Morgoil bearings used in rolling mills

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KOROGIL BEARINGS USED IN ROLLING MILLS

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SUMMARY

This subject is introduced with a description of the function of the bearings in rolling mills and the requirements of the bearings.

A description of the bearing parts is given in the next section. The next section describes the steps in the assembly of the bearing parts followed by a section on the preparation of the roll and the assembly of the bearing to the roll.

A section on lubrication of the bearings follows. It deals with the oil requirements and describes the action of the components in a typical lubricating system. The method of calibrating the oil flow to the bearing is also described.

The next section discusses the maintenance requirements of the bearings. This includes the responsibilities of both the mill and shop personnel in relation to the life of the bearings.

Some of the outstanding features found in the design of the modern bearing is given in the next section. This includes the sealing arrangement, the quick change assembly and hydrostatic lubrication.

The effect of the oil film in relation to mill performance and gauge control is discussed in the next section.

The last section involves the testing of bearings in relation to seal wear, position of oil inlets and bearing clearance. The results of some of these tests is given.
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INTRODUCTION

Morgoil is a trade name of the Morgon Construction Company in the U.S.A. who manufacture these bearings. First introduced in 1932, Morgoil bearings were welcomed by the rolling mill industry where roll neck bearing problems were hampering the development of faster and more powerful rolling mills.

The requirements of a roll neck bearing are:-

1. High continuous load capacity.
2. Low coefficient of friction.
3. A design permitting a large, strong rollneck.
5. Simplicity and ease of maintenance.
6. A design permitting quick and easy interchangeability from roll to roll.
7. Stability of operation.
8. Precision design and manufacture.

Mills using the oil-film bearing range from blooming and slabbing mills to high-speed 6-stand tandem cold mills in the steel industry and from hot mills to high-speed foil mills in the non-ferrous industry.

Morgoil bearings for use in rolling mills can be described as totally enclosed, flood lubricated, precision journal bearings that operate on a hydrodynamically generated film of oil.
The Morgoil bearing being of hydrodynamic design distributes the bearing load over a large area with no concentration points. The film of oil on which the bearing operates has immense load carrying capacity. Because of its continuous, unbroken nature, the film eliminates wear caused by metal-to-metal contact. This oil film is constantly maintained by the hydrodynamic action of the rotating sleeve to which a surplus of oil is presented at controlled temperatures.
Figure 2-1 Shows the Principal parts of the Morgeol Bearing.
DESCRIPTION OF BEARING PARTS

2-1 The Principal Parts

The principal parts of the Morgoil bearing are few and are of simple construction. Figure 2-1 shows a sectional view of the principal parts.

The rotating sleeve is an alloy steel forging, heat treated and ground to a mirror finish of approximately 2 to $4\frac{1}{2}$ micro-inches. The sleeve has a taper bore which fits on and is keyed to the tapered roll neck, thus becoming the journal.

The non-rotating bushing surrounds the sleeve. It consists of a steel shell lined with centrifugally cast, cadmium-nickel bearing metal, babbitt or a solid aluminium alloy, accurately bored to give a highly finished bearing surface which carries the radial load.

Adjacent to the outboard end of the sleeve is the thrust bearing which is either a double-acting roller or ball bearing, depending on the size of the radial bearing. The thrust bearing is independent of the radial bearing and can adjust itself to roll neck deflection and other types of misalignment common in rolling mill work.

These principal parts with enclosing chuck and the all important oil and water seals and end plates make up a compact roll neck bearing. The assembly is secured to the roll neck by a lock nut engaging a split, Threaded ring and is mounted and removed as a unit.
2-2 The Roll Neck Seal

Figure 2-2 shows a sectional view of the roll neck sealing arrangement. The roll neck seal consists of a seal end plate, the synthetic rubber neck seal with reinforcing band, the inner seal ring and the outer seal ring.

The seal rings and end plate are fastened to the chuck at the initial assembly and remain with the chuck without the need to disassemble for normal seal renewal or bearing inspection. The inner seal ring is made of high strength aluminium alloy to cut down inertia and provide corrosion resistance. The flexible neck seal may be easily installed and removed without disassembling the bearing.
Figure 2-2 Shows a sectional view of the roll neck sealing arrangement.
ASSEMBLY OF TAPERED ROLL NECK BEARINGS

3-1 Preparation of Bearing and Chuck

New bearings should be carefully dismantled and thoroughly cleaned so that all preservative coating is removed. After cleaning each part should be blown dry with clean filtered air and thoroughly coated with the same oil used to lubricate the bearings.

Chucks must be identified as top or bottom, thrust on non-thrust position. Only one Morgoil bearing with thrust is used per roll and this is used on the side of the mill as specified by each individual mill. The chuck must be thoroughly cleaned and all preservative coating removed prior to bearing assembly. The oil sumps are painted with oilproof enamel which should not be removed. The chuck is leveled on wooden blocks with the bore vertical and the outboard end on top.

3-2 With the Chuck Bore Vertical

The numbers of all the parts in the rest of this section refer to the fully assembled cross sectional view of the bearing shown in figure 3-3.

The bushing packing is a neoprene tubing inserted in the circumferential grooves in the bore of the chuck to prevent escape of oil between the bushing and the chuck.

Figure 3-1 shows the radial bearing bushing. The bushing(4) is the radial bearing and is made of a special bearing metal. Long eye bolts with a spreader bar are threaded into the lifting holes provided at the
Figure 3-1 The bushing for the radial bearing is made of a special metal.
outboard end of the bushing. The bushing is lifted and slowly lowered into the chuck bore. No force is to be used to fit the bushing in the chuck and the side holes are to be kept in line with the large tapped hole in the side of the chuck. Care must be taken to avoid scratching the bearing metal of the bushing.

Figure 3-2 shows a bushing been lowered into a chuck. The bushing lock screw(5) is a dowel to properly locate the bushing in the chuck. With the copper washer(6) it is threaded through the large tapped hole in the side of the chuck and into a hole in the side of the bushing.

The sleeve(7) which is the journal of the radial bearing is highly polished on the cylindrical outside surface. The bore is tapered to fit the taper of the roll neck. A steel bar of rectangular cross section, machined to match the tapered bore is used to lift the sleeve. The sleeve is thoroughly cleaned and oiled both inside and out, being extremely careful that the ground surfaces do not become scratched. The sleeve is then lowered at least halfway into the bushing, turned until the keyway is on the centreline at the normal top of the chuck and then lowered into place.

The sleeve ring(12) (used on thrust side only) is a spacer through which pressure is exerted by the inner race of the thrust bearing and locknut to tighten the sleeve on the roll neck. Eye bolt holes are provided in the out-board end for lowering into place. The ring is a close fit in the straight bore of the sleeve and the sleeve ring key(13) engages the keyway of the sleeve.

The thrust bearing assembly is shipped as an assembled unit. It is made up of the following parts:- roller thrust bearing, roller bearing
Figure 3-2 The bushing is lowered into the chuck bore by supporting it with a spreader bar. Note the bushing lock screw to locate the bushing in the chuck is to be fitted in the side of the chuck as shown.
housing, roller bearing end plate, end plate, coil springs and packing. This unit should be disassembled, thoroughly cleaned, reassembled and oiled.

The roller thrust bearing(8) is a steep-angle roller bearing. The outer races are a close fit in the bore of the roller bearing housing. This bearing is double acting to take the full thrust load of the mill in either direction.

The roller bearing housing(9) is machined to give ample radial clearance in the bore of the end plate to avoid any radial loading of the thrust bearing. It is keyed to the roller bearing end plate(10) to prevent rotation of the housing and outer race of the thrust bearing. The housing is thoroughly cleaned and the roller bearing carefully installed. The roller bearing housing, with its roller bearing in place, is then installed in the end plate after the coil springs(40) are in place. These coil springs are carried in the end plate and the roller bearing end plate to maintain proper alignment of the outer races of the roller bearing as the direction of the thrust load varies.

The roller bearing end plate(10) is thoroughly cleaned and the packing(38) of neoprene is pressed into the end plate groove. Coil springs are inserted in the holes provided using grease to hold them in place. Pick up the roller bearing end plate(10) with a crane using eyebolts in holes provided. Lower into place on the end plate(11) which contains the roller bearing and roller bearing housing. Be sure the roller bearing housing key engages the end plate keyway. Turn the roller bearing end plate until the oil inlet is pointing in the direction shown.
on the Morgan drawing. Insert tap bolts(39) provided and tighten in place.
Using eye bolts in the lifting holes that are provided, pick up this end
plate with thrust assembly utilizing a crane. Lower it partially into place
until all the tap bolts(29) with copper washers(32) on them can be started
in the tapped holes in the end of the chuck.

The outboard end plate packing(26) is a neoprene round body ring
which should be inserted between the end plate and the chuck outside the
bolt circle. When this has been done, the end plate assembly can be lowered
all the way and tightened in place. With the outboard end plate tightened
in place, the bearing is turned so that the axis or bore is in a horizontal
position and the drain hole of the chuck is downward.

3-3 With the chuck bore horizontal.

The seal end plate packing(28) is a neoprene round body ring which
is pressed into the circumferential groove of the seal end plate.

The seal end plate(22) is attached to the inboard end of the chuck
by cap screws. The seal end plate has two chrome plated surfaces which
afford maximum wear and corrosion resistance while contacting the rotating
neck seal.

The seal inner ring(24) is made of high strength aluminium alloy
for light weight and corrosion resistance. A resilient spacer(25) is at-
tached to the face of the inner ring. The seal inner ring maintains a tight
fit between the neck seal and the roll neck and presses against the roll
body with which it rotates. The seal inner ring is placed in the seal outer
ring for assembly on the seal end plate.
Figure 3-3 A cross-section through the bearing with the axis horizontal. All the parts are numbered as mentioned in the notes.
The seal outer ring (23) with the seal inner ring in place, is attached to the seal end plate with cap screws (21). The seal outer ring has a large opening for egress of any water passing between the seal inner and outer rings. This opening is always located at the bottom of the bearing.

The neck seal (19) is a moulded internally reinforced synthetic rubber ring with a spring-loaded lip. It rotates with the roll and is designed to ride the chrome plated surfaces of the seal end plate to form an interlocking labyrinth. The neck seal is collapsed inward for assembly, or disassembly and with the items above completes the inboard sealing arrangements, preventing loss of oil from the bearing or ingress of roll coolant to the bearing. Check that the neck seal is in position in relation to the seal end plate and also that the neck seal spring is still in place.
Figure 4-1 Method of using the gauge bar and micrometer for measuring the taper of the roll neck.
4-1 Preparation of Roll

Rolls to be used with Moigol bearings should be mounted on centres for the turning and grinding of the journal portion of the roll necks. The roll is then mounted on bearing pads for turning and grinding the balance of the necks and body. A gauge bar has been manufactured for measuring the taper of the roll neck with a micrometer as shown in figure 4-1. With proper care, it should not be necessary to regrind the roll necks during the life of the roll.

The roll is mounted on V-blocks or a specially designed assembly rack with the roll keys on top and on the vertical centreline of the roll. The roll neck should be cleaned. The projection of the keys on the tapered necks are checked, the key surface must be parallel with the roll axis and the key must not project more than that specified. Measure the roll body length and roll neck taper. The roll necks should be coated with the same type of oil used to lubricate the bearing.

The threaded half rings(15) fit in the groove of the roll neck and are held together with dowel cap screws(36). They are keyed to the roll neck to prevent rotation.

The roller bearing key(37) used on thrust bearing end of roll is a hardened roller which is a loose fit in the keyway of the roll neck and the inner race of the roller bearing.
Figure 4-2 The chuck is mounted on the roll neck by using the overhead crane and lifting the chuck on as shown.
4-2 Mounting the Bearing on the Roll.

Before mounting the bearing on the roll a check should be made to insure alignment of the sleeve keyway with the roll key and the roller bearing keyway with the roller key. The chuck can now be picked up and mounted on the roll neck, as shown in figure 4-2.

The locknut(14) is lightly coated on the threaded portion with a mixture of white lead and lubricating oil or other anti-seizure product. Thread the locknut on the threaded half rings and tighten the nut by hand. The final tightening is done by inserting a round steel pin in one of the holes is the periphery of the nut, attaching a cable to the pin and pulling up tight with a crane as shown in figure 4-3. The locknut is properly tightened at a point where the roll assembly is about to be lifted off the V-blocks.

After the locknut has been properly tightened the locknut key(16) is inserted in the keyway of the roll neck and in one of the half-round keyways of the locknut. This key is unsymmetrical and can be installed either way to suit the requirement of keeping the locknut tight. Should these not be properly aligned, keep a strain on the cable with the crane and, with a hammer, top the face of the locknut. This should make the nut move enough to align the keyways. The key will prevent the nut from backing off during operation of the mill.

The locknut key retainer(17) is a snap fit in the groove at the outboard end of the locknut and will prevent the locknut key from being dislodged during operation of the mill.

The end cover packing(27) is a neoprene round body ring and is
Figure 4-3  The locknut is tightened up by inserting a pin in a hole in the periphery of the nut, attaching a sling to the pin and pulling up tight with the crane.
pressed into the groove of the roller bearing end plate.

The end cover(18) is the last piece of the bearing to be assembled. It is bolted to the end plate with tap bolts(30).

The locking block(33) and set-screw(34) are used on the top roll only, and only when installing roll in housing or when changing rolls by "C" hook or porter bar. It is bolted into place after the end covers are removed. The purpose of this set screw assembly is to lock the chuck to the roll, thus preventing the chuck from rotating while changing rolls.

The assembly of bearings for the side opposite thrust is the same as that outlined in sections 3 and 4 except that there is no thrust bearing.
LUBRICATION OF MORGOIL BEARINGS

5-1 Oil Requirements

Morgoil bearings are lubricated with straight mineral oil, of specified viscosity delivered to the bearing at a constant temperature and in predetermined amounts. The viscosity is selected to meet the speed and load conditions of each particular installation.

The lubricant supplied to lubricate Morgoil roll neck bearings must be a high grade straight mineral oil, free from acid and other impurities, and must have a high resistance to oxidation and to the formation of sludge when subjected to hard service.

It must separate rapidly from water and other liquid impurities and must show a minimum demulsibility of 1602 at 130°F for oils up to and including a viscosity of 350 seconds Saybolt Universal at 100°F. It must show a minimum demulsibility of 1620 at 180°F for oils heavier than 350 secs. Saybolt Universal at 100°F. Demulsibility to be determined by method 320-32. Demulsibility Test for Lubricating Oils as described in Federal Standard Stock Catalogue, Section 4 Part 5.

The viscosity of the oil to be used will be specified by the manufacturer, this specification giving maximum and minimum seconds Saybolt Universal at 100°F.

Minimum viscosity index (from Dean and Davis Chart) is 80 and this index is based on actual viscosity at 100°F and actual viscosity at 210°F.
Periodic laboratory analysis of the oil by the operating company, oil supplier, or independent company should be made to assure that the original quality has not diminished. Particular attention should be paid to the water content, viscosity and neutralization number. The water content should not exceed 2% and the viscosity held within a range of 100 seconds Saybolt Universal at 100°F. The neutralization number should not exceed 2.0 but any abnormal rate of change should be investigated at once.

5-2 Typical Lubrication System.

The circulating system serving Morgoil bearings varies with different types of mill installations. A typical lubrication system is given below and shown diagramatically in figure 5-1.

Receiving tank (A)

Most systems have two separate tanks. Occasionally a single tank or a single tank with double compartments will be used. Tanks are designed so that they will allow 40-50 minutes resting time for settling out any foreign matter and temperature balancing before the oil is re-circulated. Steam coils or electric elements built into the tank are used for heating the oil in conjunction with a temperature regulator to maintain the operating tank oil temperature. The tanks are built with a floating suction. The floating suction is used at all times when supplying oil to the bearings. In small systems the suction is a fixed pipe high enough from the bottom of the tank to insure that clean oil is being drawn off. Oil Level indicators are supplied with all tanks. Oil level in the tanks should be maintained at a minimum of two thirds full at all times. In a two tank system, it is recommended that alternate tanks be used each week. The oil temperature is then raised in the idle tank to 160°F-170°F and held for a
Figure 5-1  Diagramatic arrangement of a typical lubrication system.
24 hour period, then allowed to cool. During this time a mechanical purifier can be used.

**Pumps (B)**

Two constant volume pumps are generally used in a system. One pump serves as the main operating pump and the other as a stand-by pump. The pumps are wired so that they can be switched periodically with each pump serving as the main unit for an equal period of time. The stand-by pump is automatically controlled by a pressure switch and is always ready to cut in should the main pump fail to maintain pressure. An alarm sounds whenever the stand-by pump goes into operation, so that immediate attention can be given to determining the cause of low pressure.

**Pump Relief Valves**

For pump protection, relief valves, set at 65 pounds per square inch, are installed in the line of the discharge side of the pumps.

**Main System Filter (C)**

Following the line of oil flow, the next main item after the pumps is a twin basket filter. This filter is designed so that a rapid switch can be made from one basket to the other and each basket is capable of filtering the full discharge of one pump. It is equipped with magnetic elements and stainless steel wire baskets of 100x100 mesh.

At a 5 pound per square inch pressure drop across the basket an alarm will sound indicating a dirty basket. This alarm is for emergency purposes only and a scheduled maintenance inspection should be set up so that a switch is made from one basket to the other at frequent intervals.

**Oil Cooler (E)**

The oil cooler is installed in main supply line after the filter.
The purpose of the cooler is to control the oil temperature at all times. The temperature is controlled either by manual or automatic valves.

**Pressure Tank (F)**

A pressure tank is connected to the main supply line on the discharge side of the oil cooler. The purpose of this pressure tank is to provide an even flow of oil by eliminating the pulsation effect from the supply pump. In addition, there is a quantity of oil in the tank to enable the mill to operate for a brief period of time in the event of a pump or lubrication system power failure. In general, there is enough time to clear the mill of material before shutting down.

The oil level in this pressure tank should be maintained at approximately one third of the volume of the tank, with air making up the balance. The switch controlling the standby pump and low pressure alarm are operated from the air side of this tank.

**System Pressure Control Relief Valve (D)**

The relief valve for system pressure control is operated by a pilot line connected to the main supply line on the discharge side of the cooler. The valve is located between the main filter and the oil cooler. It is set to maintain a constant pressure on the system. The final setting will be such that the differential across the mill stand pressure reducing valves will be 10–15 pounds per square inch.

**Mill Stand Pressure Reducing Valve (G)**

Pressure reducing valves are used at each stand to regulate the oil pressure required to insure the proper specified oil flow to the bearing. These regulators are set at the time of oil flow calibration and should not
require changing under normal operating conditions.

By-Pass Shut off Valve.

This valve is in a line connecting the Morgoil bearing oil supply and drain line. It is used principally for precirculation of warm oil prior to mill start up and drain back for roll or riser strainer basket charge. This valve is never open when the mill is in operation.

Mill Stand Riser Strainer (H)

Riser strainers are installed in the riser piping on each side of the mill stand. These strainers are equipped with stainless steel baskets of 100 X 100 mesh.

Mill Stand Riser Instruments

The instruments for each mill riser consist of a pressure gauge, thermometer, and a low pressure alarm switch. The switch is set after the mill riser pressure has been established.

5-3 Calibration Of Oil Flow

The purpose of calibrating oil flow to Morgoil bearings is to assure that the correct quantity of oil at the proper temperature will be available to the bearing at top operating speed of the mill. Since oil flow depends upon proper operation of the lubricating system, its set-up and operation must be checked.

A laboratory type thermometer stop watch, and two clean containers are the tools used for calibration. One container should be of sufficient capacity to hold at least 25-30% of the maximum gallons per minute (g/m) flow required to any radial bearing and it should be marked in one gallon increments. The other container should be a one gallon measure for checking oil to thrust bearings.
At each mill stand, oil flow is measured to the top bearing which is farthest from the oil cellar. Depending on the installation, the thrust bearing may, or may not, have a separate oil supply.

The lubricating system is normally set up prior to oil flow calibration. With the preliminary adjustments completed, the system should be shut down and preparations made at the mill for calibration.

At each mill stand to be calibrated, disconnect the radial supply hose at the bearing and insert it into a drain riser from which the breather cap has been removed. If the thrust bearing has a separate supply, this hose may be drained into the tee at the bearing drain. Metering nozzles should be checked for size and correct location.

With the system set for calibration, it may be desired to speed up oil circulation in order to heat the system piping. This is easily done by opening by-pass valves at the mill stands. With these open, the stand-by pump may operate since the metering nozzles will be by-passed and the oil pressure therefore, greatly reduced. By-pass valves must be closed for oil flow calibration and mill operation.

Oil temperature at the mill stand is easily checked by raising the radial supply hose from the drain riser so the laboratory thermometer can be inserted in the oil flow. This temperature should be compared with that shown on the riser thermometer to insure its accuracy.

1. With the riser temperature established and steady, note the riser pressure. Direct the flow from the radial supply hose into the calibrated container and note the time required to fill it to a known quantity. Calculation will show the actual $g/m$ delivered from this hose.
2. If necessary, oil flow can be adjusted by changes in pressure, using the regulating valve at the pressure reducing station.

3. The flow to the thrust bearing should be measured and noted if a separate inlet is used. This is done once the pressure has been established for the correct radial flow. Oil flow to the thrust bearing is adjusted only by altering the metering nozzle. Past experience indicates that such alteration is unlikely, as flow to the thrust bearing is acceptable within plus or minus 25%.

4. With calibration completed, raise the calibrating pressure one pound per square inch. This then becomes the operating pressure. The reason for this is that when the mill is operating at top speed, the riser pressure may drop slightly. When operating at top speed, the mill riser pressure gauge should not read less than the calibrating pressure.

5. Where two pressure regulating valves are used per stand, the mill supply riser pressures should be the same. Where only one regulating valve is used, the supply riser closer to the oil cellar will have a slightly higher pressure.

6. Each mill riser has a low pressure alarm switch, and these must be set and checked. Settings are made so that the alarm will silence at one pound per square inch below the operating pressure and will sound as close to this setting as possible. This is usually 1 - 1½ pound per sq. in. below the silence setting. Checking of these alarm settings is accomplished by slowly opening and then closing the by-pass valve at the mill riser.

It is recommended that a close watch be kept on all pressures while the system is new. Initially, baskets of the filter and riser strainers
should be checked once a day to establish a practical cleaning program. An alarm should not be relied upon as a signal to clean a dirty basket. Alarms are for emergencies only.
MAINTENANCE OF MORGEOIL BEARINGS

6.1 Mill Responsibilities in Regard to Bearings

Good maintenance is preventive as well as corrective and should be the responsibility of both operating and mechanical departments. Preventive maintenance, in relation to oil film bearings involves the proper mounting of the bearing on the roll necks and location of the assembly in the mill housings.

To avoid excessive axial thrust or end play, rolls should be held in close parallel alinement. Excessive end play may result in rapid wear of seal rings and rubbing of adjacent rotating parts that should run with axial clearance.

To ensure satisfactory roll alinement, periodic checks should be made along the following lines:

1. Mill housing windows should be plumb, with side faces of windows at right angles to the centreline of travel through the mill. New liners should be installed when the windows become badly worn.

2. With only the bottom backup roll in the mill, the top of the roll should be level. The liners and pressure blocks under the bottom chucks should be checked for wear.

3. The work roll and backup roll chucks should be checked regularly for wear.

4. With each roll change the fit of the locking clamps should
be checked and clamp bolts tightened.

5. If excessive thrust and end play at clamps, due to wear, is not corrected, the following may result:-

a) Excessive lateral motion of the rolls when steel enters the stand.

b) External wear on retainer rings, bolt heads, and other parts of the backup roll or bearing that should have axial running clearance.

c) Greater thrust bearing or thrust bearing bracket maintenance.

d) Bearing failures, due to the heat developed by parts, that should have axial clearance, rubbing together.

e) Greater oil consumption.

Also clean lubrication must be introduced to the bearings at the specified temperature and in the proper quantity.

The bearings must perform under exceedingly adverse conditions such as proximity to the high temperatures of the metal been rolled; the presence of roll cooling water, often under fairly high pressure, and the hazard of abrasive mill scale, sometimes driven against or toward the bearing closures by high pressure hydraulic descaling jets.

6-2 Bearing Shop Responsibility

An important responsibility of the bearing service personnel is a close dimensional check of the tapered roll necks and the roll body length. This is of particular significance since the work side and drive side chucks containing stationary bearing parts are fixed in position in relation to mill housings by housing keepers or latches, or by other means, to guarantee the desired centre - to - centre distance of bushings. Therefore, correspond-
ing accurate centreline - to - centreline distance between roll neck journals must be maintained. In the event of either a shorter or longer roll body, (or taper necks ground undersize or oversize) the taper bored rotating sleeves would position improperly and might cause endwise interference between flanges on the sleeve and the bushings of the floating side bearings.

When a bearing is being transferred from one roll neck to another, it is a good practice to examine the condition of the seals and make necessary replacements. At the same time, an inspection should be made of the chuck drains. These normally contain nothing but lubricating oil but if particles of foreign matter are found there, the bearing should be dismantled for a general inspection. During the inspection, if a bushing is found to be scored by foreign matter, the scoring can be lightly scraped. Any small scratches on the sleeve can have high, sharp edges removed by a hard Arkarsas stone. An ordinary grinder may be used to remove upset areas on the flange or shoulder, which are caused by sleeve interference at the outboard end of the bearing.

6-3 Bearing Shop Maintenance

The frequency of disassembly and inspection of Morgoil bearings can be established by experience only. Seal wear is the major factor in determining the frequency of disassembly, but it should not be the only consideration. Operating conditions may have placed excessive thrust loads, or misalinement may have caused damage to the bearing, which would require disassembly to make necessary repairs.

After the bearing has been disassembled the chuck should be thoroughly cleaned. The sleeve and bushing should be inspected for evidence of
foreign material which may have scratched the sleeve or imbedded itself in the bearing metal of the bushing. Upset areas in the soft shell of the bushing may be repaired by scraping with an ordinary bearing scraper. Any foreign material imbedded in the bushing should be dug out and the sharp edges left should be broken by scraping.

The bushing should be inspected for evidence of corrosion or etching due to acid in the lubricating oil. Most cadmium lined bushings will show evidence of corrosion or etching almost from the time they are placed into operation. A moderate amount of this type of discoloration is to be expected in all cadmium bearings and will not have a detrimental effect on its operation. If severe etching is found an immediate check of the mill oil system for excessive acid is required.

Very little work other than thorough cleaning may be done to the sleeve. The one exception to the use of abrasive materials on Morgoil bearings is in the area of the I.D. of the sleeve. Carbon like deposits collect on the I.D. of the sleeve (as shown in figure 6-1) and on the roll neck, due to rolling pressures; they are pressure oxides. These deposits must be removed with emery paper or a fire hard emery stone, inasmuch as continued accumulation will destroy the taper fit required between the sleeve and the roll neck.

The thrust bearing should be disassembled at intervals for inspection. It should not be necessary to disassemble this bearing each time the basic bearing is disassembled unless the operation is one which has either high speed or excessive thrust conditions. Evidence of wear in the thrust bearing may be observed by examining the exposed roller ends without removing
Figure 29 — Carbon-like deposits collect on the inside diameter of the sleeve due to rolling pressures.
the bearing from the case. All springs and neoprene tubing should be replaced. The cover tap bolts should be inspected for necking caused by thrust loading; they should be replaced if there is any evidence of wear.

In shops where a tank with solvent for cleaning the chuck is available, it is advisable to repaint the chuck after it is cleaned. Rocker plates should be inspected for wear and positioning before the assembly is returned to the roll neck. All parts in the chuck, as they are assembled, should be oiled with the same oil used in mill operation.

The bearing should be mounted on a roll which is the same temperature as that of the bearing. If the bearing and roll are not of equal temperature the bearing should be mounted on the roll, but the nut positioning the bearing should not be tightened for at least an hour. If the neck is much warmer than the bearing, the sleeve will become loose on the neck when the parts assume the same temperature during operation. If the bearing is much warmer than the neck, the lock nut will move the sleeve too far on to the taper; when the temperatures become equalised, the fit will be too tight and it will be difficult to remove the bearing from the neck.

It is desirable to maintain accurate service records of Morgoil bearing parts. Upon each card a complete history can be kept of the component parts of each bearing. Generally bearing life is rated as a function of operating load and operating hours; therefore, this card system affords as accurate check of these factors, since both tonnage rolled and operating hours can be recorded for each component part.

Due to operational load distribution, the relative location of the bearing in the mill is important; therefore a record should be kept of the bearing location in the mill for each particular installation. It is
particularly important that the bushings be rotated through 180 degrees periodically, insuring even load distribution. The bearing service records should include information concerning this rotation, in addition to notes on the work performed on each part and the date of service to these parts.

Figure 6-2 shows the precision measuring of the inspection equipment used by bearing shop personnel. It also shows the working tools and safety equipment.
Figure 6-2 The top half shows the precision measuring and inspection equipment used by bearing shop personnel. The bottom half shows the working tools and safety equipment.
MODERN MORGOMOL BEARING DESIGN IMPROVEMENTS

7-1 Sealing Arrangement

Although sealing is not necessarily a bearing characteristic it should be mentioned because of its relation to the bearings ability to exist in a difficult environment.

The Morgoil roll neck seal keeps oil within the bearing while roll coolant and scale are kept out as shown in figure 7-1.

The flinger portion of the synthetic rubber neck seal diverts oil into a deep sump within the chuck, from which it passes to the drain through internal passages. The one piece neck seal with no joint and a static spring - loaded lip afford positive sealing against leakage from the sleeve and roll neck interface.

A non rubbing series of rotating and stationary gutters provides ample capacity for draining away roll coolant. Interlocking deep labyrinth seal legs and end plate dam further block oil contamination. The inner seal ring is made of high strength aluminium alloy for corrosion resistance and low inertia. Neck seal legs rub against a chrome - plated surface of the seal end plate.

To prevent clogging, by mill scale, the roll coolant drain has a generous area.

Rubbing seals are not necessary on "dry" mills such as Temper mills; the spring loaded lip and flinger keeps oil in and away from the strip.
**Figure 7-1** Top left - detail of "wet" seal. Top centre - "dry" mill seal arrangement. Top right - internal reinforcing ring shown. Bottom left - seal components. Bottom centre and right shows the ease of installation of the neck seal.
The internal reinforcing ring (shown in figure 7-1) keeps the neck seal tight at high speeds. The flexible neck seal can be quickly removed and replaced without loosening a single screw (as shown in figure 7-1).

For sealing between the bushing and the chuck, neoprene tubing is used in grooves in the chuck bore at each end of the bushing to prevent oil being fed to the bearing from by-passing the chuck bore and bushing.

The sleeve packing is a neoprene tubing which seals the inboard end of the sleeve and the area between the sleeve and the roll neck. This tubing must be carefully installed in the groove provided as this is the only area of the bearing at which oil can escape under pressure. Pressure at this point is caused by oil grooves on the inside diameter of the sleeve that allow the oil, which is under great pressure from the rolling elements to enter this area. Oil loss in this area can be great enough to necessitate the removal of the rolls from service.

7-2 Quick - Change of Bearing Assembly

The quick change arrangement designed cuts the number of pieces that must be removed for a bearing change to a bare minimum just an end cover clamp ring. A sectional view of the quick-change oil film bearing is shown in figure 7-2.

The locknut is turned on the threaded ring and moves outboard until the locknut flange contacts the chuck end plate. Continued rotation of the locknut row causes the threaded ring to move inboard against the shoulder of the roll. On further rotation the locknut "jacks" the chuck outboard until the bushing contacts the sleeve flange and breaks the bearing.
Figure 7-2 A sectional view of the quick-change oil film bearing.
free. The clamp ring is now removed and the bearing can be demounted from the roll neck.

The locknut and threaded ring remain integral with the chuck and bearing when changing from one roll neck to another. In addition the locknut has been designed in such a manner that it is double acting:

1. It performs the task of pushing the bearing firmly into place on the roll neck and holding it there.

2. When removing the bearing it becomes a powerful mechanical "puller" for unseating the taper bore sleeve from the roll neck.

This "quick change" feature, combined with the traditional non-locking taper, offers the quickest and simplest means available to change heavy bearings.

7-3 Hydrostatic Lubrication

When extremely low starting torques are necessary hydrostatic lubrication can be added. Hydrostatic pressure takes over when the mill speed drops below a predetermined point and maintains the oil film in the load zone.

From the technical standpoint, the oil-film bearing traditionally used on roll necks is classified as hydrodynamic; i.e., it actually takes relative motion of the bearing elements to develop the oil film. The oil film develops in less than one half revolution of the journal and, once established, the fluid film has an extremely low coefficient of friction. As it is dynamic the bearing has the characteristic that it requires motion to sustain the oil film, however, another characteristic of the bearing is that even though the bearing comes to rest the film persists for some time.
This static persistence of the oil film permits satisfactory mill performance in the vast majority of applications—even where stalled and reversal conditions exist.

Hydrostatic lubrication is used in that distinct minority of applications where superior operating characteristics at slow speed and stalled conditions are required.

This hydrostatic principle has been in operation on a few rolling mills for a number of years, both in the ferrous and nonferrous industries. In these cases, hydrostatic lubrication was applied as an afterthought, either because the mill had extremely low starting torque capabilities, or because of special operating requirements—such as the operator desiring to start without easing rolling pressure after the mill had been at rest under full rolling pressure for an unduly long period of time.

Special bushings are machined to provide high pressure oil pads in the centres of the load zones. These pads are fed from holes in the flange of the bushing at which point the Hydrostatic oil connections are made. A check valve and 90° elbow is installed. The elbow open end faces the hose passage in the bearing chuck at an angle of approximately 8° from the normal horizontal and towards the bore of the bushing. The bushing is now placed in the chuck in the normal manner.

Figure 7-3 schematically shows a modern hydrostatic system applied to a rolling mill. Note that the system takes its oil from the normal oil-film bearing supply. The system is completely automatic and the oil film is maintained during slow speed, stalled or reversing conditions. It is important to note that the system is "fail-safe"; should there be any malfunction of the hydrostatic system a check valve immediately adjacent to
Figure 7-3  A schematic arrangement of a modern hydrostatic system applied to a rolling mill.
the bearing prevents reverse flow, therefore assuring normal oil-film characteristics.

Figure 7-4 shows a simple hydrostatic bearing that will serve to explain the principle. In this case a stationary disc is shown with a high pressure oil supply to it; on top of this stationary disc is shown a mating disc with a rebore area. The load is applied to the top disc. In order to have the lowest possible coefficient of friction it is now desired to separate these two discs and to hold them separated by an oil film. If oil is now introduced under high pressure through the stationary disc to the rebore area, it can be seen that the pressure times the area of the rebore cavity is the basic lifting capacity of this bearing.

After the initial lift, the oil then flows past the outer land in accordance with well defined physical laws. The oil is further exerting pressure over the land area as well as the rebore area. The pressure of the oil is still maintained at a high level in the rebore area, but as it flows outward over the land the pressure decreases until it is atmospheric at the outer edge. Thus the total load capacity of this bearing is the sum of the inlet pressure times the area of the high pressure rebore plus the land area times the pressure gradient over this area.

The thickness of the oil film, or the distance between the two plates, is then a function of the quantity and viscosity of oil being pumped through the system. The two surfaces are completely separated and when one surface moves with respect to the other, the only friction force is that due to the shearing of the fluid film.
Figure 7-4  The hydrostatic bearing can eliminate the effect of fluid film thickness from the gage problem.
OIL FILM EFFECT ON MILL PERFORMANCE

The effect of the dynamic characteristics of the oil film on gauge control is being investigated by modern mill designers. The oil film is developed and functions according to very precise physical laws. The equation which relates the variables that are involved, is very complex and was first defined by Reynolds in 1886.

The fundamental equation developed by Reynolds for flow in three dimensions is illustrated by the differential cube of figure 8-1 in which the volume of the fluid going into the element equals the volume going out at any instant.

Over the years many people have worked with this equation in an effort to simplify it for their particular use. The most useful approach to the solution of this equation for rolling mill applications has been the work performed by Boyd and Raimondi. The results of their computer solution of the Reynolds equation are offered in a series of graphs. These graphs, when applied to oil-film bearings used in rolling mills, give predictions that very closely check field and test stand observations.

Automatic gauge control is a significant item in mill performance. From figure 8-2 the delivered gauge ($H_2$) of a mill is defined by the equation:

$$H_2 = S_o + \frac{F}{M}$$

Where $S_o$ = unloaded roll opening

$F$ = separating force

$M$ = mill modulus
Figure 8-1 Fundamental equation developed by Reynolds for flow in three dimensions is illustrated by the differential cube in which the volume of the fluid going into the element equals the volume going out at any instant.
Therefore any quantity which affects roll gap must be analysed for its true effect on delivered gauge. In an oil-film bearing, the dominant characteristic that could affect this roll gap would be the oil-film thickness. Thus, by means of Reynolds equation, examine what would happen to an oil film under two basic conditions that are encountered in a rolling mill—namely, change in load and change in speed.

The oil-film thickness will decrease with an increase in load, and will increase with a decrease in load. The significance of the load dependent change lies in its magnitude and its effect on the total mill modulus. Theoretical predictions and experimental tests show that the oil film is the stiffest element that is involved in carrying the rolling load. Its modulus is so high in proportion to the other load carrying elements that its effect on the mill modulus is nil. It is also of interest to note that there is no known dynamic load carrying element with an equal or higher modulus that could be applied in the space available. Thus, as a function of load change, the oil film is the stiffest dynamic load carrying element available.

As a function of speed, the oil-film thickness increases with an increase in speed, and decreases with a decrease in speed. In order to put these changes in their true perspective, one must consider what other changes occur in a rolling mill during speed change; for example, roll bite friction, rolling lubricants, the rate of compression of the metal etc.

It has been found that as the mill speed increased and the rate of compression of the metal increased it took more rolling pressure to make a given reduction. There is evidence that indicates that on high-speed mills
Figure 8-2  Formula for delivered gauge ($h_2$).
as the speed increases the effectiveness of the rolling lubricant decreases, and it is necessary to increase the rolling force in order to hold gauge. It appears that from starting to maximum operating speed there are several different conditions of rolling efficiency that are reached or passed through. When the mill first accelerates from thread speed, the rolling lubricant becomes more effective and less separating force is needed for a given reduction. As the speed of the mill continues to increase, the rolling lubricant effect reverses and this together with the more pronounced increase in strain rate results in a steadily increasing separating force necessary to hold a given gauge. Eventually a plateau is reached where there is little speed effect on delivered gauge.

Figure 8-3 illustrates this phenomenon.

The data shown were obtained from the No. 2 stand of the J and L. 3-stand tandem cold mill at Aliquippa. While generating this data the incoming gauge of *0102" and the front and back tension were constant. The mill was operated at 4 speeds from 137 r/m to 3,900 r/m and the delivered gauge, the separating force and the speed at each point were recorded. The results are shown for 2 different screw settings.

A further interesting fact can be established if one now with similar experimental technique holds the speed constant, but changes the screw setting and observes the effect on delivered gauge. Figure 8-4 shows the results obtained by this technique on the No. 2 stand of the same mill.

In this case results for 2 speeds are illustrated; 500 f/m and 4,000 f/m. The entering gauge was *0102" and the target delivered gauge was *0072". Note the low roll gap transfer function for this type of mill
Figure 8-3  As speed increases, the separating force needed to hold gage gradually increases until a plateau is reached.
Figure 8-4 Curves show gage vs. screw position for the No. 2 stand of a 3-stand cold reduction mill with entering gage 0.0102 in. and delivery gage of 0.0072 in. Note the vertical scale spread is five times the horizontal scale.
"051" per inch at 500 f/m and "013" per inch at 4,000 f/m. The total screw position change was "030" for each of the 2 speeds. This relatively large screw change resulted in a large separating force change; i.e. for a mill modulus of $30 \times 10^6$, the separating force change is 900,000 lbs. but the delivered gauge changed very little. Tin plate consumers report this mill is producing a product that has superior gauge and shape.

Thus as a function of speed, these tests show that the oil film change is not dominant, is predictable and in this case moves in a direction to reduce the net control correction necessary to hold gauge during acceleration. It is of interest to note that modern tandem cold mills can get on gauge within a few hundred feet per minute. A performance review of recent thin tin-plate mills in several operating plants reveals excellent gauge performance on all. This confirms the fact that modern gauge control is doing an excellent job in controlling all mill changes that take place during acceleration.
SECTION 9

TEST STAND OBSERVATIONS

9-1 Seal Check

Due to tremendous loads, bearing size, speed, power requirements and method of loading, the best way to test a roll neck bearing is in a rolling mill. Due to the time involved and the consideration of production, these tests have been necessarily of short duration and confined to pressure tests for rolling specific material or schedules, oil flow and temperature tests and material or design evaluation: These tests have been very helpful in solving the specific question at the time, but they also emphasized the need and the desirability of further testing of bearings under laboratory conditions.

This point is well illustrated by a neck seal recently introduced to rolling mill builders and operators. The basic design of the seal was conceived in the engineering department and was given its first field test in the initial design stage at a Canadian steel plant on a 25 and 49 X 56 inch 4 high reversing hot steckel mill. A set of seals was supplied and in due course, they reported excellent sealing efficiency and little wear. This test proved the seal on a reversing hot mill having a speed of 720 to 1800 f/m, or a backup roll bearing r/m of 58 to 145 and using water as roll coolant.

Before the seal could be released for general use, two rather important questions had to be answered: how would the seal perform at 6,500 f/m mill speed and 2,500 r/m roll speed as encountered in a rod mill finishing
A testing device was made up to test seal efficiency.
train; and how would the seal material react at high rubbing speeds and without lubrication as might be encountered in a Temper mill?

To test the efficiency of the interlocking laybrinth a simple machine was built with a mock-up of the labyrinth and roll neck, as shown in figure 9-1. This machine tested the seal from 0 to 2,000 r/m with a concentrated jet of water flowing at the rate of ten gallons per minute. Under these conditions there was absolutely no visible water or mist on the stub roll neck side of the seal.

To test the seal at high rubbing speeds and without lubrication another machine was built. This simple test machine was driven by a 1 H.P. 1,800 r/m motor. The rotating disc was made of the same synthetic rubber compound that is used in the seal element. The discs used had a 9-9/16" diameter which based on an r/m of 1,750, gave a rubbing speed of about 4,400 f/m. A test load of 3lb. was used which based on the scar area gave a loading of about 35 psi.

In the actual seal the static element is a ductile-iron ring. In the test machine ductile-iron bars coated with various materials were used. A recording wattmeter was also connected to the motor. With this setup, which coating had the longest life and also the relative friction for the various materials tested could be determined.

The various coatings tested were:

<table>
<thead>
<tr>
<th>Coating</th>
<th>Friction (kw)</th>
<th>Life (hr.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bare ductile iron</td>
<td>0.5</td>
<td>--</td>
</tr>
<tr>
<td>Chrome plate 0.0005&quot;</td>
<td>0.45</td>
<td>5.5</td>
</tr>
<tr>
<td>Copper Plate 0.002&quot;</td>
<td>0.6</td>
<td>24</td>
</tr>
<tr>
<td>Cadmium plate 0.0015&quot;</td>
<td>0.6</td>
<td>2</td>
</tr>
<tr>
<td>Coating</td>
<td>Friction (kw)</td>
<td>Life (hr.)</td>
</tr>
<tr>
<td>-------------------------</td>
<td>--------------</td>
<td>------------</td>
</tr>
<tr>
<td>Chrome on flash copper</td>
<td>0.003&quot;</td>
<td>0.55</td>
</tr>
<tr>
<td>plate 0.003&quot;</td>
<td></td>
<td>Greater than 2</td>
</tr>
<tr>
<td>Acrylic 0.004&quot;</td>
<td></td>
<td>Greater than 2</td>
</tr>
<tr>
<td>Teflon 0.0015&quot;</td>
<td></td>
<td>2</td>
</tr>
<tr>
<td>Epoxy 0.0004&quot;</td>
<td></td>
<td>2 - 0.8</td>
</tr>
<tr>
<td>Teflon 0.015&quot;</td>
<td></td>
<td>0.45</td>
</tr>
<tr>
<td>Chrome 6.0025&quot;</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The life of the various coatings tested varied from less than one minute up to 60 hours and the relative friction ranged from 450 to over 2,000 watts. In all cases where the coating lasted an appreciable length of time, it was noted that the friction was high for the first 2 to 3 minutes and then reduced to its final normal level. It was found that any lubricant introduced to the rubbing parts immediately raised the friction, but not to any serious degree. These two test machines which have been briefly described, merely illustrate how design improvements have been tested.
9-2 Test Machine

Much more elaborate equipment is needed to test and evaluate a complete bearing. The Aluminium Co. of America designed and built a test machine. This machine was designed to load a test bearing up to 180,000 lb. and has a speed range of 350 to 1,400 r/m.

The following desirable objectives of a bearing test machine were set up:

1. The test bearing should be a full-sized bearing.
2. The test bearing should be mounted and loaded in the same manner as encountered in a rolling mill.
3. The test bearing should be loaded as much as 200% of rated capacity when desired.
4. The test bearing speeds should be similar to those used in rolling mills.
5. The test bearing instrumentation should be such that load, speed, friction, temperature and oil flow could be measured and recorded.

With the above objectives in mind, the 9½" 70 series bearing, which has a rating of 106,000 lb., was selected. This bearing is commonly used on rod mill finishing trains operating at speeds up to 2,500 r/m.

Figure 9-2 shows the test shaft. Note that the main and auxiliary shaft bearings have been mounted in typical rolling mill fashion on hardened rocker plates to permit alignment of the bearings for shaft deflection. The geometry of this design is such that with the test bearing carrying the maximum test machine load of 180,000 lb., the main bearing would be loaded 270,000 lb. and the auxiliary bearing 90,000 lb. The installed
Figure 9-2 A test shaft and bearings set up for a bearing load of 180,000 lb.
main roller bearing has a rated capacity of 283,000 lb. at 500 r/m and 3,000 hr. minimum life. The auxiliary bearing is rated at 8,700 lb. on the same basis.

The 9½" test bearing is top loaded through a hydrostatic breaker block so that friction characteristics of the bearing can be measured. The mounting and loading of the oil-film test bearing is similar to that of a top bearing in a rolling mill. The usual practice for oil-film bearings in a rolling mill is to design the bearing on one side with a thrust unit (flat, ball or roller) and the bearing on the other side is free floating in the axial direction. The floating design was selected so that the test results would not be affected by the thrust bearing performance.

The test machine is driven by a variable-voltage D-C drive, rated at 40 H.P. 0-1,100-2,200 r/m. With the mechanical and electrical features of the machine fixed, an attempt was made to predict the power and torque requirements of the oil-film test bearing based on earlier experiments.

Figure 9-3 shows the theoretical power and torque of the test bearing operating from 20 to 2,200 r/m. This was based on the assumption of using an oil having a viscosity of 2,600 S.S.U. for the speed range of 20 to 200 r/m, and an oil with a viscosity of 200 S.S.U. for the speed range of 400 to 2,200 r/m. Note that the 2,600 S.S.U. oil curve is shown only up to 200 r/m, which is about the normal design limits.

Figure 9-3 also shows the torque and horsepower curves of the D-C electric drive. At high speed the torque and horsepower requirements of the test bearing approach the torque and horsepower capacities of the drive. Since this graph was based on 100% loading and only considers fric-
Figure 9-3 Theoretical curves of torque and horsepower using two different oil viscosities in the test bearing. The torque and horsepower curves for the motor are also shown.
tion of the test bearing and neglects the friction of the shaft support bearings, it was realized that the testing would be somewhat limited at high speeds and high load. In actual operation the drag of the anti-friction shaft bearings was such that the top speed of the machine without a test bearing or load was 1,700 r/m. With the $9\frac{1}{2}$" test bearing loaded to 140,000lb. or 133% of its rated capacity, the top speed of the machine was about 1,100 r/m.

The test bearing and the machine bearings are lubricated by separate circulating oil systems. To lubricate the two roller bearings and the ball thrust bearing of the test machine shaft, the manufacturer recommended a 15 gallon per minute system using a straight mineral oil with a viscosity of 600 S.S.U.

The maximum oil requirement for the $9\frac{1}{2}$" test bearing at speeds up to 2,200 r/m as figured in the usual manner is 2.2 gall/min. The test program contemplated evaluating the effects of increased oil flow. A lubricating system for the test bearing, with a capacity of 5 gall/min. was installed. Thus more than two times the usual oil requirement was available.

Figure 9-4 shows a close up view of the test bearing and gauges.

Parts as numbered are:-

(2) Test bearing
(5) Load cell - for bearing load
(6) Hydrostatic breaker block
(7) Load cell - for bearing friction

For the initial series of test, three $9\frac{1}{2}$" bearings were used,
Figure 9-4 Test bearing and gauges.
two with steel-backed babbitt bushings and one with a solid aluminium bushing designated as bearing A, B and C respectively in Table 1.

**TABLE 1**

Dimensions of 9¾" Test Bearings

<table>
<thead>
<tr>
<th>Bearing Designation</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>Babbitt</td>
<td>Babbitt</td>
<td>Aluminium</td>
<td>Babbitt</td>
</tr>
<tr>
<td>Chuck Bore (in)</td>
<td>7.8745</td>
<td>7.8745</td>
<td>7.8745</td>
<td>7.8745</td>
</tr>
<tr>
<td>Bushing O.D. (in)</td>
<td>7.8720</td>
<td>7.8725</td>
<td>7.8668</td>
<td>7.8725</td>
</tr>
<tr>
<td>O.D. Clearance (in)</td>
<td>0.0025</td>
<td>0.0020</td>
<td>0.0077</td>
<td>0.0020</td>
</tr>
<tr>
<td>Bushing I.D. (in)</td>
<td>7.1320</td>
<td>7.1387</td>
<td>7.1369</td>
<td>7.1387</td>
</tr>
<tr>
<td>Sleeve O.D. (in)</td>
<td>7.1243</td>
<td>7.1243</td>
<td>7.1243</td>
<td>7.1194</td>
</tr>
<tr>
<td>I.D. Clearance (in)</td>
<td>0.0077</td>
<td>0.0144</td>
<td>0.0126</td>
<td>0.0193</td>
</tr>
</tbody>
</table>

Oil flow and oil temperature were of immediate interest, since these variables are easily measured in the field and provide a convenient means of anticipating and analysing service problems. In most bearing investigations of this type, it is general practice to measure bushing temperature by means of thermocouples or similar devices in order to define completely the bearing characteristics. Previous field test experience showed that thermocouples, which are located approximately 0.015" below the bushing surface, have occasionally been the cause of bearing failures. For this reason, thermocouples were omitted from this series of tests.

Each bearing was tested at 27,000; 54,000; 81,000; 108,000 and 140,000lb. loads corresponding to approximately 750; 1,500; 2,250; 3,000 and 4,000 psi. bearing pressures with each of the oils listed in Table 2.
### TABLE 2

**Specifications of Test Oils**

<p>| | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Viscosity, S.S.U. at 100°F</strong></td>
<td>212</td>
<td>2630</td>
</tr>
<tr>
<td><strong>Viscosity index</strong></td>
<td>93</td>
<td>95</td>
</tr>
<tr>
<td><strong>Viscosity</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Centistokes at 100°F</td>
<td>45.62</td>
<td>567.10</td>
</tr>
<tr>
<td>Centistokes at 210°F</td>
<td>6.29</td>
<td>32.40</td>
</tr>
<tr>
<td><strong>Specific Gravity</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>AT 100°F</td>
<td>0.858</td>
<td>0.879</td>
</tr>
<tr>
<td>AT 210°F</td>
<td>0.818</td>
<td>0.840</td>
</tr>
<tr>
<td>Demulsibility federal</td>
<td>1620 at</td>
<td>1530 at</td>
</tr>
<tr>
<td>method 320 - 32</td>
<td>130°F</td>
<td>180°F</td>
</tr>
<tr>
<td>Neutralization value</td>
<td>0.05</td>
<td>0.03</td>
</tr>
</tbody>
</table>

The maximum speed for each combination of load and viscosity was limited by the available torque of the drive motor at weak field. In any case, however, the maximum speed for each load was not less than 1,000 r/m. Prior to each run the machine was operated at 300 to 400 r/m under light load for approximately one hour. After warm-up, the test load was applied and the machine run at the starting speed until equilibrium conditions, as determined by oil drain temperature, were reached. The required data was then recorded. This procedure was followed for each speed change. The speed was increased in approximately 200 r/m increments until the maximum speed was reached that could be attained without exceeding the current rating of the motor. The speed was then decreased in increments until
Figure 9-5 A 90° and a 180° inlet passage were drilled for admission of oil to the bearing.
reaching the speed at which the motor stalled.

The oil flow was metered by a nozzle in the supply line. However, for this test a calibrated metering valve replaced the standard type fixed nozzle in order to easily change the calibration if desired. The valve was set to deliver one gall/min. of oil at 100°F with a 15 psi. pressure drop, which is the volume that would normally be supplied for a maximum operating speed of 1,200 r/m. The oil was admitted to the bearing from the rebore located 90° before the load as shown in figure 9.5. For ease of identification this inlet will be referred to as the 90° inlet. For the purposes of orientation this is equivalent to the left hand oil inlet of a top bearing with the shaft rotating clockwise when viewed from the outboard end. This is the standard arrangement for an installation with a single oil inlet.

Figures 9-6 and 9-7 represent typical data on the oil flow and oil temperature rise obtained from this series of tests. This particular data is for babbitt bearing A using the 200 S.S.U. oil. Oil flow increased with increasing speed but decreased with increasing load. Oil drain temperature increased with increasing load and speed. The oil flow for babbitt bearing B was greater than for babbitt bearing A, which reflects the effect of greater clearance. However, the change in oil flow between minimum and maximum loads for both bearings is approximately equal in spite of the difference in clearance. The drain temperatures for both bearings were of the same order of magnitude.

The oil flow at 27,000 lb. for babbitt bearing B and aluminium
Figure 9-6 The oil flow decreased as the speed decreased and also as the load increased.
Figure 9-7 Oil drain temperature rises as both speed and load increases.
bearing C with clearance of 0.0144 and 0.0126 in. respectively was nearly equal, but at 140,000 lb load bearing C had approximately 15% greater flow. Since the coefficient of expansion of aluminium is approximately twice that of steel, the operating clearance of the aluminium bearing is undoubtedly greater than the babbitt bearing at comparable load and speed which would account for the greater oil flow. Oil drain temperatures observed for bearing C were approximately 15 to 20°F lower than for the other two bearings.

The oil flow and oil temperature rise for those tests using 2,600 S.S.U. oil have the same general characteristics, but the oil flow is approximately 50% lower and the temperature rise is nearly double for corresponding loads and speeds. Bearing B failed while running at 900 r/m under a load of 140,000 lb. This operating condition is several times more severe than would be experienced under mill operating conditions. At the time of failure an oil drain temperature of 305°F was observed. In view of the high drain temperature, it is probably safe to assume that the bushing temperature was considerably higher, possibly exceeding the melting point of the babbitt and that this caused the failure.

One of the advantages of this type of bearing is that the sleeve can be reconditioned when necessary. Figure 9-8 shows the effect of clearance on oil flow for three babbitt bushings A, B and D. Operating with a load of 54,000 lb. and using a 200 S.S.U. oil, increasing the clearance 2.5 times results in an increase in oil flow of only 18%. In view of the fact that oil flow decreases with load and the nozzles are selected to give the required flow at maximum speed, there is a sufficient capacity in the lubricating system to accommodate the increased oil flow resulting from
Figure 9-8  Increase in oil flow due to increased clearance is approximately 18 per cent at full speed.
the permissible increase in clearance due to reconditioning.

The next series of observations were made using bearings C and D, aluminium and babbitt respectively, which were fitted with thermocouples. Each bearing was equipped with seven thermocouples, six in the load zone and one in the unloaded side, as shown in figure 9-9. Since this chuck was equipped with two independent oil inlets, each supplying a separate rebore, it was decided to observe the effect of oil inlet location upon oil flow, drain temperature and bushing temperature. As shown in figure 9-5 these inlets are located $90^\circ$ and $270^\circ$ before the load in the direction of rotation. As in the tests previously described, the metering valve was calibrated to deliver 1 g/m of 200 S.S.U. oil at 100°F and 15 psi. pressure drop. The oil flow obtained for each oil inlet with aluminium bearing C are shown in figure 9-10. The $90^\circ$ inlet as previously noted, shows a decreasing oil flow with increasing load, while the $270^\circ$ inlet indicates that oil flow increases with increasing load. For the $270^\circ$ inlet, regardless of load the oil flow increases to a maximum of approximately 1.1 g/m. In this case the flow is undoubtedly limited by the nozzle.

Figure 9-11 shows the influence of speed and load on bushing temperature for the two inlets. This data is for thermocouple no. 1 which was located in the center of the load zone. The other thermocouples showed the same characteristics, varying only slightly in magnitude. As expected, bushing temperature increases with increasing speed and load. However, the bushing temperature is as much as $40^\circ$F lower using the $270^\circ$ inlet in comparison to the $90^\circ$ inlet. This decrease in temperature can be attributed to the greater oil flow and improved distribution.

Figure 9-12 compares the bushing temperatures of bearings C and
Figure 9-9 Two bearings were fitted with thermocouples placed as illustrated.

Figure 9-10 Oil flow also varies with regard to the inlet placement.
Figure 9-11 The bearing temperature varied up to 40°F depending upon the oil inlet used.
D under the same conditions. As previously noted the bushing temperatures using the 270° inlet are lower than with the 90° inlet. The aluminium bushing operates at a lower temperature than the babbitt bushing. Since the oil flow for the aluminium bushing was less than for the babbitt bushing, the lower temperature is attributed to the better "thermal" characteristics of aluminium.

The relationship between oil drain temperature and bushing temperature at the centre of the load zone of bearing C is shown on figure 9-13 for both oil inlet positions. This data is from the tests using a standard nozzle. For the 270° inlet the bushing temperature is approximately 20° lower than for the 90° inlet for equal drain temperatures. It appears that this relationship between drain and bushing temperatures is independent of load and speed, which could make it extremely useful. The slope of these curves will undoubtedly be different for bearings operating in the field because of the effect of the roll housing and for example, the presence of roll cooling water, on the radiation capacity of the chucks.

The investigation carried out this far represents only a small portion of the total program. For the present the program has been concerned with determining the characteristics of standard bearings as a basis for comparing and evaluating other bearing designs and materials. Currently, equipment is being designed which will permit the measurement of oil-film thickness under varying operating conditions. This is extremely important considering the high operating speeds encountered in modern mills.
Under the same loading the aluminum bushing operates cooler than the babbitt bushing.

Figure 9-12
Figure 9-13 There is about a 20 degree difference between bearing temperatures, at equal oil drain temperatures, with respect to the inlet used.
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