TRIBOLOGICAL PROBLEM SOLVING IN MEDIUM HEAVY-DUTY MARINE DIESEL ENGINES PART 1: JOURNAL BEARINGS

Luca Piancastelli[1] and Stefano Cassani[2]
[1]Department of Industrial Engineering, Alma Mater Studiorum University of Bologna, Viale Risorgimento, Bologna, Italy
[2]Multi Projecta, Via Casola Canina, Imola, Italy
E-Mail: luca.piancastelli@unibo.it

ABSTRACT

Tribological problems are extremely common in classical DI (Mechanical Direct Injection Diesels) and those update to CR (Common Rail) technology. These engines are still widely used in Power Generation, Diesel-electric locomotives and Marine applications. The original design of the first prototype, in a few cases, was conceived just after the First World War The marine environment and the continuous increase of performances and TBO provoked several tribological problems that have been partially solved through the years. However, significant improvements are still possible with modern technology in journal bearings, lubrication systems and lubricants. The journal bearing should float before applying load, for this reason start up originated problems cause 90% of the bearings failure. The relative speed between journal and bearing should be over a well-defined value to avoid boundary lubrication. For this reason, compression reduction devices can be used at star or the geometrical compression ratio should be reduced in favour of turbo-charging induced compression. Start velocity should be controlled and the lubrication system should be checked. Finally, a suitable lubricant should be used.

Keywords: tribology, direct injection diesel, lubricant, journal bearings, friction, wear, cold start, seizure.

INTRODUCTION

Tribological problems are extremely common in classical DI (Mechanical Direct Injection Diesels) still widely used in Power Generation and Maritime applications. These engines have displacements from 6l up to 30l and output power from 300 up 1000 kW. Usually the maximum shaft speed is under 3000 rpm and the original design included a mechanical injection system. A few of these engines have been upgraded with an electronically controlled injection or a "common rail" system (CRDID -Common Rail Direct Injection Diesel). Engine design has been continuously updated from the first prototype that, in a few cases, was conceived after the First World War The marine environment and the continuous increase of performances and TBO provoked several tribological problems that have been partially solved through the years. However, significant improvements are still possible with modern technology in journal bearings, lubrication systems and lubricants. The duration of a combustion engine depends on its weakest component. Paradoxically reliability is a common critical factor in marine heavy-duty engines and racing engines. In fact, in aircraft engines certification and maintenance standards require very high reliability standards. Typically, the requirements are 1 out of 10 thousand of malfunction probability and failure probability of 1 in 100 thousand in 3000h [1] [2] [3]. Naval engines are similar but the minimum life is around 10,000h. The same requirement is present in races where the fate of a team often depends on the outcome of a single race. The only important difference is the life: 10,000 hours for the navy, 1,500 hours for aerial vehicles, 120 hours for Formula 1 racing. For this reason, the experimental racing experience, that is much shorter, is vital also for the other "similar" fields. In addition, lubrication and bearing problems are similar in Formula 1 and relatively large and slow diesel engines.

Journal bearing simplified fluid dynamics

In a plain journal bearing a steel base material is overlaid with a Babbitt (Babbitt-like) material and bored to a circular diameter equal to the shaft diameter plus the desired (very small) clearance. An additional extremely thin layer is added to the inner surface obtaining a tri-metal structure. The tri-metal plain journal bearing was introduced in the 1930 and continuously updated to contemporary high-performance engines. This journal bearing is formed as a laminated structure having a relatively thick steel backing layer in contact with the housing, a harder, thin middle "Babbitt" layer (copper-lead, lead-bronze, aluminium-tin, silver-bronze...) and a very thin upper layer of soft, low friction material (lead, zinc, cadmium, lead-indium, and ....). This top layer forming the bearing surface. The maximum applied pressure a bearing can carry is determined by the strength and hardness properties of the upper surface. The maximum relative velocity between the journal and the bearing depends on the capability to dissipate the heat generated by the shearing of the oil film. This heat dissipation is mainly obtained by the forced oil flow. The pressurized oil is sucked in a low-pressure area of the bearing and squeezed into the high-pressure region that bears the load. No contact should take place between the bearing surface and the shaft during normal operations. Usually the plain bearing is split into upper and lower halves (or shells), so that it can be fitted over the journal. One half fits into the crankcase, the other into the cap. Therefore, this type of bearing is referred to as plain or shell-type. Upon installation, the two shells are crushed together into a tight fit. This bearing “crush” changes the circular bush into an elliptical "lemon" arrangement. In

ARPN Journal of Engineering and Applied Sciences

fact, a bearing conforms to the shape of its elastic housing. Therefore, the two halves of the housing are bolted together in such a way that its parting line surfaces conforms to the housing, leaving the required running clearance. A slight gap is left between the housing faces, so that when the final tightening brings will deform the shells. For this reason, the bearing applies a radial load to its housing into an interference fit. The resultant lemon bearings have better dynamic characteristics over a plain bearing and are more stable. Usually the horizontal clearance is 1.5-2 times the vertical clearance. When possible, the angle of the split between shells is oriented for the best load capacity and damping. The stiffness in the direction of the greater clearance is often an order of magnitude less than across the tight clearance direction. Damping is also significantly less in the direction of the maximum clearance. In crankshafts, only one of the main bearings has also the axial bearing necessary to minimise the longitudinal (axial) movement of the crankshaft. The multiple layers have been developed to provide the multiple properties required for piston engines. In fact, the backing is invariably of high-strength steel. However, a steel bearing running directly against a steel journal would face high friction and wear in the boundary and mixed lubrication modes, and would provide little or no ability to allow foreign particles to embed in the material, but would instead turn them into cutting tools. Therefore, the upper layer is of a softer metal, designed for reduced friction and for embedding capability. The idea is to embed abrasive particles below the working surface and thereby minimise wear. In addition, it needs the capability to prevent pick up or even seizure if the oil film momentarily breaks down as at engine start and stop. The thinner is this layer the more resistant it will be to squeezing out. This is the principle of plastic constraint. At zero speed, the shaft rests on the bearing at bottom dead centre. When the shaft rotation speed (tangential velocity) reaches a certain value, the shaft “lifts off” on a layer of oil. In fluid film bearings, lubrication requires a pair of surfaces with relative motion between them. There is always a convergent wedge developed that is formed due to the relative surface speeds and the lubricant viscosity to carry the applied load. An oil pressure film develops with equal and opposite force vectors to the applied load. One surface drags the lubricant, usually oil, into a converging gap. As the space available in this gap decreases, the fluid develops a pressure gradient, or pressure hill. As the fluid leaves the gap, the high pressure helps expel it out the other side. Lubricants can be any fluid, including gasses. Applications range from heavy low speed loads (solid lubricants) to light high-speed loads (gas). In addition, water can be used as in submerged pumps. Piston engines fall in the lower part of the velocity range where oil lubricants are used. In piston engines, lubricants are usually mineral oils from petroleum with viscostatic additives, up to synthetic lubricants with special additives for low and high temperatures. In film journal bearings, the load is supported by the motion-induced, high-pressure oil region. Oil inlet ports are placed in areas of minimum pressure. This position is critical since the oil cools down the bearing. The peak pressure is significantly higher than the specific load (F/LD). The pressure at the margins returns to the boundary condition, which is the oil sump pressure. The unloaded top half should not be cavitated. In addition, it is essential that a constant flow of cool oil arrives to the bearing to avoid cavitation damage. However, cavitation damage rarely occurs in piston engine bearings. In fact, almost all the cooling is done by the oil flow. If the oil flow is insufficient, oil temperature rises to excessive values, with reduction of viscosity or oil cracking. The most pressurized area is then deprived of the film sustentation and the surfaces come in contact with immediate seizure. The load induces eccentricity. Eccentricity is the most important design parameters. For example, if the eccentricity is too high there is a risk of metal-to-metal with immediate seizure. Eccentricity is a function of both speed and load. With a constant load, as speed increases, the eccentricity decreases. Oil viscosity is not so critical. In fact, it is possible to describe oil kinematic viscosity with temperature with equation (1).

$$\nu = -0.7 + e^{\left(e^{\alpha} T^{-\beta}\right)} \tag{1}$$

α and β values for a few type of multigrade oils are summarized in Table-1.

Table-1. Viscosity with temperature (equation 1).

Sae grade	**υ40** (cSt)	**υ100**(sSt)	**α(-)**	**β(-)**
20W/50	144.8	17.8	19.1324	3.04992
15W/40	114.3	14.9	19.4716	3.11737
10W/30	72.3	10.8	20.034	3.23221
5W/30	57.4	9.9	19.1925	3.09583
0W/20	44.4	8.3	19.3722	3.13831
30	91.3	10.8	21.7011	3.51372

The minimum oil film thickness in the bearing variation with the key variables (W, r, o, L, R, c) can be calculated with equation (2). Expressions (2) and (3) are valid when h_{min}<<c (film thickness << bearing play).

$$h_{\min} = \frac{\mu^{0.5} (\omega R)^{0.5} L^{1.5} F^{-0.5}}{2} \tag{2}$$

A very important factor in film thickness is the tangential velocity ω R that equals, in importance, the viscosity μ. Viscosity is less controllable as a design parameter. In fact, it depends on temperature (see Figure-3) and on bearing cooling. Bearing cooling depends on oil flow, that depends on pump and oil sump pressure and on the amount of depression at the oil input. The position of the hole in the shell for oil input in the bearing shell is critical. The pressures values inside the bearing depend again on tangential velocity ω R (3). In particular, maximum pressures (negative and positive) are reduced

ARPN Journal of Engineering and Applied Sciences

with tangential velocity ωR. Cooling requirement increases with ωR (4). Therefore, faster engines (in term of bearings) require higher pump pressure and better venting (for oil sump pressure control).

$$p_{\max} = \frac{c^{0.5} F^{1.25}}{\mu^{0.25} R(\omega R)^{0.25} L^{1.75}} \quad (3)$$

Figure-1 shows the h_{min} (for the various multigrade lubricants of Table 1) with speed (ωR) for a 2006 Formula 1 engine. The higher values are referred to the SAE 20W/50.The combination of the angle between the vertical axis in the load direction and the line of centre with speed tells the dynamic behaviour of bearing. However, a very important additional parameter is the maximum temperature that will be generated in the fluid film. At the same load and speed, the temperature increment depends on the inlet temperature and the net flow inside the bearing. With modern lubricants; it is possible to operate journal bearings above 200 °C. However, it is better to keep the maximum oil film temperature below 180° that due to loss of journal fatigue strength and oil cracking (deposits and degradation). The film thickness follows the viscosity in a moderate way (see Figure-2).

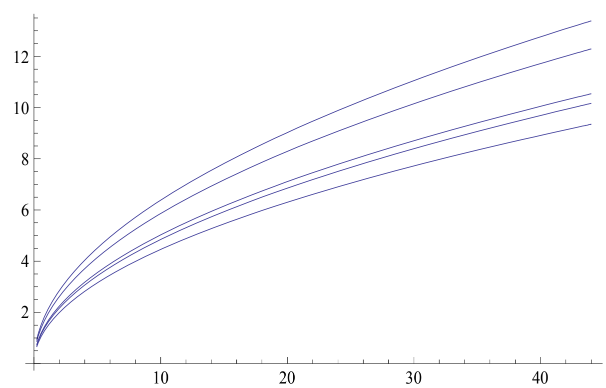

Figure-1. h_{min} (μm) vs peripheral velocity (m/s).

For fluid film bearings, viscosity is not the most controllable factor, as it can be seen in expression (1). Besides multigrade oil are very efficient in keeping the viscosity up with temperature (Figure-3).

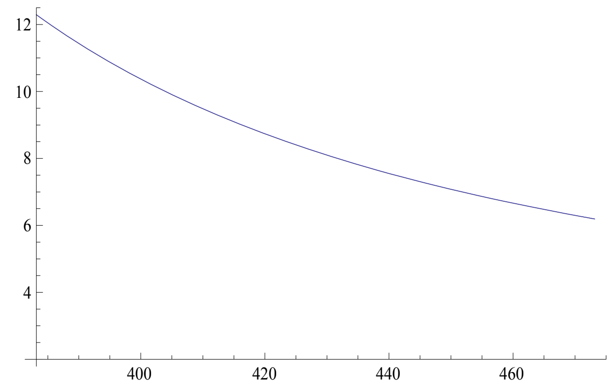

Figure-2. h_{min} (μm) vs. temperature (K) from 110C (373.15K) up to 200C (473.15K) for a 2006 F1 at full power.

As it can be seen from the comparison of Figures 2 and 3, the reduction of viscosity with Temperature is not so critical.

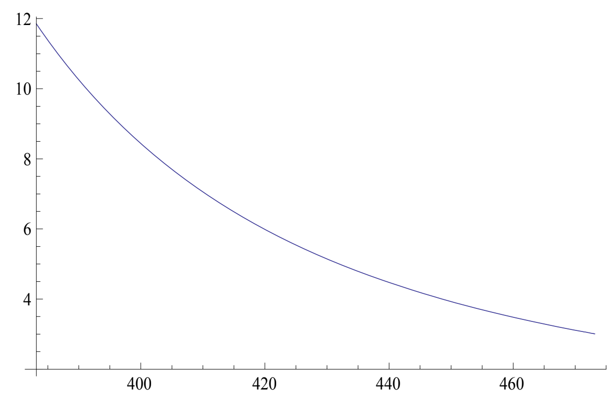

Figure-3. Viscosity C_{st} with T (K) for a SAE 15W/40.

Since the oil dissipates the power loss of a journal bearing, friction evaluation is of paramount importance (see equation 4).

$$P = 2\pi \frac{\mu^{0.75} R(\omega R)^{1.75} L^{0.25} F^{0.25}}{c^{0.5}} \quad (4)$$

As it can be seen in expression (4), the play *c* is a fundamental factor for good efficiency. The load F is nearly irrelevant while viscosity is not as controllable as tangential velocity ωR. The oil pumped-into (and sucked-by) the journal bearing cools done entirely the power P. The value of equation 4 is the minimum power loss; the true one may exceed this value by 50%. It is of paramount importance to supply a sufficient amount of lubricant to avoid an excessive temperature rise inside the bearing that unavoidably brings to oil thermal cracking with deposits and final seizure.

Bearing failure mechanisms

Oil-film bearings can fail for scoring, wiping, fretting, overheating, fatigue, and corrosion. Scoring and wiping are identified as circular parallel grooves on the bearing-lining surface. Scoring depends on lubricant contamination while wiping is caused by the displacement of the Babbitt-like material. In oil-film bearings, scoring builds a converging wedge pressure in the lubricant film of oil that keeps the surfaces completely separated. In scoring, relatively large abrasive contaminants find their way into the lubricating oil and travel with the oil through the bearing clearance, especially through the area of the minimum oil film. When the particles are not embedded into the bearing, they lead the continuous wearing and gouging of the Babbitt surface that occurs. The sizes of these particles depend on minimum film thickness. Even if these small particles are well tolerated during a full-film operating condition, they can generate wear when the shaft is running at tangential speeds less than the minimum speed required for a full oil-film development. This boundary condition happens during starts and stops. Bearings running at higher speeds require better filtering. In fact, the increased numbers of times the particles circulate through the bearing clearance accelerate the abrasion process. Furthermore, these contaminants will eventually embed into the Babbitt and form a harder lining surface compromising further embedding capability. Wiping is the displacement of Babbitt from the damaged zone to another location in the bearing. Usually, a temporary breakdown in the oil film causes boundary lubrication. The following direct contact between the shaft or thrust runner and the shell inner surface causes Babbitt melting with wiping. Also excessive peak pressure and temperature may cause localized plastic deformation of the Babbitt. The squeezed material is then pushed into the clearance and displaced by the shaft or thrust runner. Misalignment, unbalanced loading, excessive vibration, overloading and oil starvation cause wiping. This condition can also lead to bearing overheating. This problem is more prevalent in cold start-up where heat generated in the oil film may cause the shaft to grow more rapidly than the liner and bearing housing. Lemon bearings with the same average clearance of a cylindrical bore profile will have a larger horizontal clearance, therefore increasing oil flow and heat dissipation. Misalignment overloads locally one side of the bearing. Unbalanced load imposes a rotating dynamic force in the system, which results in an offset bearing pressure centre with an alternating edge loading of the bearing. Excessive vibrations are also induced from instability in the oil film. This is mainly due to manufacturing tolerances.

Lubricant degradation mechanisms

Common fluid degradation mechanism is time, oxidation, thermal breakdown, micro dieseling, additive depletion, contamination and dilution. Time: any oil of any type will last one-year time in a piston engine sump. It is mandatory to change lubricant at least once a year. Oxidation: it can be responsible for varnish, sludge and sediment formation, additive depletion, base oil breakdown, filter plugging, loss in foam properties, acid number increase, rust and corrosion. Oxidation is a prime factor in lubricant's life. Oxidation may also increase viscosity. Thermal Breakdown: the lubricant must separate the moving parts of the machinery and should also dissipate heat. Overheating over its recommended stable temperature can cause the evaporation of light ends and, in extreme cases, the decomposition. This can cause certain additives to be removed from the lubricant or the viscosity of the lubricant may increase. If the temperatures greatly exceed the thermal stability point of the lubricant, thermal cracking will occur. In this case, larger molecules will break apart into smaller molecules. This thermal cracking or thermal breakdown, can initiate side reactions, induce polymerization, produce gaseous by-products, destroy additives and generate insoluble by-products. In many cases, thermal degradation will cause a decrease in viscosity. Microdieseling is a pressure-induced thermal degradation. In this process air bubbles transit from a low-pressure to a high-pressure in a nearly adiabatic compression. This compression rises locally the temperatures in excess of 1,000°C, resulting in the formation of carbonaceous by-products and accelerated oil degradation. Additive Depletion: most additive systems are sacrificial. Lubricant substitution intervals depends on the specific degradation mechanisms. Occasionally but not commonly also electrostatic spark discharge may occur especially in mechanical filters. In this case, friction within the oil can generate static electricity and may accumulate to the point where it produces a sudden discharge or spark. This occurrence is extremely rare in modern filters. Contamination: metals such as copper and iron are catalysts to the degradation process. Water and air can provide a large source of oxygen to react with the oil. Lubricant dilution by fuel is a common cause of contamination. This may happen due to several reasons: excessive blow-by, frequent start-stop, overcooling and post-injection. It is an important factor in low emissions automotive diesel engines.

Problem solving: design issues

In many cases, these engines were designed many years ago and continuously updated. Old design methods for the crankshaft assembly began with the dimensioning of the various parts for strength. Crankcases were usually designed by similitude with similar engines with increased thicknesses and ribs. Journal bearings were dimensioned starting form specific pressures. A major problem of old diesel engines was the cold start. For this reason, the compression ratio was kept as high as possible. Since working temperature should be very high for good combustion and efficiency, the play of the piston assembly is usually quite large at start. When the engine heats up the play is recovered by the thermal expansion and the blow-by value assumes the final-correct values. Oil pressures are usually quite low to modern standards and the oil flow is limited by this value. Venting system of the oil sump or of the dry sump is not as efficient as in modern engines and sump pressures may be much higher than atmospheric ones. This situation is worsened by turbo charging that has

VOL. 12, NO. 22, NOVEMBER 2017 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

become common in this type of engines. The normal result is that journal bearing tangential velocity and oil flow are insufficient for modern design standards. For this reason,wiping caused by excessive loading, especially during starts and stops, is common. This problem is particularly important in pure mechanical injection systems, where the fuel is injected immediately during start at relatively high pressure [4] [5] [6]. At cold start, the blow-by is very high due to the excessive play in piston assembly. The temperature in the combustion chamber is then very low. Diesel combustion is controlled by temperature as it can be seen in equation from Wolfer (1938) (5). Low temperatures delay the ignition with fuel accumulation in the combustion chamber, irregular firing and low combustion speed.

$$\tau_{ign} = 0.44 P_{0}^{-1.19} e^{\frac{4650}{T_0}} \quad (5)$$

To increase the problem, a certain amount of fuel and gas pass through the large (cold) play between rings and liners. This play is necessary to compensate the differential dilatation of piston and cylinders at working temperature. This fuel dilutes the lubricant, while the gas increases sump pressure. The crankshaft-assembly bearings are then penalized by depleted lubricant and smaller oil flow. In fact, this latter depends on the differential pressure between pump outlet and sump. Historically, a hydrostatic-lubrication system is required to prevent boundary lubrication problems at start. Pressurized oil flows through channels and pockets into the bearing clearance to lift and support the shaft before the system starts or stops. Once the shaft reaches the speed necessary to develop a full hydrodynamic-oil film, then the hydrostatic-oil system shuts downs. However, the hydrostatic system requires the presence of holes in the bottom part of the journal bearing. Through these holes, the pressurized oil lifts the crankshaft and avoids shaft to shell contact. However, holes in the higher-pressure part of the bearing reduce film bearing dynamic load capacity. In most of cases, this will reduce journal-bearing life. Another typical solution was to use high viscosity oils. However, an oil-ring-lubricated bearing in low ambient temperatures may not deliver sufficient oil during start-up, which leads to boundary lubrication and wiping.

Problem solving: lubricant choice

Film bearings works when the film is dynamically formed between shaft and bearing. This condition happens only at a certain minimal tangential speed. In old engines, bearing diameters are quite small, it is then necessary to avoid loading before this film is formed. The first condition it to provide oil to the bearing. At the always (for a diesel engine) low temperature at start a viscostatic oil should be used. Typically, a 15SAE40 for temperate climatic zones, better if synthetic for higher stability and lubricity. To reduce warm up time, the oil can be heated with an external heating system. Also cold starting intake air heating system is beneficial, since the blow-by is reduced by these systems. The more regular combustion will reduce vibration and reduce bearing loads. The crankshaft assembly will always start with boundary lubrication condition s. For this situation, cold additives are highly beneficial; some of them are specifically designed for oil adhesion at cold start.

Problem solving: manufacturing tolerances and balancing

Manufacturing tolerances and crankcase stiffness are critical. Main journal bearing alignment is extremely important the main journal seats should be manufactured with special boring machine designed for the purpose. Crankcase stiffness is most critical for bearing alignment. Displacements goes with the cubic power of length and the larger the engine is the more difficult is to obtain sufficient stiffness. Old designs could not be checked for stiffness. In fact, with modern Finite Element Analysis it often possible to improve bearing alignment with minor modifications on the crankcase castings. Another important geometrical parameter of an engine bearing is the oil clearance or play. The importance of the clearance parameter is outlined in equation (3), where reducing the clearance reduces the maximum pressure.Typical values of oil clearance are summarized in Table-2. These values are measured with cold engine. Real values are reduced by the differential dilatation of the shaft and the housing. Usually the housing is colder than the shaft. The true "warm" clearance or play is then less than indicated in Table-2.

Table-2. Typical journal bearing parameters.

	Automotive	Racing
Cmin	0.001 R	0.002 R
Cmax	0.0015 R	0.003 R
e (mm)	0.005	0.03
Ra	0.2	0.2
Rz	0.6	0.4

The variation in normalized pressure with clearance is outlined in figure 4 for a 15SAE40@110 °C.

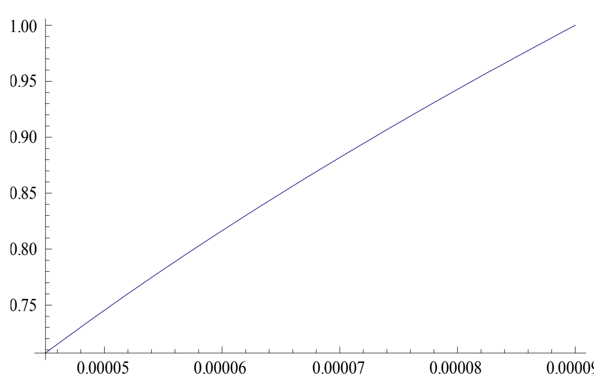

Figure-4. p/pmax vs clearance (mm) in a film bearing.

The maximum pressure is reduced also by tangential speed as shown in Figure-5.

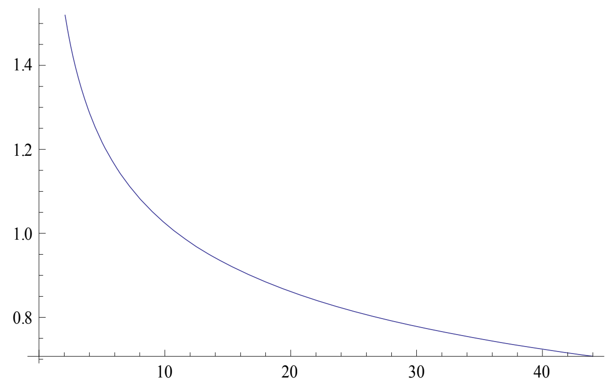

Figure-5. p/pmax vs tangential speed [m/s].

Figure-6 summarizes the two concepts of speed and clearance for a film bearing.

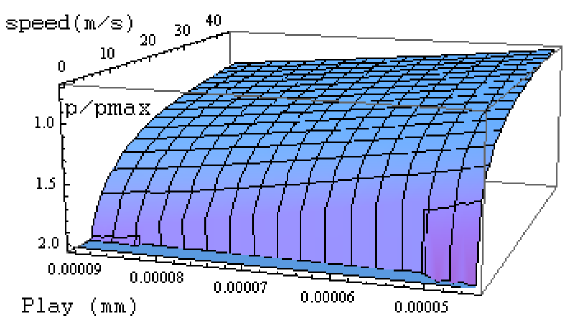

Figure-6. 3D graph of figures 4 and 5.

Reduced clearance reduces maximum pressures at any tangential speed. It is then of paramount importance to keep play c at the minimum. Therefore, roughness of journal bearings surfaces should be kept at the minimum possible. Journal of worn engines will fail more easily than new ones. The inside bearing surface is not round. It has a lemon shape due to the varying thickness of the bearing radial clearance having maximal value at the centreline and gradually decreasing towards the shell parting line. The difference between the maximum and minimum radial clearance is called eccentricity e (see Table-2). The position of the eccentric measurement is over the parting line by a certain amount h and this value should be specified. In the case of Table-2, h is 8 (mm). Tolerances are critical also for the journals. The fundamental geometrical parameter of a journal is its diameter that commonly have a tolerance within 10-20µm. The actual journal shape deviates from the perfect cylinder geometry. As the journal pin diameter varies in the axial direction, the journal shape forms one of the taper (conical), barrel (convex) or hourglass (concave) pattern. The best is the convex pattern since it is more stable. The journal diameter deviation on the axial direction should be within 0.001 L. Roundness defect (out of round deviations) should be lower than 1µm along the radial direction. In fact, grinding chatter marks produce variable shape of the oil film during the rotation. This variation tends to break the oil film between the bearing and journal surfaces. The main pins of a crankshaft should be aligned (concentric). Deviations from concentricity facilitates the direct contact between the bearing and the misaligned journal pin. Misalignments are particularly dangerous for modern tri-metal bearings due to thin overlays, which are easily removed by the contacting journal. Maximum misalignment of the main pins should be inferior to 3 µm as an overall value and 1.5 µm for adjacent journals. Excessive wear of a bearing surface may also be caused by the direct metal-to-metal contact during start and stop due to the journal surface roughness. Surface quality is particularly important in bearings operating with low oil film thickness as modern piston engines. Maximum Ra (average roughness) and Rz (average maximum height of the profile) for journals are shown in table 2. Crankcase bores have a tolerance on diameters of30µm, a roughness Ra<0.8; maximum ovality is 0.2µm only if the diameter along the parting line) is larger than that the other direction. Elastic radial displacements should be kept to half the above values. The maximum overall/adjacent misalignment of the bores 0.4/0.2µm. The diameter tolerance for the connecting rod is 10µm with Ra of 0.8. The ovality is allowed again only along the parting line with a maximum of 25µm. The taper/barrel/hourglass diameter deviation should be not greater than 3µm with a maximum twist between rod bore and wrist pin hole of 25µm. Vibrations cause the journal to orbit in the bearing. This causes an oscillatory dynamic pressure to act on the bearing surface. It is common that the peak hydrodynamic pressure to reach 3 to 5 times the calculated one (3). Bearing fatigue develops in a manner similar to potholes in a road surface. In the journals, surface cracks propagate to the bond line below. The detached piece of Babbitt is unable to leave the area because of the close clearance to the shaft. Therefore, it breaks down into smaller bits, which are carried away by the oil film. This hammering action smooths the sides of the pit leaving a rounded smooth hole. Therefore, it is of paramount importance to balance at the best values all the parts of the engine including the accessories. Unbalanced parts may cause bearing failures in positions far away from the origin of the vibration. In fact, vibrations run into parts with waves at the velocity of sound (Mach 1).

Problem solving: lubrication system

The lubrication system is of paramount importance in the bearing life. Filters should be of the topmost technology available, since they not only filter particles and dirt but also water and acids that may be found in the oil. It is extremely important to keep the oil flow to the required value. For oil flow control, in modern ECU controlled engines it is convenient to use an electrically driven oil pump. Old engines with a mechanical driven pump tend to have insufficient oil and pressure output especially at the end of the TBO. A continuous check of the oil pressure is convenient and, in a few cases, it is necessary to add a boost pump especially at cold start when oil viscosity is at the top. Other critical parts are the oil gaskets and seals. In fact, in classical designs, the gaskets limit the pressure to an acceptable value to lubricate the valve train and its transmission. If these components are worn out, the pressure will be kept at the required value but the oil flow will be subtracted to the journal bearing with starvation and overheating risks. Finally, the oil flow depends on the difference between the pump outlet and the sump pressure. Insufficient sump venting is a classical problem in engines as erroneous position of the suction strainer. In this case, it is possible to have both low pressures or even near vacuum at suction position or air entrapment. Foam and air entrainment problems are quite common, but are traditionally hard to treat. Almost all lubricating oil systems contain some air. Air is found in four phases: free air, dissolved air, entrained air and foam. Free air may be trapped in the system, such as an air pocket in a lubrication line, and may have minimal contact with the fluid. It can be a contributing factor to other air problems when lines are not bled properly during engine start and free air is drawn into circulating oil down to the bearings. Dissolved air becomes a problem when temperatures rise rapidly or pressures drop. Petroleum based oils may contain as much as 12 percent in volume dissolved air. When the engine starts up or when a part overheats, the dissolved air changes into small bubbles. If the bubbles are less than 1 mm in diameter, they are stable inside the liquid phase of the oil, particularly in high viscosity oil at start-up (air entrapment). Air entrainment is treated differently than foam, and is most often a completely separate problem. Some of the potential effects of air are pump/bearing cavitation, erratic operation of hydraulic tensioners, vibrations, oil oxidation, component wear, switching of oil pressure switches, micro-dieseling due to the compression ignition of the bubbles, loss of head if the oil pump is of the centrifugal type. Depending on application, lubricating oils are susceptible to contamination from combustion results and wastes, unburnt fuels, particulates, sealing materials, process fluids, rust preventives, cleaning compounds, airborne contaminants... Particulates act as seeds or nucleation points on which bubbles grow. Antifoam additives may also be attracted to their surface, reducing their effectiveness in the bulk oil. In practice, dirty oil can foam more than clean oil. Water generates foam, but it will dissipate faster. In engine oils, particularly highly additivated oils, oil and additive breakdown may result in an increased foaming tendency. Sump design plays a significant role in controlling foam and air entrainment. Keeping the reservoir inlet below the surface of the fluid to prevent splashing is an obvious way to prevent foaming. If this is not possible, install a wire screen that acts as a nucleation site for bubbles and prevents foam from entering the outlet (suction line). In dry sumps, scavenger pumps often mix large amounts of air into the fluid. In this case, instead of keeping the inlet below the level of the fluid in the reservoir, raise the inlet so that air has time to dissipate before it reaches the bulk oil. If the inlet is too close to the outlet, install baffles or wires to increase residence time. Loose fittings on the suction side of a pump are a common source of air. Oil separators and inclined surfaces favor the elimination of foam.

Problem solving: cold starting

Studies in laboratory engines equipped show that seizure probability due to tribological problems is highest during at cold startup (up to 90% probability). The higher viscosity of the oil retards the oil flow/pressure grow up and increases the time of boundary-condition lubrication. Engine blow-by is has its highest value due to the large play of piston assembly. If in this condition fuel injection takes place, it is highly probable that small amount of fuel mix in the sump increasing the foaming effect. Cavitation may easily occur, in the pump and in the oil ducts. Sump pressure is also very high and the condition can worsen if irregular combustion takes place. In this case, vibrations and local overloads are common. It is a typical situation of journal bearings seizure. In many cases, the engine will finally start and you will run it for a few time with slightly increased noise. Then suddenly the engine will stop. It may restart again with increased noise down to final failure. It is then of paramount importance to assure the oil supply and to make the bearing float before applying load. Suitable oil should be used; a good synthetic base with cold start low-friction additives is the best solution. Since loading is critical, both compression loads and combustion ones should be strictly controlled. If it is possible, it is better to preheat oil and air before starting the engine. The disconnection of the load from the engine is also beneficial. When applicable, the compression ratio can be reduced by a valve-lifter of a pressure relief mechanism in the cylinders or in the head. In turbocharged engines, it is possible to reduce the compression ratio and to recover pressure and power by applying more boost during warm engine operation. Since the bearings should float, it is necessary that the starting devices provide a sufficient tangential velocity to the bearings. In old engines, bearings have a too small diameter. In this case, it is necessary to increase the starting speed. As the crankshaft assembly floats and oil pressures reach the required value, it is possible to begin fuel injection. Preheating devices should be used and low injection pressures with proper injection mapping. Regular combustion is very important. Fuel quantity should be as low as possible to avoid overpressures. The almost instantaneous start of the modern automotive CRDID (Common Rail Diesel Engines) are required by the emission requirements and asked by customers who do not like to wait. Powerful starter motors and extremely efficient low voltage glow plugs (that reach temperatures up to 1300K) obtain the extremely short start. The heat is then drained where possible, from fuel return piping, from alternator, in order to reduce the warm up time as much as possible. From 1997 up to now, CRDID compression ratios have been reduced from 22:1 down to 14:1 in many engines. This is also for start-up seizure problems. However, it is better to take the time necessary to be sure that the bearings are floating and the oil film is well formed before starting the engine. In this time, the play will begin to be reduced by

ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

the progressive heating of the engine. Even with slow starting, load reduction at start is of paramount importance. If the problem is particularly acute, it is possible to treat the journal and bearing surface. The most efficient treatment for the journal is grinding to tight tolerances and to the barrel shape, honing to the best finish possible and then the DLC treatment. For the journals several antifriction treatments are available. Beware that the particles that are released by the antifriction treatment are compatible with the lubricant you are using. To preserve the adhesion of the oil film on the bearing when the engine is at rest, it is better to start the engine regularly. This operation also prevents corrosion and adhesion of parts. Finally, the oil should be replaced at least once a year. In addition, tolerances and stiffness should be checked.

Problem solving: overheating

Theoretically, overheating can be solved by increasing the bearing clearance. This is generally false since increasing bearing clearance means increasing pressure rise in bearing hydrodynamic film. Since the oil always contains air, microdieseling takes place with deleterious effects on bearing Babbitt and bearing TBO (Time between Overhaul). Another common concept is to increase oil viscosity. The effect of this operation may be partially beneficial, but, in many cases, the oil flow is reduced and the true viscosity increase is limited. The best operations that should be done are to check tolerances, balancing procedures for all engine components and to increase oil pressure. If necessary, a priming pump can be used. The entire oil system should be checked to increase the net oil flow down to the unluckiest bearing of the system. In many cases, it is the suction system that it defective and should be revised. Finally, it is possible to change the Babbitt-like material. Several improved materials are available on the market. However, tolerances and oil are the most critical factors. In addition, the inlet port position of the oil should be checked. Finally, the oil cooler efficiency should be enhanced to have lower outlet temperatures. In some cases, also the dry sump solution reduced overheating by freeing the sump of the warm oil.

Problem solving: short bearing TBO

In most cases short bearing TBO is caused by the oil. Lubricant optimization, filtering and substitution intervals are most critical. In addition, floating surface of the bearing can be increased by working on the eccentricity of the lemon shape. Harder materials can be used for the Babbitt, if cold starting is not a known problem.

CONCLUSIONS

Tribological problems are extremely common in classical DI (Mechanical Direct Injection Diesels) still widely used in Power Generation, Diesel-electric Locomotives and Marine applications. These engines have displacements from 6l up to 30l and output power outputs from 300 up 1000 kW. Usually the maximum shaft speed is under 3000 rpm and the original design included a mechanical injection system. A few of these engines have been upgraded with an electronically controlled injection or a "common rail" system. In any case, engines have been continuously updated from the first prototype, that, in a few cases, was conceived after the First World War The maritime environment and the continuous increase of performances and TBO provoked several tribological problems that have been partially solved through the years. However, significant improvements are still possible with modern technology in journal bearings, lubrication systems and lubricants. Often cold starting problems are confused with overheating problems since final failure occurs during ordinary operations. However, 90% of the failures originate during cold starting. It should be understood that the journal should float before being loaded. It is of paramount importance to reach the proper tangential speed before starting combustion. The use of high viscosity oil is not beneficial and pre-lubrication (hydrostatic lubrication) add holes in positions that will deplete bearing full load capacity and duration. It is better to choose good oil with the right additives, control the tolerances and take the required time to start the engine. This paper indicates tolerances and stiffness necessary for correct operation. In old engines, journal diameters are often insufficient; tighter tolerances, reduced clearance and higher oil pressure can solve the overload of the bearings. In addition, the suction system is extremely critical and should be checked for trapped air, foam and leakage. Finally, the oil cooler true efficiency should be verified. As a last solution, several surface coatings are available for journals and bearings. However, their high costs are justified only by the unfeasibility of other more economical solutions.

Symbols

Symbol	Description	Unit
τ_{ign}	Ignition delay	ms
p_o	Combustion chamber pressure	MPa
T_o	Combustion chamber temperature	K
hmin	Minimum film thickness	m
T	Oil temperature	K
ωR	Tangential velocity (relative velocity journal-bearing)	m/s
ω	Angular velocity	1/s
D	Journal diameter (D=2R)	m
μ	oil viscosity	Pa s
υ	Oil kinematic viscosity	Cst
L	Journal width	m
F	Load	N
c	Bearing radial clearance	m

REFERENCES

[1] L. Piancastelli, L. Frizziero, E. Pezzuti. 2014. Aircraft diesel engines controlled by fuzzy logic. Asian Research Publishing Network (ARPN). Journal of Engineering and Applied Sciences. ISSN 1819-6608, 9(1): 30-34, EBSCO Publishing, 10 Estes Street, P.O. Box 682, Ipswich, MA 01938, USA.

[2] L. Piancastelli, L. Frizziero, E. Morganti, A. Canaparo. 2012. Fuzzy control system for aircraft diesel engines. International Journal of Heat and Technology, ISSN 0392-8764, 30(1): 131-135.

[3] L. Piancastelli, L. Frizziero and I. Rocchi. 2012. Feasible optimum design of a turbocompound Diesel Brayton cycle for diesel-turbo-fan aircraft propulsion. International Journal of Heat and Technology. 30(2): 121-126.

[4] L. Piancastelli, L. Frizziero, N.E. Daidzic, I. Rocchi. 2013. Analysis of automotive diesel conversions with KERS for future aerospace applications. International Journal of Heat and Technology, ISSN 0392-8764, 31(1).

[5] L. Piancastelli, L. Frizziero, E. Pezzuti. 2014. Kers applications to aerospace diesel propulsion. Asian Research Publishing Network (ARPN). Journal of Engineering and Applied Sciences. ISSN 1819-6608, 9(5): 807-818, EBSCO Publishing, 10 Estes Street, P.O. Box 682, Ipswich, MA 01938, USA.

[6] L. Piancastelli, L. Frizziero, E. Morganti, E. Pezzuti. 2012. Method for evaluating the durability of aircraft piston engines, Walailak Journal of Science and Technology, ISSN 1686-3933, 9(4): 425-431.