Marine Gear Type Couplings in Service

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SYNOPSIS

The introduction outlines the main function of gear couplings and their application in marine plant, with reference to those features in design which, from the user's standpoint, influence performance.

A summary of Shell companies' general experience includes brief case histories which are intended to highlight the discussion which follows, covering alignment requirements, materials selection, methods of manufacture, and lubrication considerations. Both static and dynamic applications are included.

Finally, in the light of operational experience, consideration is given to the possibilities for improvement in future designs.

INTRODUCTION

Gear type couplings have been in use for many years in marine applications, and it is only comparatively recently that increased interest has been shown by merchant ship operators in membrane types. The latter are used now in Naval applications, but their use is still relatively small in the merchant marine.

This is believed to be partly because of natural conservatism on the part of ship owners and turbine builders, but perhaps mainly because of the more specialised nature of membrane designs.

In this paper, operating experience is described for gear type couplings, not only for turbine/gear applications, but also in the more particular application of the coupling systems for epicyclic units. In Shell ships the powers transmitted vary between around 3000 kW (4000 shp) for each of the HP and LP units in general purpose ships up to 13,400 kW (18,000 shp) for VLCC. Most of the experience related will be familiar to other users, so it is the hope of stimulating discussion, out of which may come suggestions for improved performance.

Design details of gear type and indeed membrane type coulings[1] are fully covered elsewhere in this Conference, and therefore only general reference to these is made in this paper.

The main requirements which should be fulfilled by flexible couplings are that they should:-

Operate with complete safety. The failure of a high speed coupling in service can result in disastrous damage to surrounding plant, and very likely serious injuries or even loss of life to engine room personnel.

Perform adequately. The design should allow the machine to operate with limited misalignment, and without loss of operational efficiency.

Operate for long periods without replacement. The average merchant ship life is 20 years, and a high speed coupling should give safe and adequate service for at lest half this time without deterioration in condition to the point when replacement is required.

As for as geared coupling rings for epicyclic units are concerned, these should be regarded as part of the gear system, and therefore be expected to last for the life of the ship.

SERVICE EXPERIENCE

Turbine/Gear Drives

The life obtained from different designs has varied widely. In older ships one particular design has been changed regularly every 4 or 5 year for reasons of excessive clearance due to wear, and general condition showing heavy scoring and fretting. In similar applications, other designs have given a life in excess of 14 years. In all cases it has been found that if the turbine/gear units are incorrectly aligned, wear and surface damage inevitably result. Indeed, the couplings often provide the clue as to whether the units are aligned within reasonable limits.

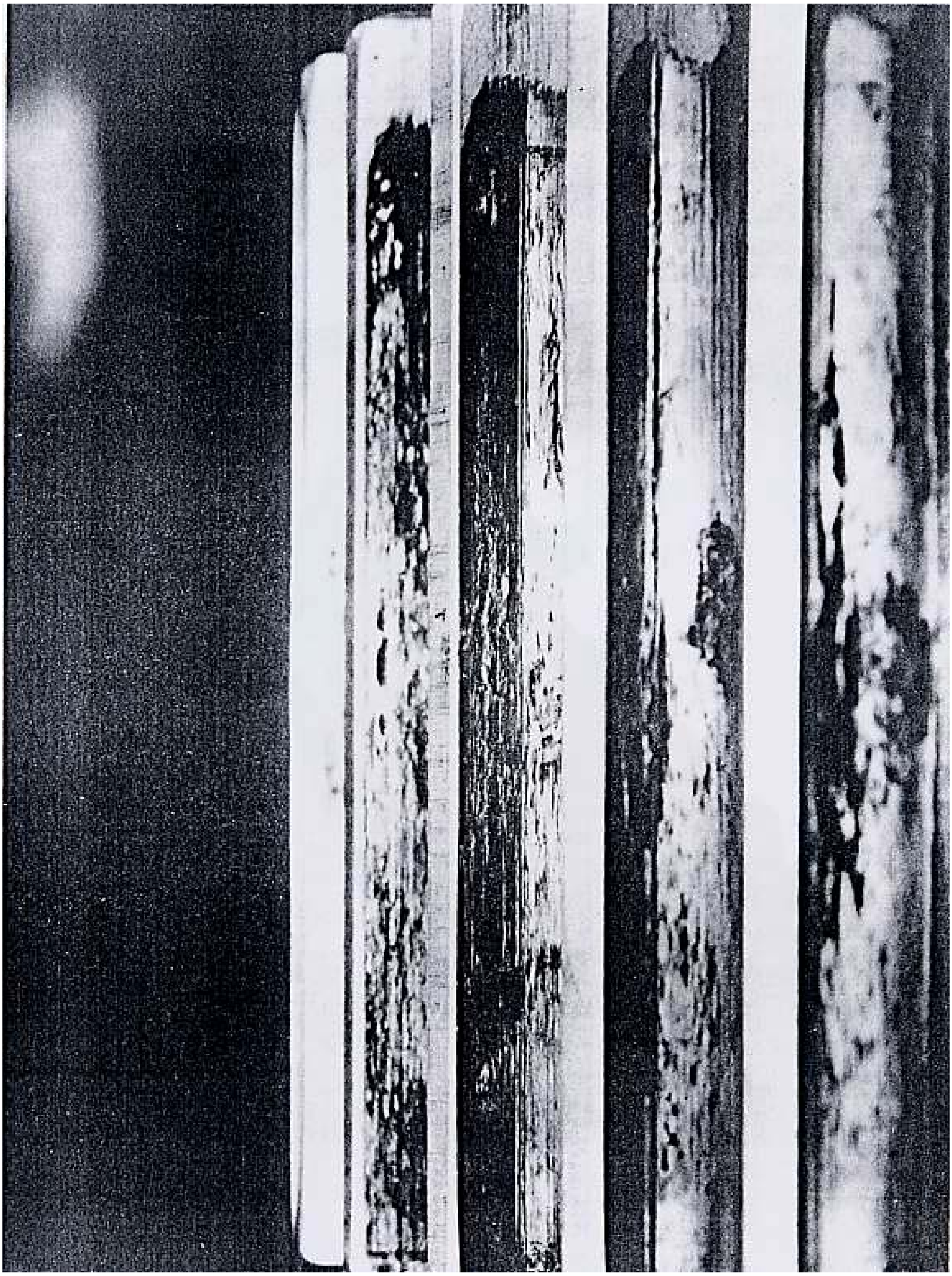
In later ships, built in 1966 and after, coupling performance has been markedly improved, but there have been cases of early replacement due to excessive wear and surface damage. One example is illustrated in Figure 1, which shows axial scoring and heavy fretting after only about 2 years in service. Some, but not all, of the damage was due to alignment being outside specified limits. In this and another case which was also subject to extensive misalignment, the design was altered by introducing barrelling and the tooth surfaces were nitrided. Subsequent performance in these and all other ships having the same modified couplings has been completely satisfactory. In one other design, coupling condition was indifferent after 2 years in service, mostly at reduced power. It is likely that both HP and LP couplings in this ship will be replaced after a further 2 years service. The manufacturer's opinion in this case was that at the time of examination the couplings were satisfactory. Surface damage, again scoring and fretting, was considered as being normal, and it was stated that such damage generally stabilized, getting no worse in subsequent service. This tacit acceptance of indifferent performance is not shared these days by other manufacturers.

Service periods in a large number of VLCC now exceeds 10 years without coupling replacements.

As to safety, accidents have happily been rare. One occurred in 1962 when a coupling failed (see Figures 2a and 2b), leading to a disastrous engineroom fire, serious injuries and loss of life. The cause of failure was the development of a corrosion fatigue crack originating between two teeth on the inside of the pinion coupling sleeve. Examination of the failed components suggested that the failure was caused by the combined effects of large out-of-balance forces due to heavy fretting, and severs corrosion of the inner surface of the coupling sleeve.

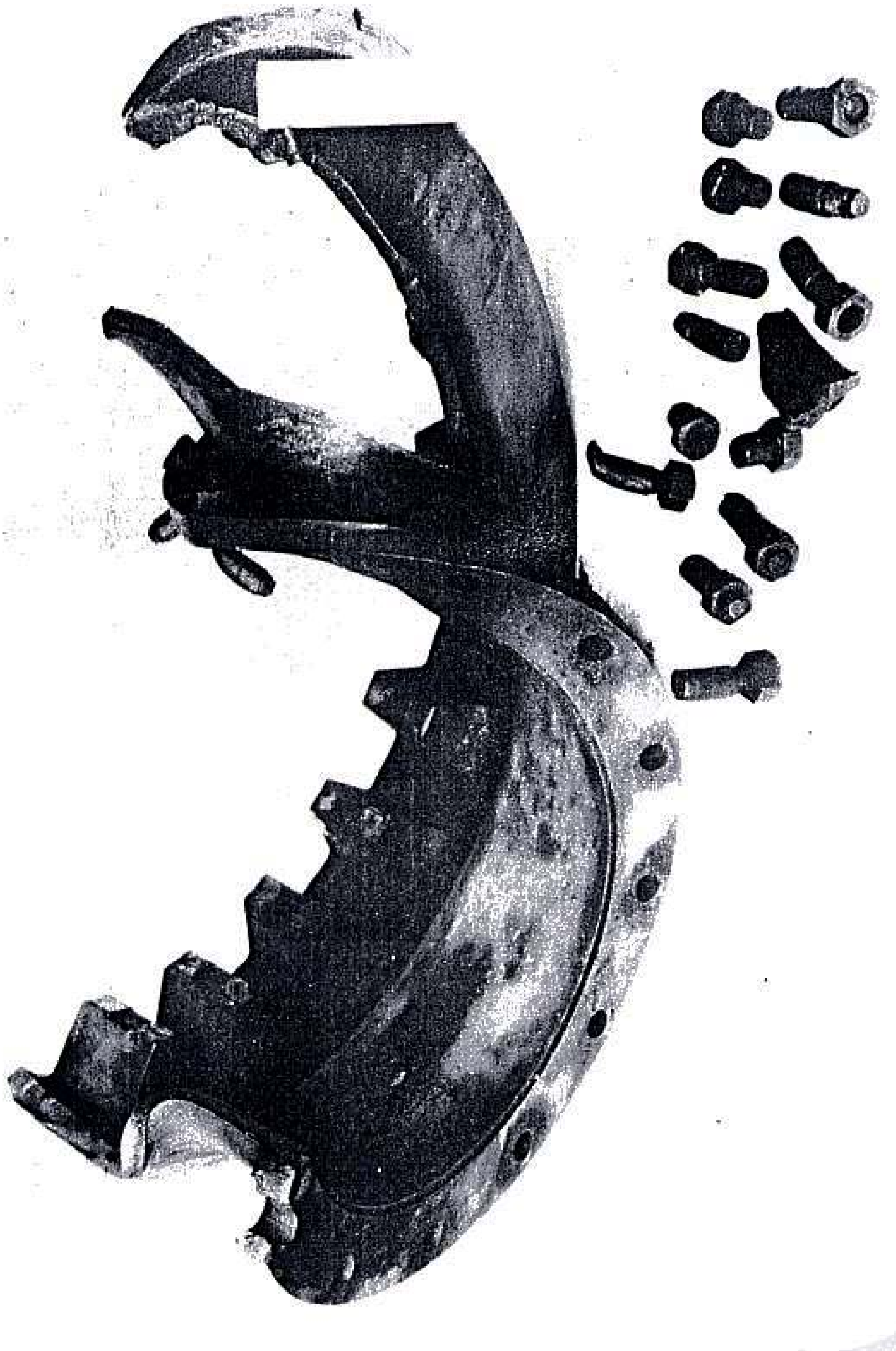
Another "near miss" occurred when a 200,000 dwt ship came in for dry docking and inspection of engine components showed that the gear side female half of the coupling had fractured in four places, and cracks were present also in the turbine half coupling. Two teeth had been broken from the male coupling sleeve, which also contained a crack (see Figures 3a and 3b). The basic cause of failure was corrosion fatigue which had started at corrosion pits in the roots of the gear teeth. There is no doubt that had the ship continued steaming, the coupling would have failed completely and the case illustrates how it is still possible for accidents to occur with relatively modern designs. It should be added that this particular turbine also suffered alignment problems, but these were not considered to be the central cause of coupling failure. Subsequent experience with replacement couplings to the same design has not been entirely satisfactory, and it has been necessary to keep a close watch on the condition of both the HP and LP units, thus assign to the maintenance costs and the work load of the ship's staff. A final example of failure was the case of a coupling actually sliding out of mesh whilst in operation; the turbine tripped on overspeed and no damage was sustained. The cause on this occasion was failure of a spring washer followed by fracture of the retaining circlip.

Turning to the static application of geared coupling rings in epicyclic units, operating history is confined to a particular make of engine incorporating epicyclic reduction gear



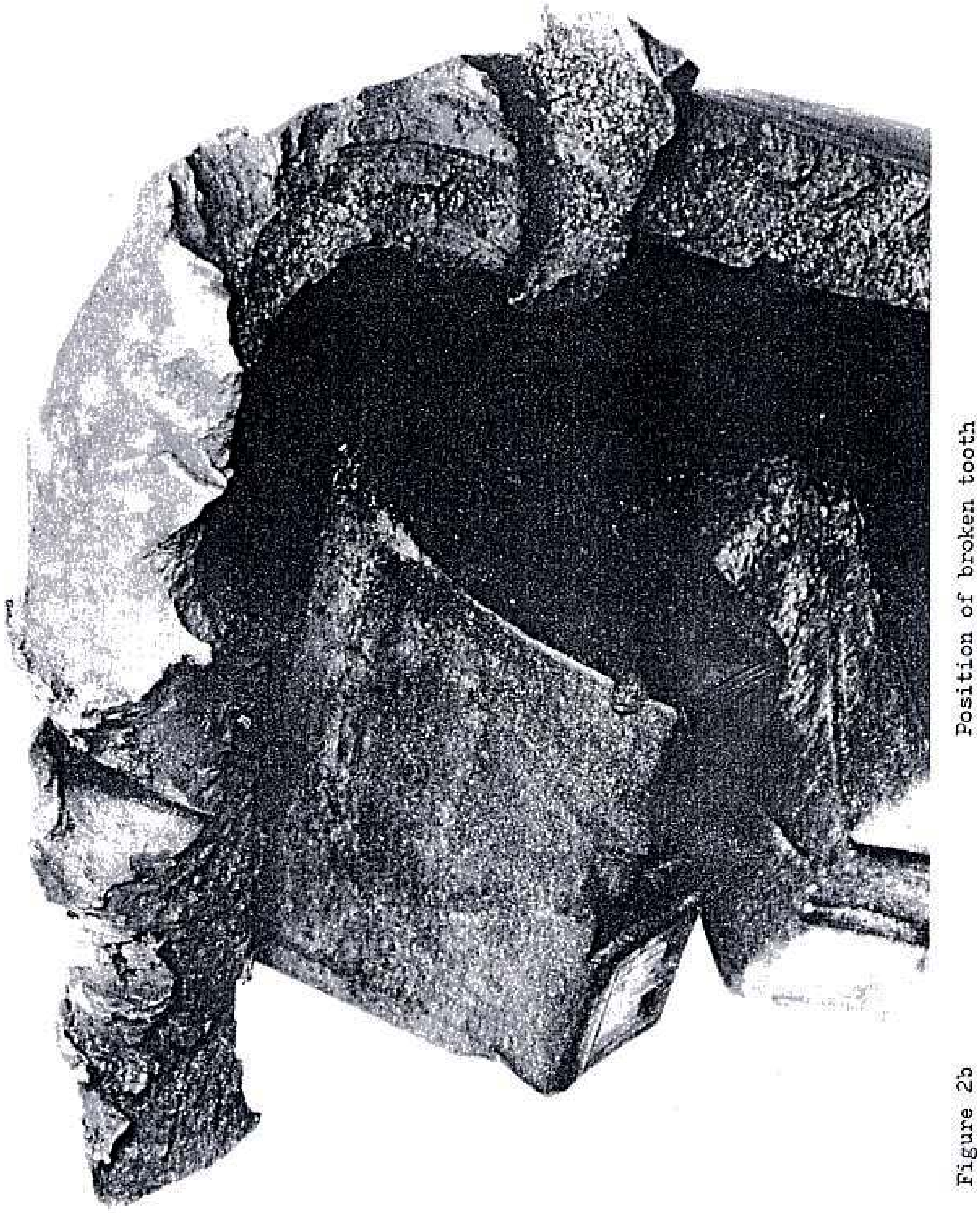
Axial scoring and fretting

Figure 1



Failed L.P. turbine coupling

Figure 2a



Failed L.P. turbine coupling

Figure 2b

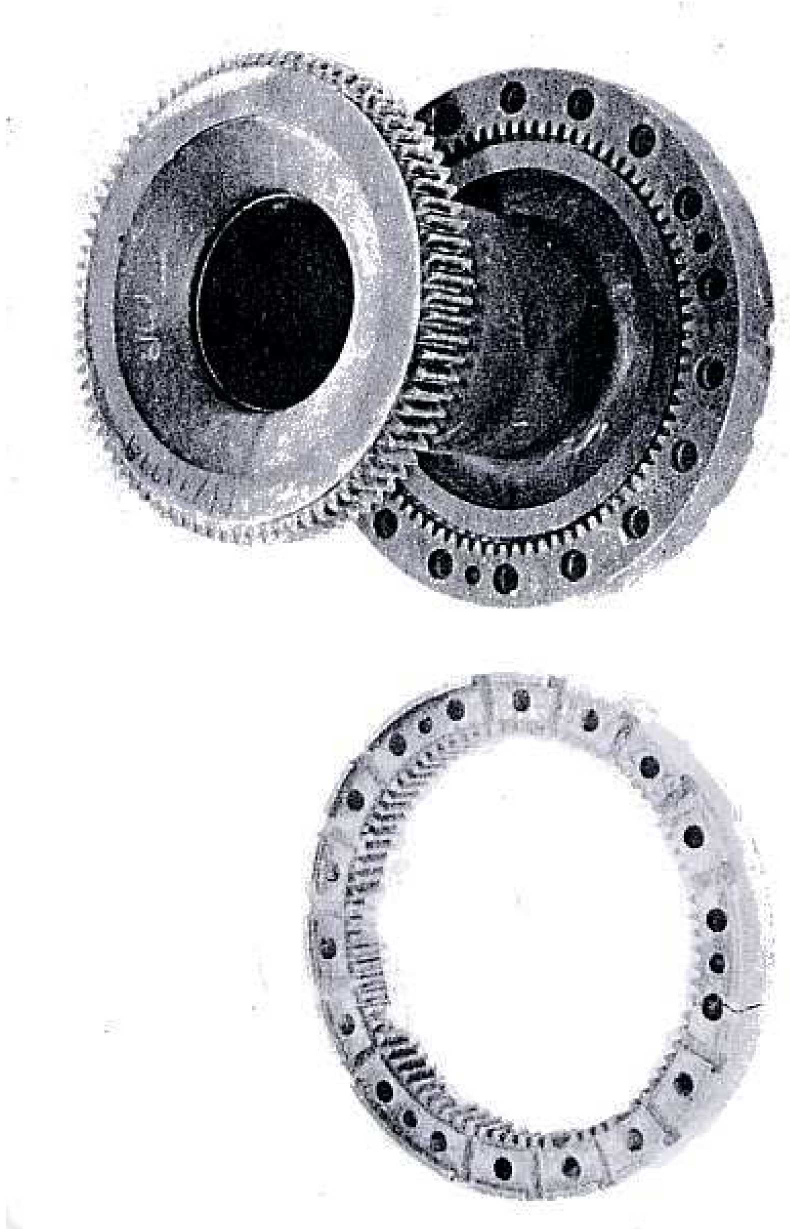


Figure 3a General appearance of failure

Figure 3

Figure 3b Close up of fatigue fracture

Failed H.P. turbine coupling

Figure 3

units. This is the well known Stal-Laval AP design, which was introduced in 1965. It is interesting to note that modern Stal-Laval engines incorporate not only gear type flexible couplings, but also the membrane type in the output between the first and second reduction epicyclic gearing and, finally, geared rings in the coupling system of the epicyclic units. The coupling rings serve to support the epicyclic gears, and provide a connection to the gear casing. Service experience with these rings over periods up to 11 years has shown that fretting develops in the early stages, and often the condition of the components appears unsatisfactory. However, the condition stabilizes, and coupling replacements have not been found necessary in the large majority of ships (see Figure 4).

Wear, fretting, and axial scoring were excessive on the LP unit of a 310,000 dwt VLCC after only one year in service, and the rings were replaced. A second ship of the same class recently dry docked and, again, the rings were found to have deteriorated more than was desirable for the period in service, in this case 15 months at less than full power. The rings in the second case were not replaced, it being judged that there was no operational hazard. One possible cause of this abnormal condition is believed to be vibration emanating from the LP turbine, induced by the propeller blades. Early measurements showed that the turbine was vibrating axially at 2nd order blade passing frequency and an amplitude of 0.87 mm maximum. The amplitude varied considerably with the response of the ship in the seaway. It was found that the LP gear unit was also vibrating in phase with the turbine, but that the coupling rings themselves showed very small relative movements. The oscillograph traces in Figure 5 illustrate the transient variation in amplitude of the vibration.

DISCUSSION

There are several points arising out of the operation experience related briefly above, which warrant further discussion.

Basic Design

The various couplings in service have differences in diametral pitch, all designs following a similar tooth form. What, therefore, is considered to be the optimum, and how does this affect performance? Two factors which must be relevant are strength and manufacturing facility. As far as the user is concerned, however, he wishes to have adequate safety margins in strength to take account of misalignment possibly developing in service, and machining accuracy is therefore very important. Potential strength in design is of no use if machining accuracy in practice is of a low standard. In general, the couplings used for high speed applications have shown satisfactory load sharing between the teeth, but in the static case this has not always been so. Here, however, dynamic forces do not play the same part, and the manufacturer takes account of limited accuracy in the design safety margin.

Barreling of high speed coupling teeth is now widely adopted, and there is no doubt that improved performance results. Caution is required, however, not to exceed limits.[2] Barrelling increases capacity for misalignment before lock-up, but will not alleviate the effects of excessive angles of misalignment. Further, even modest amounts of barreling increase the surface stress on the teeth very appreciably.

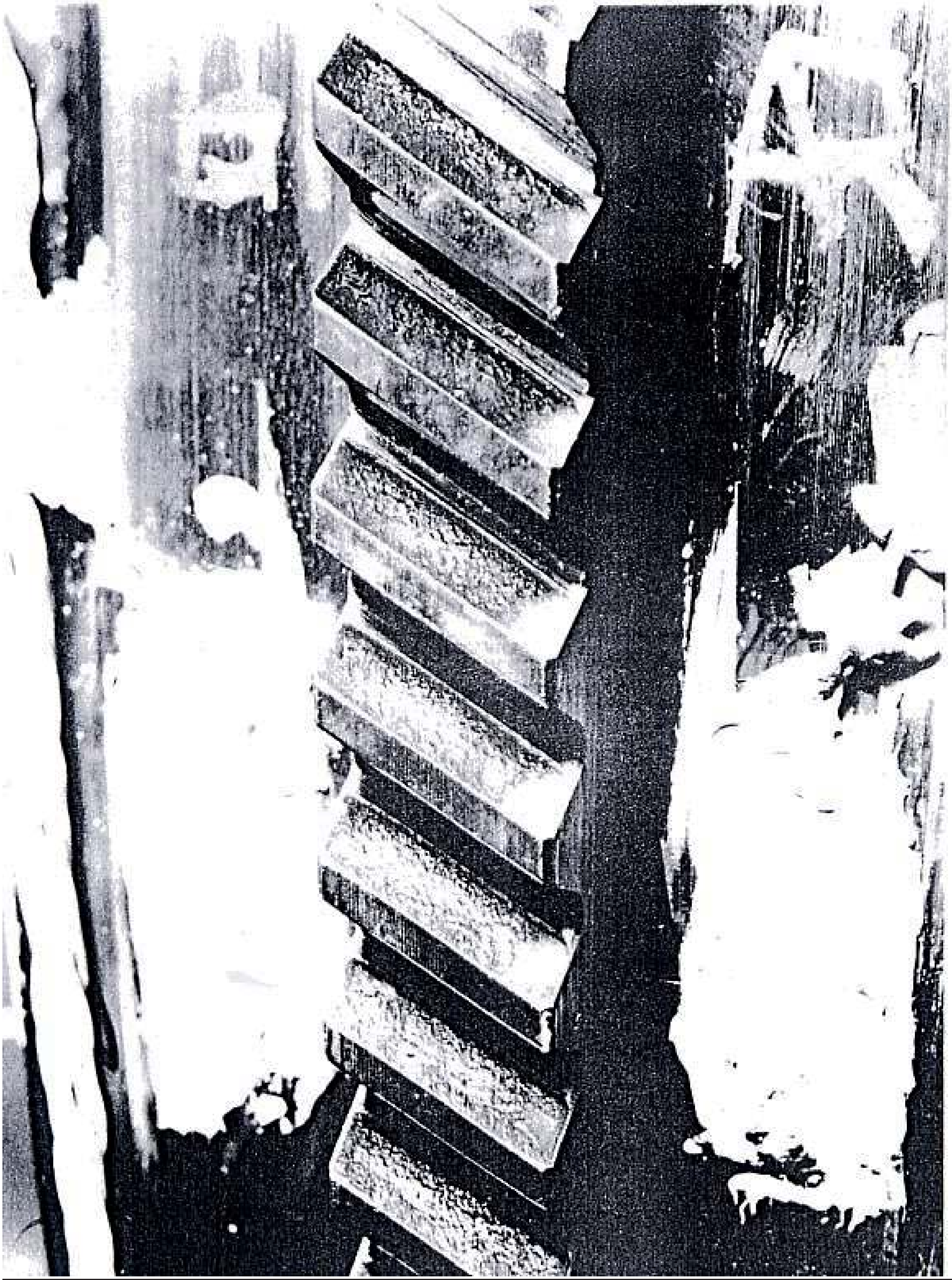
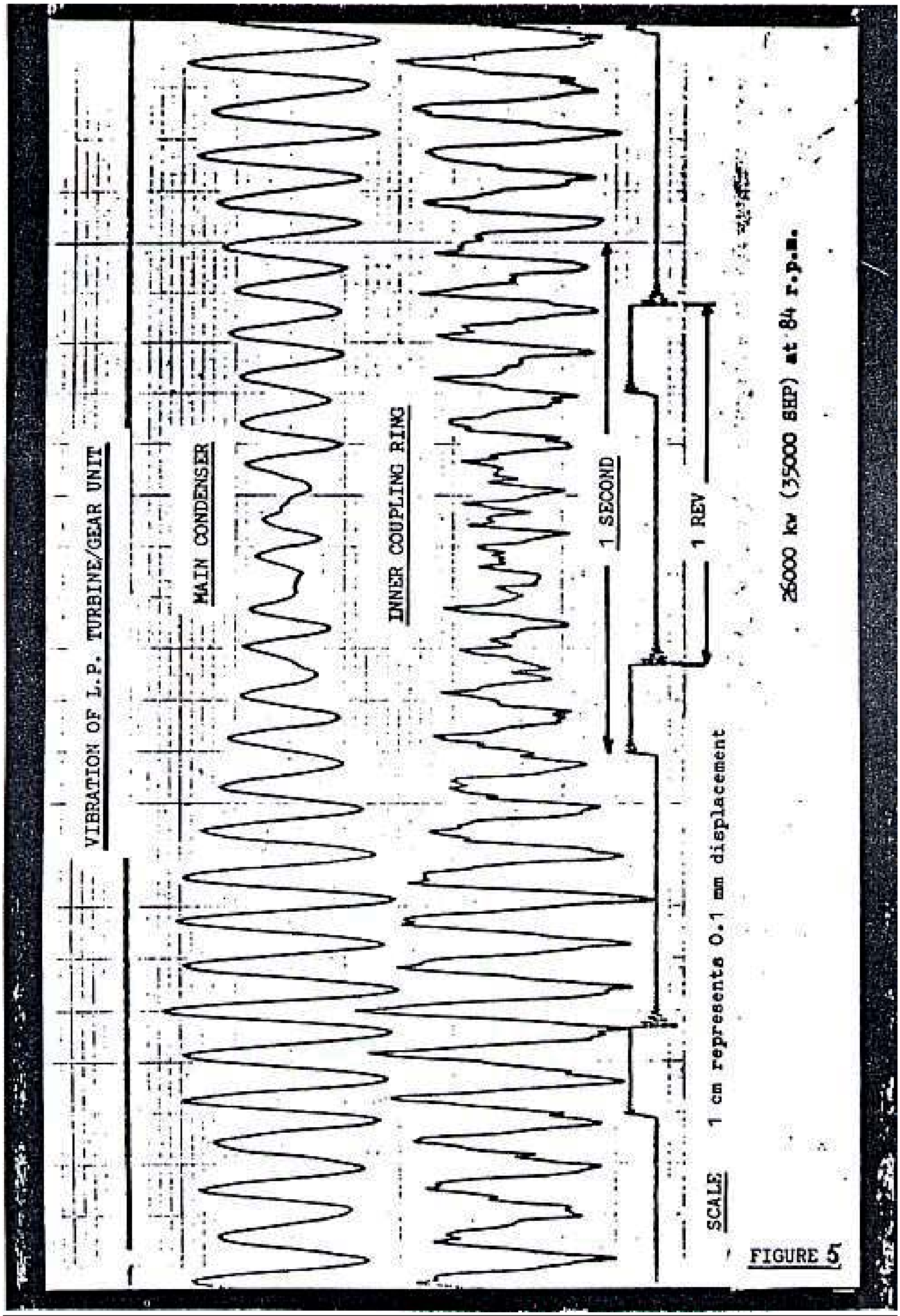


Figure 4 Fretting of coupling ring teeth


VIBRATION OF L.P. TURBINE/GEAR UNIT
MAIN CONDENSER
INNER COUPLING RING
1 SECOND
1 REV
SCALE 1 cm represents 0.1 mm displacement
26000 kw (35000 SHP) at 84 r.p.m.
FIGURE 5

Alignment and Balance

The purpose of any flexible coupling is to transmit power, and at the same time absorb such misalignment as must inevitably be present in a turbine/gear unit, the amount depending on engine and ship operating conditions. There can be no perfect alignment for all operating conditions, nut the static alignment of turbine to gear must be optimum for the normal running condition if the coupling teeth are to survive without excessive wear.[3] The vibration behaviour of the main engine will also influence coupling condition, and here again good coupling performance can only be achieved if rotating parts, including the coupling, are properly balanced.

In the case of geared coupling rings, again the gear/gear alignment has to be a compromise as the gears will take a different load line depending on whether they are running ahead or astern. In this application it is likely that as the engine load increases, the rings tend to lock under torque and thus will remain in one attitude for the running period of the ship which, in the case of a VLCC, is usually at least 4 weeks at, or near, full power.

Vibration

The problems experienced with the geared coupling rings in the 310,000 dwt class ships did not arise with the smaller 210,000 dwt class, and it is pertinent to ask why. One explanation could be that with the broader beam of the larger ships, hull resonant frequency is lower and is therefore more prone to excitation by propeller forces. It is notable that the LP turbine vibration measured in the 310,000 dwt ships fitted with 6 bladed propellers has been less than that experienced with the same hull fitted with a 4 bladed propeller. It may be added also that this feature is not peculiar to one make of engine. However, it could be more significant in the case of the Stal-Laval machines if there is any relationship between this phenomenon and wear of the coupling rings. It has been mentioned that measurements showed only a very small relative movement between the two coupling rings, and this appeared insufficient to cause serious fretting and, particularly, scoring. However, it is conceivable that very small movements such as have been measured are, in fact, sufficient to produce scoring if small wear particles are trapped between the teeth and “scuffed” back and forth in the gear mesh. This particular problem is still under investigation, and it is hoped that a solution may be found by making improvements in coupling ring finish and applying special surface treatment.

Surface Treatment and Nitriding[4]

Surface damage to gear coupling teeth can take several forms, one or more of which will be present, depending on operating conditions. The three main forms appear to be:

scoring or galling due to axial forces;
fretting due to small angular movements under load;
pitting due, it is suggested, to vibration and/or corrosion.

Over many years attempts have been made to obviate damage by attention to various forms of surface treatment or hardening. None of the surface treatments has proved really effective, and the best approach up to the present time has been found to be nitriding. This is an attractive form of surface hardening as it can be carried out as a one step heat treatment without excessive distortion of the component. Thus, machining to correct growth and inaccuracies is not necessary.

However, the quality control applied during nitriding is important as is also the correct choice of nitriding steel. Lack of attention to temperature control and distribution can result in poor finish and a shallow case depth. The nitriding depth should be of the same order as that for gear elements, about 0.6 mm. Further, subsequent handling of nitrided components must be carefully watched as the case is vulnerable to impact damage.

Nitriding cannot at the moment be applied to gear coupling rings as the components, of largish diameter, present a difficult section to handle during heat treatment. An alternative approach has been to phosphate the rings in order to provide a better chance for the tooth surfaces to run in during the critical period in the early operating history of the engine, when the rate of deterioration id highest. Phosphating is a straight-forward and relatively cheap way of providing assistance for running-in, but there are other possibilities which are being considered. At the moment there is a set of phosphated rings in service in a Shell vessel of 540,000 dwt. Results from this experimental set of rings will be available sometime in the middle of this year.

A question often raised is whether or not flexible couplings, whether in static or dynamic applications, remain flexible in an axial sense when under full or nearly full load conditions. General experience with all tooth couplings, including quillshaft applications, suggests that this is not so. It is felt that the inability of the couplings fully to absorb axial movements could be the cause of some operating problems, and could also be responsible for some cases of wear in service.

Lubrication

Both turbine/gear couplings and toothed coupling rings are difficult to lubricate adequately. In both cases there is only relatively small movement between the mating elements, and it is questionable, therefore, whether hydrodynamic film lubrication plays a significant part in preventing wear. It is believed that most lubrication is effected under boundary conditions when penetration of the oil film by asperities can take place. In addition, the small relative movements under load encourage fretting, whilst vibration and axial movement can cause pitting and scoring.

In geared turbine applications the choice of oil viscosity is limited by high sliding speeds of a number of components. Again, therefore, protection of the surfaces by using an oil of high viscosity is precluded. Further, additives of the anti-wear or boundary lubrication type are often incompatible with system operating conditions and can, as a result, be responsible for side effects. This is particularly so in the case of incompatibility with either one of the inhibitors (rust and oxidation) normally present in a double inhibited turbine oil. One of the most promising boundary lubricants is sulphurised oleic acid (SOA). This has been used effectively for other “difficult” applications, and is regarded as an anti-wear additive. Only very small quantities are required in order to be effective in a lubricating oil, and there are no known side effects. As far as is known this has not been used specifically to assist lubrication of flexible couplings, but it seems worth considering. Assistance from dry MoS_2 treatment before commissioning has been suggested in the past, but it is not general in marine turbine practice, the modern approach being to use case hardened teeth to combat wear.

At one time the lubrication system, so-called, for flexible couplings was a pure act of faith. Far more attention is mow paid to making sure that ample lubricant reaches the teeth, and that a flow exists to flush away any debris and the inevitable sludge resulting from the centrifuging action of the coupling at high speed. Even so, there is a tendency for the oil to flow preferentially through the unloaded side of the mesh where there is less resistance to flow. A good design of oil system, made by a Japanese turbine manufacturer, is illustrated in Figure 6.

In present designs, lubrication of the geared coupling rings is by loss from oil leaving the plant gear spindle bearings. For reasons given above, it is felt that a more positive feed to the coupling teeth would give better results, but it is recognized that this is not easy to achieve in a static application.

One final point in connection with lubrication is accessibility. It is not always possible easily to check that all is well with the coupling assembly, and that the oil flow is satisfactory. Some means of being able to observe these points would be of benefit to operators.

CONCLUSIONS

In general, it seems that gear coupling designs still follow conventional lines, but improvements have been made in three major respects:-

Manufacturing accuracy and finish
Resistance to surface damage by nitriding.
Improved lubrication with particular respect to ensuring flushing of the tooth mesh.

There is no doubt that, as a result of the above attention, coupling performance in all respects has improved in ship applications over the past 10-15 years. There are still instances, however, of manufacturers failing to recognize that indifferent standards of design and manufacture are simply not acceptable for what the user regards as a vital component in high speed applications.

The performance of geared coupling rings can, it is considered, be improved in respect of:-

1. Manufacturing technique.
2. Resistance to fretting
3. Lubrication.

The introduction of epicyclic gearing into ships main propulsion plant is comparatively new, and much still has to be learnt from service experience. It takes time to accumulate operational data, and the final design is evolved, therefore, over several years. It is believed that the ring support system for the epicyclic units could be simplified, and it is known that the manufacturers are already considering new designs. In the meantime, however, means of improving the present performance to the rings should be studied and applied.

Stal-Laval use full scale back-to-back rig tests for studying ring performance. It would be interesting to know whether other manufacturers have studied the performance of high speed gear type couplings in back-to-back rigs. It is considered that a great deal could be learned from this type of testing without excessive expenditure.

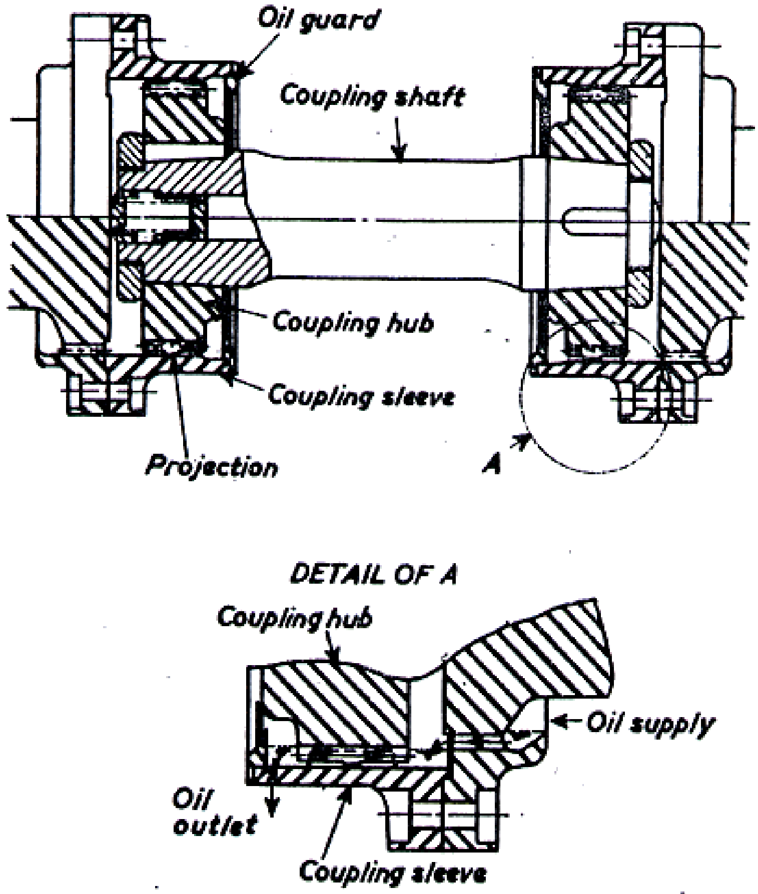


Fig. 6 —*Sectional view of flexible coupling*

Finally, safety has been stressed in this paper, and fairly recent experience underlines the continued need for vigilance both by the manufacturer in design, and by the user in applying good housekeeping, with meticulous observation of coupling condition during refits.

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