Self Contained Bearings and their lubrication

SELF CONTAINED BEARINGS AND THEIR LUBRICATION

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Conveyor bearing performance testing

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SYNOPSIS Non-precision ball bearings are examples of self-contained bearings used in humble applications such as conveyors, trolleys, and farm implements. There would be a far greater market for these low cost bearings if their performance potential and lubrication requirements were better known and understood.

1 INTRODUCTION

Non-precision (also known as semi-precision) ball bearings are well established components used where precise location is not needed. See Table 1. They differ from the much better known precision ball bearings in being made without finish grinding and from cheaper steels to keep the cost down.

Manufacturers of non-precision bearings are generally small companies with low overheads and limited technical resources. As a result the bearing life, ratings, and other detailed performance data which is available for precision bearings does not exist for their non-precision counterparts.

Some work carried out at Leeds has allowed the wear and lubrication of these bearings to be studied in the laboratory.

2 GRAVITY CONVEYOR ROLLERS

A typical gravity conveyor roller comprises a welded steel or plastic tube with a bearing at each end. The bearing has an inner ring machined with a circular section groove to form the inner raceway. A pressed steel ring formed to a shallow U shaped cross section serves as the outer raceway. A one piece plastic moulding retains the balls. Figure 1 shows the various components that make up the bearing. Simple plastic or steel rings act as shields to keep the grease in and dirt from contaminating these grease-lubricated-for-life bearings.

Raceways of these bearings are used in the as-pressed or as-profile-turned conditions with salt bath casehardening and a metal anti-corrosion flash being the only subsequent treatments.

Performance or design guidance provided for the bearings is generally limited to a statement of load capacity at a certain speed. For example, a 20mm ball pitch diameter bearing with nine balls of 4mm diameter had a stated capacity of 100 kg at 40 r/min. The bearings were lubricated on assembly with a number two penetration grade premium quality EP grease.

Data from the Leeds tests substantiated this claim by giving a rating load for one year's operating life of these gravity conveyor roller bearings of 9.2 kN each.

3 OVERHEAD CONVEYOR TROLLEY WHEEL BEARING

An example of a trolley wheel bearing designed to run on an overhead steel joist is shown in Figure 2. Inner and outer rings are machined from free-cutting steel to form a deep groove ball bearing.

Performance data for these 'heavy duty' bearings is given as a guide by the manufacturer in the form of, for example, '850 lb load capacity at 140 r/min or '500 lb load capacity at 250 r/min. The former applied to a 44.4 PCD double row ball bearing and the latter to a 12.7 mm bore 47.6mm outside diameter conveyor bearing.

Trolley wheel bearings of the type shown were life tested at 110 r/min under a load of 2 kN. Lives in excess of 1000 hours were achieved, accompanied by wear at the rate of, typically, 0.25 μm per hour.

Bearing failure was more the result of the grease drying up and ceasing to lubricate effectively than due to fatigue. The bearings tested had bulky sintered iron cages which limited the amount of grease which could be packed into the bearing to act as a reservoir of lubricant. The lubricant used was molybdenum disulphide loaded grease. Metal shields were fitted to both sides of the bearing.

Dynamic load rating C values were calculated for the bearings using the approach established for precision bearings given in ISO R281. This immediately caused a problem in the definition of bearing life. For precision bearings it is the number of revolutions which the bearing is capable of enduring before the first sign of fatigue flaking occurs. With non-precision bearings, as described later, fatigue flaking occurs as part of the bedding-in process and is present at an early stage in the bearing's life without any detriment to the performance of the bearing. Failure of non-precision bearings was therefore taken to be evidence of distress such as continual noise, vibration, rough running, or excessive wear.

Using these criteria the C value (the load to cause failure in one million cycles) derived from the tests was a quarter that of an equivalent precision ball bearing. Failure was caused by inadequate lubrication due to deterioration of the grease.

4 WEAR OF NON-PRECISION BALL BEARINGS

Precision rolling element bearings are manufactured with great accuracy to close tolerances, from special steels, and to high quality surface finishes. As a result very little running-in is necessary and any wear beyond slight polishing is considered detrimental to the performance of the bearing.

For non-precision bearings, on the other hand, the running-in process acts as a vital post hardening mechanical treatment. In place of grinding, the raceways are subjected to rolling burnishing. Rolling, or roller burnishing, is a metal-working technique employed to strengthen the surface of a component, in much the same way as shot peening does. In both processes the residual stress distribution is rendered more favourable with a resulting increase in resistance to fatigue (1).

Figure 3 shows that the higher the carbon content of the steel, or the greater the initial surface hardness, then the greater will be the hardening effect (2).

In addition to this metallurgical effect the running-in process will improve the surface finish of the raceways and enhance the contact geometry between ball and raceway, so reducing the surface stresses. The process is typified by surface pitting and polishing, similar to the behaviour of gear teeth. This initial pitting is due to surface fatigue which continues only until the overstressed local high areas have been reduced, thus increasing the area of contact to support the applied load. Such pitting is not considered serious on gears since it is corrective and non-progressive. Running-in of non-precision bearings is probably helped by the use of leaded free-cutting steels. The dispersed lead phase would help to initiate fatigue cracks and speed up the removal of high spots in the bearing raceways. This, of course, is entirely contrary to the philosophy of the precision bearing manufacturers, who continually search for ever 'cleaner' steels with the metallurgical structure entirely free from inclusions.

With greased-for-life shielded non-precision ball bearings there is the problem of the steel detritus created during running-in, a problem not faced by gears where the wear debris is continually washed away by the lubricant. Eventually the grease in the bearing becomes over-loaded with solid matter, to its increasing detriment as a lubricant.

Figure 2 and Figure 4 illustrate the polishing effect of the running-in process, showing examples of surfaces before and after running.

5 COMMENTS

Life tests on non-precision bearings have shown that they have the potential for much wider application than is found at present. Bearing in mind that the great majority of rolling element bearings fail for other reasons than fatigue; due to contamination by dirt, due to corrosion, to misalignment, mishandling and inadequate lubrication, for example; then the lower fatigue rating of a non-precision bearing is not an overriding factor.

The lack of precision is exemplified by out-of-roundness values being about ten times greater than would be expected for a precision bearing. This, together with the initial surface finish of 1 μmRa or greater, would limit the use of non-precision bearings to low and moderate speed applications.

An area of great interest for the development of this type of bearing is its use for large diameter bearings. Precision bearings become increasingly expensive in large sizes, where the advantage of economy of large scale manufacture is lost. Most of the shrinkage and distortion of the bearing rings as a result of the surface hardening process can be predicted and allowed for by careful design and by collaboration with the heat treatment specialist, who now has a wide variety of processes to offer. A low distortion process like ion nitriding which has been successfully used for hardening gears without the need for subsequent grinding is probably still too expensive for non-precision bearings.

Effective lubrication of non-precision bearings will always be more difficult than precision rolling element bearing lubrication. During the early stages of running, when the raceway surfaces are rough and osculation is poor, there would be no chance of elastohydrodynamic fluid films being generated between the metal surfaces. Wear would occur with the production of ferrous wear debris. By the time the bearing had run-in a substantial amount of fine solid particles would have become mixed in with the lubricant. Ideally this contaminated lubricant should be regularly displaced by fresh, clean grease or oil throughout the life of the bearing.

Oil wick lubrication has attractions. The wick would absorb the wear particles and filter the oil automatically before it was recirculated back to the bearing rubbing surfaces. Good design could make the wick act as a lubricant reservoir, a filter and as a bearing seal.

Grease, however, is almost universally used as the lubricant for non-precision bearings. To optimise the performance of the bearings the following recommendations are offered:

(a) Use a high viscosity base oil, suitable for low speed, boundary lubricated bearings.

(b) Use a base oil which contains anti-wear additives.

(c) A grease meeting the above requirements will often be marketed as a number 2 NLGI EP premium grade grease.

(d) Because much oxide and metallic solid matter will be created by the bedding-in wear process, lubricants already containing solid matter, albeit solid lubricant, should be avoided.

(e) Design the bearing to leave as much space as possible around the rolling elements which can be filled with grease to act as a lubricant reservoir for the bearing. A pressed steel cage is better in this respect than moulded plastic or sintered metal cages.

(f) Provide means of periodically expelling the contaminated grease and introducing fresh grease into the bearing.

6 REFERENCES

(1) Rolling bearings - methods of evaluating dynamic load ratings. ISO Recommendation R281, 1962 (E).

(2) Parrish, G. The influence of microstructure on the properties of case-carburised components. Part 8. Post hardening treatments - mechanical. Heat Treatment of Metals, 1977.4, p107-116.

(3) Papshev, D. D. Increasing the fatigue strength of high tensile steels by work-hardening. Russian Engineering Journal, 1970, Vol L, No 1, p38-42.

Table 1.
Applications of non-precision bearings

Conveyors	Gravity conveyor rollers
	Overhead conveyors
	Production line track conveyors
	Return rollers
Farm implements	Disc harrows
	Hay rake tine bars
	Power take off roller
General engineering	Cam followers
	Plummer or pillow blocks
	Idler pulleys and sprockets.
	Rope or belt sheaves
	Take up units
	Sliding doors, windows and shutters
	Lawn mowers
	Industrial cleaners
	Skate wheels
	Orthopaedic chair wheels
	Trolley wheels

Table 2.
Typical non-precision ball bearing specifications

Balls (AFBMA 200)	Roundness tolerance	5 μm
	Diameter tolerance	±25 μm
	Surface roughness	0.2 μmRa
Inner and outer ring (machined from tube or solid)	Free-cutting mild steel, casehardened (1.0 to 1.2 mm case depth, 58 to 63 HRC hardness). Raceway profile machined with a form tool. Inner and outer raceway diameters gauged and matched to give required clearance, eg 0.1 to 0.18 mm for a 44 mm PCD bearing. Raceway out-of-roundness, up to 0.25 mm. Bore and outside diameter tolerances 0.08 and 0.13 respectively.	
Cage	Injection moulded nylon or sintered iron.	
Lubricant	No 2 grade grease with EP or solid lubricant additives.	

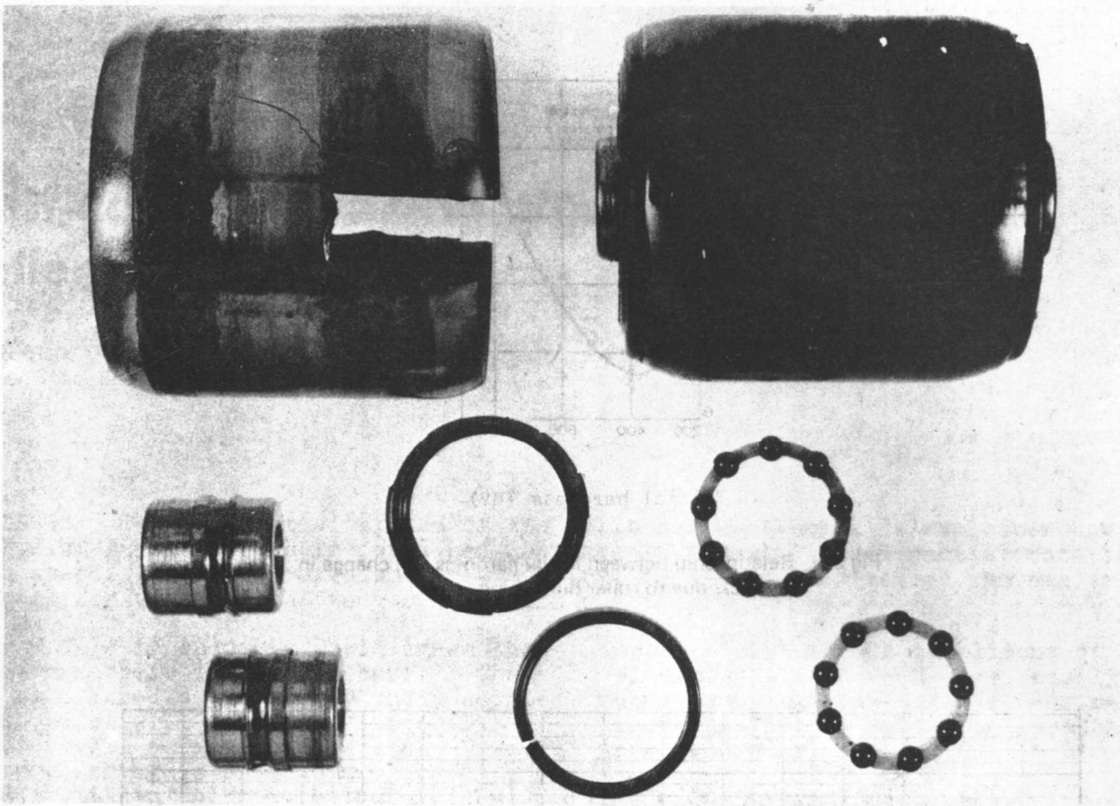


Fig 1 Tested gravity conveyor roller and bearing components

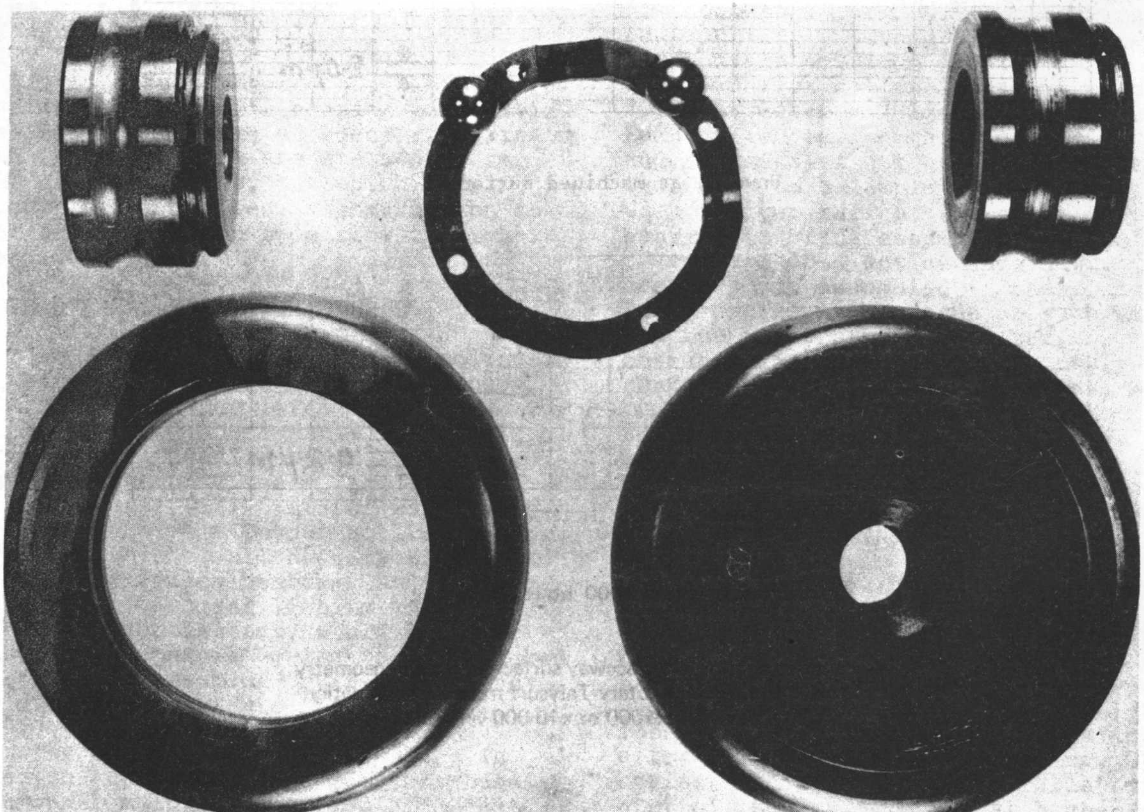


Fig 2 Tested trolley wheel bearing components

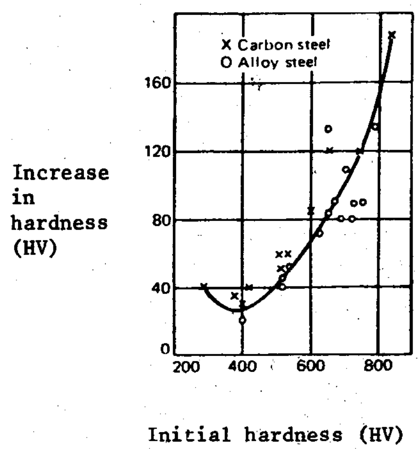
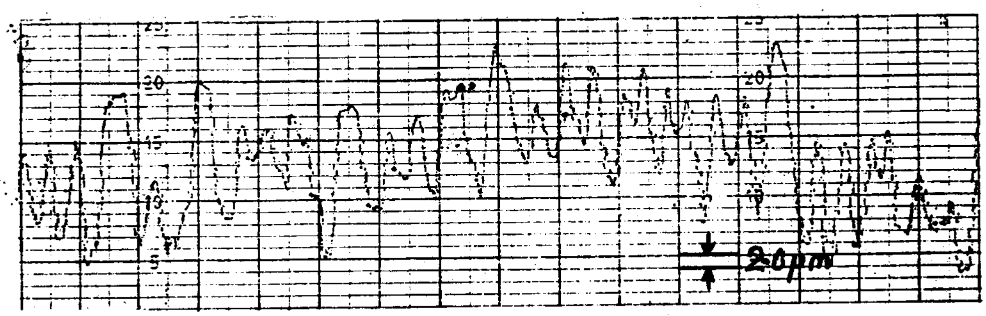
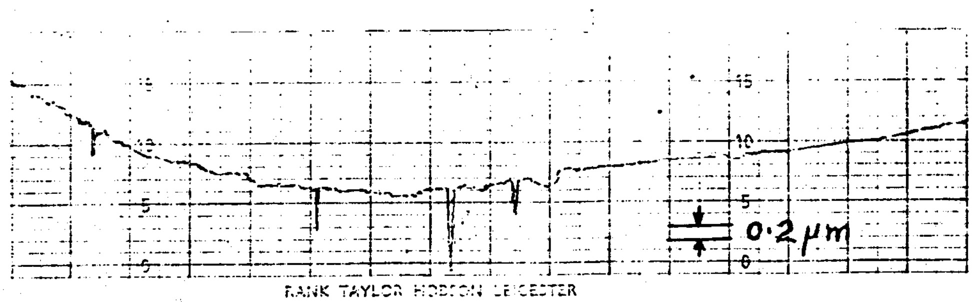


Fig 3 Relationship between initial hardness and change in hardness due to roller burnishing



Unworn, as machined surface



Surface after 1000 hours running

Fig 4 Change in bearing raceway surface finish and geometry due to running (Rotary Talysurf traces: magnification x20 horizontal; x1000 or x10 000 vertical)

Self contained lubrication system for split roller bearings

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SYNOPSIS The lubrication system of the split roller bearing is explained and the method of lubricating self contained bearings in pedestals from partial rotation to high speed. Examples also show how bearings and their lubrication systems are adapted for vertical shafts and on thrust bearings.

Most people in industry would agree that it serves their industries best if they can be offered a package by their suppliers, which will work satisfactorily between well defined parameters. However as engineering is both an art and a science, it is inevitable that the extremes of the parameters are hazy and capable of being extended. This is the case with self contained rolling bearings in pedestals, flanges and other types of enclosures.

It is probably most convenient to consider the types of bearings and their lubrication in terms of speed, starting with bearings which are stationary for most of their lives, but which turn through a few degrees, and moving on to continuous rotation from slow to high speed.

1 ZERO SPEED TO PARTIAL TURNS

Rolling contact bearings in pedestals can be packed full of grease when they are the 'sealed for life' type with factory charged lubricant. In normal environmental conditions and provided they have adequate seals, the single pack of grease will last for some years. For example a 75mm Cooper roller bearing, 01 series which would be very similar to NU or N1015 roller bearing, and is lubricated with a normal lithium grease of number two consistency, would last for 60 000 hours before having to be repacked with grease, assuming Cr/Pr of 5 (C rating divided by dynamic radial load).

The estimation of grease life is based on the Engineering Sciences Data Unit publication, number 78032 'Grease Life Estimation in Rolling Bearings'. It is probably the most comprehensive study of grease life that has been produced, taking into account every conceivable parameter, except contamination of the grease through dirt or moisture.

For rolling bearings in the regime of partial rotation, it is often best to operate with a full compliment of rolling elements, which can be most easily achieved by using split roller bearings for which no filling shots or loose end rings are necessary.

One typical application for this type of bearing is the trunnion support bearing for large bulk oxygen steel converters, where there is a tilting action of the vessel for pouring, charging and 'de-sculling'. These bearings range in shaft diameters from 700 to 1150mm and have parallel rollers in them up to 100mm diameter x 150mm long and over. Each roller may weigh in excess of 10 Kg, and so, special but quite simple techniques have to be employed to ensure that, when the top half of the split roller bearing is being assembled, the column of rollers does not cascade down. The bearings are assembled in pedestals and fully packed with grease able to withstand the temperatures involved. Relubrication needs to be made reasonably frequently i.e. once a month, to help keep the grease free from contamination, and to lubricate the seals adequately.

2 FROM PARTIAL ROTATION TO FULL PACK OF GREASE, dn = 50 000 (SHAFT DIAMETER IN MM X REV/MIN).

The full row roller bearing is capable of rotating at speeds up to dn = 15 000 (shaft diameter in mm x rev/min) with a full pack of normal lithium based ball and roller bearing grease giving only minimum wear, typically a weight loss on each roller of only .014 gms after 200 hours for a 6" bearing, 01 series. This is a loss in roller diameter of .0012mm. To run at higher speeds it is necessary to use much softer greases or oil with a higher viscosity, i.e. in the region of 300/400 centistokes at 40 degrees C., and having EP (extreme pressure) additives. One such application is the main bearings on a large reciprocating diaphragm pump, where the bearings are split parallel roller bearings up to 320mm, rotating at

50 rev/min. This is a self contained bearing in a pedestal, using a hypoid back axle oil of SAE90 viscosity.

It is always desirable to fill a rolling bearing and its housing with as much grease as possible provided the bearing does not heat up through excessive churning of the grease. The less air space left within the housing, the less likely it is that the bearing will become contaminated by moisture from condensation.

The Cooper split roller bearing mounted in its cartridge can be packed full of grease up to dn = 50 000 (shaft diameter mm x rev/min) and beyond. This is a high enough speed for most mechanical handling applications. However, although a full pack of grease may last for a number of years, if the bearings are operating in a very dirty environment it is best to relubricate more often, so as to push the old grease out past the seals as shown in Figure 1. As can be seen the fresh grease enters the Cooper bearing centrally onto the roller path and forces the old grease out.

An additional reason for lubricating more often is so as to replenish the grease on the thrust walls of the fixed bearing, shown in Figure 2. Normally, if the fixed bearing has to take a certain amount of axial thrust, it is desirable to relubricate every 100 hours or longer according to duty and experience.

Because most maintenance engineers are familiar with the fact that rolling bearings can heat up, if they are overfilled with grease, there is a tendency not to use enough grease on assembly. One such application which suffers this way is the intermediate shaft bearings in ships, between engine and propeller. Because some of these bearings are large, 400 to 600mm rotating at about 100/200 rev/min, it is assumed that the bearing parts should only be smeared with grease and that the housing need not have much in it either. But because of the extremes of humidity and temperature causing condensation, it is best to fill the bearing and cartridge with as much grease as possible during assembly, provided excessive churning does not take place. A useful rule for the amount of grease to use in a Cooper split roller bearing, assembled into its standard cartridge is shown below:

dn (shaft dia. mm x rev/min) over	dn (shaft dia. mm x rev/min) up to	% of full cartridge
	50 000	100
50 000	100 000	70
100 000	150 000	50
150 000	200 000	40
200 000		20

A guide to enable the bearing user to judge whether the correct quantity of grease has been used is shown in Figure 3. With the correct amount of grease in the bearing and cartridge, the temperature climbs to a maximum after 2 to 3 hours, depending upon speed and then drops away, as shown.

3 FROM A FULL PACK OF GREASE, dn = 50 000 TO THE LIMITS OF GREASE LUBRICATION.

All that has been said above applies to medium to high speed grease lubrication. For the parallel roller bearing it is much more difficult to define the upper limit of speed for grease lubrication. If the user is prepared to lubricate a 'little and often', the upper limit can be very high. For example a light series parallel roller bearing designated 04 series for the Cooper split roller bearing supports the hollow tubes of tubular stranders as encircling bearings and operates at speeds up to dn = 900 000 (mm x rev/min). This is a 900mm bearing running at 1000 rev/min. It has to be lubricated with two to three shots of grease from a grease gun every 8 hours. A number three lithium grease is normally used.

In order to overcome the problem of supplying a small amount of grease to a rolling bearing at frequent intervals or continuously, the spring loaded external lubricator has been improved by using the chemical reaction of two fluids which form increasing amounts of gas which gradually expands against a piston, to extrude the lubricant at a controlled rate into the bearing. This type of lubricator can be made to handle a number of different types of greases and oils. In the case of the Cooper split roller bearing, it is probably most useful for medium to high speed bearings on horizontal shafts, but has a more definite use on any application where a continuous supply of lubricant is essential.

For example full row bearings and vertical shafts are two types, where it is difficult to lubricate with oil without complicated means to recirculate it, and it is no longer self contained.

Another type of bearing which requires a continuous supply of grease is a parallel roller thrust bearing. As soon as the speed is faster than dn = 5000 (mm x rev/min) it is advisable to specify oil lubrication or an external grease lubricator as described above and made under the trade name 'Greasomatic'. Obviously there are great advantages in being able to lubricate with a continuous supply of grease rather than oil. Even an oil bath can be more difficult than grease.

4 OIL LUBRICATION IN A SELF CONTAINED UNIT.

For speeds above approximately dn = 220 000 (mm x rev/min) for medium or heavy (01 or 02 series) split parallel roller bearings, it is usual to use oil for lubrication, because of the difficulty of controlling the supply of grease in small enough quantities and often enough, when using an ordinary hand gun. Clearly overgreasing at high speeds is more critical.

Types of self contained oil lubricators.

Probably the most common self contained lubrication system for rolling bearings in pedestals is the oil bath. This normally has an oil level sight-glass as shown in Figure 4. The level of oil is fairly critical for high speed bearings and is, therefore, not the easiest type to set up. For example, an oil level of half way up the bottom roller of a roller bearing is too high for speeds greater than about dn = 150 000 (mm x rev/min). In order to keep the operating temperature of the bearing to within reasonable limits of 30/35 degrees C. above ambient, it is necessary to reduce the level of oil until it is just above the lips of a ribbed outer race at bottom dead centre as shown in the figure. This might be about 1/5th of the way up the bottom roller. It is also best to drill a passage to connect the oil reservoir on either side of the outer race as shown.

The traditional 'sight gravity feed oiler' can be used in conjunction with pedestal mounted roller bearings, but it must have a good needle valve capable of regulating the flow of oil to no more than a drop every two to three minutes, otherwise it will not last long. For example, a typical sight gravity feed oiler has a reservoir of 6.5oz = 185cc. If one drop of oil in this type of oiler at 'normal temperature and pressure' is .025cc, then a drop of oil every two minutes will last 245 hours. This is not an unreasonable time.

To prevent oil accumulating in the bearing housing it is usual to drill it at the correct oil level for the speed and then pipe it away. The lower the safe level, the less likely is it to leak and the cooler it will run.

There are no self contained oiling systems for slow moving bearings on high temperature applications, because the volume of oil required would be more than can be built onto a standard pedestal. However, a gravity feed oil tank can be used, but this is hardly more self contained than a separate oil pump and reservoir.

The Greasomatic lubricator can be used containing oil and could be useful for high speed applications. For example the minimum feed rate from a Greasomatic lubricator lasting 12 months is .013cc per hour, the maximum rate being four times from one unit. If four units are connected in parallel, this will increase to .21cc per hour. These rates are perfectly adequate for the average rolling bearing at speeds above dn = 200 000 (mm x rev/min), and it would be necessary to have an oil outlet, preferably piped to a suitable drain.

The constant level oiler is another type, which is mounted outside the bearing as shown in Figure 5. It is somewhat equivalent to having a separately mounted oil bath outisde the pedestal, but because they are mounted on a separate bracket attached to the bearing pedestal, the level within the bearing can be more difficult to control, resulting in the bearing either running too hot because there is too much oil in it or else being starved of oil.

5 EXAMPLES OF STAND ALONE UNITS IN HEAVY ENGINEERING.

See Figure 6 showing a photograph of a pinion shaft manufactured by David Brown Gears for a Blue Circle Northfleet Cement Kiln, 660ft. long. The interesting point about this application is the way the thrust has been contained by a parallel roller thrust bearing on the end of the shaft as shown in Figure 7. The whole bearing is grease lubricated and the spherical surfaces of the cartridge and pedestal are separately lubricated with grease under pressure to ensure correct alignment. The outer race has a single lip to counteract any reverse thrust, when the motor is switched off and inertia from the kiln is driving the pinion gear.

Figure 8 shows a vertical shaft stirrer mounted over a 15 000 gallon fermenting vessel at Boots Pure Drug Company at Nottingham. A cross section of the bearing, Figure 9, shows the radial Cooper split roller bearing, 02 series 8" diameter and the split flat thrust bearing catering for the axial thrust due to the stirring action. The bearings are rotating at 100 rev/min and are grease lubricated. It is, however most important to lubricate all parallel roller flat thrust bearings from the bore to the o.d., because they have a natural outward pumping action when rotating.

For bearings supporting gears in large gear boxes it is sometimes convenient to have units which have their own self contained lubrication system as shown in Figure 10. This type of design allows for the gear box casing to be very straightforward, having no complicated geometry for bearing outer races and it can also contain far less oil for lubricating the gears. Notice that the width of the gear box casing is only slightly greater than the two pinion gears. The pedestal bearings shown can be relubricated according to duty or on a regular maintenance basis.

Figure 11 shows electric motors by GEC Machines driving the Light Plate Mill at B.S.C. S.Teesside Works. The sizes are 760mm, 24" and 20". The large 760mm bearing supports a radial load of approximately 445 kN (100 000 lb) at 30-75 rev/min. On the inside of the cartridges of these bearings, there are split bronze thrust pads which are designed to take an emergency thrust load of 500 kN (50 Tons). Clearly the pedestal itself has to be capable of withstanding this axial load and hence the holding down bolts centres are as wide as possible. However the length on shaft of the bearing and cartridge itself is only 406mm. It is a free standing unit, lubricated once a month with five or six shots from a grease gun.

Figure 12 shows a photograph of a large 03 series 280mm bearing in cartridge and pedestal mounted on the headshaft pulley of a raw material conveyor built by Babcock Moxey for B.S.C. Port Talbot. Because the pedestal has been vertically mounted, the load imposed on it is through the base, which is at an optimum angle for minimum deflection. However the design of the Cooper pedestal is such that it will resist load at any angle, but somewhat lighter loads in the top 140 degrees of arc. The bearing is also designed to take load at any angle.

Because these bearings are fairly slow moving, rotating at about 60 rev/min, it is possible to assemble them with a full pack of grease and to regrease at yearly intervals or longer for correct lubrication of the bearing only. However to prevent the seals and the grease near them from becoming contaminated, it is advisable to relubricate much more often.

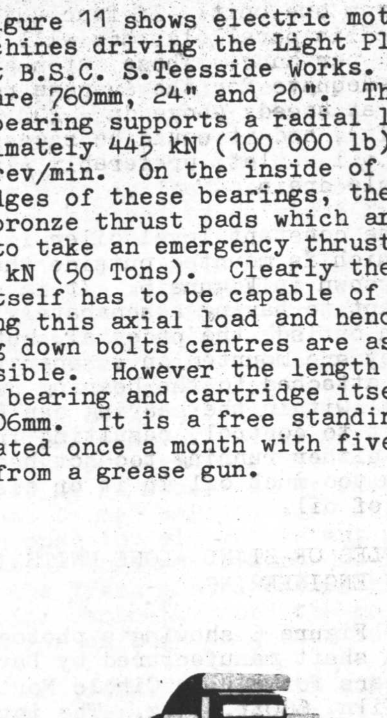

Fig 1 Expansion EX. Bearings for radial loads only. The inner race and shaft have axial freedom

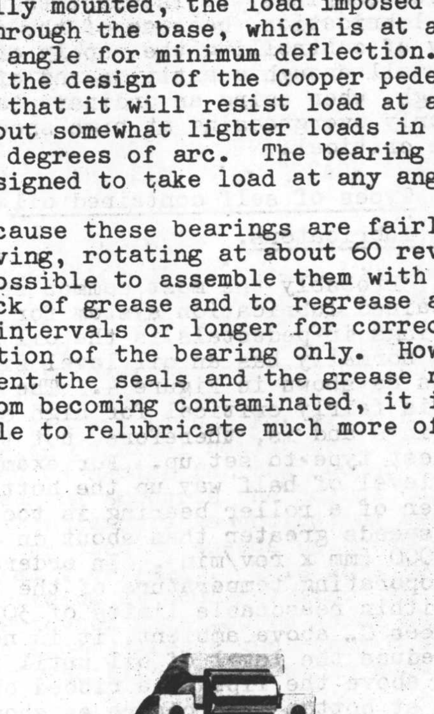

Fig 2 Fixed GR. Bearings for radial and axial loads. Position the shaft endways and resist axial load by cycloidal contact of the roller ends within the inner and outer race grooves

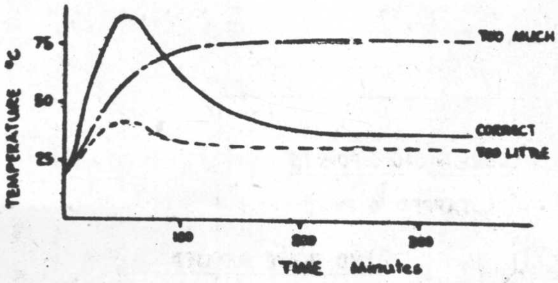

Fig 3 Time—temperature graph of grease lubricated rolling bearings

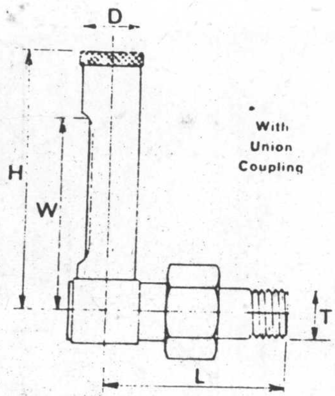

Fig 4 Typical oil-level sight glass

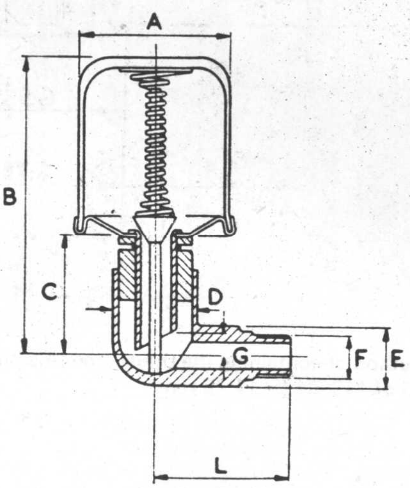

Fig 5 Constant level oiler. As the oil level drops below that indicated by the vertical arrow below letter 'D', air enters the scarfed pipe shown, allowing oil from the reservoir above to feed into the bearing and keep the level constant

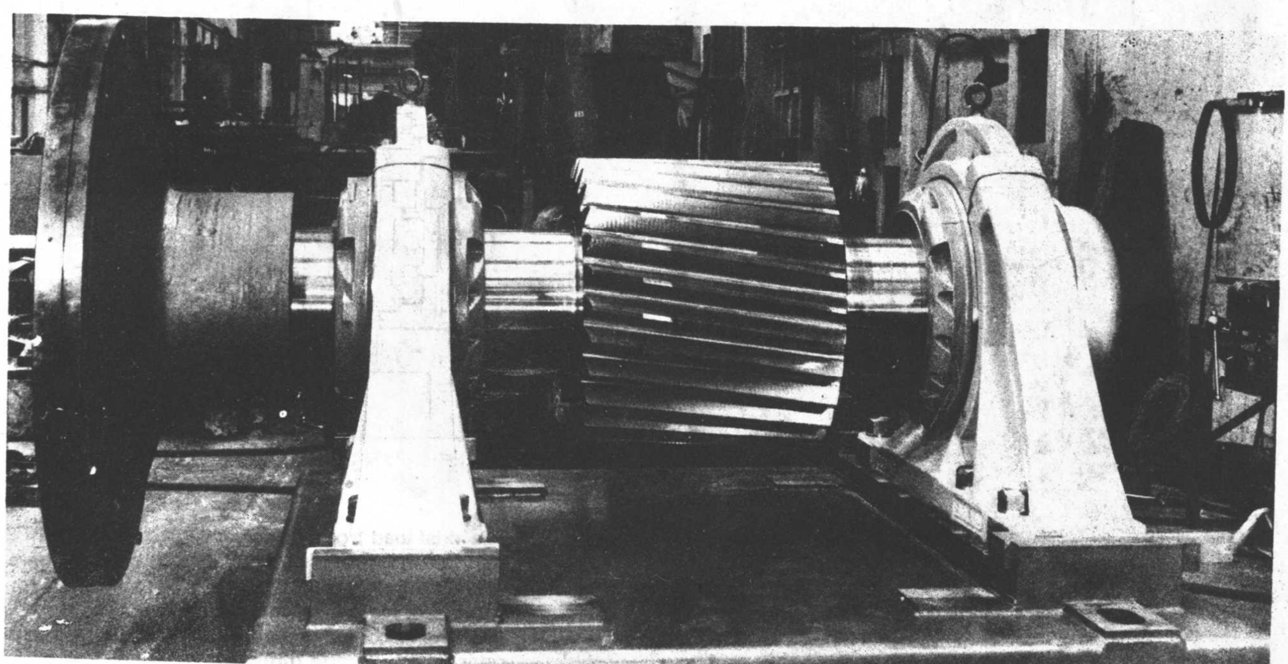

Fig 6 Pinion shaft manufactured by David Brown Gears Ltd. One of 12 supplied for the 660 ft cement kilns at Blue Circle, Northfleet works. Each pinion transmits 775 hp at 15 r/min and is supported by two 18 in series 03E Cooper bearings in angled pedestals

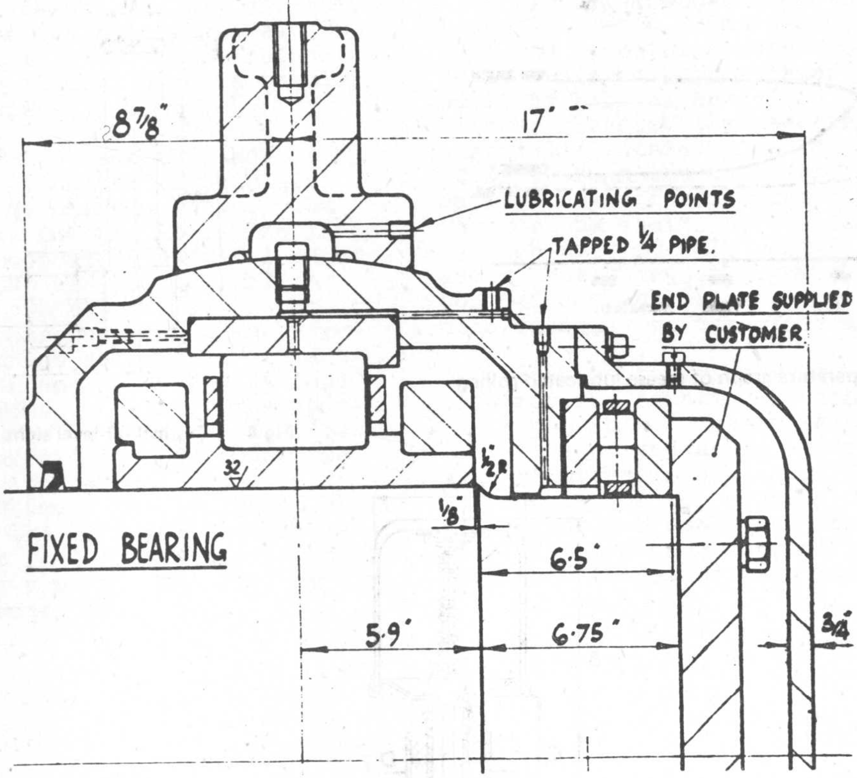

Fig 7 A method of containing the thrust from the pinion gear shown in Fig 6 of 118 kN (≈ 11.75 tons)

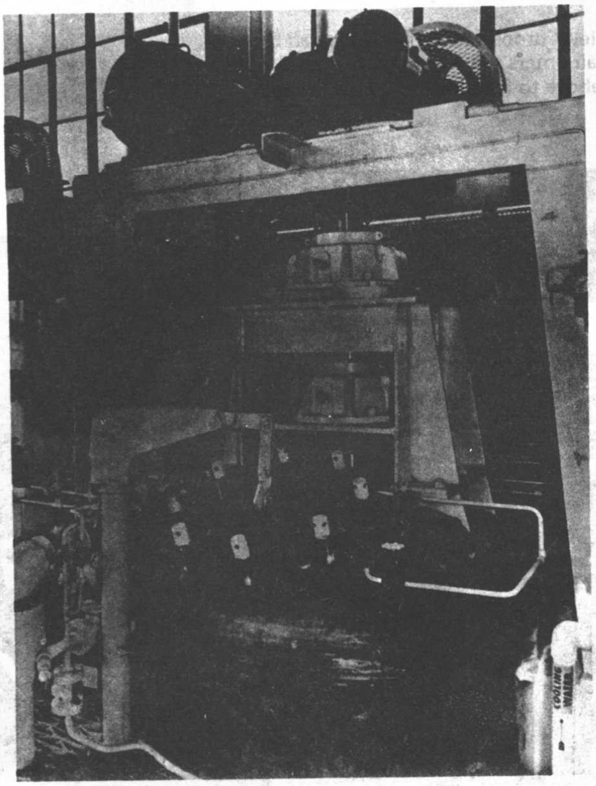

Fig 8 15 000 gallon fermenting vessel at Boots Pure Drug Co. Ltd., Nottingham. The vertical shaft is supported by 8 in. series 02 Cooper bearing in flange mountings incorporating split roller thrust bearing.
Speed 100 r/min

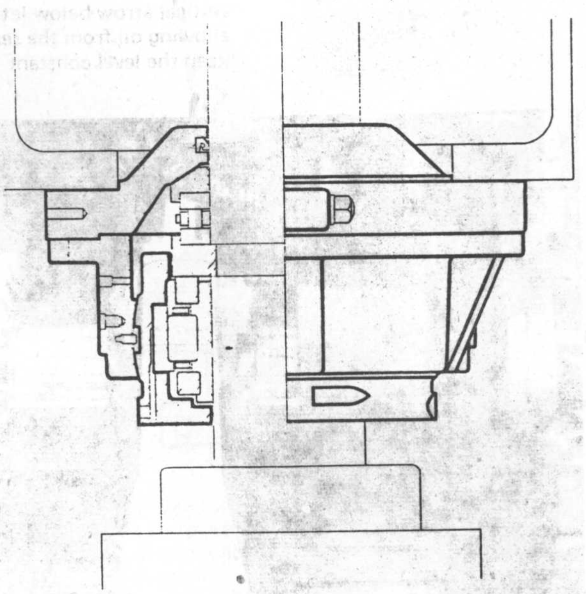

Fig 9 The axial load from the stirrer is taken by the 8 in split flat thrust bearing, lubricated through the bore of the stationary race, which fits into a shroud with a spherical seating for correct alignment. Note the single-lipped outer race of the split radial bearing to cater for any reverse thrust

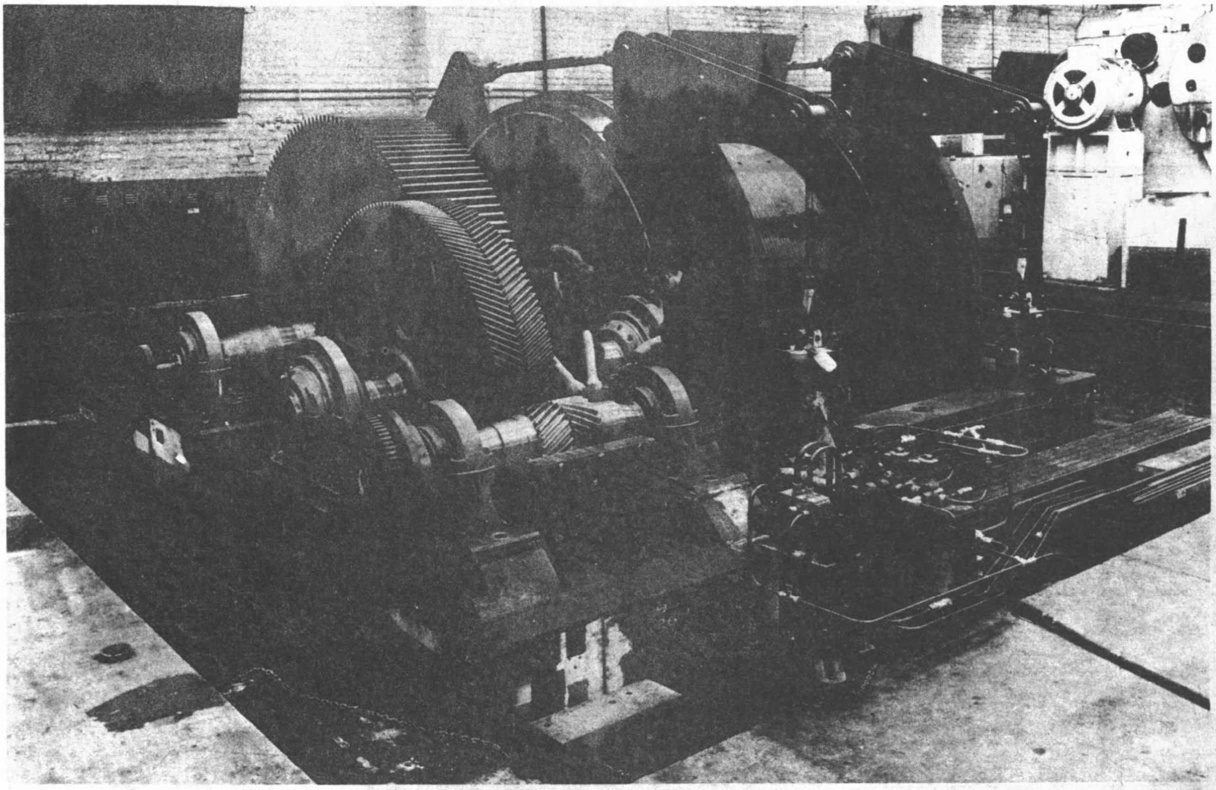

Fig 10 Shop assembly of 1000 hp mine hoist manufactured by Markham and Co. Ltd., for Lingan Mine, Nova Scotia. The 8 ft 6 in diameter x 6 ft wide drum designed for a rope pull of 12½ tons at 700 ft per minute. Cooper split roller bearings incorporated: Drum shaft — 17 in series 02; 15 in series 01; 12 in series 01, second motion shaft — 11 in series 03E, Pinion shaft — 7 in series 02

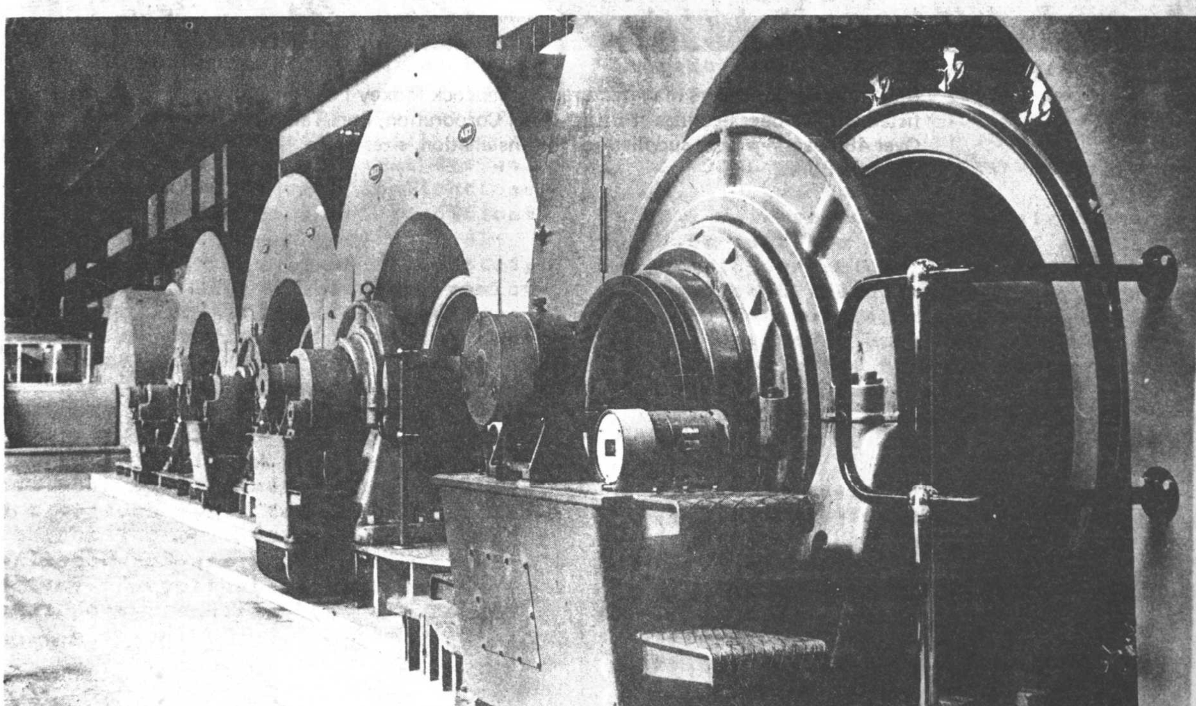

Fig 11 Motors by GEC Machines Ltd. on the Light Plate Mill at the British Steel Corporation Works, South Teeside. 760 mm, 24 in, and 20 in Cooper bearings in high-rise pedestals are fitted to the variable speed motors