Original Research Article

# Prediction of Cavitation Failure in Crankpin Bearings

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**Abstract**

Cavitation failure is a common failure in bearing shells. Due to the generation and immediate collapse of small gas bubbles, causing high-pressure pulses, the bearing surface is being locally damaged. Cavitation failure is also observed in IC engines due to highly dynamic loading, oscillation of pins, the turbulence of oil flow, and other factors. In this paper, cavitation failure in the crankpin bearing of an IC engine is studied. In order to calculate the bearing lubrication characteristic such as oil fill ratio and maximum oil film pressure, the Elasto-Hydrodynamic Lubrication (EHL) method to consider the effect of stiffness of the bearing shell housing in the model is utilized that incorporates mass conserving algorithms. In order to investigate the effect of some design parameters, such as clearance height between shaft and bearing shell, oil supply temperature and pressure, and oil bore position, on the cavitation failure, a parametric study was also done. The results showed that the cavitation failure in crankpin bearing is not critical and it is slight.

**Keywords:** Cavitation Failure; Crankpin Bearing; Elasto-Hydrodynamic Lubrication; IC engine.

## 1. Nomenclature

| | |
|---|---|
| $C_R$ | Bearing radial clearance |
| $e_x$ | Eccentricity in the X direction |
| $e_y$ | Eccentricity in the Y direction |
| $Z$ | Axial position of bearing nodes |
| $\alpha$ | Pressure viscosity coefficient |
| $\alpha_x$ | Misalignment around X-axis |
| $\alpha_y$ | Misalignment around Y-axis |
| $\beta$ | Circumferential coordinate of bearing |
| $\delta$ | Radial deformation of the bearing surface |
| $\eta_0$ | Viscosity at ambient pressure |
| $\eta$ | Viscosity |
| $\theta$ | Fill ratio |
| $\sigma_s$ | Composite surface asperity height (r.m.s) |
| $v$ | Poisson ratio |

## 2. Introduction

Engine designers are increasingly using more advanced simulation techniques to reduce design time and costs and at the same time to improve the accuracy of the results to limit the number of validation tests required. In recent years, the demand for higher specific power has enforced a higher bearing load in IC engines. On the other hand, due to competition in the market, engine parts should be durable and low damage should be occurred in critical parts [1]. Cavitation is a common failure in engine-bearing shells, therefore the assessment of that and introduction of the methods in order to prevent this failure are important. Connecting rod of an IC engine is subject to high firing and inertial loading. Design limitations for weight reduction of the connecting rod have been caused to lighter and thus more flexible bearing support structures, especially in crankpin bearing. This would highlight the importance of elastic deflection of bearing support and its consequent effects on clearance height between pin and shell [2]. In addition, this will have significant effects on the lubricating performance of bearings. Cavitation can take place if the pressure in a liquid is reduced sufficiently that causing the formation of vapor-filled voids or bubbles. When the liquid that contains bubbles is subsequently subjected to higher hydrostatic pressure, the bubbles collapse suddenly. Bubble collapse causes cyclic pressure that can result in surface failure [3]. Figure 1 depicts cavitation failure in an engine bearing. Dowson and Taylor [4] believed that most of these have been concerned with aqueous systems, and the erosive damage to surfaces bounding a cavitating flow has received much attention. Components that have proved susceptible to cavitation erosion damage include pump impellers, valves, marine propellers, pipes, and cylinder liners. Karimaei et al. [5] studied wear and cavitation damage in big end bearings of two different designs of connecting rods structure. In order to calculate the bearing lubrication characteristic such as minimum oil film thickness and maximum oil film pressure, EHD was

applied. They showed that housing structure has an effect on the failure and should be considered in calculations.
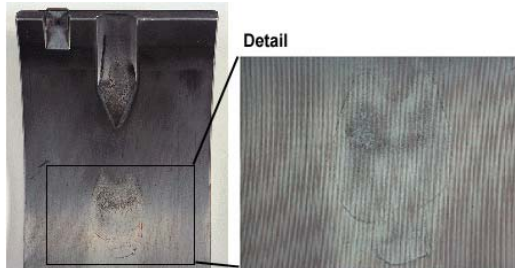


**Figure 1.** Cavitation failure in bearing [3]

Flageul et al. [6] proposed a method to predict cavitation damage from cavitating flow simulations. For this purpose, a numerical process coupling cavitating flow simulations and erosion models was developed and applied to a 2D hydrofoil test. Two different erosion models were used and compared. Based on these models, two aggressiveness parameters were introduced and evaluated using CFD results. The simulated qualitative influence of flow velocity, sigma value, and gas content on cavitation damage agreed well with experimental observations. Significant discrepancies between simulated and experimental results were found for upstream unsteady cavitating flows. The CFD tool will have to be enhanced to improve simulation in this zone. To meet high-reliability bearings, Ligier and Noel [7] reviewed several technical solutions. Damage risks relating to customer uses were also presented to check whether these risks are negligible or not. Nikolakopoulos et al. [8] believe that accurate simulation and modeling in machine components are dependent on realistic lubricant rheology and lubricant properties, where especially the latter may change as the machine ages. The key role of the action of pressure and temperature in engine oils' aging is described. The aim of their work was to evaluate changes due to temperature and pressure in the viscosity of engine oil over its lifetime and to perform an uncertainty analysis of the measured values.

Ferretti [9] This activity deals with the problem of simulating lubricated contacts, mainly focusing on the development of a proper approach to the analysis of the asperity contact problem. The hydrodynamic problem will be tackled by adopting a mass-conserving algorithm, which uses a linear complementary formulation of the Reynolds equation to calculate the hydrodynamic pressure distribution within the lubricant film and to identify the cavitated region at a given oil film height. Allmaier and Sander [10] investigated the rotational dynamics and lubrication of the piston pin of a Gasoline engine in their work. The clearance plays an essential role in the lubrication and dynamics of the piston pin. A thermoelastic simulation is conducted for the piston at the full-load firing operation and then the calculated temperature field of the piston and the piston-pin clearance are used in the simulation of the piston-pin journal bearings. It is found that the piston pin rotates mostly at very slow rotational speeds and even changes its rotational direction between different operating conditions. Several influencing effects on this dynamic Behaviour are investigated to see how the lubrication of this crucial part can be improved.

In the literature, the works that have been done in the field of cavitation assessment of reciprocating engine bearings are really few and the majority of assessments are qualitative and not quantitative. In this study, cavitation failure assessment in crankpin bearing using the Elasto-Hydrodynamic Lubrication (EHL) model that incorporates mass conserving algorithms is studied. Therefore, a detailed discussion on cavitation failure in the crankpin bearing of an IC engine is performed. Also, the effects of some design parameters on the occurrence of such a failure are investigated. These parameters include clearance height between the shaft and bearing shell, oil supply temperature and pressure, and oil bore position.

## 3. Elasto-Hydrodynamic Lubrication Model

Elasto Hydrodynamic Lubrication (EHL) is a lubrication regime (a type of hydrodynamic lubrication (HL)) in which significant elastic deformation of the surfaces takes place and it considerably alters the shape and thickness of the lubricant film in the contact. EHL method is based on Reynold´s equation (1) [5] solving in the bearing surface. Equation (1) includes a mass conserving cavitation model, which is reflected by the additional variable, clearance fill ratio $\theta$. For $\theta = 1$, the equation will become the ordinary Reynold's equation. Reynolds equation is solved for pressure p in the lubrication region and filling factor $\theta$ in the cavitation region. The filling factor serves to model the cavitation effects and is defined as the fraction of volume filled with oil to the total volume [11]. The fill factor equals one indicates the gap is filled with oil and zero indicates an empty gap.

$$\frac{\partial}{\partial x}\left(\frac{\theta h^3}{12\eta}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{\theta h^3}{12\eta}\frac{\partial p}{\partial z}\right) = \left(\frac{u_1+u_2}{2}\right)\frac{\partial(\theta h)}{\partial x} + \frac{\partial(\theta h)}{\partial t} \quad (1)$$

For an EHL analysis, the effect of elastic displacements of the bearing surface has to be included. The oil film thickness $h$ is expressed by eq. (2) in consideration of the initial geometrical clearance, misalignment of the shaft, and elastic deformation.

$$h(\beta,z) = C_R - (e_x + \alpha_y Z)cos\beta - (e_y + \alpha_x Z)sin\beta + \delta(\beta,z) \tag{2}$$

Radial deformation of the bearing surface is obtained from the nodal displacements of the bearing surface along the radial and circumferential axes. The nodal displacements of the bearing surface are determined by solving the equations of motion for the condensed bearing structure. In the current work, Barus' equation is utilized in order to define the oil viscosity.

$$\eta = \eta_0 e^{\alpha p} \tag{3}$$

where, $\eta_0$ is the viscosity at ambient pressure and $\alpha$ is the pressure viscosity coefficient.

## 4. Cavitation Failure Mechanism

Cavitation in liquids occurs when the local fluid pressure falls less than the vapor pressure. Its effect is boiling the liquid by pressure reduction rather than an increase in temperature. This type of cavitation is called *vaporous cavitation*. But the other type of cavitation can appear at higher the vapor pressure, which is called *gaseous cavitation*. In this state, due to pressure decrease, some gas solves in the liquid and as pressure rise, again, the bubbles diffuse back to a dissolved state [12]. Rapid and violent collapse of vaporous cavities adjacent to the surfaces can cause damage to the solid surfaces so-called *cavitation failure*. Figure 2 shows a schematic of the bubble collapse mechanism [13]. In most cases, cavitation phenomena are well known in pump wheels, turbine blades, propellers, and valves as well as bearings. However, this phenomenon in bearings seems much more complicated than others in terms of analysis of flow within the gap between the bearing and its pin. The highly dynamic load acting on the bearing consists of compressing and rotating components leading to the creation of a nonstationary lube oil flow. In the current study, the probable regions subjected to cavitation failure are estimated using the EHL method and some cavitation failure criteria will discuss thoroughly.
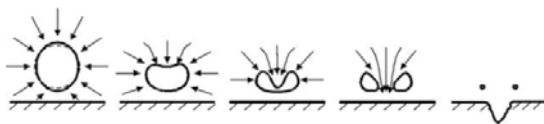
**Figure 2.** Schematic of bubbles collapse mechanism

Cavitation failure in the oil-lubricated plain bearing is observed both in medium/slow speed IC engines for marine or power station applications and in high-speed automotive engines [14]. Because of fluctuation of

radial force from crankshaft and instability of lubricant flow, variation of oil pressure can be sufficient to produce bubble inception, collapse, and pressure pulses formation process. Four main cavitation mechanisms including suction, discharge, flow instability, and impact mechanism in plain bearing are generally recognized. To predict the cavitation damaged regions, some criteria have been used which are:

1) The hydrodynamic pressure development must not go through poorly filled regions. Inside those regions, HD pressure cannot be grown. As a consequence, the pressure development splits up into regions with an insufficient fill ratio. The resulting flow turbulences then may initiate a cavitation attack.

2) The HD peak pressure development must not be intersected by journal oil supply bores. The rapid decay from peak HD pressure to the oil supply pressure of the bore and vice versa will cause severe cavitation.

3) Rapid suction of oil out of bearing grooves or bores to refill an empty or increasing gap has to be avoided by large elastic housing deformations or a radial pin orbital path movement is made [13].

## 5. Simulation and Analysis Model in AVL-Excite Software

In this study, the bearing analysis has been performed using the Elasto-Hydro Dynamic Lubrication (EHL) method. AVL\Excite software [15]is a powerful tool for bearing analysis and in the present work has been employed for EHL analysis of the crankpin bearing of an IC engine. The crankpin bearing parameters of the engine under study which is a 1000 kwIC engine at 1500 Rpm is illustrated in table 1 [16]. The nodal displacements of the bearing surface are determined by solving the equations of motion for the condensed bearing structure. Thus, the mass and stiffness matrixes of the condensed bearing and connecting rod should be extracted using the sub-structuring method. In this study, ABAQUS software has been used for this purpose. Linear tetrahedral elements were used for the meshing of the connecting rod. NONL joint type is employed in piston pin bearing and EHL joint type with 33 axial and 210 circumferential FD mesh nodes is used in the crankpin bearing.

**Table 1.** Crankpin bearing parameters

| Items | Unit | Value |
|---|---|---|
| Diameter | mm | 100 |
| Width | mm | 50 |
| Nominal diametric clearance | μm | 120 |
| Lubricant | - | SAE 15W40 |
| Maximum compressive loading | kN | 300 |
| Maximum tensile loading | kN | 65 |

In order to consider the effect of crankshaft flexibility on oil film performance of crankpin bearing, a model consisting of a crankshaft, rigid crankcase, and connecting rods was created. Figure 3 shows the analysis model of crankpin and piston pin in AVL-Excite software. To reduce the computation time, the bearing analysis has been focused on just one crankpin bearing therefore one of the connecting rods is finely meshed and condensed on piston pin and crankpin bearing nodes. The first boundary condition requests the pressure at the bearing's edges to be 0. The second boundary condition for the rotation requests the pressure in the widest gap to be 0. At the end of the pressure distribution of the rotation (after the narrowest gap), the pressure must become 0 there, where the pressure gradient in the circumferential direction becomes 0. The oil flow inlet is also from the oil hole by adjusting its supply pressure. The walls mentioned earlier are not rigid and are flexible. Compressive and tensile loadings on the bearing can be also seen in Table 1. Of course, their distributions are also obtained from the same dynamic analysis. However, in this analysis, oil supply temperature, clearance height, and oil bore position are also given as the inputs.
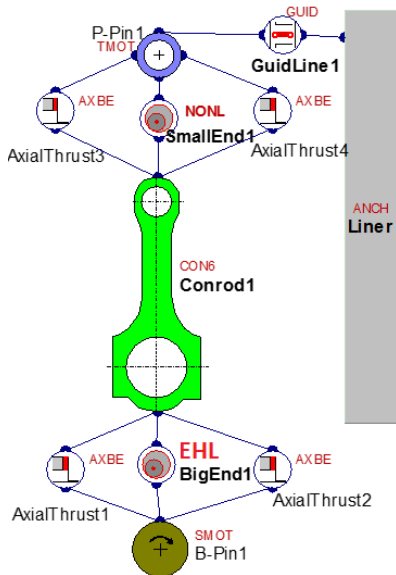


**Figure 3.** Bearing analysis model in AVL-Excite

# 6. Results and Discussions

Figure 4 demonstrates EHL results at 12 degrees after top dead center (TDC) which is the position of maximum firing load. The first plot shows the gap-fill ratio, and the second and third plots show the oil film pressure in two color-bar ranges. The maximum range of the third plot is reduced to illustrate the region with

pressure lower than oil vapor pressure to indicate potential cavitation areas. The fourth plot indicates the oil bore position (the angular position of the oil bore in bearing) and the last plot is related to clearance height. Red and blue colors indicate the high and low quantity regions respectively. Although a large poorly filled region is seen in Figure 4, none of the above-mentioned criteria is satisfied for cavitation occurrence, therefore this situation is uncritical with respect to cavitation failure.
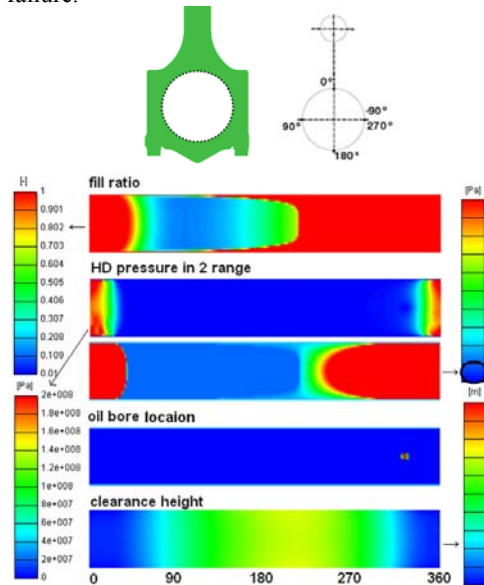


**Figure 4.** Contours of EHL analysis; first case study (oil supply pressure=5 bar, oil supply temperature=100 c, clearance height=100micron, oil bore position= 40 deg before TDC), $P_v$ = 0.5 bar (it is marked by circle)

Figure 5 shows the gap-fill ratio distributions at different special crank angles (CA). Oil flow behavior in the bearing as appears by fill ratio reveals that the cavitation phenomenon can be predicted. The oil supply bore in the crank pin is moving around the bearing and consequently, a depressed oil region is developed in the depleted oil zone around the oil supply rapidly. This rapid process causes a local cavitation phenomenon that can result in the erosion of the bearing surface. In this case, the clearance height is small, as a result, the bubbles haven't an opportunity to escape. Therefore, these bubbles collapse and cause cavitation failure. Due to the consequence of the development and movement of the bubbles, cavitation failure is expected. Henceforth, the regions which are marked by oval are competent for cavitation occurrence, either vaporous or gaseous cavitation. According to Section of Cavitation Failure Mechanism, the regions with the pressure above the vapor pressure also can be competent for cavitation occurrence. In fact, pressure growth due to depressed oil development tears the starveling region to extend the fill ratio to an adequate value, therefore the flow will be turbulent and a cavitation attack will be initiated. As Figure 5shown, the growth of the full flooded region in

starveling regions is faster at the crank angles between 180 and 250, and also the clearance height variation is apparent. Therefore, slight erosion at the range of 90 to 180 degrees of bearing angle (in the lower shell) is feasible. In the entire mentioned region, pressure is less than oil vapor pressure. It is necessary to mention that the bearing angle reference is set on TDC and in a trigonometric direction as shown in Figure 4. Totally, in this case, the development of depressed oil from the oil supply bore into the depleted regions isn't very rapid, therefore due to medial pressure pulses, the cavitation phenomenon is assessed to be medium subversive and this amount is usual for an IC engine bearing. Some other regions in other various crank angles also have pressure less than the oil vapor pressure, but these are not critical concerning the cavitation failure.
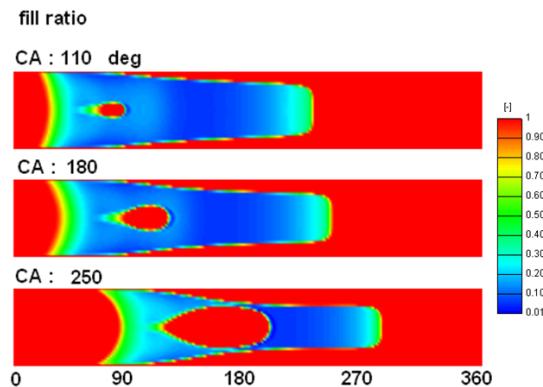


**Figure 5.** Contours of oil fill ratio at three crank angles

# 7.  Parametric Study

The main parameters which can influence the occurrence of cavitation failure are such as clearance height between the shaft and bearing shell, oil supply temperature, and pressure, and oil bore position.

Figure 6 illustrates the effect of an increase in the oil supply pressure on the oil fill ratio. As shown in this figure, an increase in the oil supply pressure causes the cavitation-prone area to become smaller, however, the damaged regions are similarly possible on the lower shell. Therefore, enhancement of oil supply pressure decreases the possibility of cavitation failure. In this case, the oil supply pressure is 6 bar whereas in the first case study this value was 5 bar. In Figure 7 and Figure 8, the effect of a decrease in the oil supply temperature on cavitation failure is investigated. These two figures show the critical situations from different crank angles. In this case, the oil temperature is 70 °c whereas in the first case study oil temperature was 100 °c. Temperature reduction has an undesired effect so which leads to the growth of the cavitation-damaged regions. The occurrence location of the cavitation phenomenon, in this case, is in the middle of the lower shell and the erosion is predicted to be partial. Totally, in all of the above-mentioned cases, since the basis of cavitation

formation is a rapid variation of the fill ratio, therefore if this variation is rapid, the cavitation failure will be more destructive.
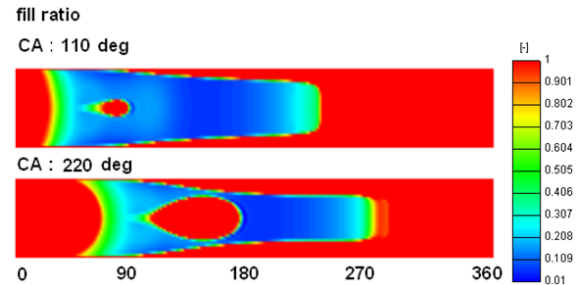


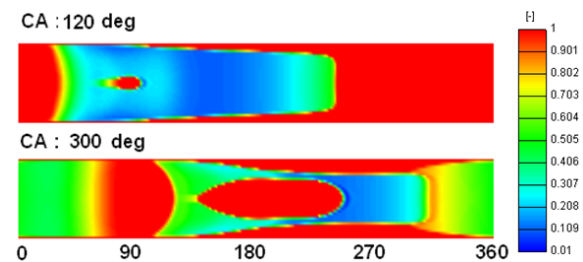**Figure 6.** Contours of oil fill ratio; oil supply pressure = 6 bar



**Figure 7.** Contours of oil fill ratio; oil supply temperature = 70 °c
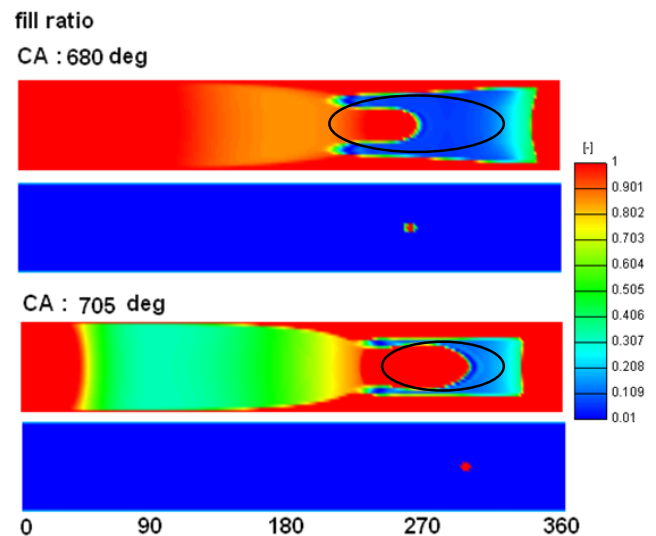


**Figure 8.** Contours of oil fill ratio; oil supply temperature = 70 °c

Figure 9 to Figure 11 show the effect of change in oil bore position on cavitation occurrence in probable critical situations. In this case, the oil bore is placed at 10 deg before TDC whereas already it was at 40 deg before TDC. Figure 9 shows that the peak pressure is intersected on oil bore position, therefore sudden pressure reduction from peak to oil supply pressure and vice versa causes severe cavitation that happens in the middle of the upper shell. Figure 10 shows a different situation with cavitation probability due to the rapid oil development in the starveling region that occurs in the lower shell. This phenomenon happens more rapidly

than in the first case study, therefore bubble collapse pulse is predicted to be stronger in such a case to cause cavitation failure. In Figure 11 choking phenomenon is appeared. It means that oil flow cannot develop from the oil supply bore into the starveling regions and full flooded zones and inevitably becomes to be smaller instead of growing. Just behind the full flooded zone, rapid movement of the oil flow in the starveling zone makes the fully flooded zone no longer full. On the other hand, because the flow regime is turbulent and cavitation bubbles also develop, therefore cavitation failure would be happening. Consequently, the new oil bore position is not good at all and it can be concluded that the oil bore position is a very important factor from the aspect of cavitation occurrence.
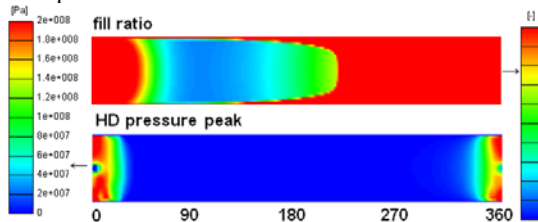


**Figure 9.** Contours of oil fill ratio; Effect of changing the oil bore position in peak pressure state (situation 1); oil bore is set 10 deg before TDC
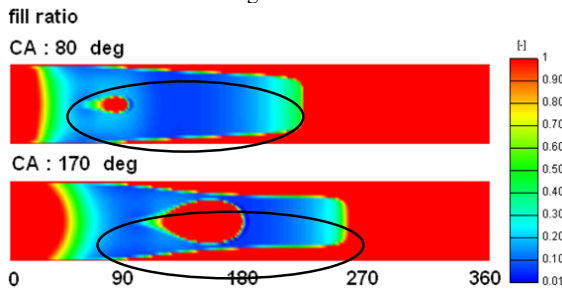


**Figure 10.** Contours of oil fill ratio; Effect of changing the oil bore position (situation 2); oil bore is set 10 deg before TDC
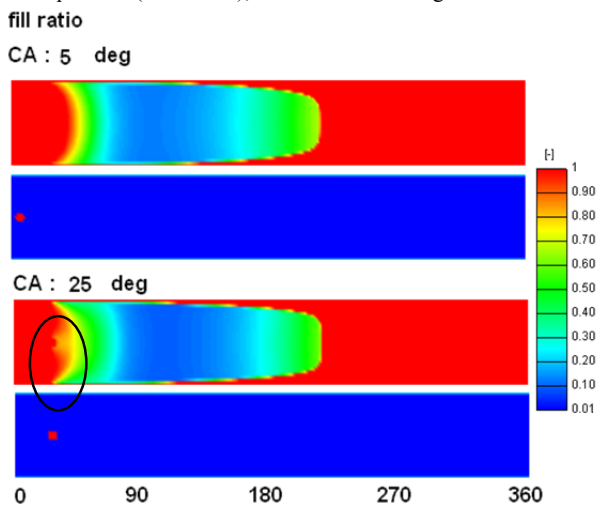


**Figure 11.** Contours of oil fill ratio; Effect of changing the oil bore position (situation 3); oil bore is set 10 deg before TDC

Figure 12 presents the effect of an increase in clearance height between the bearing shell and its pin.

Clearance height, in this case, is set to be 185 microns, whereas in the case of minimum clearance (i.e. the first case study) clearance height was 100 microns. In this case, rather than in the first case study, the cavitation generating source will not be changed but its occurrence location and the involved area will be changed. The involved area with cavitation, in this case, is bigger than the case of minimum clearance (the first case study), because the spatial gap is bigger than in the previous.
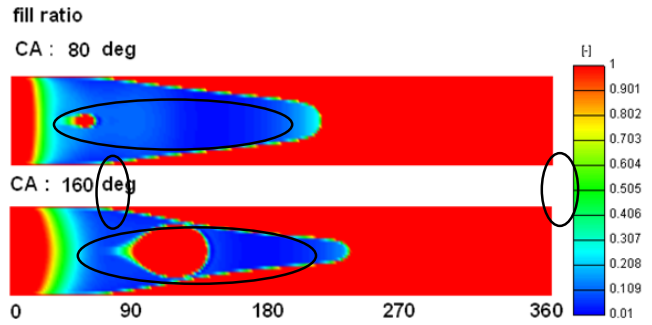


**Figure 12.** Contours of oil fill ratio; clearance height = 185 microns

Given the above discussions, to improve the bearing condition to avoid cavitation failure, some suggestions can be presented such as increasing the bearing oil pressure, decreasing clearance height, and being constant oil flow speed to the prevention of the flow turbulence and flow instability, and using a bearing with hard surface material.

## 8. Conclusion

The following conclusions were obtained from cavitation failure assessments in crankpin bearings. High-speed IC engines generally use the un-grooved crankpin bearings. Elasto-Hydrodynamic Lubrication (EHL) method is utilized to consider the effect of stiffness of the bearing shell housing in the model to obtain the required characteristics of the bearing lubrication. Results show that in the crankpin bearing, cavitation failure is not critical and it is slight. The depressed oil development from the oil supply bore in the depleted region isn't very rapid, therefore due to the weak pressure pulses, the cavitation phenomenon is assessed to be medium subversive which is usual for crankpin bearings of IC engines.

In order to investigate the effect of some design parameters, such as clearance height between shaft and bearing shell, oil supply temperature and pressure, and oil bore position, on cavitation failure, a parametric study was also done. An increase in the clearance height leads to getting bigger the involved area with the cavitation phenomenon than in the case of minimum clearance. Enhancement of the oil supply pressure decreases the possibility of cavitation failure. Temperature reduction has an undesired effect that leads to the growth of the cavitation-damaged regions in the

middle of the lower shell and the erosion is predicted to be partial. Also, it is included that the oil bore position is a very important factor for cavitation occurrence. Consequently, in order to improve the bearing condition to avoid the cavitation failure, some suggestions can be presented such as increasing the bearing oil pressure, decreasing the clearance height, being constant oil flow speed to the prevention of the flow turbulence and flow instability and using a bearing with hard surface material.

# 9. Funding

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