FRICTION LOSSES AT CRANK MECHANISM BEARINGS OF A DIESEL ENGINE

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Abstract

From the point of user it is important to minimize engine friction losses because it means the reduction in fuel consumption and emissions. Among all losses encountered during energy transformation a significant part relate to the friction in crank mechanism, in particular in bearing nodes.

This paper presents results of calculation of averaged friction losses generated at the main and crank bearings of a diesel operating in a wide range of rotational speeds. A main bearing served as an example for demonstration of utilization of obtained results to an analysis of friction losses from the perspective of their relation against engine certain constructional (dimensions, clearances, lubricating oil viscosity) and operational (speed, load, oil properties) parameters. Total power required for overcoming friction resistance at main and crank bearings has been presented in relation to the variable rotational speed.

Keywords: *friction, IC engine, friction losses, slide bearings*

1. Introduction

It is important to minimize the friction losses of a running engine because they substantially affect fuel consumption and emissions. One of the ways toward the effective reduction in fuel consumption and improvement of a number of indices like engine durability and reliability is minimization of friction losses through proper engine design and operation.

In the literature on friction losses one can find various percentage share of individual friction nodes in total losses generated by the engine. According to Martin [9] losses encountered at the “piston-rings-pin-cylinder” assembly account for about 44%, bearings 22% and valve train another 6%, which gives about 72% of all mechanical losses. Other authors associate the volume of friction losses with rotational speed. Fig. 1 presents a typical division of friction losses in relation to the crankshaft rotational speed taking into account a number of considerable factors [1]. A significant shear in total mechanical losses of modern engine have an auxiliaries drive. Losses relative to the valve train drive prevail at low speeds, while those connected with piston-cylinder assembly and crank mechanism quickly increase with the increase in rotational speed.

Both constructional and operational factors affect friction losses connected with main and crank bearings operation. The most important are: bearing dimensions, clearance, mass of engine moving parts, rotational speed, lubricating oil (its viscosity in particular). Friction losses can be determined analytically utilizing relevant computational procedures, above all forces in crank mechanism and hydrodynamic parameters of slide bearings as well as carrying the test stand measurements.

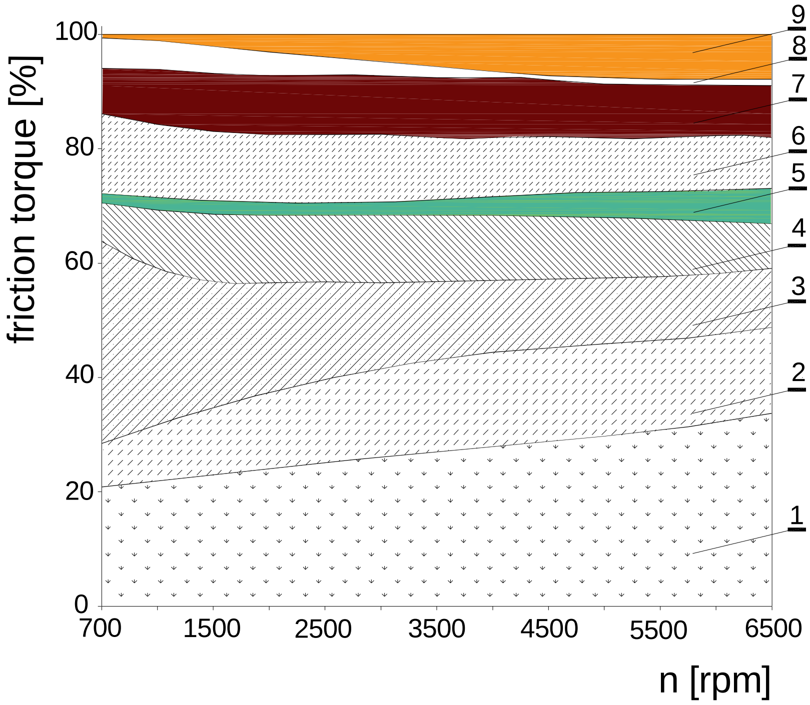

Fig. 1. Friction losses of engine individual subassemblies and auxiliaries: 1 – piston-cylinder node, 2 – crank system, 3 – valve train, 4 – oil pump, 5 – alternator, 6 – water pump, 7 – auxiliary pump, 8 – vacuum pump, 9 – balancing shafts, 10 – fuel pump (GDI engines) [1]

2. Friction losses at crank mechanism bearings

Besides the hydrodynamic lift due to lubricating wedge, dynamically loaded bearings of engine crank mechanism experience hydrodynamic lift due to lubricant squeeze when the journal displaces relatively to the bush. Superposition of both these phenomena is utilized to determine the characteristic parameters of bearing operation (see Fig. 2), including friction losses.

Total friction losses of dynamically loaded bearings equal:

$$N_t = N_k + N_w, \tag{1}$$

where:

N_k – friction losses due to rotation (wedge effect),
N_w – friction losses resulting from displacements (squeeze effect).

The friction losses understood as the flux of energy absorbed as the result of overcoming the resistance to motion, have the dimension of friction power. The friction power N_k (wedge) consists of a part resulting from shear due to speed gradient in lubricant layer between surfaces of journal and bush for angular velocities ω_1 and ω_2, respectively. The friction power N_w (squeeze) results in turn from the work of dumping the journal displacement relative to bush performed by the lubricant squeezed from bearing.

In order to determine the friction forces acting on surfaces of journal and bush, balance of forces acting on an element of lubricant in lubricating gap should be analyzed. This balance of friction forces and pressures acting on an element of lubricant has been presented in Fig. 3.

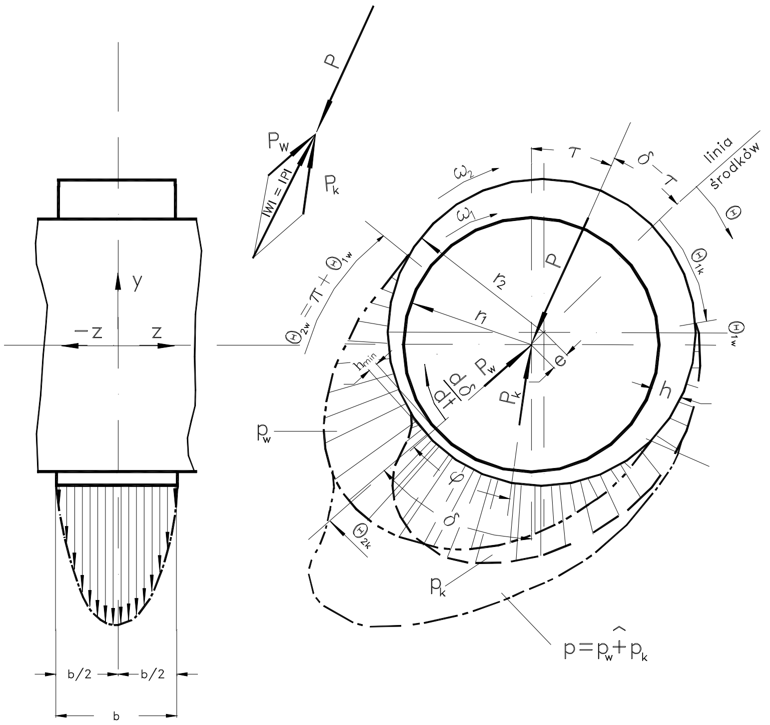

Fig. 2. Schematic of a cylindrical slide bearing, geometric marking and a principle of lift summation: P – bearing load, P_k – oil film reaction due to wedge effect, P_w – oil film reaction due to squeeze effect, r_1 – journal radius, r_2 – bush radius, e – eccentricity, h – lubricating gap variable height, b – bush effective length, τ – angle of load direction, δ – angle of centers line direction, φ – angle between P_k and P_w, Θ – angular coordinate measured from the thickest layer of lubricant (indexes mark respectively beginning and end of the region of individual effects), ω – angular velocity of journal (1) and bush (2), $d\partial/dt$ – angular velocity of line of centers

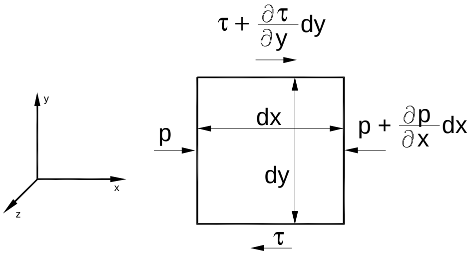

Fig. 3. Balance of forces acting on a lubricant element in lubricating gap

According to Fig. 3:

$$p \cdot dy + \left(\tau + \frac{\partial \tau}{\partial y} dy\right) dx - \left(p + \frac{\partial p}{\partial x} dx\right) dy - \tau dx = 0 \ , \tag{2}$$

where:

p – hydrodynamic pressure,

$\tau = \eta \frac{\partial u}{\partial y}$ – shear in lubricant layer (η – lubricant dynamic viscosity, $\frac{\partial u}{\partial y}$ – velocity gradient in direction of lubricant layer thickness).

From Eq. (2) it comes:

$$\frac{\partial \tau}{\partial y} = \frac{\partial p}{\partial x}, \tag{3}$$

and after substitution of τ to (2)

$$\frac{\partial^2 u}{\partial y^2} = \frac{1}{\eta}\frac{\partial p}{\partial x}. \tag{4}$$

As result of double integration of (4) relative to y, we receive:

$$u = \frac{1}{2\eta}\cdot\frac{\partial p}{\partial x}\cdot y^2 + C_1\cdot y + C_2. \tag{5}$$

Integration constants can be defined using boundary conditions that are as follows:
for $y = 0$ $u = u_2$ which corresponds to the bush circumferential speed,
for $y = h$ $u = u_1$ which corresponds to the journal circumferential speed.
Hence

$$u = \frac{y(y-h)}{2\eta}\cdot\frac{\partial p}{\partial x} + \frac{h-y}{h}\cdot u_2 + \frac{y}{h}\cdot u_1. \tag{6}$$

Elementary force of friction in the layer of lubricant

$$dS = \tau\cdot dx\cdot dy. \tag{7}$$

Substituting (6) to (7), we receive:

$$\begin{aligned} dS &= \eta\cdot\frac{\partial u}{\partial y}dx\cdot dz = \eta\cdot\frac{\partial}{\partial y}\left[\frac{y(y-h)}{2\eta}\cdot\frac{\partial p}{\partial x} + \frac{h-y}{h}\cdot u_2 + \frac{y}{h}\cdot u_1\right]dxdz = \\ &= \eta\cdot\frac{\partial}{\partial y}\left[\frac{1}{2\eta}\cdot\frac{\partial p}{\partial x}\left(y^2 - yh\right) + (u_1 - u_2)\frac{y}{h} + u_2\right]dxdz, \end{aligned} \tag{7}$$

and after differentiation

$$dS = \left(\eta\frac{u_1 - u_2}{h} - \frac{h}{2}\frac{\partial p}{\partial x} + y\frac{\partial p}{\partial x}\right)dxdz. \tag{8}$$

For $y = 0$ – bush slide surface:

$$dS_2 = \left(\eta\frac{u_1 - u_2}{h} - \frac{h}{2}\frac{\partial p}{\partial x}\right)dxdz, \tag{9}$$

and for $y = h$ – journal slide surface:

$$dS_1 = \left(\eta\frac{u_1 - u_2}{h} + \frac{h}{2}\frac{\partial p}{\partial x}\right)dxdz. \tag{10}$$

Friction force on the journal surface is bigger than on the bush surface, hence the friction power should be calculated using the S_1 force.

Taking into consideration the difference in circumferential speeds of journal and bush that equals $u_1 - u_2 \approx r(\omega_1 - \omega_2) = r\cdot\omega_s$ and introducing the circumferential coordinate $\partial x = r\cdot\partial\Theta$, the friction force in the lubricant layer adjoining to the journal surface

$$S_1 = r\int_{-\frac{b}{2}}^{+\frac{b}{2}}\int_{\Theta_{1k}}^{\Theta_{2k}}\left[\eta\cdot\frac{r\cdot\omega_s}{h} + \frac{h}{2}\cdot\frac{\partial p}{r\cdot\partial\Theta}\right]d\Theta dz. \tag{11}$$

Hence [8]:

$$S_1 = \frac{b \cdot d \cdot \eta \cdot \omega_s}{\psi} \cdot \frac{\pi}{\sqrt{1-\varepsilon^2}} + \psi \cdot \frac{\varepsilon}{2} P_k \cdot sin\varphi , \qquad (12)$$

where – besides earlier introduced indications, ψ is a bearing relative clearance and ε – relative eccentricity.

Due to periodical changes in position of journal center relative to bush center (dynamically loaded bearing) the friction force changes periodically as well.
Friction power:

$$N_t = \frac{1}{m \cdot \pi} \cdot \frac{d}{2} \cdot \omega \cdot \int_0^{m\pi} S_1(\alpha) d\alpha , \qquad (13)$$

where: m = 2 for two stroke engine and m = 4 for four stroke engine, d – bearing nominal diameter, ω – crankshaft angular velocity, α – crankshaft angle. The friction force S is being numerically integrated.

3. Results of calculation and analysis

In order to determine the friction losses (friction power) at the engine crank mechanism bearing a method presented in the previous chapter has been utilized. The computations have been carried out using earlier formulated computer programs. A turbocharged diesel of nominal power N_e = 66 kW and maximum torque M_o = 195 Nm has been selected for calculations [5]. A part of technical data have been provided by the manufacturer, other were found during engine tests at test bed (indication diagrams) and previously conducted computer calculations of bearing loads and hydrodynamic parameters of bearing operation [4, 10]. Computations have been carried out for following speeds: idle run (800 rpm), maximum torque (2500 rpm) and maximum power (4100 rpm).

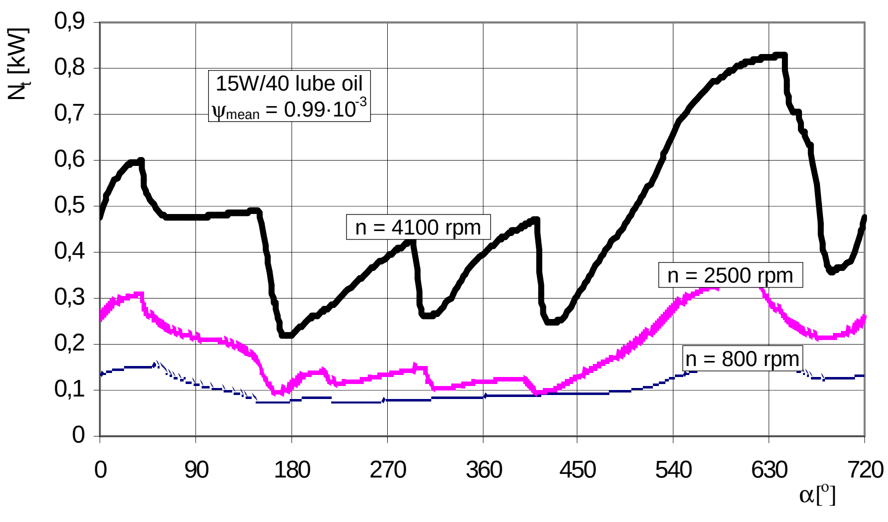

Fig. 4. Course of momentary friction power at B main bearing vs. crank angle for various speeds

Eventually presented results concern the highly loaded main bearing B. Fig. 4 presents the course of momentary power dissipated at the bearing as a result of friction against the crank angle within the full cycle. The mean friction power is: N_{tmean} = 0,10 kW, 0,19 kW and 0,47 kW at n =800, 2500 and 4100 rpm, respectively.

For the further analysis of the relation between certain parameters and magnitude of friction losses at the bearing, an averaged value of friction power within the entire cycle has been assumed. When analyzing the effect of bearing size on friction losses a length to nominal diameter ratio (b/d) was taken into consideration. Since the bearing half bush has the circumferential groove, an effective length (b_c) has been taken into account. Fig. 5 presents the course of friction loss change vs. bearing length b_c (and b/d ratio as well) for various rotational speeds.

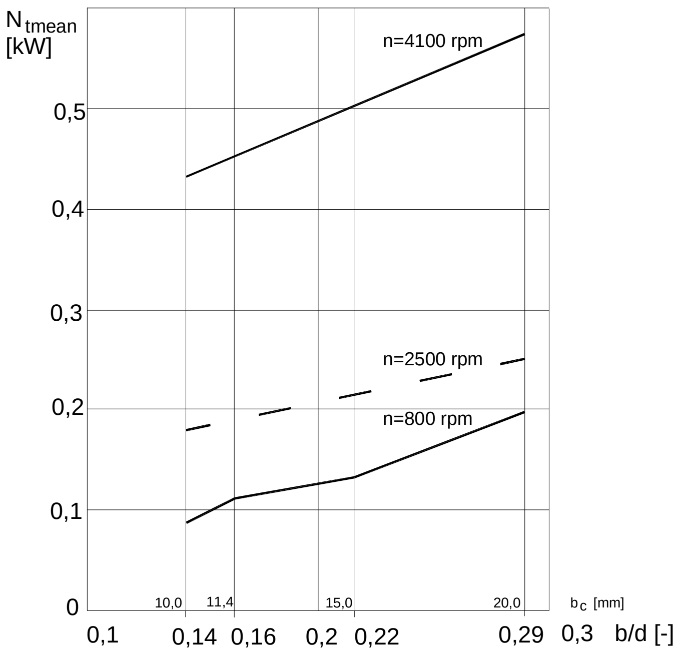

Fig. 5. Mean friction power vs. length of main bearing (bearing nominal diameter d = 0.070 m, half of grooved bearing effective length b = 0.01142 m)

The bearing clearance is another parameter influencing the friction losses. For the speeds analyzed, there is a close relation between the value of bearing clearance and friction losses as it has been presented in Fig. 6. Along with the increase in bearing clearance the friction losses insignificantly decline. A considerable drop in friction losses can be observed in the range of minimum and medium clearances for the speed of idle run (n = 800 rpm). All calculations have been carried out for the SAE 15W/40 mineral lube oil. In the case of minimal bearing clearance a replacement of SAE 15W/40 oil with the synthetic 5W/40 one would be advantageous because the friction losses could be diminished by about 60% (from 0.3 kW to 0.12 kW).

The grade of lubricating oil, especially its viscosity is an important factor influencing the friction losses at bearing. Three types of the Lotos lube oil, namely synthetic, semisynthetic and mineral ones (the last fresh and used) have been chosen for calculations. Their viscosity corresponds to the manufacturer's data and in the case of used oil its viscosity remains within the predicted range (increase or drop in viscosity by 40 and 20 percent, respectively) [6]. Fig. 7 presents the friction losses depending on oil grade (calculated for bearing mean clearance) for different rotational speeds. Despite the oil grade friction losses at the bearing are very similar for the same speed (2500 and 4100 rpm). In the case of idle run use of the 5W/40 synthetic oil brings about definitely lower losses. Comparing losses resulting from application of fresh and used 15W/40 mineral oil, an insignificant fall in their value could be observed for the used oil.

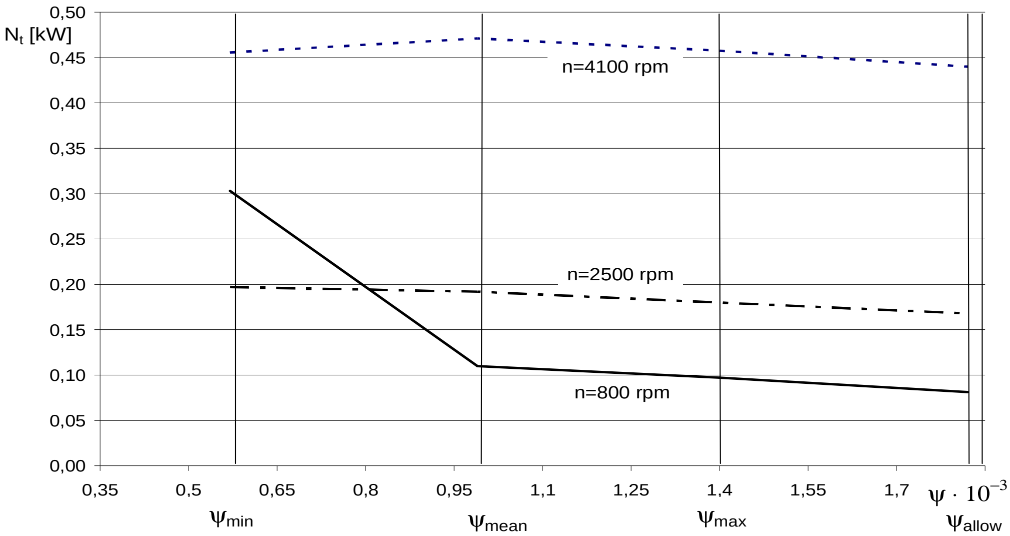

Fig. 6. Mean friction power vs. bearing clearance for various crankshaft speeds

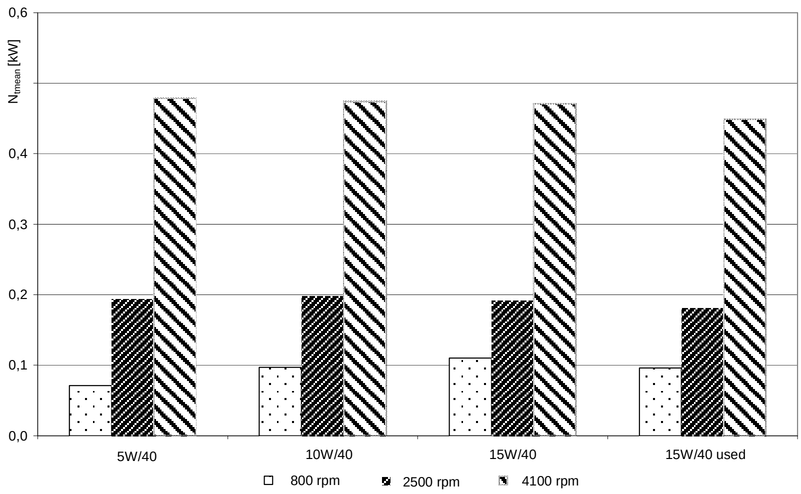

Fig. 7. Bearing mean friction losses for tested oils and speeds

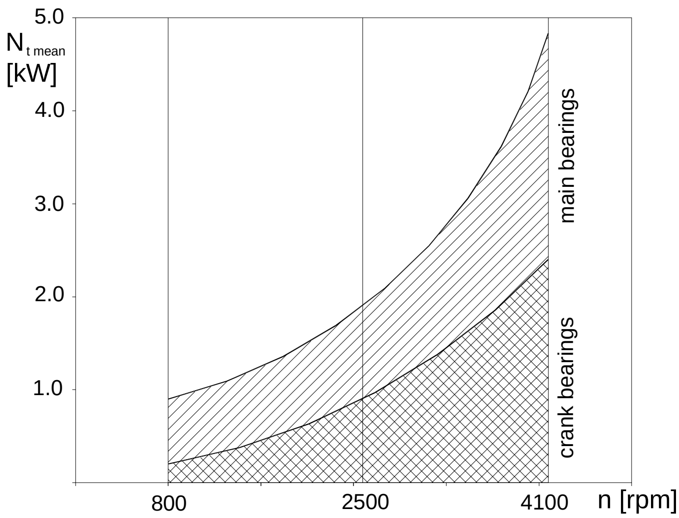

Fig. 8. The course of change in friction losses at engine main and crank bearings

Fig. 8 presents a diagram summarizing friction losses at all main and crank bearings vs. crankshaft speed. Curves representing friction losses combine friction power calculated for individual main (five fold) and crank (four fold) bearings. Calculations were carried out for a mean relative clearance and for SAE 15W/40 grade lube oil.

4. Conclusions

The friction losses calculated for running crank mechanism bearings allow to conclude that the rotational speed affects them to the highest degree. For the idle run speed the losses at crank bearings are a half of those generated at main bearings. However, along with the increase in speed they increase as well and for the speed of 4100 rpm are comparable to the losses at main bearings.

For the most loaded bearing (the B main bearing) the friction losses:

– increase with the increase in bearing length,
– decrease slightly with the increase in bearing clearance, while the significant drop in their value could be observed for 800 rpm within the clearance range ψ_{min} and ψ_{mean},
– are almost the same for oils analyzed at the same crankshaft speeds; in case of synthetic oil far lower friction losses correspond to the idle run speed. This is why this kind of oil is recommended for cold starts.

Eventual works on friction losses should take into consideration higher number of factors affecting the power necessary to overcome friction resistance at main and crank bearings. Then the decrease of those losses could be achieved thanks to the proper design and optimization, what in turn results in lower fuel consumption and lower toxic emissions.

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