of this influence was only later fully appreciated.

About the first set of complete pistons was installed in the 12D engine then operating as a prototype in a freight locomotive at 2400 HP for traction on the Santa Fe. These pistons performed in heavy service without failure. Much later two of them were fully examined after 257000 miles and found in perfect condition. Gradually more of these pistons went to selected customers. During 1963 about 600 pistons were made and no failures were reported. All these pistons were machined and welded to the same drawings as the then current cast carrier pistons except that the shrink fit had been increased to the old .019" mean interference.

The end of 1963 attention was called to a small gap between insert and carriers on piston, made from latest forgings received. After careful consideration the piston were accepted, a decision which later events proved wrong. Ref. 42-8.

In 1964, the forged carrier, still without metallurgical bond, gradually replaced t cast carrier in full production.

Not long after failures in two distinctly different areas began to appear. One area concerned the insert itself also its tightness in the carrier, the other concerned the weld. These two areas will be discussed sepatately.

Insert And Interface Deficiencies

The typical defect starts with loosening of the insert in the aluminum carrier. and proceeds in stages to the point where the insert flops back and forth in a groove perhaps 1/4" wider than the insert. The insert and piston ring at this point may or may not have broken. Ref. 42-9. The most startling instance was a piston removed from an engine where the insert had disappeared but the piston ring itself was still in the wide groove of the aluminum where the insert had been. In some cases the fragments of insert or piston ring might work upwards into the combustion chamber or inward into the cooling oil ducts.

Early some inserts then made of Niresist #1 were found cracked before the pistons went into an engine. Investigations showed high stresses introduced during forging and the insert material was changed to Ductile Niresist #2. Loosening of inserts continued. At some later date Ductile #2 inserts were found to have cracked in the field, it was believed due to a combination of reduced wall thickness, explained later and high lead contamination of the iron alloy. Material control at Koppers the suppliers of all inserts for Alco pistons was improved. (Many pistons were recalled).

Many of these problems arose or were believed to have arisen from unintended, uncontrolled changes in the forging process. The first prototype forgings were made on a small upsetter, the early production forgings, which performed well in the field on a fairly light press. This press was transferred to the L.A. plant and became unavailable.

The first occurrences which showed up in significant numbers were traced with high probability to a change in press handling. In the earlier forgings the press was kept in down position for a few seconds which allowed the aluminum to make more intimate contact. Later the man on the press omitted this delay since the forging appeared to meet size inspection. It took months until such carriers went to the field, failed and started the investigation (many pistons were recalled). Later practice, believed still valid is to close the die momentarily release pressure, reapply pressure and keep it on for a few seconds.

In about 1966 the die broke several times, it was only then released by Alco, that these transfers to different presses had taken place. T he forging was then done on a very large press. The pressure exerted exceeded the strength of the existing die. None of these presses has means of varying applied hydraulic pressure. To reduce the unit pressure on the slug a new die was made based on a step turned solid slug of aluminum rather than the original turned down tubing.

By then much had been learned about the forging of carriers, but even so the switch to the new die introduced unexpected consequences in the forgings.

Several different cross-sections of the insert were used in attempts to improve the mechanical interlock between insert and the forging. Due to the pressures involved in the forging process, premachining of the groove was out of the question it was found that the originally machined I.D. could not be relied on to give adequate wall thickness at the bottom of the groove, during forging the insert expands to the diameter of the die. When at one time the dies had worn opposite the area of the insert, the inserts got stretched so that the wall at the bottom of the groove became not only intolerably thin but in addition not uniformly thin over the circumference. Four small holes were drilled in the bottom of the groove of pistons then in stock here and abroad to probe the wall thickness. (Many pistons were recalled).

Residual stress in the inserts was investigated, it was found that there was a distinct tendency for the inserts to be held under circumferential compression by

the aluminum with a small gap between both pieces on the insert ID. When the carrier was shrunk on the piston and the aluminum thereby expanded the insert lost some or all of its precompression; the insert therefore did not only not act as a shrunk on piece but in addition had lost some or all pressure from the aluminum on the flanks of the insert. This, of course, contributed to loosening in service. These phenomena can be explained by the heat treatment which involves temperatures at which the differential expansion of the two materials is significant, also where there is a great difference in stiffness between a nearly malleable state of the aluminum and the still quite high strength of the ductile Niresist.

After years of resistance ALCOA introduced a coining operation to improve the contact at the interface. ALCOA had been apprehensive that this operation might cause forgings to fly apart and create personnell hazard.

Opposite the lower inner corner of the inserts small discontinuities in the aluminum were frequently found. They are euphemistically referred to as laps rather than cracks. They can be found only by destructive examination. No failure was ever traced back to these laps.

The welding proceedure also has an influence on insert tightness. If too much welding heat is introduced the aluminum above the insert is likely to be pulled away in cooling.

Based on recollection and not claiming completeness; ring carrier forgings which have given quite acceptable performance in recent years were made as follows:

A solid slug of 2219 aluminum DC cast stock is turned down to a diameter giving just enough clearance to let a preheated insert be dropped over the heated slug. The step in the slug has a fillet and conforms reasonably well to the insert contou The insert is trapezoidal with two semicircular grooves at the flanks, and 4 rounde grooves at the ID. Press manipulation includes a significant time delay with die closed. The core is bored out of the forging, it is coined at some elevated temperature, but not reaching forging temperature, the coining being controlled by reduction of height. After coining a solution heat treatment followed by aging is applied. Inspection includes some examination of gap between insert and forging which is very uncertain at this stage due to aluminum flash. Also inspected is the extent of bellying out during coining.

Weld Failures with Forged Ring Carriers

After the first few carriers were made of 2014 ALCOA recommended the use of 2219 for the ring carriers and also suggested changing the bodies from low expansion 4032 to 2219. A tabulation of material properties is appended at the end of the piston section. Principally 4032 has low expansion, is fairly light weight, seems to have good bearing properties, but it is not highly resistant to the influence of elevated temperatures for long periods of time; 2219 is very good in temperature resistance but less desirable than 4032 in the other aspects mentioned.

It was then stated that the forging and eventual bonding of the insert had much better chance of success if 2219 were used for the ring carrier than 4032. Changing the body material as also recommended by ALCOA when it was still questionable whether or not the project could be carried through was a step Alco did not care to make, but accepted rather the discrepancy in coefficient of expansion between the 4032 body and the 2219 carrier.

In 1965 failures in the weld area, never as numerous as insert failures, had occurred in sufficient number to be taken seriously; Ref. 42-10. Briefly the mechanism was both body and carrier expand, the body is at higher temperature than the carrier. The carrier tries to retain the body, in frequent heating and cooling the body of lower hot strength gradually yields, becomes smaller in diameter longer in length than the carrier. The interference fit is reduced, where the interface between body and carrier meets the weld, it acts as a crack which gradually progresses through the weld, resulting either in a lube oil torch or separation of carrier and body.

42-B-71152-1 (6 Ring)

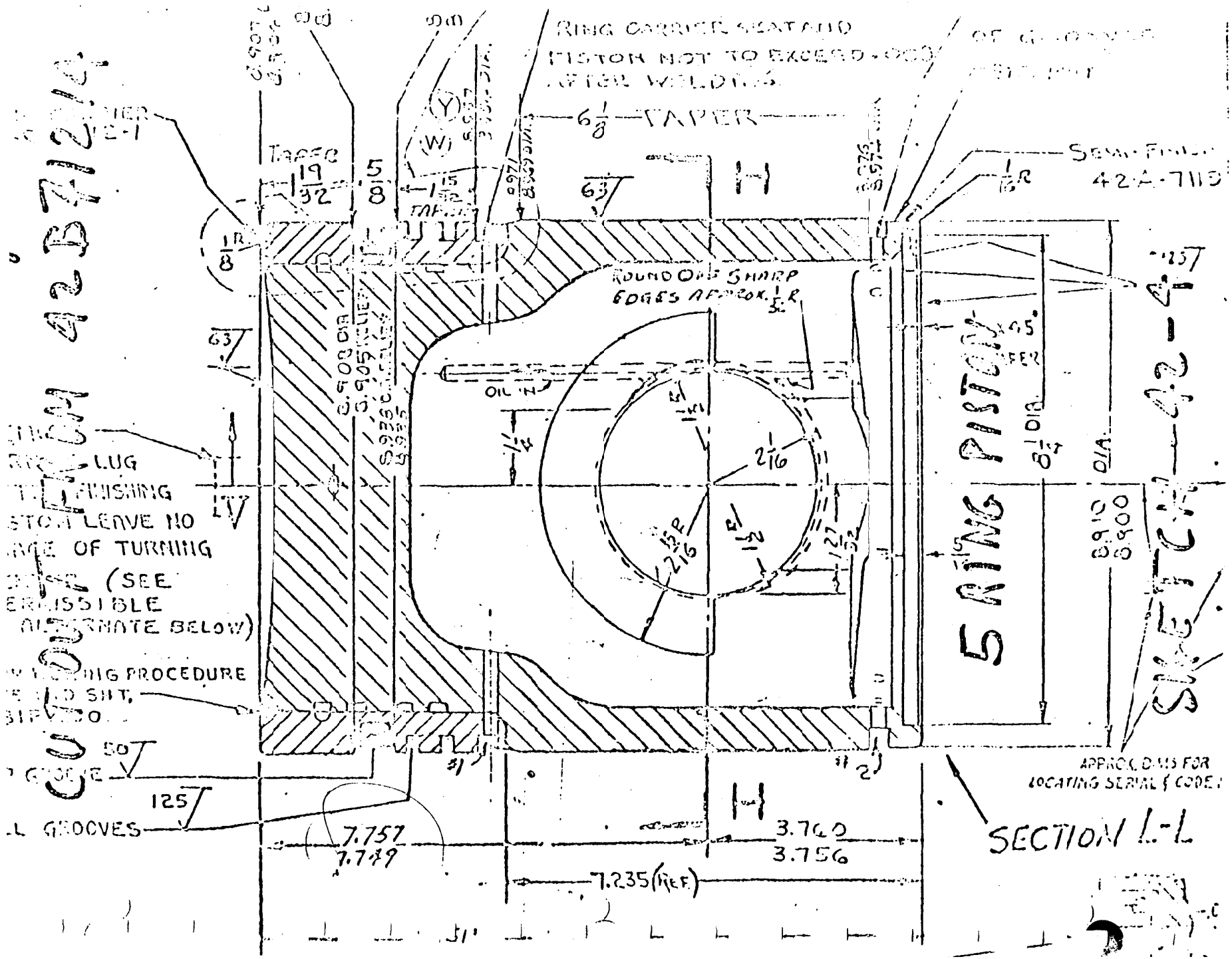
In 1966 the decision was reached to change from the 4032 body to 2219 bodies; thi change eliminated the weld failures described.

42-B-71214-1 (5 Ring)

The material change did , however, force certain other compensating changes. To keep piston weight the same and thereby permit indiscriminate piston replacements in spite of a 5% difference in density, the body was cut shorter and one scraper ring was eliminated. It should be noted that the bearing surface of the skirt was not shortened only the reduced diameter below the lowest oil ring; see sketch 42-4. To compensate for omitting the scraper ring in the 5th groove, the oil ring in the fourth groove was made a conformable ring under the assumptic that one conformable ring would give about the same oil control as two self energized scraper rings.

While this was tested out in short lab runs it was not proven in controlled extensive endurance runs and presents to date 1972, an area of uncertainty. Some customers e.g. West Pakistan have blamed excessive liner wear on this change.

Difference in thermal expansion forced changes in running clearances; the 4032 pistons had a straight skirt with .026 mean clearance; the 2219 pistons have a tapered skirt with clearance at the top of the skirt .030" and .025" at the bottom. When judging piston clearances it must be remembered that the liner is both waisted in and somewhat out of round opposite the lower insert in the cylinder block.


RING CARRIER SEAT AND
PISTON NOT TO EXCEED
AFTER WELDING
6 1/8 TAPER
SEMI-FINISH
42A-7119
TAPER 19/32
5/8
1 15/32
TAPER
ROUND ONE SHARP
EDGES APPROX. 1/32 R
OIL IN
2 1/16
2 15/16
1 5/32
1 27/32
5 RING PISTON
8 1/4 DIA.
6.925
6.900
42-4-125
SECTION 1-1
APPROX. DIMS FOR
LOCATING SERIAL & CODE
LUG
FINISHING
STON LEAVE NO
NCE OF TURNING
(SEE
ERISSIBLE
ALTERNATE BELOW)
NISHING PROCEDURE
D SHT.
50
125
L GROOVES
7.757
7.749
3.760
3.756
7.235 (REF)

In this connection welding procedures 31-P-7600 should be noted; they are the result of extensive development work. They have contributed markedly to make the inserted forged carrier piston finally quite usable. As mentioned earlier, the welding procedure does influence insert tightness since the aluminum adjacent to the weld has a tendency to shrink away from the insert on cooling. In this respect the dished crown 12.5:1 compression ratio piston is most favorable because the ring groove and insert is farthest away from the weld. It is 5/16 further away than on the 244 piston which, during the time of the cast ring carrier, seemed to result in proportionally fewer weld failures on the 251 than on the 244. With the advent of the flat top 11.5:1 compression ratio piston which brought the insert 3/16 nearer to the weld, most of this gain was lost. This may be a contributing reason for having reduced piston failures when then flat top piston was changed to steel cap pistons exclusively.

Near the end of Alco's sole reliance on the forged ring carrier piston, relations with Alcoa became somewhat strained and it was felt that another source should be opened. Negotiations were started with Harvey in Torrance, it is not recalled exactly what forging method they planned to use, it was different from Alcoa's but sounded promising. Harvey made prototypes MET 6821 perhaps a dozen, they were run in a laboratory engine for a few hundred hours and would have merited field trials; but by that time the steel cap piston had proven itself viable and the connection with Harvey was dropped.

PISTONS WITH REVISED DIRECTION OF OIL FLOW

Also in 1966 the idea was conceived that piston crown and insert temperatures would "obviously" be reduced if the oil flow were arranged so that the entering, coolest oil would be brought to the top and flow downwards through the cooling grooves. See sketch 42-5 which also shows the first flat top piston.

This was the reverse from the long time practice in which the oil is brought in at the bottom. The change was made in production before the "obvious" improvement was verified in the laboratory and about half a year's production went out this way before the decision could be reversed in the light of lab results. The only means then available for temperature measurements of pistons were hardness checks of sectioned pistons which had operated for certain time at controlled loads. Ref. 42-11 showed that the original oil flow gave lower temperatures. It is suspected that hole drilled across the crown to lead to oil from the labyrinth to the drain hole gives a noticeable benefit in the original arrangement

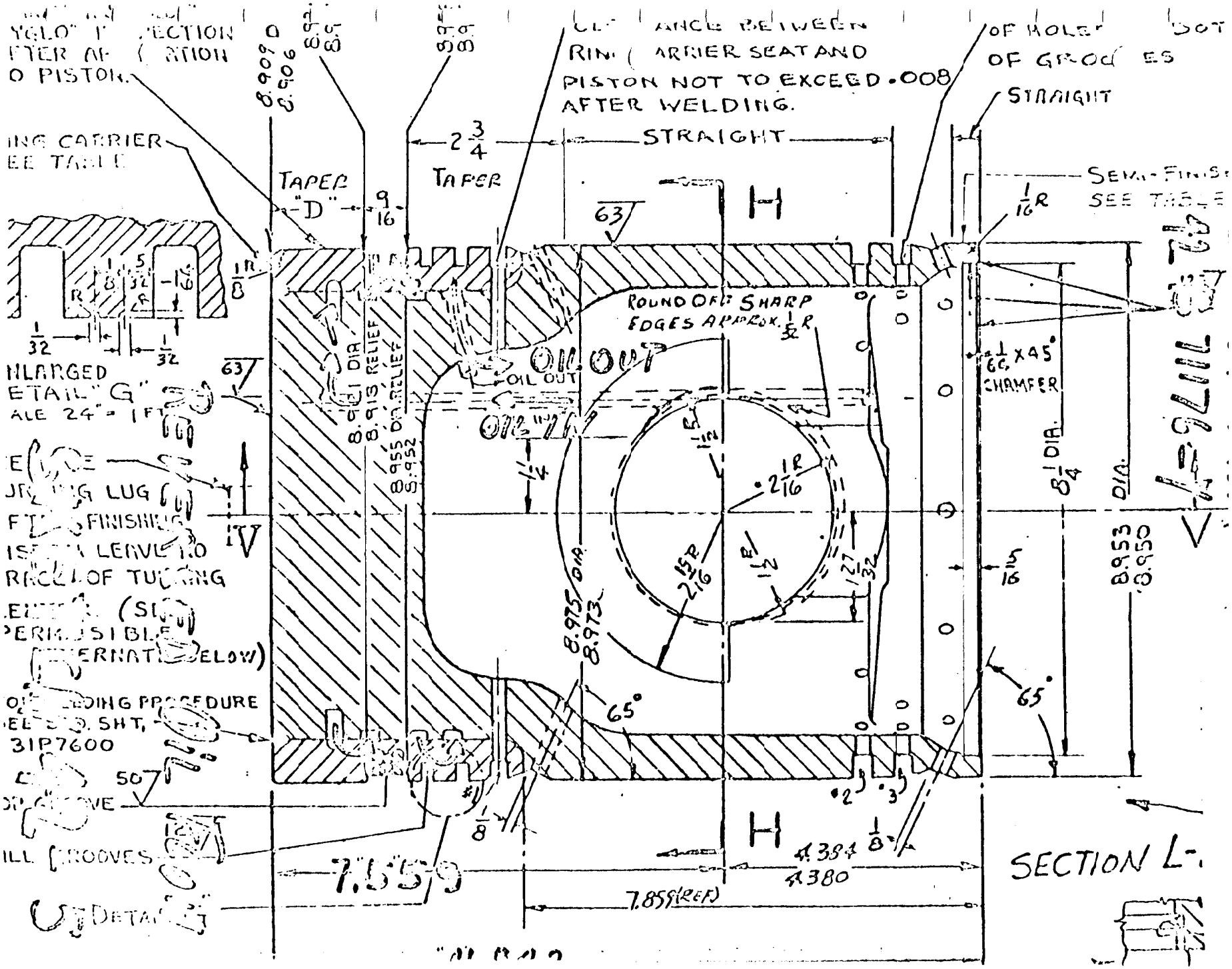
While this was one instance of an arbitrary and unnecessary change, virtually all other changes made on forged ring carrier piston were made under the pressure of having to keep production going for new engines as well as for renewal parts to keep locomotives running. Monthly piston requirements were seldom fewer than 1000, a return to cast carriers was soon declared to be totally unacceptable to customers, no other alternative except buying Mahle pistons seemed to exist and this alternative also was distasteful.

OTHER APPROACHES

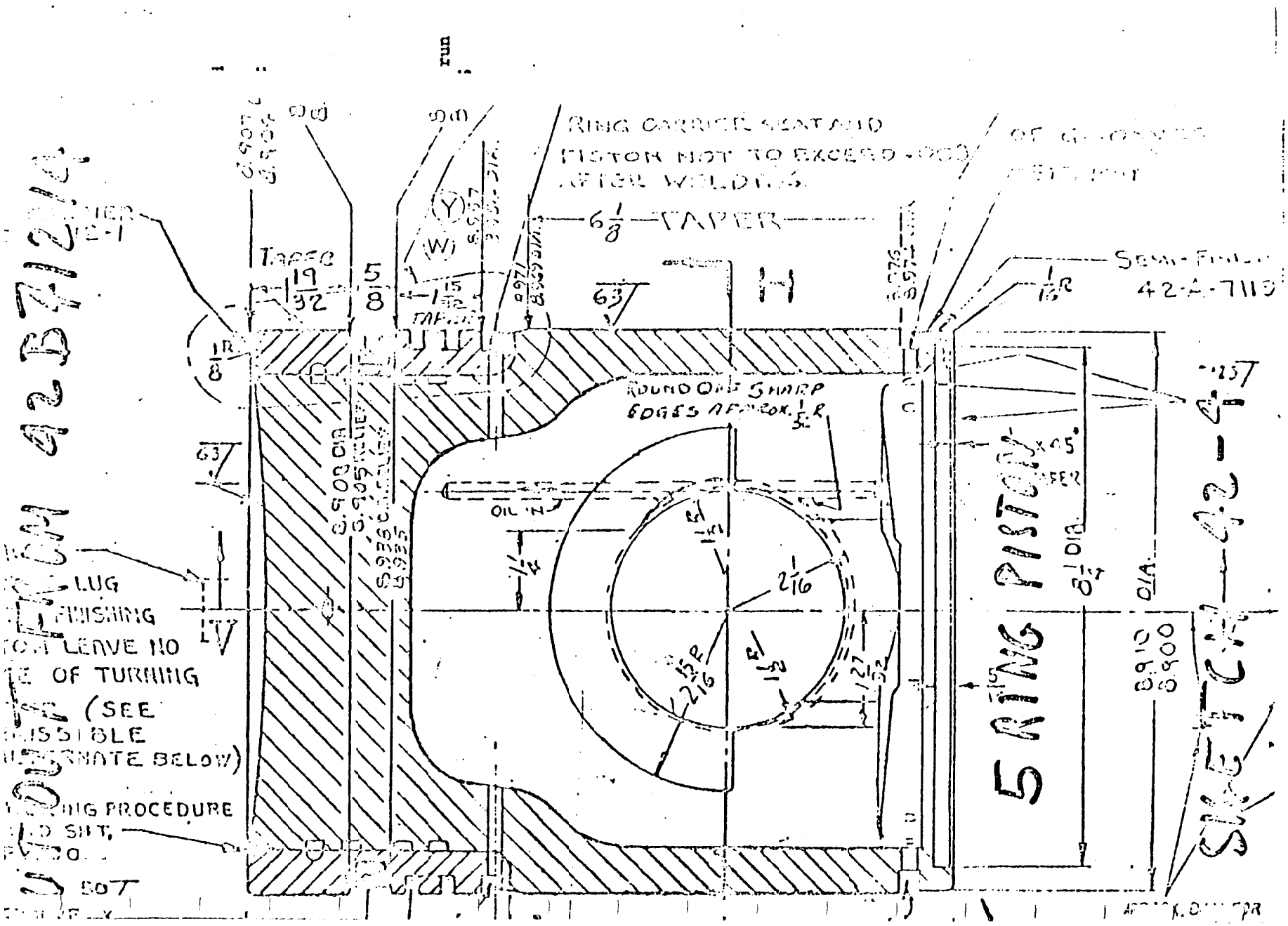
At the end of 1964 when the pattern of failures had become quite clear, a large meeting between Alco and Alcoa was arranged; it was attended by management and responsible metallurgical and mechanical men of both companies. Status and outlook was reviewed. Alco left the meeting with the impression that a metallurgical bond between an aluminum forging and any kind of iron casting was not attainable in the forseeable future. Alcoa promised and subsequently kept the promise to try to achieve some bond, perhaps with high strength castings; at a meeting several years later a comprehensive report of their efforts was rendered but no useful new approach was discovered. But the meeting did result in more engineering support and probably helped to bring the inserted carrier forging to the current reasonably acceptable state.

Alcoa could achieve a metallurgical bond between stainless steel and aluminum forgings under laboratory conditions with chemically clean metal surfaces. The same result was achieved by Battelle who had a contract from Alco and bonded a stainless steel ring to an aluminum cylinder in an evacuated autoclave under high pressure and temperature. No attempt was made to translate either results into production methods.

After the meeting Alco reached the decision to renew efforts to develop backups for the forged ring carrier. Exploratory discussions were


TAPER "D"
2 3/4 TAPER
STRAIGHT
63
H
ROUND OFF SHARP EDGES APPROX. 1/32 R
OIL OUT
OIL OUT
1/16 X 45° CHAMFER
8.909
8.906
8.915 RELIEF
8.955
8.952
8.975
8.973
65°
65°
8 1/4 DIA.
8.953
8.950
2 1/16 R
1 1/2 R
1 27/32
1/8
4.384
4.380
7.855(REF)
SECTION L-

initiated both with Mahle and Gould. No suitable commercial basis for cooperation could be found. The Mahle subject was pursued further. See next section. The concept of the steel cap piston was found shortly after and then assumed first priority. Other approaches included uninserted high hardness ring carriers, metal sprayed pistons, neither showed any promise.


RING CARRIER SEAT AND
6 1/8 TAPER
TAPER
19/32
SEMI-FINISH
42-A-7119
ROUND ONE SHARP
EDGES APPROX. 1/32 R
OIL
2 1/16
1 27/32
5 RING PISTON
8 1/2 DIA.
LUG
FINISHING
E OF TURNING
(SEE
ISSIBLE
NATE BELOW)
ING PROCEDURE
D SHT.
50T

	White Pigeon Materials		Similar Alcoa Alloys	
Class Alloy	Al - Si 12 - C - Ni Hypereutectic	Al - Si 18 - C - Ni Hypereutectic	Cast Alcoa A-132	Forged Alcoa 4032
Designation	Table 124	Table 130		
Composition	Alum. Basis 12% Si. 1% Cu 1% Ni 1% Mg	Alum. Basis 18% Si. 1% Cu 1% Ni 1% Mg	Alum. 12 Si. .8 Cu 2.5 Ni 1.2 Mg.	Alum. 12.2 Si .9 Cu. .9 Ni 1.1 Mg.

Condition	Chill Cast Age Hardened	Forged Age Hardened	Chill Cast Age Hardened	Forged Age Hardened	CAST T65	FORGED T6
Weight lbs/Cu. In.	0.097	0.097	0.098	0.098	0.098	0.097
Coefficient of Expan.						
68° F basis	0.32	0.34	0.28	0.30	0.20	0.33
Mean Linear						
1/1/1 68° F to 212° F x 10^{-6}	11.4	11.4	10.3	10.3	11. ?	10.6 ?
Hardness						
BHN 68° F	105	110	105	115		120 BHN
212° F	95	98	95	98		
572° F	83	82	87	83		
Ultimate 68° F	33,700	51,000	23,200	35,000	30,000	50,000
Tensile						
572° F	17,000	19,000	14,250	16,500	10,000 ?	8,000 ?
Elongation 68°	1	3	0.5	2	1 ?	0 ?
in Percent						
572° F	8	8	1	5	15 ?	60 ?
Alternating 68°	13,700	19,000	11,000	14,000	12,000	18,000
Bending						
Fatigue 572°	7,100	12,000	7,100	12,000		
Use Range.						
PSI	11,000,000	11,000,000	12,000,000	12,000,000		11,000,00

? AREA DATA FOR SPECIFIED CONDITIONS USED SAME DATA
AREA GIVEN TO ALLOY VERY CLOSE THAN DES SPECIFICATION

42-17-A

42-17-A

107-1.3
3-11-43

In the meantime the steel cap piston had proven itself and any attempt to recreate the Mahle piston was dropped

Mahle themselves have since developed a steel cap piston for Alco engines which is being offered in competition with Alco. For description, see ref. 42-13.

STEEL CAP PISTON

The following thought, went into the basic design of the steel cap piston:

Its weight should not be too much more than the piston it replaced.

High stresses can be tolerated provided the stress range between installed condition and during exposure to firing pressures can be kept within reason.

If high mechanical stresses are permissible metal thickness can be kept thin to minimize thermal stresses. To further minimize thermal stresses local thickening for bosses should be avoided as much as possible.

If high stresses are employed, it must be possible to control thicknesses and all transitions must be filleted. This requirement virtually makes a fully machined solid of revolution mandatory.

If the crown is to be made of a material resisting high stresses, the retaining stud can be formed integrally with the cap.

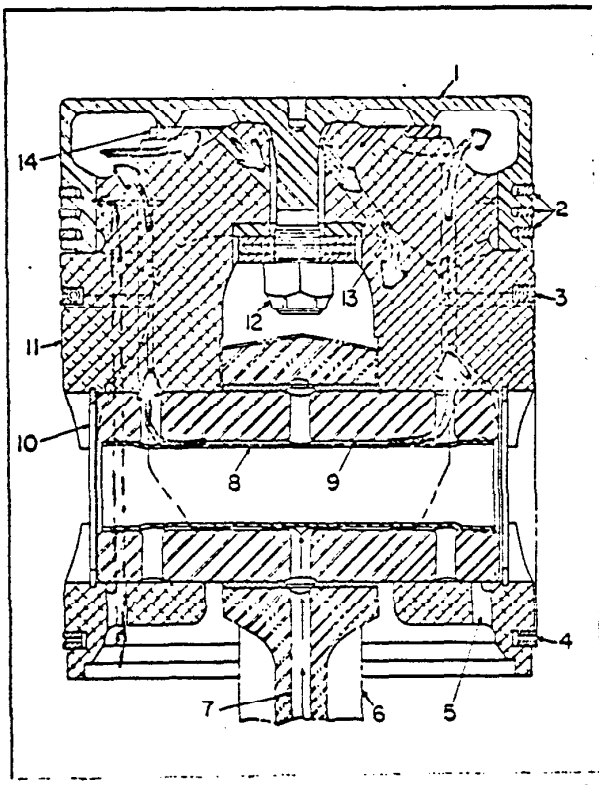
Assuming for the moment that the supporting piston body were truly rigid, further that the crown is a flat plate without the intermediate supporting annulus, this plate is now deflected by the single force of the stud until it contacts the washer face in the center. This creates a prestress in the same direction as that from the firing pressures, the two stresses do not add, because no additional deflection can take place, the center being supported by the washer face of the rigid piston body. The same reasoning can be followed in considering the intermediate support annulus which contacts part of the plate at a certain deflection while the center is pulled further down. While this is an idealized assumption on which design calculations were based, straingage test data , ref. 42-15 , and performance in the engine tend to support the assumption in general terms. Sketch 42-7 shows a cross section for reference.

Steel for the cap was chosen for strength at about 600°F, a special steel AMS 6302 heat treated to 300 BHN mean. The bodies are made of the well tried 4032 aluminum, since it is shielded by the cap from excess heat and all other properties are more desirable than most other piston materials.

Development proceeded steadily but without skimping time for careful testing. The first complete cap drawing is dated February, 1965; the first field test began November, 1966. In the interim prototype pistons were made ,straingage tested, endurance tested in a laboratory engine for over a thousand hours, cooling oil flow conditions were investigated and several changes to related parts were worked out to increase oil flow.

The most important was the design and development of a separate oil feed to the vibration damper to increase cooling oil flow to the number

STEEL CAP PISTON



→ Cooling Oil

→ hydraulic pressure at assembly

1. Steel Crown
2. Compression Rings
3. Conformable Oil Scraper Ring
4. Oil Scraper Ring
5. Oil Drain Hole
6. Connecting Rod
7. Oil Supply Hole
8. Piston Pin Sleeve
9. Piston Pin
10. Snap Ring
11. Piston Body
12. Nut and Washer Assembly
13. Oil Outlet
14. Washer

SKETCH 42-7

one cylinders. Main bearing cap, oil grooving, connecting rod oil drilling and grooving was altered to increase oil flow. Techniques to measure piston temperatures were developed; unfortunately they became operational only after the oil flow increase development was nearly completed. In reviewing the mass of test data, ref. 42-16 there are some numbers which suggest that increased oil flow does not necessarily and not always result in lowered temperatures.

All engines leaving the Alco plants at ratings E and F have the changes to related parts incorporated and the oil pressure raised to above 75 psi header pressure with clean filters and should not be operated at any pressures below 62 psi with dirty filters. At first the changes to all related parts were considered mandatory on engines of all ratings; as field experience increased recommendation for some of these changes were dropped. On rating B, C and D increase in lube oil pressure is now considered mandatory and change to separate oil supply to the filter is recommended to be done at the next overhaul.

Changes to the piston from inception to field test were:

1. Addition of Belleville washers, note that they need tool mark-free bores and rounded ends at the bores, some cracking was experienced.

2. Addition of the steel washer under the intermediate support annulus, some digging into the aluminum body at the inner edge of the annulus was noted during lab testing.

3. The original straight shrink fit between body and cap was changed to a taper fit with means of expanding the cap hydraulically at assembly and disassembly.

4. A second oil discharge hole which allowed some oil to bypass the ducting around the stud was closed. Oil flow remained the same at high speed, temperature was not measured. It may have changed violent oil shaking to laminar flow in the inner annulus (to 42-16 tabulation Q and B respectively).

5. The outside of the cap was lubrited to resist oxidation. The inside is blasted to obliterate tool marks.

6. The first pistons duplicated the dished crown; all later ones were made as flat top. Refer to next chapter.

During operation in field and laboratory a small number of failures occurred. They all started as cracks in the fillet between the crown and the center boss (not boss to stud fillet) and strain gage result showed this the area of highest stress. The first occurrence permitted clear evaluation of the mechanism. A valve broke in the laboratory engine, the piston appeared in perfect condition and was reused. Some 40 hours later this cylinder started to smoke. A partial circumferential crack was seen on the crown surface with evidence that oil had passed which caused the smoke. On opening up the crack it was found that on

the underside a .001" deep crack had been formed, presumably by pounding of the fractured valve, which then progressed as a fatigue failure through the crown deck.

Fractures of similar appearances occurred on pistons on which the washer under the intermediate annulus had been left out, pistons which had been subjected to hydraulic lock. Most alarming were cases where wear between washer and annulus had taken place so that the washer became loose. To correct this the fillet radius was increased from 1/2 to 1", this took effect June, 1970. It resulted in an increase in thickness at the critical section and tends to increase the clamping force between annulus and washer. Sometime in 1971 the hardness of the washer was increased to equal that of the cap. No failures of parts with the large radius have been reported. The change was not proven by straingages.

By now many thousands of steel cap pistons are in operation with only a small number of failures (all as described above) having been reported.

In 1971 and 1972 two modifications to the body are being tested, primarily to affect oil contamination; one is changing the tapered skirt to a straight skirt extending the current minimum clearance at the bottom for the whole length; the other changing the drain hole drilling to exit below the wrist pin. The cap OD has recently been reduced by .010 to insure absence of contact between cap and liner.

DISHED TOP VERSUS FLAT TOP PISTON

Since approximately 1965 there are two crown shaped in production use. The original one has a shallow spherical dish about 1/4" deep in the center; it is dimensioned to give a nominal compression ratio of 12.5:1. (The word nominal reflects a tolerance of about ±0.5:1 due to accumulation of the many dimensional tolerances involved and also a probable discrepancy caused by great number of changes since the original calculations were made to individual parts, valve seat insert, piston hand clearance, etc.)

The later crown shape is completely flat and gives a nominal 11.5:1 compression ratio.

The change in compression ratio was made to keep firing pressure low in spite of increase in engine ratings and was adopted when increasing MEP above 215 (C rating). Per (MEL 3473) the change in compression ratio from 12.5 to 11.5:1 alone, no change in crown shape, drops firing pressures by about 110 psi. There is some indication that the change from dished top to flat top, without change in compression ratio, also lowers the firing pressures by about 50 psi (MEL 3307).

The change from dished top to flat top was made to combat valve failures due to valves sticking, getting hit by the rim of the dished piston and thereby getting the stem bent which would lead either to burnt out or broken valves. It was reasoned that contact between a stuck valve and the flat top piston would not result in damage to the valve. This seemed true until valve yokes started to break in Australia in 1971.

These are the primary reasons for the two changes to the piston.

A very important secondary effect of the change relates to the height of the piston crown above the wrist pin and to the minimum distance between piston and valve during operation. If the valves were fully open while the piston is at top center actual interference would occur. The closest approach between air valve and piston occurs at between 3 and 9 crank degrees past T.C. depending cam profile and timing. The exhaust valves do not come close to the pistons if they follow the cam; if they stick, contact may occur.

Lowering the compression ratio alone would have decreased the height approximately 3/32"; while lowering compression ratio and simultaneously changing from dished top to flat top decreased the height by approximately 3/16".

While original conditions, 12.5:1 CR and dished top resulted in a minimum acceptable clearance between air valve and piston with an air cam timed to give 123 3/4° valve overlap for scavenging, the reduction in piston height permitted an increase in this overlap. The new overlap was chosen at 140° which resulted in a decrease in turbine inlet temperature of about 100°F.

Appended to this chapter are two tabulations giving pertinent valve piston clearance data. They are marked 42-8 and 9.

The change from dished top to flat top as such without compression ratio change does not affect combustion to any demonstrable degree.

Changes of compression ratio and crown shape took place on the aluminum pistons before the steel cap went into production.

As soon as the steel cap had proven itself, the flat top all aluminum piston which had been used only on the higher rated engines was discontinued and superseded by steel cap pistons; these had to be flat and of low compression ratio. Most, if not all, flat top aluminum pistons were either recalled or took themselves out by failures, there are probably very few now in the field.

Due to expected valve interference if a high compression piston were used with a 140° overlap cam shaft, a clear distinction independent from any stamping is desirable. The difference between dished top and flat top is unmistakable and was, mostly for this reason, retained. All dished top pistons, regardless whether all aluminum or steel top, have 12.5:1 compression ratio. All flat top pistons have 11.5:1 compression ratio.

It was considered undesirable from a production and distribution point of view to make two different steel caps; up to 1971 there seemed to be no disadvantage in using the flat top lower compression ratio steel cap piston in any engine regardless of rating. All steel caps for production and field replacement made up to July, 1972, have been flat top 11.5:1 CR.

Evidence gradually accumulating indicates that the flat top 11.5:1 CR piston greatly increases contamination of lub oil, presumably by carbon from imperfect combustion. For this reason dished top 12.5:1 CR steel caps will be produced starting in summer, 1972; this is expected to be shortly followed by discontinuation of production of all aluminum pistons for 251 engines.

BY 5-22-71 DATE SUBJECT Clearances valves to piston. SHEET NO. 5 OF

CHKD. BY DATE JOB NO.

Tabulation to show minimum distance between valve and wristpin centerline under different valve timing conditions and accumultation of tolerances.

Valve overlap ,crank degrees	123 3/4 ^{0}	140 ^{0}	140 ^{0}
Camshaft assembly drawing	16 B 71045-1/2	16 B 71052-1,2	16 B 71052-3,4
Air cam profile	16 A 72023	16 A 72036	16 EE72048
Distance valve to wristpin c/L ref : page 1	7.6075" ±.043	7.6075" ±.043	7.6075" ±.043
Closest approach occurs at crank ref: page 4	6^0 after TC	4 1/2^0 after TC	3 1/2^{0}after TC.
Closest approach,if dimensined so that valve open kisses piston in TC. ref : page 4 (valve to piston)	.290	.122	.197
Closest distance valve to wrist-pin c/L (sum of preceding lines)	7.8975 nominal	7.7295 nominal	7.8045 nominal
Clearance reductions due to various tolerances .			
Assume angular relation tolerances result in advance of camshaft by 1/2 tooth ,3 1/2^{0}crank,Ref page 4 resultant closest approach valve to piston	.253	.098	.173
Closett approach valve to wristpin c/L (7.6075 plus preceeding line)	7.8605	7.7055	7.7805
No tappet clearance set,(deduct .035")	7.8255	7.6705	7.7455
All static dimensional tolerances add unfavorably -.043	7.7815	7.6275	7.7025

These are the minimum distances between valves and piston , all tolerances have been taken into account. Not cosidered were thermal expansion and dynamic deflection.(differential thermal expansion estimated at less than .003", dynamic deflection-extension- etsimated at less than.005")

Per assembly drawing tolerances laft bank camshaft trails right bank by about 1 crank degree, due to gear mesh conditions.This will increase the valve to piston clearance and has therefore been neglected.

BY _____ DATE 5-22-71 SUBJECT Clearance valves to pistons SHEET NO. 0 OF 0

CHKD. BY _____ DATE _____ 251 Engine JOB NO. _____

S U M M A R Y ,

MINIMUM DISTANCES VALVES TO PISTON UNDER CONDITIONS ACTUALLY USED, OR PROPOSED.

Piston # & description compression ratio.	Camshaft # & description	Minimum clear. nominal	Minimum clear. worst combinatio
42 C 71214 ,Alum dish 12.5 : 1	16 B 71045 123 3/4 overlap paraboloid cam	.144 " (- .145 ")	.024 " (-.227 ")
42 B 71200 Steel cap flat 11.5 : 1	16 B 71052-1,2 140 overlap paraboloid cam	.162 " (+.040 ")	.054 " (-.044 ")
42 B 71200 Steel cap flat 11.5 : 1	16 B 71052-3,4 140 overlap polynomial cam	.237 " (+.040 ")	.129 " (-.044 ")
42 C 71226 Steel cap dished 11.5 : 1	16 B 71052-3,4 140 overlap polynomial cam	.166 " (-.031 ")	.058 " (-.115 ")
42 B 71200 Steel cap flat 11.5 ; 1	16 B 71045 123 3/4 overlap paraboloid cam	.330 " (+.040 ")	.2075" (-.044 ")

(....) Numbers in paranthesis are clearances (interfefences if negative) between pistón at TDC and valves fully open.

45 -9

SALVAGE

Starting with the first solid aluminum forged ring carriers, through the inserted cast carriers and well into the time of the inserted forged carriers rebanding of pistons was routinely done by Alco, its licensees and many railroads.

Typical drawings are 50-B-73751 of the fully machined ring carrier which required only shrinking on and welding to a piston body prepared per 50-A-73847. This print as well as the pertinent welding instruction 31-P-7600 show one point which proved extremely important in rebanding. The area at the top of the body where the original weld had been was usually slightly porous and had gotten impregnated by carbon and/or oil. Until this was recognized and this metal was fully removed and built up again before machining for the new weld groove, it had caused many failures of rebanded pistons.

The print also shows a repair for the lifter holes; cracking of this area was very much time related. By limiting the depth of the cracks it was attempted not to let too old pistons go back into engines.

Early pistons had pressed in die cast plugs to close the wrist pin bore; they used to become loose and the key slot used to wear out. The print shows remachining for oversize plugs. Later pistons got Truarc snap rings, unfortunately it was not possible to remachine old bores to accept the snap rings.

The snap rings were introduced in production after trials in the field mostly in an attempt to check influence on oil consumption. As usual adequate control was lacking on these railroad tests. But all indications are that oil control was not affected. There were one or two cases where the snap rings had come out, it was traced back to a change in Truac ring and recommended groove dimensions which were not called to Alco's attention.

In 1966 at the height of difficulties Alco stopped rebanding 251 pistons and refused to furnish rebands to customers. This refusal was breached in some cases of overseas customers. Rebanding of 244 pistons is still being done.

To date no instruction has been issued to the field for opening up and rebuilding steel cap pistons. In view of past experience with field rebanding, the steel cap pistons will experience fewer failures if field reconditioning can be avoided. A drawing establishing condemning limit has been issued internally.

Before using inserted carriers, regrooving for oversize rings was frequently done; 1/64 oversize rings are still marketed; this practice may merit reviving for steel cap pistons, conceivably even in the field, without disassembly of cap from the piston.

Alco also recommended cleaning tanks to circulate cleaning fluids through the cooling ducts of pistons. They could still be useful for steel cap pistons.

PISTON RINGS

Except for the conformable oil ring, all other rings were adopted in the mid-fifties. They are made of hardened malleable iron Koppers designation K26, have given excellent service on 200 MEP engines, recent reports of engines at this rating still show extremely little wear unless some other influences mainly oil formulations interfere. The more modern hardened ductile iron K28 was operated in the 12D engine in a short trial with results inferior to K26.

On higher rated engines especially with high dispersant lub oils tho rings can not anymore be considered satisfactory, but nothing else has been proven out.

Most of the Alco experience was obtained with rings furnished by Koppers; rings made by Robb in Canada to Wellworthy patents are now considered acceptable alternates. The decision to so consider them was made mainly on the basis that they had been used by MLW for long time without showing any apparent difference from Koppers rings. They were not subjected to extensive lab tests.

Alco used for many years two slightly different set ups. The one in general use was: two straight grooved; one taper compression rings; three scraper rings. Only for lightly loaded switches, the set up was: one straight grooved, two taper compression rings and three scraper rings. In cases unplated liners where they are used, the top ring is straight but chrome plated followed by two taper compression, one conformable, one self-energized scraper, all rings are unhardened.

When the piston was changed from three oil ring grooves to two, the upper oil ring was made an unhardened conformable ring while the bottom groove retained the self-energized scraper. The compression rings remained as before.

To combat souping the compression ring arrangement was changed in 1968 on all general applications to: one straight grooved, two taper compression rings, one conformable, one scraper. It seemed to increase liner wear and was changed in 1970 on locomotive engines to: two straight grooved, one taper compression and two self-energized scrapers. At this time, summer 1972, it appears as if this is currently the most desirable combination until some other combination has been proven out.

TABLE I

TYPICAL MECHANICAL PROPERTIES

	Ultimate Tensile Str. (psi)	Yield Str. (psi)	Elongation in 2" 1/16" Thick %	Elongation in 2" 1/2" Dia. %	Hardness BHN 500 Kg Load 10 mm Ball	Shear Str. (psi)	Fatigue Limit (psi)	Modulus of Elasticity (psi)	Compressive Yield Str. (psi)
2014-0	27,000	14,000	-	18	45	16,000	13,000	10.6×10^6	-
2014-T4, T451	62,000	42,000	-	20	105	38,000	20,000	10.6×10^6	-
2014-T6, T651	70,000	60,000	-	13	135	42,000	18,000	10.6×10^6	-
2218-T72	48,000	37,000	-	11	95	30,000	-	10.8×10^6	-
2219-0	25,000	10,000	20	-	-	-	-	-	-
2219-T31	54,000	37,000	17	-	100	33,000	-	10.3×10^6	-
2219-T37	60,000	49,000	11	-	117	37,000	-	10.3×10^6	-
2219-T62	61,000	42,000	11	-	115	37,000	18,000	10.3×10^6	-
2219-T81	70,000	53,000	11	-	130	41,000	18,000	10.3×10^6	-
2219-T87	70,000	58,000	10	-	130	41,000	18,000	10.3×10^6	-
A032-T6	55,000	46,000	-	9	120	38,000	16,000	11.4×10^6	-
A132-T551	36,000	28,000	-	0.5	103	28,000	13,500	-	28,000
A132-T65	47,000	43,000	-	0.5	125	36,000	-	-	43,000
F132-T5	36,000	28,000	-	1.0	105	-	-	11.2×10^6	-

TABLE II

TYPICAL TENSILE PROPERTIES .. AT VARIOUS TEMPERATURES

Temp. °F	2014-T6 Tensile Str.(psi) Ult.	2014-T6 Tensile Str.(psi) Yield	2014-T6 Elong. in 2" %	2219-T62 Tensile Str.(psi) Ult.	2219-T62 Tensile Str.(psi) Yield	2219-T62 Elong. in 2" %	2219-T81 Tensile Str.(psi) Ult.	2219-T81 Tensile Str.(psi) Yield	2219-T81 Elong. in 2" %
-320	84,000	69,000	14	78,000	52,000	14	86,000	63,000	13
-112	74,000	63,000	14	65,000	44,000	12	73,000	56,000	12
-18	72,000	61,000	13	63,000	43,000	11	72,000	54,000	11
75	70,000	60,000	13	61,000	42,000	11	70,000	53,000	11
212	63,000	56,000	14	53,000	39,000	14	63,000	50,000	13
300	40,000	35,000	15	44,000	30,000	15	48,000	38,000	15
400	16,000	13,000	35	33,000	24,000	18	34,000	29,000	16
500	9,500	7,500	45	27,000	20,000	18	28,000	23,000	16
600	6,500	5,000	65	9,000	7,500	35	7,000	6,500	40
700	4,500	3,500	70	4,500	3,500	100	4,500	3,500	100

TABLE II (concluded)

TYPICAL TENSILE PROPERTIES - AT VARIOUS TEMPERATURES

Temp. °F	4032-T6 Tensile Str.(psi) Ult.	Yield	Elong. in 2" %	A132-T551 Tensile Str.(psi) Ult.	Yield	Elong. in 2" %	F132-T5 Tensile Str.(psi) Ult.	Yield	Elong. in 2" %
-320	66,000	48,000	11	-	-	-	-	-	-
-112	58,000	46,000	10	-	-	-	-	-	-
-18	56,000	46,000	9	-	-	-	-	-	-
75	55,000	46,000	9	36,000	28,000	0.5	36,000	28,000	1
212	50,000	44,000	9	35,000	25,000	1	33,000	27,000	1
300	37,000	33,000	9	31,000	22,000	1	31,000	24,000	2
400	13,000	9,000	30	26,000	15,000	2	25,000	16,000	3
[illegible]0	8,000	5,500	50	18,000	10,000	5	19,000	12,000	6
600	5,000	3,000	70	10,000	4,000	10	12,000	8,000	15
700	3,500	2,000	90	5,000	3,000	45	8,000	6,000	25

2218-T72 - Not Available

January 31, 1963

TABLE III

TYPICAL PHYSICAL PROPERTIES

Alloy & Temper	Spec. Gravity	Weight #/in^3	Melting Range °F	Electrical Conductivity at 20°C (68°F) Cu Std.	Thermal Conductivity at 25°C (77°F) CGS Unit
2014-O				50	0.46
2014-T4	2.80	0.101	950-1180	34	0.32
2014-T6				40	0.37
2218-T72	2.80	0.101	940-1175	40	0.37
2219-O				44	0.40
2219-T31,37	2.83	0.102	1010-1190	28	0.27
2219-T62,81,87				32	0.30
4032-O	2.69	0.097	990-1060	40	0.37
4032-T6				35	0.33
A132-T551	2.72	0.098	1000-1050	29	0.28
F132-T5	2.76	0.100	970-1080	26	0.25

TABLE IV

CHEMICAL COMPOSITION LIMITS

	Si	Fe	Cu	Mn	Mg	Cr	Ni	Zn	Ti	Each Other	Total Other	V	Zr
2014	.5/1.2	1.0	3.9/5.0	.4/1.2	.2/.8	.10	-	.25	.15	.05	.15	-	-
2218	.9	1.0	3.5/4.5	.20	1.2/1.8	.10	1.7/2.3	.25	-	.05	.15	-	-
2219	.20	.30	5.8/6.8	.2/.4	.02	-	-	.10	.02/.10	.05	.15	.05/.15	.10/.25
2319(1)	.20	.30	5.8/6.8	.2/.4	.02	-	-	.10	.1/.2	.05	.15	.05/.15	.10/.25
4032	11.0/13.5	1.0	.5/1.3	-	.8/1.3	.10	.50/1.3	.25	-	.05	.15	-	-
4043(1)	4.5/6.0	.8	.30	.05	.05	-	-	.10	.20	.05	.15	-	-
A132	11./13.	1.3	.5/1.5	.10	.7/1.3	-	2.0/3.0	.10	.20	.05	.15	-	-
F132	8.5/10.5	1.2	2.0/4.0	.50	.5/1.5	-	.50	1.0	.20	-	.50	-	-

(1) When in form of welding electrode, beryllium content .0008 max.

TABLE IX

COEFFICIENT OF THERMAL EXPANSION AT VARIOUS TEMPERATURES

TEMP. °F	ALLOYS				
	2014	2218	2219	4032	A132
68-212	12.8	12.4	12.4	11.1	10.5
68-392	13.1	12.9	13.1	11.3	-
68-572	13.6	13.4	13.6	11.7	11.6

January 31, 1953

PISTON GROUP 42 - REFERENCES

42-1 Late Gillette & Eaton piston examination MET 3144

42-2 Most cost inserted carriers were bought to specification 31-P-5418 issue 1957. It was expanded but not issued anymore because cast carriers were then obsoleted. Specification 31-P-5414 issued in 1970 to cover the procurement of a few cast cast carriers for 12 1/2" pistons reflects the latest practice.

42-3 Cast aluminum carriers intermediate layer failure MET 3189
Typical carrier drawings
Casting 42-B-71131
Inserts 42-C-72052, 42-C-72056
Premachined carrier 42-C-71105
Premachined body 42-A-71101
Piston complete 42-B-71097
Typical cast carrier failure MET 2676, MET 2974

42-4 Special thick carriers 42-C-71115

42-5 Ductile iron pistons
Scoring MET 3103
Thermal stress cracking MET 3148

42-6 Properties of DC casting MET 3081

42-7 Report on first carriers after 555 hours operation MET 3158

42-8 Memo Tromel to Gaenger 12/16/63
See 863 387 folder, or piston file 1963

42-9 Typical loosening of insert
MET 3352, MET 3310 Item 3
Cracked insert MET 3330

42-10 Typical weld failure MET 3352
Also interim report Young to Gaenger 11/15/65 in
Forged carrier correspondence folder

42-11 Reversed oil flow MET 3354, 3348

42-12 Mahle piston examination
MET 3169 Interim and final 1963
MET 3342 - 1966

42-13 Mahle piston temperature report of 12/20/66, also description of steel cap piston - Mahle correspondence file

42-14 Drawings of Mahle type piston 98-EE-42413, 98-EE-42425, 98-EE-42426 also material MET 3390

42-15 Data of first strain gage test per MEL 3483 are questionable data recorded in monthly lab report 1-7 to 1-21 and 2-12 to 3-10-68 are more reliable, see folder Steel Cap Stress Analysis 1967

42-16 Oil flow MEL-3522.
Piston temperature MEL-3522-A.

SKETCH 46-1
GOODMAN DIAGRAM
120 000 psi
100 000
80 000
60 000
40 000
20 000
20 000
40 000
60 000
80 000
100 000 psi
SERIES 46 D 72003-72015
72020 PROTOTYPE FOR TYPES
16 0.5 RELIEF SUCCESSFUL
PRACTICE
Valve 2SI 72020
CURRENT OF 72028
1972
Valve 2SI-72015
Endu. Limit 72015
Fatig. Limit
Yield Strength

SPRINGS

Air and exhaust valve and yoke spring, 46-D-72023, and the former auxiliary fuel pump spring, 46-D-76085, are typical of what have been considered highly stressed but very satisfactory springs.

They are calculated in accordance with Alco engineering practice sheet and material follows specification 31-P-5606. Both were developed in cooperation with Raymond Spring, currently Alco's sole supplier. The stress range falls within the published Goodman diagram, Sketch 46-1, without shot peening. Shot peening provides the desired margin.

All important Alco springs call for shot peening, The notation on the drawing "shot peen to obtain maximum fatigue resistance" is loose; but Raymond claimed that they do determine peening variables by checking fatigue resistance on complete springs.

The 1/32 gap, specified between the spring ends and the adjacent coils, is to insure that no part of the adjacent coil is shielded from peening by the spring end.

Under normal circumstances failures of these springs are extremely rare; but two instances, one during 1971 and one many years ago, are recalled when a large number of springs on a single engine failed within short time. In the earlier instance, it was established that the springs had been laying during an engine overhaul in a pan in which a little water had somehow been left. The recent instance was after installing St. Louis overhauled heads on some seagoing vessel. The heads with the failure prone springs were changed out and no more failures were reported. Examination of some of the springs showed surface defects (corrosion pits?) and it is suspected that these heads, either at the dock or during shipment, had been exposed to sea air without adequate protection.

The torsion springs at the overspeed stop mechanism have worked sufficiently well so that the mechanism as designed could be continued in production, but it is not the most happy application of a torsion spring. In order to provide the torque needed to turn the control shafts, the springs are as highly stressed as possible. They are likely to have yielded slightly when they were cocked the first time.

In the early days when the control shafts were supported by needle bearings, which had a tendency to require high force to move the shafts (due to brinelling and cocking of needles), much time was spent on increasing work capacity of the springs (e.g. rectangular wire, also larger diameter wire). When the needle bearings were replaced by sintered bushings made of graphite containing bronze, efforts to change the springs could be terminated.

At the collapsible link, the drawing calls for setting the torque of the torsion spring on the governor support at assembly before clamping and pinning. It is known that the shop misunderstood this instruction in the past and sent out engines with this spring insufficiently wound up. This became clear on one of the more recent standby units; the setting was then explained to the shop people.

Experience with these two torsion springs leads to the recommendation to avoid torsion springs for any important functions in new designs.