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ANALYSIS OF BEARING CLEARANCE AND WIDTH OF AN OIL GROOVE ON THE CHARACTERISTIC OF FLUID FILM IN HYDRODYNAMIC MAIN CRANKSHAFT BEARINGS

ANALIZA WPŁYWU WARTOŚCI LUZU ŁOŻYSKOWEGO ORAZ SZEROKOŚCI ROWKA SMARUJĄCEGO NA CHARAKTERYSTYKI FILMU OLEJOWEGO HYDRODYNAMICZNYCH ŁOŻYSK GŁÓWNYCH WAŁU KORBOWEGO

Key words: hydrodynamic bearing, oil groove, bearing clearance, shaft, bevelling of the shaft, computer simulation.

Abstract Paper represents an analysis of the influence of oil groove width and bearing clearance on the characteristics of oil film in a hydrodynamic journal main crankshaft bearing. Analysis was performed to define the influence of bearing clearance and dimensions of oil grooves on static characteristics and manufacturability of half-shelf bearings in industrial high-volume production. Computer simulations were made using an ARTbear program, which was developed in Department of Machine Construction and Vehicle Design in Gdańsk Technical University. The analysis allowed defining the correlation between groove geometry and load capacity and working conditions of bearings.
Potential opportunities of possible shortening of production time in high-volume production were also indicated.

Słowa kluczowe: łożysko hydrodynamiczne, rowek olejowy, luz łożyskowy, ukosowanie czopa, symulacja komputerowa.

Streszczenie Przedstawiono analizę wpływu szerokości rowka olejowego oraz luzu łożyskowego na charakterystykę filmu olejowego w hydrodynamicznych łożyskach poprzecznych wału korbowego. Analizę przeprowadzono w celu określenia wpływu luzu łożyskowego oraz wymiarów rowka na charakterystyki statyczne łożyska oraz technologiczność wykonywania panewek w przemysłowej produkcji wielkoseryjnej. Symulacje komputerowe wykonano, wykorzystując autorski program komputerowy ARTbear opracowany w Katedrze Konstrukcji Maszyn i Pojazdów Politechniki Gdańskiej. Przeprowadzone analizy pozwoliły określić zależności pomiędzy geometrią rowka smarującego a nośnością i warunkami pracy łożyska. Odniesiono się również do potencjalnych możliwości skrócenia czasu operacji wykonywania rowka w trakcie produkcji seryjnej panewek.

INTRODUCTION

Half-shell slide bearings have been produced and used in road vehicles for more than one hundred years and their reliable work is one of the key factors of proper functioning of complete vehicle. Despite very long time of the existence of these sliding bearings in internal combustion engines, it is still developing rapidly in search of new bearing materials and layers, more efficient processes, new construction solutions, etc. [**L. 1–10**]

This is due to the high demands of automotive market, i.e. required reliability and working in highly loaded applications with low price production cost. Modern half-shell bearings consist of a steel backing, a layer of bearing material, and additional sliding layers, which are helping in bearing performance under mixed lubrication.

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The main crankshaft bearing is supporting the crankshaft and consist of two halves – lower which is full (without groove) and upper with an oil groove. During work, the upper half is less loaded than the lower one, because it is carrying loads not associated with the working stroke of cylinder during combustion. One of the key elements of upper bearing is the oil groove, which has the main function of delivering oil to the lower main bearing of conrod bearings [**L. 11**]. In modern bearings produced by the automotive industry, there are many different solutions of groove construction.

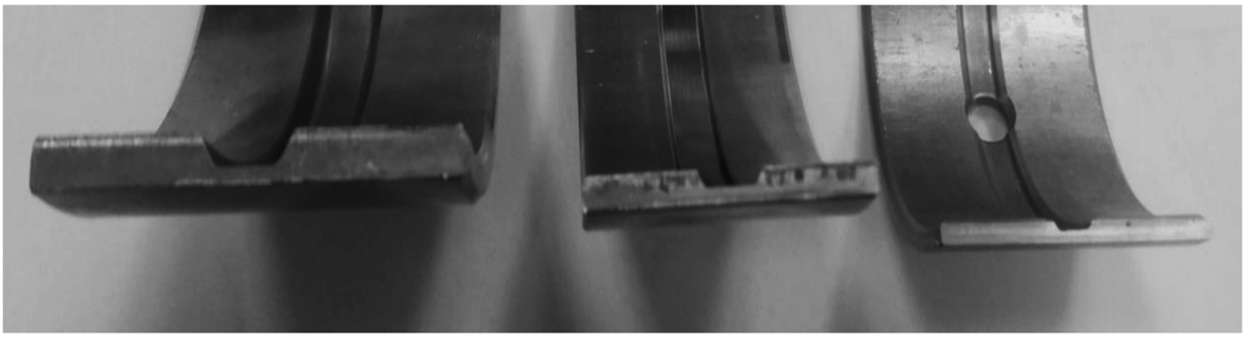

Fig. 1. Oil grooves with different types of groove geometry in modern automotive slide bearings (from author archives)
Rys. 1. Widok rowków olejowych o odmiennej geometrii stosowanych w nowoczesnych łożyskach ślizgowych silnika spalinowego (z archiwum autora)

Oil grooves in bearings (shown on **Fig. 1**) are commonly made by milling (mostly by profile milling). Grooves are usually relatively wide (sometimes more than 20% of bearing width). During machining of the groove, a lot of material must be removed, which often makes this machining stage the longest operation of the whole machining process, which, assuming that parts are produced in hundreds thousands, is an important factor impacting the whole production capability and costs.

Influence of the bearing dimensions on its tribological performance was investigated by few researchers [**L. 12–15**], but they did not take into account angular misalignment of shaft and some real condition of bearing work and geometry of the bearing itself.

Main reason for making the analysis was to define exact influence of groove width in static film characteristics for different values of bearing clearance (which are result of machining tolerances) and with the possibility of shaft angular misalignment (which are result of shaft deflection of the shaft and misalignment deviations).

BEARING FOR ANALYSIS AND CALCULATION METHOD

For calculation representative half-shelf bearing was chosen that has internal diameter 125 millimetres with a complete groove (180 degrees which is shown in **Fig. 2**). Bearing clearance and their tolerances was taken from producers catalogues [**L. 11, 16–17**]. Tolerances are defined by the precision of machining shaft, housing, and bearing.

Fig. 2. Axonometric view of half-shell bearing with oil groove of bearing used in calculations
Rys. 2. Poglądowy rzut aksjonometrycznypół panewki z rowkiem analizowanego łożyska

Calculations of bearing clearance influence on characteristics of hydrodynamic film were made for three clearances: minimal, maximal, and medium (which was mean value from maximal and minimal clearance). Analysis of groove influence was made for medium clearance and different grooves with angular misalignment of shaft. For the calculation process, the physical properties of oil were taken as producer recommends.

The decision was made that calculations will be performed for static load but with a very broad range of loads. This reflects the actual working conditions of bearings when load is changing cyclically, depending on working parameters and engine constructions. Analysis was made for 1500 revolution per minute (engine of large displacement with spontaneous ignition). Angular misalignment of a shaft was set as 10% of bearing clearance.

Main parameters of investigated bearing are shown in **Table 1**.

Table 1. Parameters of bearing
Tabela 1. Parametry badanej panewki

Internal diameter	125 mm
Bearing Wall thickness	3 mm
L/D correlation	0.28
Lubricant	Oil SAE 30
Supply oil temperature	95°
Bearing clearance minimum	0.00055
Bearing clearance maximum	0.00105
Bearing clearance mean	0.0008
Rotational speed	1500 RPM

The calculations were performed using ARTbear computer software, which has been developed over many years at the Machine Design and Vehicles Department [**L. 18**]. It employs an adiabatic model of the hydrodynamic film and allows for determination of all important parameters of a hydrodynamic bearing, such as pressure and film thickness distribution, lubricant expenditure, stiffness, and damping coefficients, journal position, etc. The software enables the simulation of cylindrical and multi-surface bearings of complex axial groove geometry with accommodation of the influence of the shaft's angular misalignment and bearing shell imperfections. The adiabatic model of lubrication film is not taking into consideration heat transfer through shaft [**L. 18**] which should be remembered during results analysis.

INFLUENCE OD GROOVE WIDTH ON OIL FILM CHARACTERISTICS

Analysis of loss of bearing load capacity in function of its width

Bearings with a partial groove (less than 180 degree) were not investigated, because literature studies [**L. 11**] shows clearly that a full groove (groove cut through complete angular area of the bearing) is the optimal solution in the case of power losses in the engine and minimal oil film thickness (shown on **Fig. 3**).

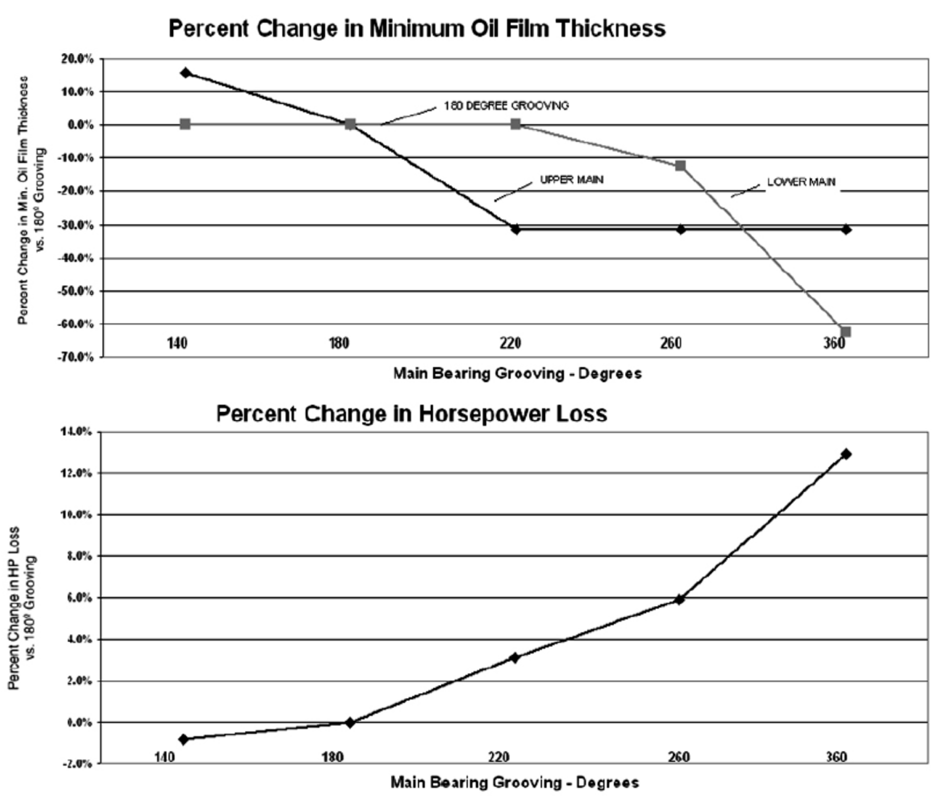

Fig. 3. Changes of oil film thickness and power losses in internal combustion engine (plot taken from MAHLE catalogue [L. 11])
Rys. 3. Zmiany grubości filmu olejowego oraz straty mocy w silniku spalinowym (wykresy zaczerpnięte z katalogu producenta MAHLE [**L. 11**])

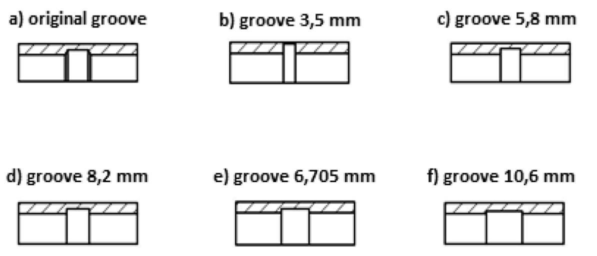

Fig. 4. Different variants of geometry of analysed oil groove (shown in cross section through bearing wall)

Rys. 4. Różne warianty geometrii analizowanych rowków olejowych (rowki pokazane w przekroju przez ściankę panwi)

Firstly, the groove cross area was calculated to propose different groove shapes and dimensions, which have the same area as the original groove (which is mandatory to keep oil flow rate). Investigated types of grooves are shown in **Fig. 4**. For some examples, like a groove of 3.5 mm width, the groove depth is almost equal to total wall thickness of the bearing, which obviously makes this solution impossible to use in any practical application, and the calculation was made only in purpose of comparison to other variants.

Calculations were performed with bearing load and pressure controlled for the criteria of minimal oil film thickness and maximum temperature. The original groove was trapezoidal shaped with sides inclined by 30 degrees to the symmetry axis of the groove (Type a) presented on **Fig. 4**). A short calculation of cross-sectional area of oil groove showed that the replacement of the original trapezoidal groove by square one (advantages of such solution was described in [**L. 11**]) lowers loads acting on bearings by more than 3% with the same groove depth that corresponds to load capacity increase of 0,8 kN for eccentricity 0,9.

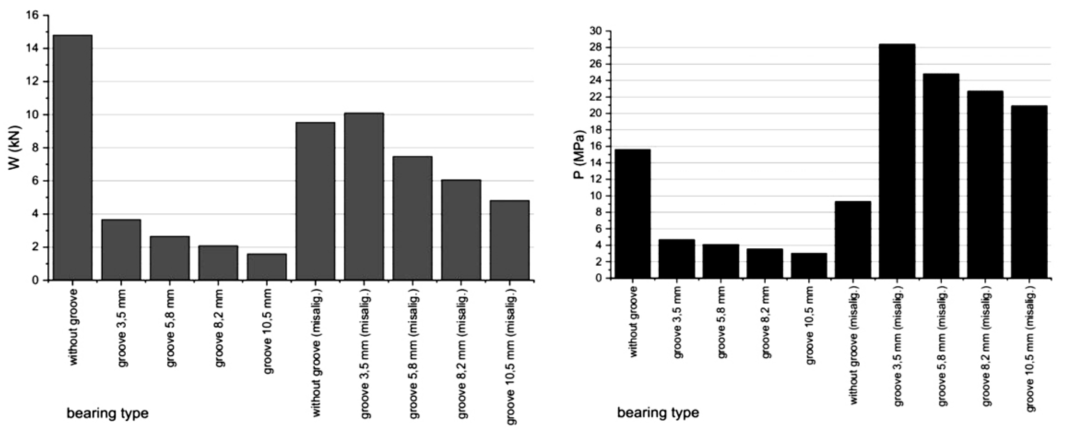

Fig. 5. Load capacity and maximal pressure in film in function of groove width and bevelling of the shaft for medium bearing clearance 0.0008 and eccentricity 0.9

Rys. 5. Nośność i ciśnienia maksymalne w łożysku w zależności od szerokości rowka oraz ukosowania czopa przy bezwzględnym luzie łożyskowym 0,0008 oraz mimośrodowości względnej czopa 0,9

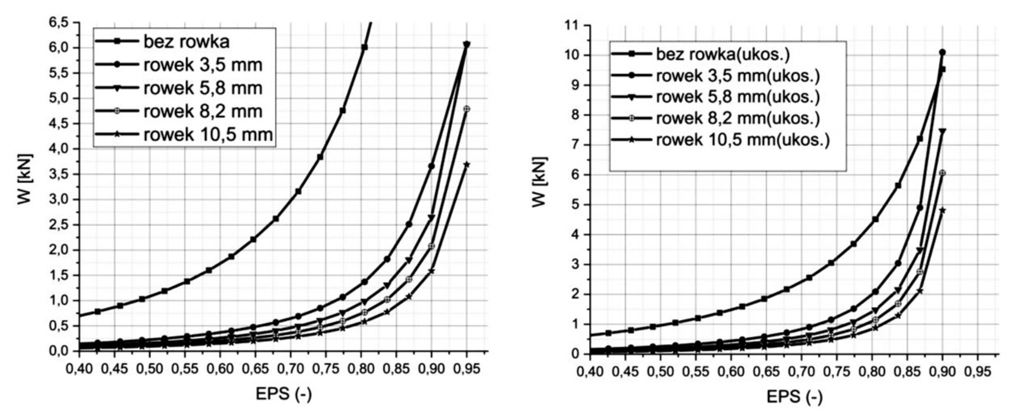

Fig. 6. Load capacity in function of eccentricity for different groove widths. Left side: parallel shaft; right side: with angular misalignment of a shaft

Rys. 6. Nośność łożyska funkcji mimośrodowości względnej dla różnych wariantów szerokości rowka. Z lewej czop równoległy, z prawej z ukosowaniem wału

Calculation results for mean bearing clearance and different types of groove geometry are shown on **Figs 6–9**.

A thin groove (3.5 mm) for the analysed eccentricity of 0.9 cause almost a 4-fold loss of bearing load capacity (for eccentricity 0.95, the load capacity without groove is more than 5 times higher). The relation between groove width and load capacity is not linear, and the difference in loss of bearing load capacity is decreasing with the width of the groove. For example, between grooves of 3.5 and 5.8 mm (difference in width 2.3 mm), the difference in load is 1 kN; however, for grooves of 8.2 and 10.5 mm (difference in width also 2.3 mm), the difference in load is only 0.49 kN.

Temperatures in the oil film for half-shell bearings without groove are much lower than for grooved ones. An especially large increase in temperature is seen for bearings with grooves that work under angular misalignment of a shaft. For such working parameters, film temperatures exceed the maximum allowable temperatures quickly. Taking the temperature of 140 degree as a limit of safe operation, it is clear that, for bearings without a groove, shaft angular misalignment does not change the maximal temperature by much. However, temperatures in the oil film increases rapidly for grooved bearings and exceed the criterion of maximum temperature quickly. The load capacity of bearing with a groove of 3.5 mm with an angular misalignment of the shaft is 1.5 kN; however, for a bearing with a groove of 10.5 mm has a load capacity of less than 0,6 kN.

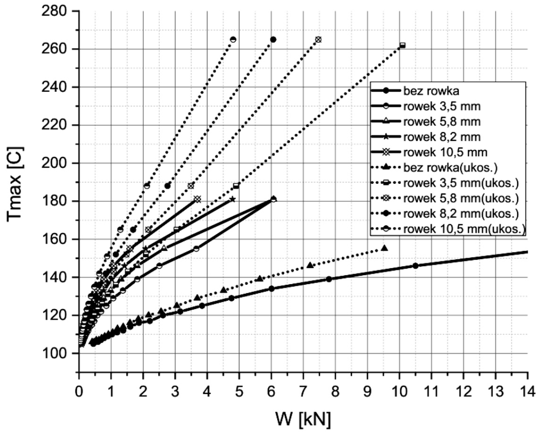

Fig. 7. Temperature in oil film for different widths of groove with angular misalignment of a shaft

Rys. 7. Rozkład temperatury w funkcji obciążenia łożyska-dla różnych wariantów szerokości z uwzględnieniem ukosowania czopa wału

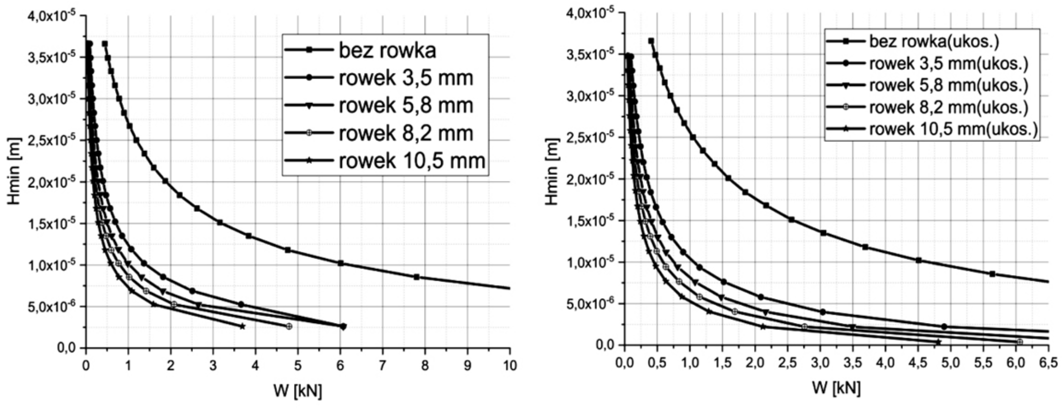

Fig. 8. Minimal film thickness in function of load for different types of grooves and angular misalignment of the shaft

Rys. 8. Minimalna grubość filmu w funkcji obciążenia dla różnych wariantów szerokości kanałka i ukosowania wału

In real conditions, the shaft is working as a heat recuperator [**L. 19**] and equalizes the temperature distribution in really thin oil films (which are typical for such bearings). Additionally, heat is transferred through the housing and lowers the maximum temperature of the oil. Nevertheless, rapid increase in temperature as a function of load shows that the presence of an oil groove is critical if shaft is bevelled, which always exist to some extent in modern combustion engines caused by heat deflections, and it is impossible to avoid geometry imperfections [**L. 20–22**].

For all analysed grooves, minimum film thickness criteria was fulfilled practically in the whole range of analyses (minimal film thickness was set on level of 3–5 micrometres because of the surface finish of elements and real values of film thickness found in literature [**L. 20–22**] in these types of bearings and using the author's best knowledge). For a bearing working with angular misalignment of a shaft, the criteria of minimal film thickness of oil film failed in the range of loads of 2.2 to 3.5 kN, depending of the groove width.

Analysis of possible shortening of production process time

A separately analysed issue was the possibility of shortening production time during groove machining. Recommended working parameters to provide the required quality of surface finish and dimensional stability defines feed per tooth of the cutting tool. Additionally, the producer of cutting tools defines optimal cutting speeds for milling process. This causes that tool speed is closely correlated with its rotational speed. Because of that fact, groove depths have a critical influence on complete time of the machining cycle, which is about 65% of the total cycle time (from loading to unloading the part).

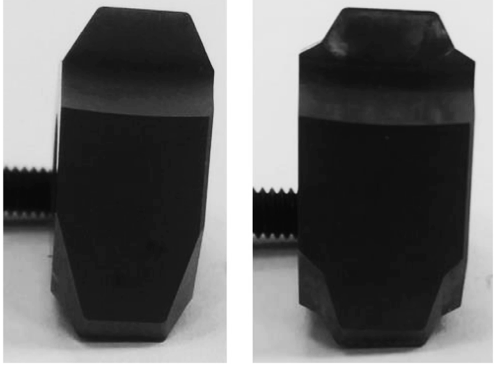

Fig. 9. Example of profile cutter which is used for milling the oil groove in bearings – left side: new part; right side: used one with wear marks (from author archives)

Rys. 9. Przykładowy nóż kształtowy służący do frezowania kanałka olejowego w łożysku – po lewej nowy, po prawej noszący ślady użycia (z archiwum autora)

Increasing the width of the profile tool (An example of the cutting tool is shown in **Fig. 9**) that is machining the groove (machine has enough power margin), it is possible to keep same feed which is defined by technological parameters but with better material removing efficiency proportional to the increase of cutter width.

The potential increase of groove width in relation to this decrease of groove depth causes the essential shortening of production time cycle. Shortening cycle time by 40% (difference in groove width between 5.8 and 8.2 mm is 41%) reduces the complete cycle time by 28% with a simultaneous loss of load capacity by 27% (for eccentricity on level of 0.9).

Taking into consideration the fact that the main crankshaft upper bearing is not heavily loaded and its main function is oil feeding for more loaded bearings, the loss of load capacity can be negligible for the proper working of the bearing, but shortening of production time by 27% lowers the production costs. Additionally, this working cycle will increase the service life of cutting tools proportionally to cycle time shortening. Obviously, a wider cutter will be slightly more expensive, but the additional cost is negligible compared to production profits and the longer service life of tool.

This short analysis was included to shown one of the main reason for which the analysis was performed and shows potential possibilities that will be investigated more broadly in the future.

ANALYSIS OF BEARING INFLUENCE OF BEARING CLEARANCE ON CHARACTERISTICSOF OIL FILM

Bearing clearance as well as oil viscosity and rotational speed are key factors that influence hydrodynamic bearing performance. Tolerances defined by producers for shaft diameters, housings, and bearings provide information that theoretical bearing clearance is in range between 0.00055 to 0.00105. Twice the difference between maximal and minimal clearance makes a huge difference in the load capacity of bearings.

Fig. 10. Pressure distribution in bearing: left side: minimal clearance, right side: maximal clearance

Rys. 10. Dwa pierwsze rysunki od lewej: rozkłady ciśnienia dla panwi o luzie minimalnym; dwa rysunki od prawej: rozkłady ciśnienie dla panewki z luzem maksymalnym

The influence of bearing clearance on film characteristics is shown in **Figures 10–13**. The calculation was performed for three different clearances for bearings without grooves and with 8.2 mm wide grooves.

Fig. 11. Pressure distribution in bearing for medium clearance (drawings on the right with shaft misalignment)
Rys. 11. Rozkłady ciśnień dla panewek z luzem średnim (rysunki po prawej z ukosowaniem wału)

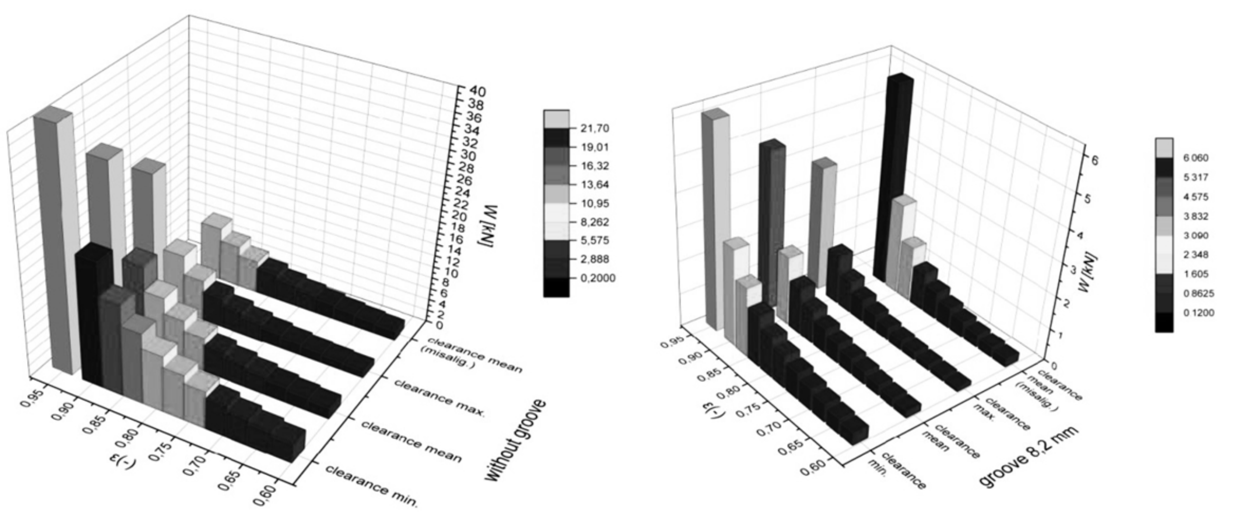

Fig. 12. Load capacity as a function of eccentricity in different clearances and with angular misalignment of shaft left side: bearing without groove, right side: bearing with 8.2 mm groove)
Rys. 12. Nośność w funkcji mimośrodowości względnej w zależności od luzu i ukosowania wału (z lewej strony dla panewki bez rowka, z prawej strony dla panewki z rowkiem o szerokości 8,2 mm)

Automotive manufacturers use dimension selections that provide optimal clearance for all bearings. However, during service repairs, exact selections of shafts, housings, and bearings are unknown. Replacement in a vehicle under such circumstances must carefully use proper bearing clearance selection [**L. 11**]. Because of the critical influence of bearing clearance on the characteristic of hydrodynamic film, this paper was extended with a short analysis of bearing clearance influence on its performance.

Results show that, for eccentricity 0.9 in a bearing without a groove, and maximal clearance load capacity is about 8.5 kN. For a 8.2 mm grooved bearing with minimum clearance, the load capacity is somewhere around 2.9 kN, which is almost 35% of load capacity for a bearing without a groove. Comparing two situations where both analysed bearings will have same clearances, the load capacity of the grooved part for medium clearance is only about 12% of the full bearing (without groove). We can see clearly that, in individual highly demanding applications where manufacturer can use product selections from wide range of bearings (which is impossible in mass production because of high cost), there are possibilities for bearing load capacity

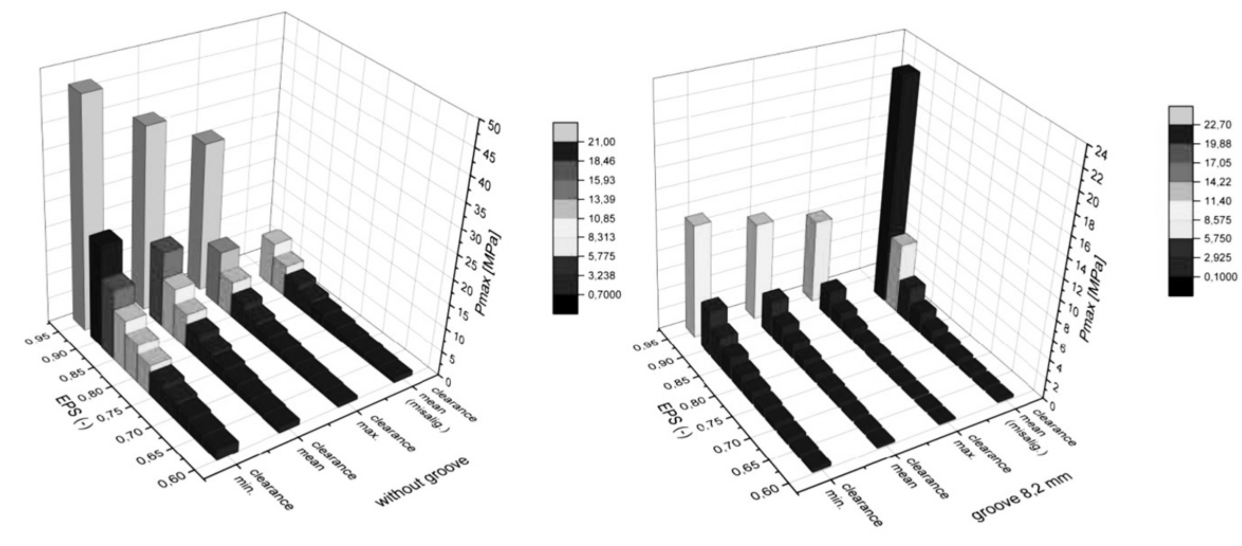

Fig. 13. Maximum pressure in film as a function of eccentricity for different bearing clearances and angular misalignment of a shaft (left side: bearing without groove, right side: bearing with 8.2 mm groove)

Rys. 13. Rozkład max. ciśnienia w filmie w funkcji mimośrodowości względnej w zależności od luzu łożyskowego i ukosowania wału (lewa strona – panewka bez rowka, prawa – z rowkiem o szerokości 8,2 mm)

manipulation. With angular misalignment of a shaft, the load capacity is theoretically high; however, in practical application, it will cause quick wear on the edges, and previous calculations show that it causes high temperatures in the oil film, which results in a narrow area in which bearing can be used.

The calculations and analysis allow the formation of the following conclusions:

Replacement of typical trapezoidal shaped groove with square version can lower the calculated pressure by more than 3% and increase load capacity by 0.8 kN (for eccentricity 0.9).

The relationship between groove geometry and load capacity is not linear, and difference in the loss of bearing load capacity is decreasing with width of the groove.

Temperatures in oil film for bearings with a groove and a bevelled shaft increase rapidly and quickly exceed the safe operation range of the bearing.

Decreasing the groove width can significantly increase bearing load capacity and allows the bearing's use in a wider range of eccentricity and provides better performance when shaft is bevelled, but this can badly influence on fatigue strength of the bearing, which is not part of the scope of this analysis. Wider analysis of this topic is needed.

Analysis of the influence of bearing clearance shows clearly that there is possibility of a clearance selection that can significantly help to compensate the load capacity decrease caused by the presence of an oil groove.

CONCLUSIONS

The performed short calculations of the influence of groove geometry and bearing clearances show that there are solutions for load capacity increase with the preservation of oil flow through the groove.

There are also potential possibilities of lowering production costs of bearings through the use of a wider groove and lower depth, which can essentially shorten production time. A change of groove width must be preceded by precise analysis of working conditions of specific bearings in specific engines.

On the other hand, in heavily loaded engines, a narrower groove provides a significant increase in load capacity and its resistance to shaft bevelling. This increases the production time of the bearing; however, for engines produced in a low volume series, it can be sometimes negligible. Reduction of groove width and the increase of groove depth can cause other negative effects on bearing performance, which are hard to define in theoretical analysis (for example, lower fatigue strength because of a local notch). This requires further more complicated CFD calculations and the use of a test rig that simulates work conditions of crankshaft bearings.

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