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IMPACT OF OIL VISCOSITY ON FUNCTIONAL PARAMETERS OF JOURNAL BEARINGS IN INTERNAL COMBUSTION ENGINES

Abstract

The IC engine crnakshaft bearings, especially at the crankpin journals, are very loaded elements. For their proper functioning it is necessary to define carefully the working conditions, where the quality of oil is one of the most important factors. From the aspect of the working conditions of journal bearings it is important to ensure a minimum change in oil viscosity in a wide IC engine operating range. Selection of lubricating oil from the standpoint of physical characteristics, primarily the appropriate viscosity, has an important place in the area of reducing friction losses in journal bearings and increasing IC engine efficiency.

This paper deals with the impact of lubricating oil viscosity on the working conditions of vehicle turbocharged diesel engine crankpin journals. Analyzed parameters are: an oil temperature in the bearing, a change of the maximum value of the average pressure in the oil, the minimum carrying oil layer and the share of mixed friction, all in the function of oil viscosity at the bearing inlet, with different clearances between bearing and crankpin journal. Results were obtained using the model and own developed software to simulate the working conditions of journal bearings with an experimental verification.

1. Introduction

Today there are several basic goals for the development of internal combustion engines:

- reducing the emission of contaminating substances in exhaust gases and reducing the noise,
- increasing the efficiency of transformation of fuel internal energy into mechanical work at the outlet of internal combustion engine and
- the use of non-conventional fuels.

The previous requirements, with constant complying with legal limits on the emissions of contaminating substances and on the noise as well as on reducing specific fuel consumption, included:

- massive introduction of turbocharged engines with high level of air compression behind the compressor and variable geometry at the turbine inlet and
- increasing number of electronically controlled processes in the engine (mechatronical systems in the engine and its equipment).

Apart from many advantages, the use of turbocharging with higher level of air compression caused the increase of mechanical and thermal loading of the engine, of piston mechanism especially, which is not favorable from the point of the engine design. Due to increased mechanical load and certain design modifications on the crankshaft bearing, the journal bearings of internal combustion engines became more prone to malfunction. This especially relates to crankshaft bearing which are exposed to highest loading. Increased mechanical load at the engine, resulting from forcing the engine power (increasing the specific engine power) inevitably leads to reducing the ratio of the bearing width and the diameter, the increase of lubrication oil temperature, etc. In the conclusion we can say that for the proper operating of journal bearings the following optimizations need to be performed:

- of constructive bearing characteristics (dimensions, clearances) and
- of lubrication system characteristics (oil type, oil flow, inlet oil temperature, etc.).

The paper presents the functional values analysis of journal bearing in a truck turbocharged diesel engine crankpin as compared to the functional parameters of one main bearing in terms of changing lubrication oil viscosity. Modelling methods with own developed softwares were used for the analysis of oil viscosity influence on functional parameters of journal bearing, along with appropriate verifications of these results based on experimental research.

2. Model for determining movement parameters of a crankpin in the bearing

A physical model of crankpin-bearing system, along with all the relevant parameters, is shown in the Figure 1. The most important parameters for defining the working conditions of bearing and crankpin journal are:

- oil pressure distribution in the bearing,
- oil temperature in the bearing and
- bearing eccentricity.

According to the physical model in the Figure 1 for the calculation of these parameters some preconditions were introduced:

– oil is viscous Newtonian fluid with constant viscosity at the appropriate moment,
– the flow is laminar and non-inertial,
– journal and bearing surfaces are smooth and absolutely hard.

By using the equations about mass and momentum maintenance for the fluid flow of the bearing gap, with mentioned preconditions we can produce a well-known Reynolds equation of journal hydrodynamic behaviour in the following form:

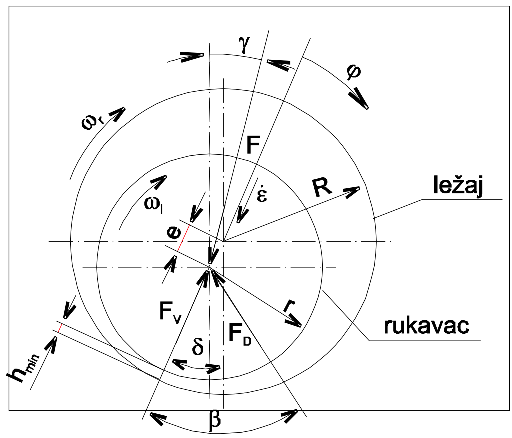

Figure 1: Physical model of crankpin-bearing with marked characteristic values

$$\frac{\partial}{\partial\varphi}\left[(1+\varepsilon\cos\varphi)^3\frac{\partial\Pi}{\partial\varphi}\right]+\left(\frac{2R}{B}\right)^2\frac{\partial}{\partial\bar{z}}\left[(1+\varepsilon\cos\varphi)^3\frac{\partial\Pi}{\partial\bar{z}}\right]=$$

$$=6\frac{\partial}{\partial\varphi}(1+\varepsilon\cos\varphi)+\frac{12}{\overline{\omega}}\frac{\partial}{\partial t}(1+\varepsilon\cos\varphi) \tag{1}$$

with: $\Pi=\frac{p\psi^2}{\eta\overline{\omega}}$ - dimensionless pressure,

p - oil pressure,

$\varepsilon=\frac{e}{R-r}$ - relative eccentricity,

$\overline{\omega}=\omega_r+\omega_l-2\frac{\partial\delta}{\partial t}$ - effective angular velocity,

$\bar{z}=\frac{z}{B/2}$ - dimensionless coordinate by bearing width,

$\psi=\frac{R-r}{R}$ - relative clearance,

D = 2R - bearing inner diameter

B - bearing width,

η - dynamic oil viscosity,

φ - angle of rotation,

δ - angular position of minimum oil film,

r - crankpin radius,

z - coordinate by bearing width.
t - time.

Based on the balance of external load (F force) and the pressure force, the equations define the eccentricity at the crankpin as:

$$\frac{d\varepsilon}{dt} = \dot{\varepsilon} = \frac{F\psi^2}{BD\eta S_{OV}}\left[\cos(\delta-\gamma) - \frac{\sin(\delta-\gamma)}{tg\beta}\right] \tag{2}$$

$$\frac{d\delta}{dt} = \frac{1}{2}\left[(\omega_r + \omega_l) - \frac{F\psi^2}{BD\eta S_{OD}}\frac{\sin(\delta-\gamma)}{\sin\beta}\right] \tag{3}$$

with:

$S_{OV} = \frac{F_V\psi^2}{BD\eta\dot{\varepsilon}}$ - Sommerfeld number for translation

$S_{OD} = \frac{F_D\psi^2}{BD\eta\overline{\omega}}$ - Sommerfeld number for rotation

F_V i F_D - resulting forces of bearing oil pressure in the translation and rotation.

Due to constant oil viscosity on one hand and unreliable boundary conditions on the other hand in these models we do not usually use energy equation in the original form for the calculation of current oil temperature. The appropriate approach in these models is setting balance at both bearing inlet and outlet, where the heat produced by friction in the bearing is released through oil (neglected heat release through bearing and crankpin journal). On the basis of this, the average bearing oil temperature can be defined as:

$$\vartheta_{sr} = \vartheta_1 + \frac{\mu F D\overline{\omega}}{4\dot{Q}\rho c_p} \tag{4}$$

with:

$\dot{Q}$ - oil flow through the bearing,

μ - friction coefficient is commonly taken for the conditions of hydrodynamic behaviour (μ) and the mixed friction conditions (μ_m),

c_p - specific oil heat,

ϑ_1 - oil temperature at the bearing inlet

ρ - oil density.

The correlation between friction in the hydrodynamic lubrication and mixed lubrication was used from [4] as:

$$\mu_m = \mu\left(2{,}25\frac{h_o}{h_{\min}} + 3\right) \tag{5}$$

with:

h_o- sum of maximum heights of rough surfaces on the bearing and crankpin,
h_{min} - minimum thickness of the oil film.

The loading of crankpin journal was calculated for the specific case of turbocharged diesel engine at the rotation speed of n = 2200 °/min and at maximum power. The results are shown in the polar diagram for loading with the coordinate system related to the connecting rod, the Figure 2.

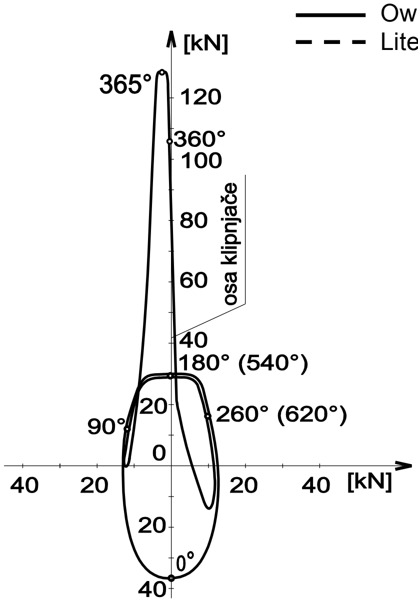

Figure 2: Polar diagram of loaded crankpin journal - coordinate system related to a connecting rod

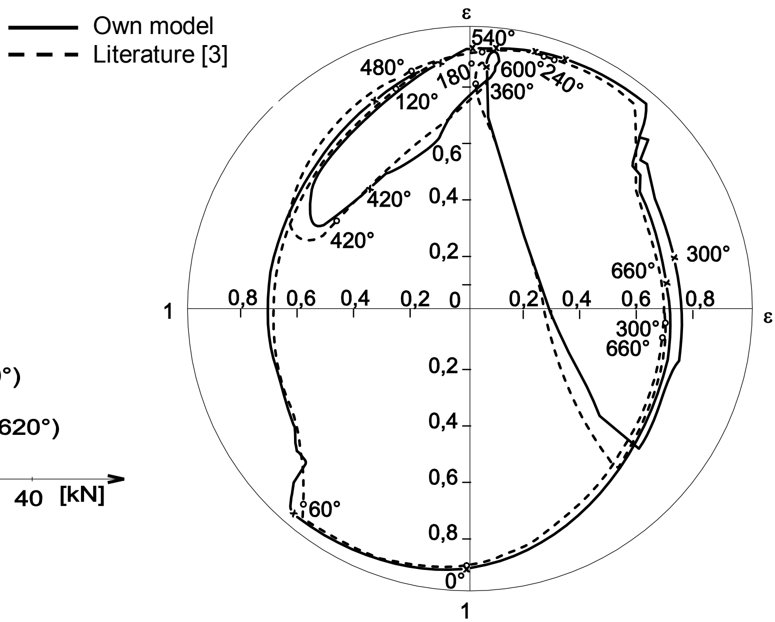

Figure 3: Polar diagram of crankpin journal eccentricity - coordinate system related to a connecting rod

The equation system (1), (2), (3) and (4) is composited and can be numerically solved as described in [2, 3]. Since the original conditions are unknown we assume them at the beginning of the calculation and at the end they are compared to the final results. If there is a big difference between presumed and calculated results in a given cycle, the inlet information are corrected. The procedure, known as iterative, is repeated many times until a satisfactory deviation between presumed values set prior to the calculation and the results at the end of the cycle is obtained.

The presented model is verified through polar diagram of eccentricity shown in the Figure 3 along with the results given in the literature [3], for engine rotation number *n* = 2200 o/min and kinematic oil viscosity at the bearing inlet ν = 23.6 mm^2/s.

Matching the results obtained by our own software with the results from [3] is satisfactory which led us to the conclusion that, by using this model, the influence of different working conditions on functional parameters of journal bearings can be analyzed.

3. Result analysis

A crankpin journal of the medium speed diesel engine as the most loaded bearing with relative clearance of $\psi_l = 0{,}6$ ‰ was analyzed. Also some results were obtained for a crankshaft of the same engine with relative clearance of $\psi_g = 0{,}967$ ‰. Lubrication oil used in the analyses is SAE 15W-40 with parameters of kinematic (ν) and dynamic (η) viscosity measured in Modriča oil refinery and shown in the Figure 4. By using the equation (4) and the Figure 4, the Figure 5 shows the comparative diagram of oil viscosity (ν_{sr}) in the bearing, defined on the basis of average temperature (ϑ_{sr}) of bearing oil, and oil viscosity at the bearing inlet (ν) for different relative bearing clearances (ψ).

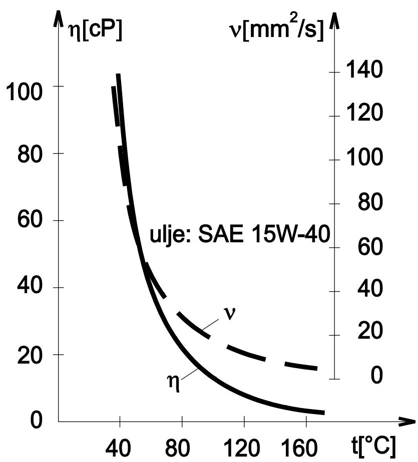

Figure 4: Diagram of kinematic (ν) and dynamic (η) oil viscosity in the function of oil temperature (t) under the pressure of *p* ≈ 1 bar

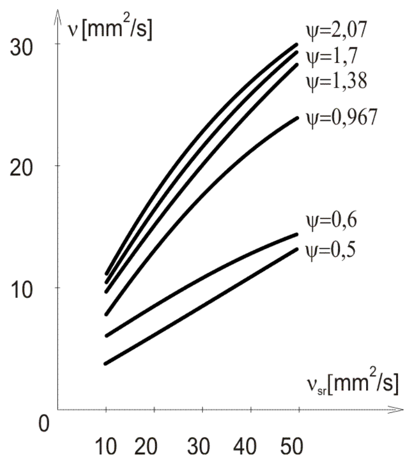

Figure 5: Relation between oil viscosity at the bearing inlet (ν) and medium bearing oil viscosity (ν_{sr}) for different clearances (ψ)

Due to real influence of oil pressure on viscosity, and by using the measurement results in the Figure 4, correlation for dynamic oil viscosity SAE 15W-40 was analyzed in the form of:

$$\eta = ae^{\left(\frac{b}{c+\vartheta}+\frac{d}{e+\vartheta}\cdot p_{sr}\right)} \quad [Pa\ s] \tag{6}$$

with $\vartheta[°C]$ - as oil temperature,

p_{sr} [Pa] - as medium oil pressure,

$a = 1{,}7\cdot 10^{-7}; b = 720; c = 71; d = 0{,}2\cdot 10^{-5}; e = 54$ - as constant values,

With the referent literature [1] as the basis for the correlation (6).

The analysis of the influence of oil viscosity (ν) on the bearing inlet as independently flexible value, which can be regulated in the engine by cooling the oil, on the functional parameters of journal bearings was also performed. Relative bearing clearance (ψ) was also varied and analyzed by its influence on minimum oil film thickness (h_{min}). The results of values calculation of minimum oil film thickness in the function of oil viscosity at the bearing inlet and relative clearance are shown in the Figures 6 and 7 for the bearing of crankpin journal, and the Figures 8 and 9 show the crankshaft bearing.

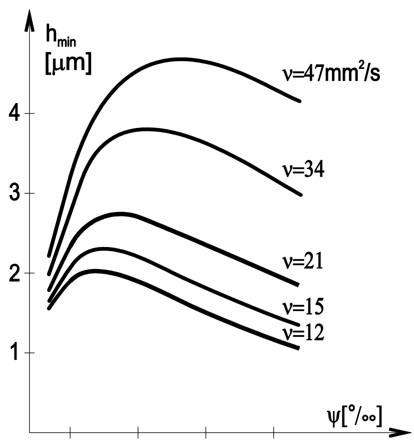

Figure 6: Change in value of h_{min} as clearance (ψ) for different viscosity parameters (ν) for journal bearing of an engine (n = 2200 o/min – max power)

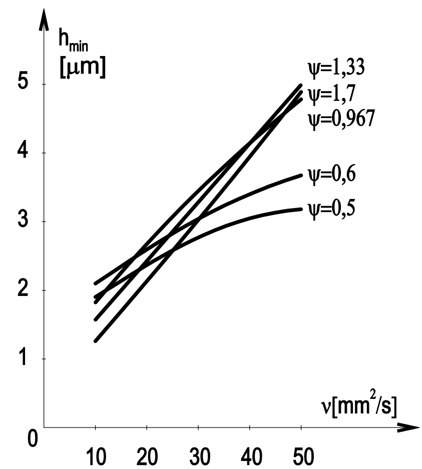

Figure 7: Change in value of h_{min} as viscosity (ν) for different clearance parameters (ψ) for journal bearing of an engine (n = 2200 o/min – max power)

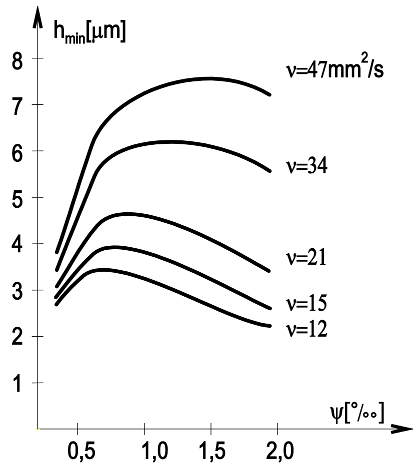

Figure 8: Change in value of $h_{\min}$ as clearance (ψ) for different viscosity parameters (ν) for main bearing of an engine (n = 2200 o/min – max power)

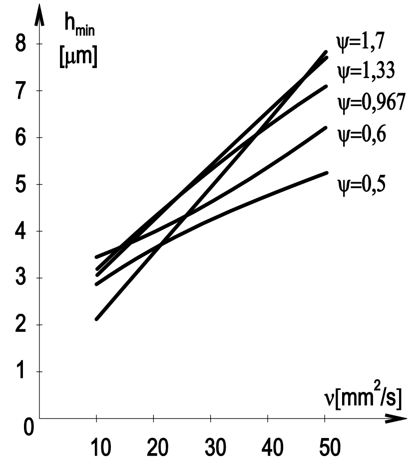

Figure 9: Change in value of $h_{\min}$ as viscosity (ν) for different clearance parameters (ψ) for main bearing of an engine (n = 2200 o/min – max power)

There are obvious differences ($h_{\min}$) between crankpin journals and crankshaft journal mostly because of the load, and the bends are similar in character. Maximum values ($h_{\min}$) are set according to the change of clearance (ψ) and viscosity (ν).

All these results show a decrease in values of minimum thickness of oil film ($h_{\min}$) with reduced viscosity or, in other words, with increased oil temperature at the bearing. Apart from oil flow characteristics in front of the bearing (flow, temperature…) oil viscosity also depends on regimes of speed and engine load as well as physical oil properties.

Considering the bearing oil inlet temperature, the oil inlet viscosity is within the limits of ν=16-25 mm^2/s. The bearing oil viscosity depends on a clearance between bearing and crankpin journal (journal bearing in the Figure 5). This parameter directly influences the minimal bearing oil layer ($h_{\min}$), the portion of mixed friction which causes the increase of bearing oil temperature, the decrease of viscosity and the increase of mechanical losses. This indicates the fact that the lubrication oil viscosity in the engine is very important parameter from the point of defining working conditions as well as from the point of regulating mechanical losses in an engine.

4. Conclusion

The paper presents a model for the calculation of functional values of bearing and crankpin journal based on given working conditions of the bearing system. The model is verified by the experiment results which confirm their practical application for the analysis of functional parameters of bearing and crankpin journal.

The results of measuring the influence of oil viscosity on functional values show all the complexity of this problem. Reduced bearing oil viscosity directly influences the reducing of minimum thickness of oil film (h_{min}) and the reducing of movement resistance of journal in the bearing. It results in increasing the portion of mixed friction due to the decrease in h_{min}, rough surfaces on the bearing and its deformations which also contribute to significant increase of mechanical losses. Therefore, it is very important to select lubrication oils with appropriate viscosity quality of stable operating temperature in the engine.

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UDK	ključne riječi	key words
621.822.573	hidrodinamički klizni ležaj	hydrodynamic plain bearing
.004.14	gledište funkcionalnih svojstava i konstrukcije	functional performance and design viewpoint
519.283.001.57	iterativni matematički model	iterative mathematical model
621.436.75	predpunjeni dizelov motor	supercharged diesel engine
621.4.016.4	radna temperatura i opterećenje motora	engine operating temperature and load
532.135	reološka svojstva motornog ulja	engine oil rheologic properties
621.891.275	čvrstoća mazivog sloja	lubricating film strength

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