Research Report

Preventive Design and Analysis of Cavitation Noise on Diesel Engine

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ディーゼルエンジンにおけるキャビテーションノイズの発生メカニズムとその防止法

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Abstract

This paper presents an analysis of noise occurrence at a diesel engine, and a design to prevent the noise which occurred unperiodically with frequency over 5kHz. The mechanism of noise occurrence was assumed to be as follows. The noise occurred when the following conditions were combined: (1) cavitation appeared in the oil film at the main bearing, (2) main journal vibration in the radial direction induced further appearance and collapse of the cavitation. The mechanism was verified by the following items derived from numerical analyses and experimental results, (a) the existence of cavitation at the time of noise occurrence, (b) the instability of the main journal-oil film, (c) the simultaneous fluctuation of the combustion pressure.

Finally, the observation of excited oil film and the measurement of noise were conducted simultaneously by making a test rig based on the mechanism in order to investigate the relation between the cavitation and noise. The noise occurred when the cavitation reached the atmosphere at the end of oil film. The reaching of cavitation was prevented by cutting a ring groove at the end of a circular piece, and the noise was reduced. The noise reduction was confirmed on an actual engine by using a main bearing with grooves cut at both ends.

Keywords Diesel engine, Noise, Cavitation, Main bearing, Oil film

要　　旨

エンジンの振動・騒音は，多くの研究や技術開発によって低減されてきたものの，未解決の課題も存在する。その一つに，ディーゼルエンジンにおける主軸受油膜のキャビテーションの発生・崩壊に起因する高周波数かつ断続的なノイズがある。このノイズの解析法および防止法を解説する。このノイズは，以下の３つの条件 (①軸受油膜のキャビテーションの存在，②軸 - 軸受系の不安定状態の存在，③軸の高周波加振力の存在) が重なったときに発生するというメカニズムを仮定し，数値解析 (質量保存を考慮した弾性流体潤滑解析と軸 - 軸受系の不安定性解析) および実験によって妥当性を確認した。次に，上記ノイズ発生メカニズムを基に，キャビテーションとノイズの関係を明らかにするために，油膜を加振した際の油膜可視化実験を行った。その結果，加振によって崩壊したキャビテーションが，大気に到達した際にノイズが発生することを明らかにした。また，軸受端部に油を満たした溝を設置しキャビテーションの大気への到達を防止することで，ノイズを防止できた。このノイズ防止法は実機においても効果を確認できた。

キーワード　ディーゼルエンジン，ノイズ，キャビテーション，軸受，油膜

1. Introduction

In recent years, attempts to reduce noise of diesel engines have contributed to the reduction of combustion noise and mechanical noise. However, the insolvable problem of other noise yet exists. For example, the noise with a high-frequency of over 5 kHz unperiodically occurred from the main bearing of a diesel engine.

This paper presents an analysis of the noise occurrence at the 7th main bearing of an in-line six-cylinder diesel engine, and proposes a design to prevent the noise from occurring. Firstly, the cause of the noise was considered and then clarified by EHL analysis and stability analysis. Finally, a visualization test of the excited oil film and measurement of the noise were conducted simultaneously to investigate the relation between the cavitation and noise, and prevention of the noise.

2. Condition of noise occurrence

The operational condition of noise occurrence is shown in **Table 1**. The noise occurred at the 7th main bearing in the cylinder block at the time of the 5th cylinder combustion as shown in **Fig. 1**. The frequency of noise was over 5 kHz, and this noise occurred unperiodically.

3. Assumption and verification of the cause of noise

3. 1 Assumption of noise occurrence

The cause of noise was investigated on an actual engine. No contact between the crankshaft and bearing was observed. On other hand, cavitation erosion was observed on the bearing surface as shown in **Fig. 2**. Therefore, the cavitation seemed to be one factor of the noise occurrence. The mechanism of noise occurrence was assumed by using the following three conditions as shown in **Fig. 3**, and the assumption was verified as described in the following sections.

Table 1 Condition of noise occurrence on in-line six-cylinder diesel engine.

Engine speed	2200 rpm
Output torque	90 N-m
Crank angle	610-620 deg. ATDC (No.1 cyl.)
Oil temperature	90 ℃

3. 2 Elastohydrodynamic lubrication analysis

Elastohydrodynamic lubrication analysis which considered mass conservation of the oil flow in the oil filled and cavitation areas[1, 2] was conducted for confirmation of cavitation in the oil film. The analysis was defined by using the relation between the inward and outward flows which considered a fill ratio of oil, ψ, in a control volume as shown in

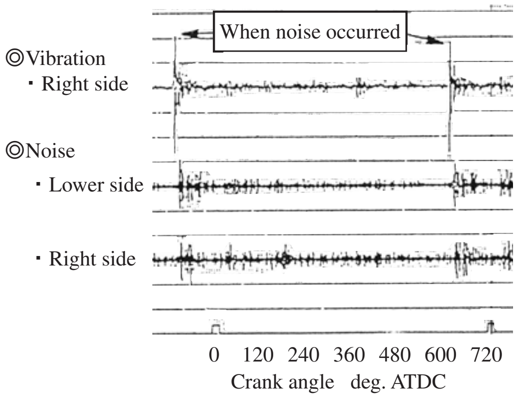


Fig. 1 Cylinder block vibration and noise.

Fig. 2 Cavitaion erosion on bearing surface.

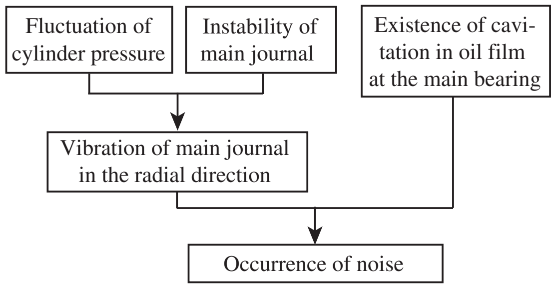


Fig. 3 Assumption of noise occurrence mechanism.

Fig. 4(a). The cavitation in the oil film was calculated by considering the equation of continuity including the fill ratio and the oil film thickness including elastic deformation of the main bearing. The bearing loads Wx, Wy of input data were calculated by the engine vibration analysis.[3)]

The cavitation in the 7th main bearing under the noise occurrence condition was calculated using the load data and the calculating condition in **Table 2**. A large degree of cavitation existed at the time of noise occurrence as shown in **Fig. 5**(a). This agrees with the assumption shown in Fig. 3. The cavitation existed in the cap side of the bearing as shown in Fig. 5(b). If the crankshaft vibrated in the radial direction in this situation, it caused the further appearance and collapse of the cavitation in the oil film of the main bearing.

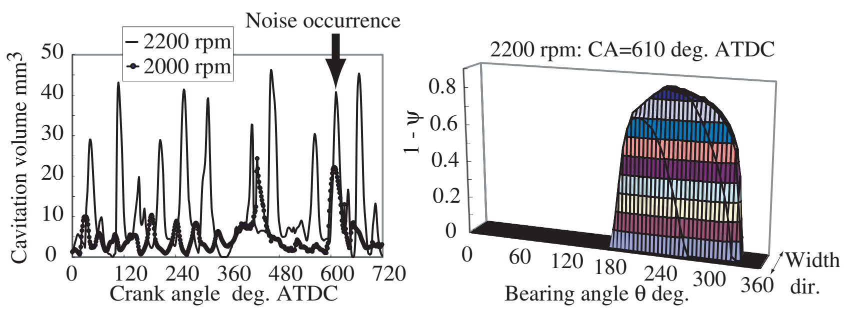
3. 3 Vibration of main journal in radial direction

When the 7th main journal is unstable and an exciting force exists simultaneously, the main journal vibrates. Stability analysis of the main journal was introduced to investigate the vibration of the crankshaft at the time of noise occurrence. The bearing oil film model shown in Fig. 4(b) was expressed by Reynolds' equation.[4)] The linear stiffness $[K_b]$ and damping $[C_b]$ of the oil film were defined as Eqs. (1) and (2) by deformation of Reynolds' equation. Two bearing models were defined to express the translational and rotational displacements of the main journal as shown in **Fig. 6**. When the displacement vector of the crankshaft journal was $\{X\}$ and the equivalence mass matrix was $[M]$, the characteristic equation could be described as shown in Eq. (3). Here, the equivalence mass matrix of the crankshaft was decided from the dimensions of the main journal. The introduction of Eqs. (1), (2) and (3) is shown in the reference[5)] in detail.

$$[K_b] = \begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix} = \begin{bmatrix} \dfrac{\partial(df_x)}{\partial(dx)} & \dfrac{\partial(df_x)}{\partial(dy)} \\ \dfrac{\partial(df_y)}{\partial(dx)} & \dfrac{\partial(df_y)}{\partial(dy)} \end{bmatrix} \quad \ldots\ldots (1)$$

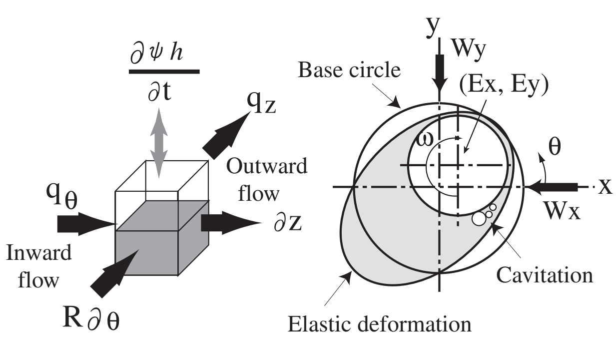
Table 2 Calculating condition for EHL.

Compressibility factor	500 MPa
Oil viscosity	10.1 m Pa s
Bearing housing	205.8
Young's modulus	GPa
Poisson's ratio	0.26
Density	7.9×10^3 kg/m^3
Engine speed	2200 rpm



(a) Record of cavitation volume (b) Distribution of cavitation (1-ψ) in oil film

Fig. 5 Cavitation in the main bearing.



(a) Control volume model (b) Bearing model

Fig. 4 Elastohydrodynamic lubrication model.

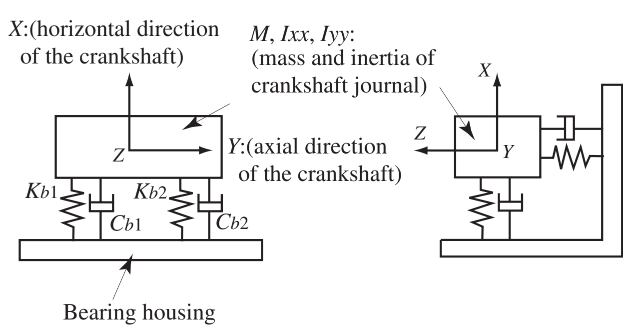


Fig. 6 Bearing model by linear approximation.

$$[C_b] = \begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix} = \begin{bmatrix} \dfrac{\partial(df_x)}{\partial(d\dot{x})} & \dfrac{\partial(df_x)}{\partial(d\dot{y})} \\ \dfrac{\partial(df_y)}{\partial(d\dot{x})} & \dfrac{\partial(df_y)}{\partial(d\dot{y})} \end{bmatrix} \quad \cdots\cdots (2)$$

$$\begin{Bmatrix} \dot{X} \\ \ddot{X} \end{Bmatrix} = \begin{bmatrix} O & I \\ -M^{-1}K & -M^{-1}C \end{bmatrix} \begin{Bmatrix} X \\ \dot{X} \end{Bmatrix} \quad \cdots\cdots\cdots\cdots (3)$$

The instability of main journal was defined by the sign of the real part of the complex eigenvalue. The maximum of the real part in the mode vector of the complex eigenvalue was selected at every time step, and the vertical component of the mode vector was shown in **Fig. 7**. The eigenmode of the journal-bearing system became unstable at the time of noise occurrence. Therefore, when an excitation force such as a fluctuation in the combustion pressure acted on the crankshaft, the crankshaft probably vibrated violently in the vertical direction.

The combustion pressure fluctuated with high-frequency at the time of the noise occurrence as shown in **Fig. 8**. The high-frequency fluctuation of the combustion pressure changed cycle by cycle, which was considered to be related to unperiodical noise. The noise occurrence condition at 2200 rpm was constructed by combining the three conditions as shown in Figs. 5, 7 and 8.

4. Verification of mechanism of noise occurrence

The conditions under which the noise did not occur were applied to the above mechanism of noise occurrence. For example, the noise did not occur at 2000 rpm as shown in Figs. 5(a) and 7(b). The noise did not occur unless the three parameters were combined simultaneously, thus strengthening the validity of the proposed mechanism as shown in **Table 3**.

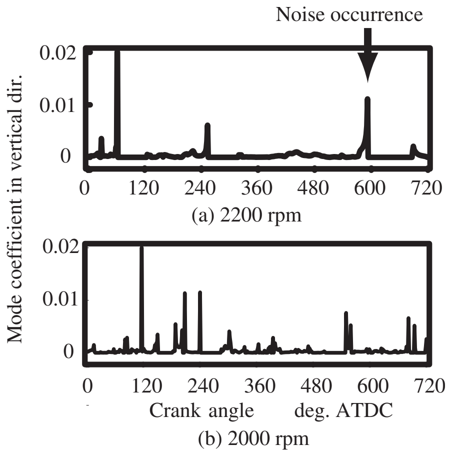


Fig. 7 Mode coefficient of the most unstable mode.

Table 3 Verification of the mechanism applied to the conditions without noise occurrence.

	Parameter A	Parameter B	Parameter C
2000 rpm	√		√
2600 rpm	√		√
Stopping fuel supply to 5th cylinder	√	√	
Small oil clearance	√		√
Large oil clearance	√		√
High oil viscosity			√

√ : satisfies the parameter
Parameter A : Cavitation in the oil film (Fig. 5)
Parameter B : Instability of 7th main journal (Fig. 7)
Parameter C : Fluctuation of cylinder pressure (Fig. 8)

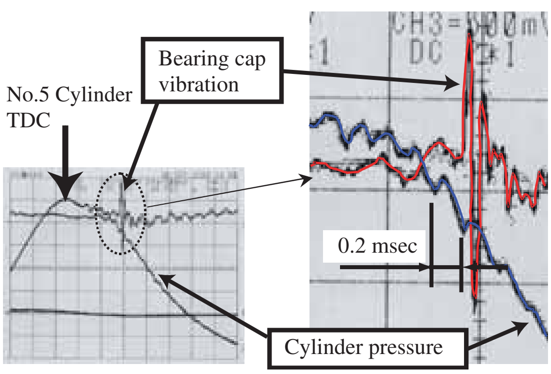


Fig. 8 Cylinder pressure and block vibration.

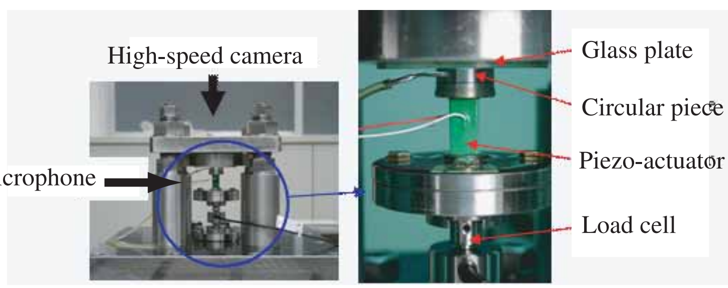


Fig. 9 Test rig for cavitation observation and noise measurement.

5. Experimental verification and prevention of noise

5. 1 Relation between cavitation and noise

The observation of the excited oil film and the measurement of the noise were conducted in a test rig based on the mechanism of noise occurrence in order to investigate the relation between the cavitation and noise as shown in **Fig. 9**. The test rig was composed of a circular piece and a glass plate with the same clearance as the bearing clearance, and the clearance was filled with engine oil. The diameter of the circular piece was the same as the width of the main bearing of the engine. When the circular piece was excited in the vertical direction by using a piezo-actuator, the oil film was observed through the glass plate using a high-speed camera. At the same time, the noise was measured using a noise meter through a microphone, and the oil film load was measured using a load cell simultaneously.

The relation between the cavitation and noise was shown in **Fig. 10** when the circular piece was excited with half the maximum clearance and the frequency of 60 Hz. When the clearance was maximum, the oil film load was negative and cavitation appeared as shown in Fig. 10(a). Next, as the clearance became small, the cavitation size also became smaller as shown in Fig. 10(b) and the cavitation collapsed by the positive oil film load which was observed to be "lightning shape" as shown in Fig. 10(c). However, the noise did not occur because the collapsed cavitation had not yet reached the outer edge of the circular piece. When the cavitation reached the edge of the circular piece, noise occurred as shown in Fig. 10(d).

5. 2 Prevention of noise

Preventive methods were presumed: (1) preventing the appearance of cavitation, and (2) preventing the

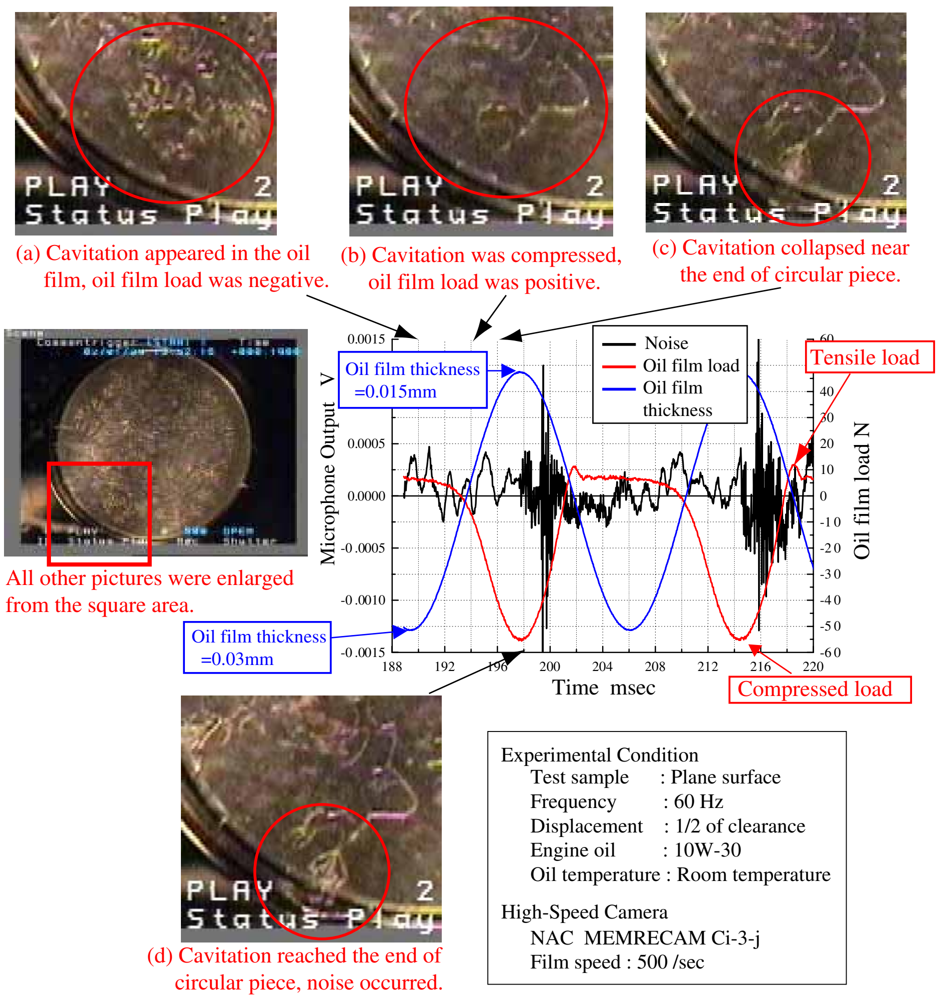


Fig. 10 Relation between cavitation and noise.

collapsed cavitation from reaching the atmosphere. Firstly, an experiment to prevent the appearance of cavitation was conducted using modified shapes of the surface of the circular piece. Several kinds of modified shapes were tested, for example, making a hole, micro grooves, concave shape and others. However, the cavitation and noise still occurred for all the surface shapes.

Secondly, a ring groove was cut at the periphery of the circular piece and filled with engine oil to prevent the collapsed cavitation from reaching the atmosphere. The cavitation stayed in the groove and did not reach the outside atmosphere, as shown in **Fig. 11**(a). As the result, the noise did not occur. Figure 11(b) shows the noise reduction for frequencies over 4 kHz which was achieved by using the ring groove.

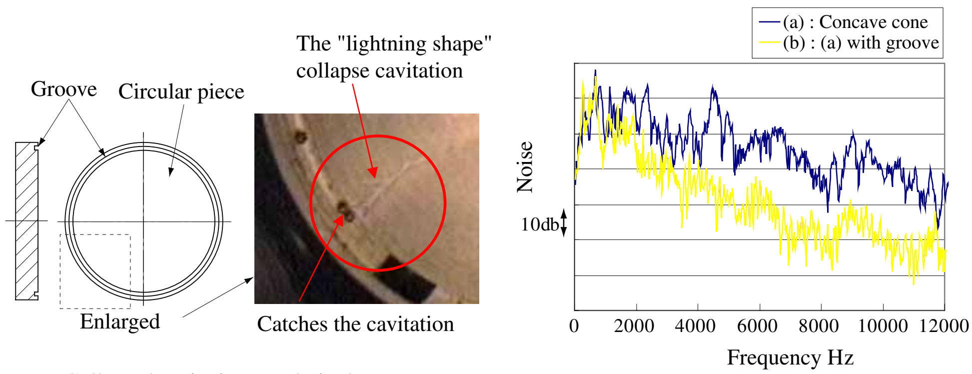
Finally, the bearing with grooves was made and applied to an actual engine to investigate the effect. The grooves were cut at both ends of the bearing, as shown in **Fig. 12**(c). The noise which occurred unperiodically at the bearings without the grooves as shown in Fig. 12(a) did not occurred at the bearings with the grooves as shown in Fig. 12(b), confirming the preventive effect of the grooves.

6. Conclusions

(1) Cavitation noise occurred only when the following three conditions were combined: existence of high-frequency fluctuation of the cylinder pressure, existence of unstable eigenmode of the crankshaft journal-bearing system, and appearance of cavitation in the main bearing oil film. Therefore, the noise occurrence can be estimated using these three conditions.

(2) Cavitation appeared at the negative oil film load and collapsed at the positive oil film load, which was clarified by the visualization test of the excited oil film based on the above mechanism. Noise occurred when the collapsed cavitation reached the atmosphere.

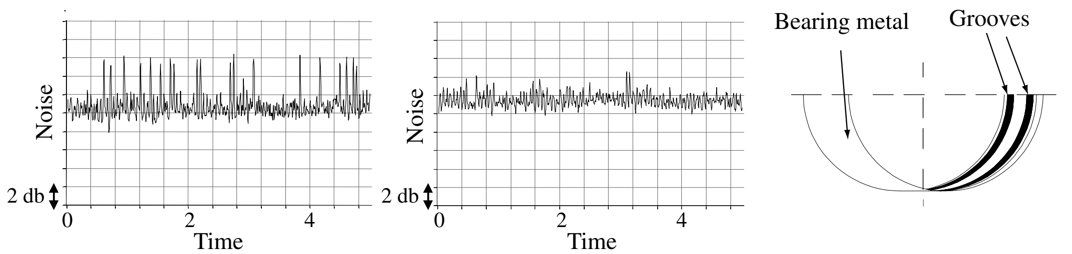
(3) The noise was prevented by cutting ring groove at the periphery of the circular piece and filling with engine oil to prevent the collapsed cavitation from reaching the atmosphere. The cavitation stayed in the groove and the noise did not



(a) Collapsed cavitation caught in the groove

(b) Effect of the piece with groove on noise

Fig. 11 Observation of cavitation in the groove.



(a) Without grooves

(b) With grooves

(c) Bearing with grooves

Fig. 12 Effect of the bearing with grooves on noise.

occur. Bearings with grooves cut at both ends were applied to an actual engine, and tests showed that the noise did not occur, thus confirming the noise-preventive effect of the grooves.

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