

## Whirling vibration characteristics due to lubricating oil pressure in diesel generator shafting system

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**Abstract:** Diesel generator is the essential component on ships and offshore structures. Not only supply the electric power for the auxiliary systems but also in the case of special purpose vessels, the diesel generator is widely used as the primary power source because of the flexibility, efficiency and effective cost in its initial setup. To ensure the stability and prevent from leading to failure, the vibration characteristics of shafting system should be controlled especially the whirling vibration.

In this paper, the influence of shaft bearings lubricating oil pressure to the vibration characteristics of diesel generator shafting system is concerned. When the lubricating oil pressure increases, the oil film stiffness increases. In the case of oil film stiffness variation, the vibration resonance changes. Through analysis results, critical bearing stiffness with the referred lubricating oil film pressures can be identified in order to control the vibration resonance. The resonance of diesel generator structural vibration combined with hull foundation should be avoided from synchronized speed. Therefore, it is necessary to analyze and control the shaft whirling vibration.

One of methods to reduce shaft whirling vibration of diesel generators is to select suitable pressure for lubricating oil film in bearings. Analyzing and experimental results showed that the adjustment of suitable lubricating oil film pressure could improve the vibration characteristics especially the whirling vibration of diesel generator shafting system.

**Keywords:** shaft whirling vibration, oil film stiffness, diesel generator, lubricating oil film pressure.

### 1. Introduction

Dynamic characteristics of diesel generators are variable. The unbalance masses (i.e. piston - connecting rod groups, misalignment-shafting system...) and excitation forces (i.e. gas pressure in cylinders, rotating movement of propeller in water...) can lead to the structure vibration especially shaft whirling vibration. Whirling vibrations is excitation vibration. When excitation vibrations coincide with the natural frequency of engine structure, the resonance occurs resulting violent vibration. The excessive whirling vibration adversely influences particular shafting system elements and system integrity in whole. **Figure 1** shows a destroyed shaft line at high speed due to whirling vibration.



**Figure 1** Destroyed shaft line at high speed due to whirling vibration

The problems caused by whirling vibration are as the following:

- Complete shafting system destruction;
- Shorten shaft and other elements fatigue life;
- Fatigue cracks at crankshaft and journal bearing;
- Excessive noise, hull and superstructure vibrations.

In order to reduce the structure vibration, most diesel generators are mounted on a flexible bed, while the engine and generator are coupled by employing a flexible coupling. Shaft whirling vibration is closely related to the oil film pressure in shaft bearings. When the lubricating oil pressure increases, the oil film stiffness increases. In the case of oil film stiffness variation, the vibration resonance changes.

This paper presents whirling vibration characteristics due to lubricating oil pressure variation in the diesel generator shafting system. The influence of shaft bearings lubricating oil pressure to the vibration characteristics of diesel generator shafting system is concerned.

## 2. Theory for whirling calculation

### 2.1 Vibration of diesel generators by the moving parts and the expanding gas forces

Structural vibration is generally caused by the dynamic properties of the structure and excitation forces. Mathematically the relationship between the exciting forces, structure properties and the vibration response is normally presented by the general equation of motion [1]:

$$\mathbf{M}\mathbf{x}(t) + \mathbf{C}\mathbf{x}(t) + \mathbf{K}\mathbf{x}(t) = \mathbf{F}(t) \quad (1)$$

Where  $\mathbf{x}(t)$  = solution of the equation (deflection at the system nodes);  $\mathbf{M}$  = mass matrix;  $\mathbf{C}$  = damping matrix;  $\mathbf{K}$  = stiffness matrix;  $\mathbf{F}(t)$  = excitation force vector.

Mass of moving parts of diesel generator can be calculated by equation as [2]:

$$m = \frac{(m_p + 0.3m_c)}{\pi r^2} + 0.7m_c \quad (2)$$

Where:  $m$  = total mass of a piston-connecting rod mechanism,  $m_p$  = mass of the piston,  $m_c$  = mass of the connecting rod,  $r$  = rotating radius of crankpin.

The matrices  $\mathbf{M}$  and  $\mathbf{K}$  represent the dynamic characteristics of the structure. Modifying these characteristics or excitation forces can result reducing the vibration response.

The matrix  $\mathbf{M}$  is not only the total masses, but represents also the masses distribution over the whole structure. The same applies to the stiffness matrix  $\mathbf{K}$ . From the vibration point of view, it may be very important where the mass or stiffness is located.

$\mathbf{C}$  denotes the damping, which in practice is not only a uniform number. In real structures the damping normally varies depending on the frequency and mode shape, as well as on the location. In complex structures like engines, several different damping types can be found.

**Figure 2** shows a schematic diagram for researching the dynamic properties of the structure and the excitation forces [2]. The crank shaft is driven by the expanding gas in the cylinder. The expanding gas creates on the piston top a pressure force  $F$ , which is transmitted to the crankshaft through the connecting rod. The transmitted gas pressure acting on crankshaft can be divided into two components: *centripetal force* resulting shaft axial vibration and *tangent force* resulting torque, which tends to rotate the crankshaft.

**Acceleration of the piston.** The pressure force  $F$  exists on the top of piston and makes piston moving downward. If we consider the origin of the  $x$  - axis as the uppermost position of the piston, the

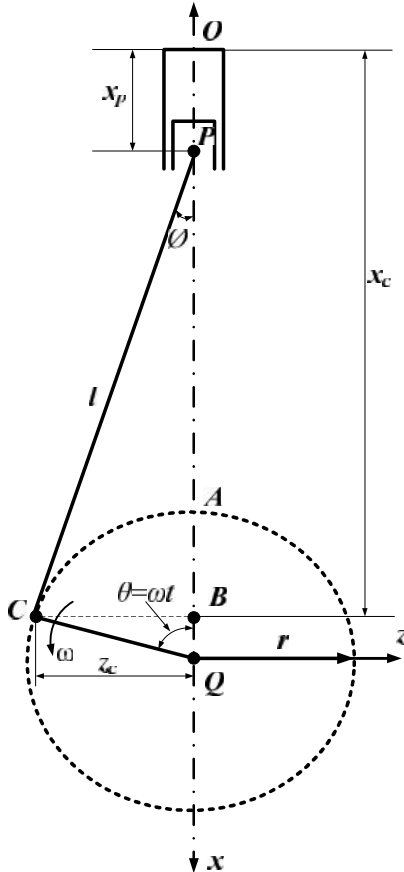
displacement of the piston corresponding to an angular displacement of crank of  $\theta = \omega t$ . The displacement of the piston from its topmost position can be expressed as [2]:

$$x_p = r + l - r \cos \omega t - l \sqrt{1 - \frac{r^2}{l^2} \sin^2 \omega t} \quad (3a)$$

Also:

$$l \sin \phi = r \sin \theta = r \sin \omega t \quad (3b)$$

$$\cos \phi = \left( 1 - \frac{r^2}{l^2} \sin^2 \omega t \right)^{\frac{1}{2}} \quad (3c)$$



So that:

$$x_p = r + l - r \cos \omega t - l \sqrt{1 - \frac{r^2}{l^2} \sin^2 \omega t} \quad (4)$$

In general,  $r/l < 1/4$ , **Eq. (4)** can be approximated as:

$$x_p = r(1 - \cos \omega t) + \frac{r^2}{2l} \sin^2 \omega t \quad (5)$$

Or, equivalently:

$$x_p = r \left( 1 + \frac{r}{2l} \right) - r \left( \cos \omega t + \frac{r}{4l} \cos 2\omega t \right) \quad (6)$$

**Eq. (6)** can be differentiated with respect to time to obtain expression for the velocity and acceleration of the piston:

$$\dot{x}_p = r \omega \left( \sin \omega t + \frac{r}{2l} \sin 2\omega t \right) \quad (7)$$

$$\ddot{x}_p = r \omega^2 \left( \cos \omega t + \frac{r}{l} \cos 2\omega t \right) \quad (8)$$

Where:  $x_p$  = vertical displacement of the piston in the  $x$  direction from its topmost position,  $\omega$  = angular velocity of the crankshaft,  $l$  = length of connecting rod.

**Figure 2 Schematic diagram of a cylinder of diesel generator**

**Acceleration of the Crankpin.** With respect to the  $xz$  – coordinate shown in **Fig. 2**, the vertical and horizontal displacements of the crankpin are given by [2]:

$$x_c = OA + AB = l + r(1 - \cos \omega t) \quad (9)$$

$$z_c = CB = r \sin \omega t \quad (10)$$

Where:  $x_c$  = vertical displacement of crankpin in the  $x$  direction,  $z_c$  = horizontal displacement of crankpin in the  $z$  direction.

Differentiations of **Eqs. (8)** and **(9)** with respect to time give the velocity and acceleration of the crankpin:

$$\dot{x}_c = r \omega \sin \omega t \quad (11)$$

$$\dot{z}_c = r\omega \cos \omega t \quad (12)$$

$$\ddot{x}_c = r\omega^2 \cos \omega t \quad (13)$$

$$\ddot{z}_c = -r\omega^2 \sin \omega t \quad (14)$$

**Inertia Force.** Although the mass of the connecting rod is distributed throughout its length. It is generally idealized as a massless link with two masses concentrated at its ends, the piston end and the crankpin end. The vertical component of the inertia force ( $F_x$ ) for one cylinder is given by [2]:

$$F_x = m_p x_p + m_c x_c \quad (15a)$$

or:

$$F_x = (m_p + m_c)r\omega^2 \cos \omega + m_p \frac{r^2\omega^2}{l} \cos 2\omega \quad (15b)$$

Similarly, the horizontal component of inertia force for a cylinder can be obtained:

$$F_z = m_p z_p + m_c z_c \quad (16a)$$

Where  $z_p = 0$  and  $z_c$  is given by **Eq. (14)**. Thus:

$$F_z = -m_c r\omega^2 \sin \omega \quad (16b)$$

The unbalanced and inertia force on a single cylinder are given by **Eqs. (15b)** and **(16b)**. The mass  $m_p$  is always positive, but  $m_c$  can be made zero by counterbalancing on the crank. Therefore, it is possible to reduce the horizontal inertia force  $F_y$  to zero, but the vertical unbalanced force always exists. Thus, a single cylinder engine is inherently unbalanced.

In a multi-cylinder engine, it is positive to balance some or all of the inertia forces and torque by proper arrangement of the cranks. The lengths of all the cranks and connecting rods are taken to be  $r$  and  $l$ . For force balance, the total inertia force in the  $x$  and  $y$  directions must be zero, so that:

$$(F_x)_{total} = \sum_{i=1}^N (F_x)_i = 0 \quad (17)$$

$$(F_z)_{total} = \sum_{i=1}^N (F_z)_i = 0 \quad (18)$$

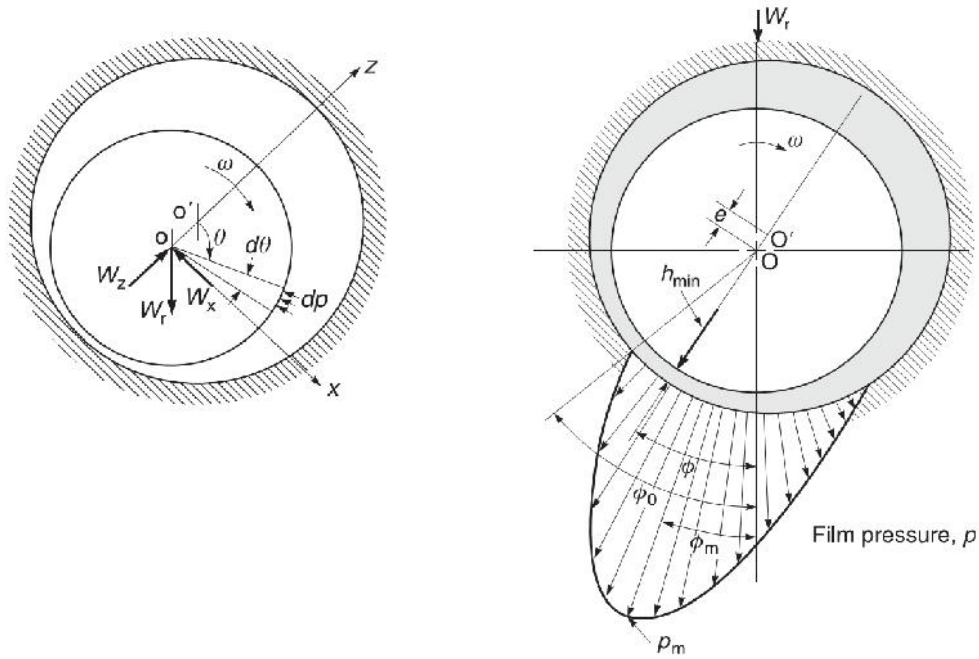
Where  $(F_x)_i$  and  $(F_z)_i$  are the vertical and horizontal component of inertia forces

Thus, if the moving parts of the cylinders are equivalence and the power of cylinders are balance, the vibration due to the expanding gas forces and inertial forces of cylinders are suppressed, so the engines will not vibrate.

## 2.2 Stiffness and damping calculation for journal bearing

Lubricating oil is supplied into bearings of shafting system to lubricate its working surfaces and then reduce the friction. It plays a key role in determining the dynamic characteristics of diesel generators shafting system.

The schematics of a journal are shown in **Fig. 3**. Letter  $O$  is the center of journal whereas  $O'$  is the center of bearing. The radial clearance,  $c$ , between the bearing bore of radius  $r_I$  and of the journal radius,  $r$ , equals  $r_I - r$ .



**Figure 3 Schematics of plain journal bearing geometry**

When the shaft speed is fast enough to create hydrodynamic pressure larger than the applied load, the journal can separate and move around the bearing until it reaches an equilibrium. The eccentricity  $e$  is the distance between the centers of shaft and bearing. When the journal touches the bearing and when the load equals zero. The ratio of eccentricity to clearance is presented by eccentricity ratio :

$$\varepsilon = \frac{e}{c} \quad (0 \leq \varepsilon < 1) \quad (19)$$

The oil film thickness at any position can be calculated approximately by [3]:

$$h \sim c(1 + \varepsilon \cos \theta) \quad (20)$$

The minimum thickness is given as:

$$h_{min} = c - e = c(1 - \varepsilon) \quad (21)$$

Journal bearings are short bearing in this case. Following Reynolds equation [4], the pressure of oil film can be calculated by the equation:

$$P = \frac{3\mu\omega\varepsilon}{c^2} \left( \frac{b^2}{4} - y^2 \right) \frac{\sin \theta}{(1 + \varepsilon \cos \theta)^3} \quad (22)$$

Maximum pressure of oil film at angular position  $\theta_m$  and  $y = 0$  can calculate by [5]:

$$p_m = \frac{3\mu\omega\varepsilon L^2 \sin \theta_m}{4c^2(1 + \varepsilon \cos \theta_m)^3} \quad (23)$$

with:

$$\theta_m = \cos^{-1} \left( 1 - \frac{\sqrt{1 + 24\varepsilon^2}}{4\varepsilon} \right) \quad (24)$$

The load capacity is solved by integration of the pressure wave around the bearing:

$$W_x = 2 \int_0^\pi \int_0^{L/2} pr \sin \theta dy d\theta \quad (25)$$

$$W_z = 2 \int_0^\pi \int_0^{L/2} pr \cos \theta dy d\theta \quad (26)$$

By using the Sommerfeld substitution we can obtain:

$$W_x = \frac{\mu}{4} \left(\frac{L}{c}\right)^2 \frac{\pi}{(1-\varepsilon^2)^{3/2}} \quad (27)$$

$$W_z = \frac{\mu}{4} \left(\frac{L}{c}\right)^2 \frac{4\varepsilon^2}{1-\varepsilon^2} \quad (28)$$

$$W_r = \sqrt{W_x^2 + W_z^2} = \frac{\mu}{4} \left(\frac{L}{c}\right)^2 \frac{\varepsilon \sqrt{16\varepsilon^2 + \pi^2(1-\varepsilon^2)}}{(1-\varepsilon^2)^2} \quad (29)$$

The Sommerfeld number  $S$  is given by equation:

$$S = \frac{\mu}{P} \left(\frac{r}{c}\right)^2 = \left(\frac{L}{c}\right)^2 \frac{(1-\varepsilon^2)^2}{\pi \sqrt{1-\varepsilon^2 + \pi^2(1-\varepsilon^2)}} \quad (30)$$

And average pressure is calculated by equation [6]:

$$P_{average} = \frac{W_r}{2rL} \quad (31)$$

The bearing stiffness can be obtained by equation:

$$K_f = \frac{dW_r}{dh} \quad (32)$$

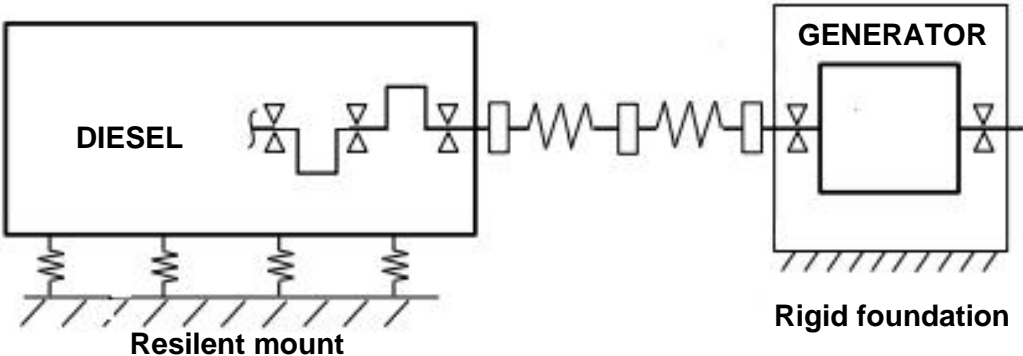
Where:  $r$  = journal radius (m);  $r_I$  = bearing bore radius (m);  $L$  = bearing length (m);  $c$  = radius clearance (m);  $e$  = eccentricity (m);  $\varepsilon$  = eccentricity ratio;  $h$  = oil film thickness (m);  $h_{min}$  = minimum oil film thickness (m);  $\theta$  = angular position (rad);  $h_v$  = oil film thickness at angular position  $\nu$  (m);  $p_m$  = maximum pressure position (rad);  $P$  = oil film pressure (Pa);  $P_m$  = maximum oil film pressure (Pa);  $W_x$ ,  $W_y$ ,  $W_r$  = load capacity (N);  $P_{average}$  = average oil film pressure (Pa);  $K_f$  = bearing stiffness (N/m);  $\mu$  = lubricant absolute viscosity (Pa-s).

### 2.3 Whirling vibration in diesel generator shaft system

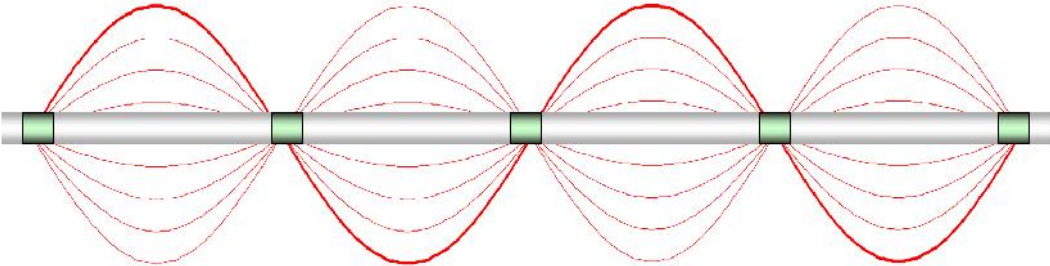
Whirling vibration due to shaft static and dynamic unbalance is a forced vibration caused by internal centrifugal force. It is well known that a diesel generator shafting system will never be perfectly balanced. Due to material inhomogeneity the center of gravity does not coincide with the rotation axis. Due to the center of journal and bearing does not coincide to make eccentricity. Eccentricity depends on oil film pressure of journal bearing. When the shaft rotates, the eccentric masses produce exciting centrifugal force. Unbalance excites forward synchronous whirling.

Alignment-related vibration of shaft system of diesel generator is possible if the shafts are not properly coupled due to assembly or manufacture errors. It is the case when shaft system loses the rotation axis continuity and smoothness. Alignment-related errors excite first order synchronous whirling vibration.

Excitations having alignment errors nature normally are not taken into account in whirling vibration calculation. Such errors must be prevented by adherence to the technological specifications and standards of shaft manufacture and alignment processes. The schematic of a diesel generator arrangement is shown in **Fig. 4**.



**Figure 4 Schematic of a diesel generator arrangement**



**Figure 5 Schematic of whirling vibration of rotating shaft and bearing**

At the same time there are no obstacles for rotating shaft to vibrate as above-mentioned beam in two orthogonally related planes if correspondent alternating forces are applied. As a result, the shaft rotation axis will move in the space along the beam vibration orbit. **Fig. 5** shows the schematic of whirling vibration of rotating shaft and bearing (in the blue color).

Thus, whirling is a poly-harmonic motion of the rotating shaft caused by two plane excitation forces. Every time when ‘whirling vibration’ term is used with regard to the shaft system of diesel generator we should bear in mind shaft lateral vibrations in two planes simultaneously. The difference between two plane lateral vibration and whirling vibration consists in shaft section proper rotation only.

Sometimes, people concerned with “vibration analysis” of machinery lose sight of a very simple truth: vibration is merely a response to other conditions in a machine, it is not (and should not be) the fundamental concern for the machinery engineer. Instead vibration should be thought of as nothing more than a ratio of the forces acting on the machine to its stiffness. Mathematically, this is summarized as:

$$V(r) = \frac{F}{D \cdot S} \tag{33}$$

A change in unbalance by eccentricity is a force changing in diesel generator shaft system. In this case, force increases, stiffness stays the same, and vibration increases as a result.

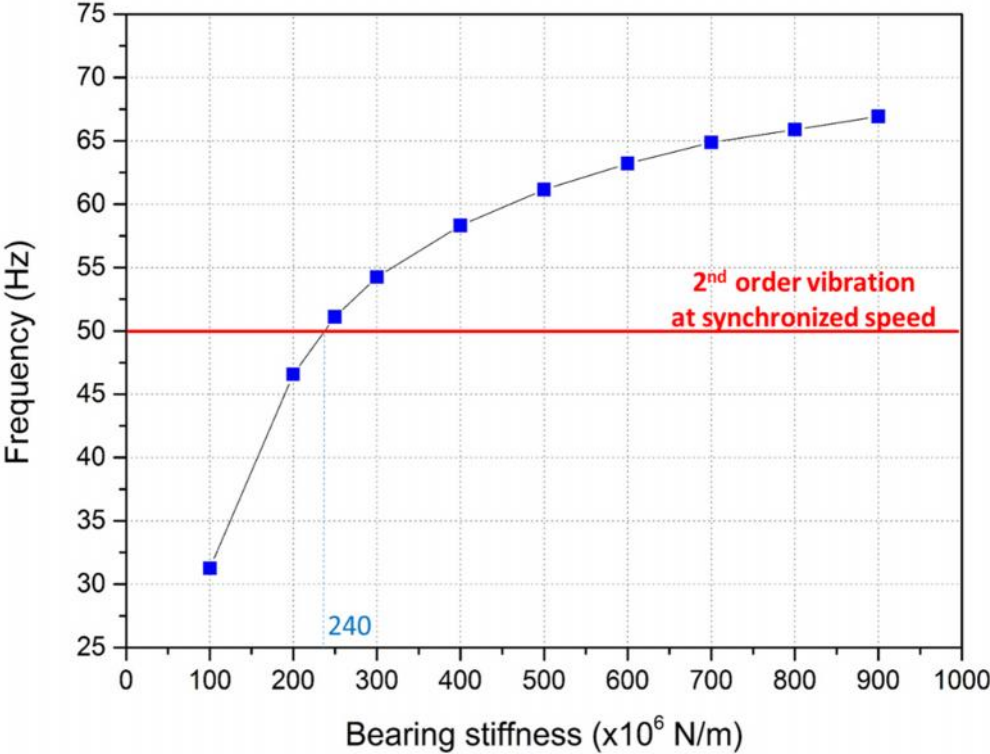
In order to reduce the vibration, along with reducing the exciting sources, adjusting the dynamic stiffness to control the vibration response is the very valuable method. One of methods to adjust the dynamic

stiffness to control shaft whirling vibration of diesel generators is to select suitable pressure for lubricating oil film in bearings.

**2.4 Results and Discussion**

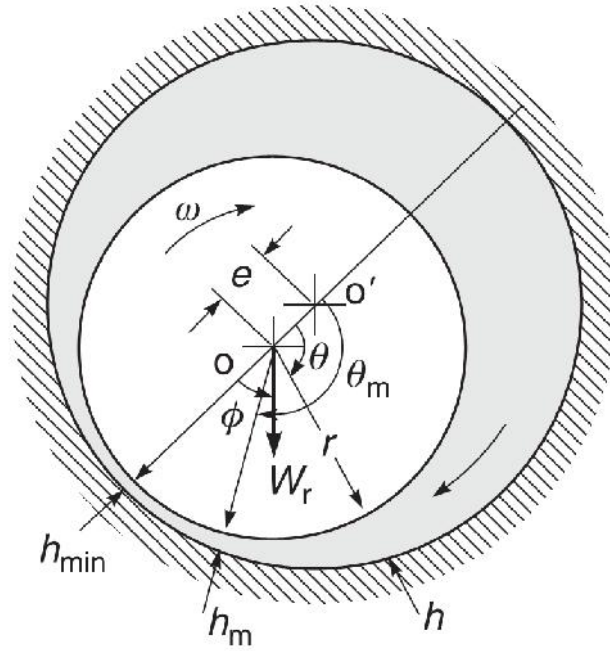
This study concerns about **6 12/14** diesel generator: nominal power = 50 kW; number of cylinder = 6; cylinder bore = 0.12 m; piston stroke = 0.14 m; maximum expansion gas pressure  $p_z = 7.5$  MPa; revolution  $n = 1500$  r.p.m.; shaft diameter  $d = 0.14$  m; length of bearing  $L = 0.075$ m; bearing clearance  $c = 0.1$ mm, lubricating oil viscosity  $\mu = 0.0185$  Pa-s.

According to the local structure vibration measuring results, at synchronized speed (1500 r.p.m.) the vibration is most heavy at 2<sup>nd</sup> order (50Hz - Mode 1). The relationship between bearing stiffness and resonance frequency is shown in **Fig. 6** obtained by theoretical calculation base on masses spring system. According to this figure, the resonance of shafting system intersects the frequency of 2<sup>nd</sup> order vibration at synchronized speed when bearing stiffness is about 240 MN/m. That is critical dynamic stiffness of the shaft bearing. The final goal in this study is identifying the oil film pressure corresponding to critical dynamic stiffness in shaft bearing, on purpose to avoid the 2<sup>nd</sup> order resonance structural vibration of the generator combined with hull foundation from synchronized speed.



**Figure 6** The relationship between bearing stiffness frequency





**Figure 6 Schematics of shaft bearing geometry**

The attitude angle :

$$= \tan^{-1} \left( \frac{\pi \sqrt{1-\varepsilon^2}}{4\varepsilon} \right) \quad (34)$$

The oil film thickness :

$$h \sim c(1 + \varepsilon \cos \theta) \quad (35)$$

At the lowest bottom position ( $\theta_v = 180 - \phi$ ) :

$$h_v = c(1 - \varepsilon \cos \theta_v) \quad (36)$$

The load  $W_r$  can be obtained:

$$W_r = \mu \left( \frac{r}{c} \right)^2 \left( \frac{L}{d} \right)^2 \frac{\varepsilon \sqrt{1 - \varepsilon^2 + \pi^2(1-\varepsilon^2)}}{(1-\varepsilon^2)^2} \quad (37)$$

The bearing stiffness:

$$k_f = \frac{dW_r}{dh} = \frac{W_r(\varepsilon) - W_r(\varepsilon - 0.0)}{h_v(\varepsilon) - h_v(\varepsilon - 0.0)} \quad (38)$$

And average pressure is calculated by:

$$P_a = \frac{W_r}{2r} = \frac{W_r}{d} \quad (39)$$

All of the other parameters can be calculated and are shown on **Table 1**.

According to the calculating results, the critical dynamic stiffness  $240 \times 10^6$  N/m occurs when the eccentricity ratio = 0.456 and the average pressure  $P_{average} = 0.5$  MPa. The critical pressure of lubricating oil film is identified. This critical oil film pressure should be avoided on purpose to reduce the 2<sup>nd</sup> order resonance of structural vibration of generator combined with hull foundation from synchronize speed.

**Table 1 Calculation sheet of lubrication oil film parameters with various oil pressures**

Parameters		Sym.	Unit	Value				
<i>Eccentricity ratio</i>			-	0.3	0.4	<b>0.456</b>	0.5	0.6
<i>Average Pressure</i>		$P_{average}$	MPa	0.24	0.38	<b>0.50</b>	0.61	1.04
<i>Stiffness</i>		$K_f$	$\times 10^6$ N/m	167	200	<b>240</b>	288	498
<i>Load</i>		$W_r$	N	2509	4006	<b>5206</b>	6440	10921
<i>Sommerfeld number</i>		S	-	0.95	0.59	<b>0.46</b>	0.37	0.22
<i>Attitude angle</i>		$\emptyset$	degree	68	61	<b>57</b>	54	46
<i>Minimum oil film thickness</i>		$h_{min}$	mm	0.07	0.06	<b>0.05</b>	0.05	0.04
<i>Lowest position</i>	<i>Location</i>	$\theta_v$	degree	112	119	<b>123</b>	126	134
	<i>Oil film thickness</i>	$h_v$	mm	0.09	0.08	<b>0.08</b>	0.07	0.06
<i>Location of maximum pressure</i>		$m$	degree	119	112	<b>110</b>	109	107
<i>Maximum pressure</i>		$P_m$	MPa	0.51	0.74	<b>0.87</b>	0.98	1.26

### 3. Conclusion

Whirling vibration may result damage on shafting system in some case and the structural vibration of diesel generator acts as the excitation force causing whirling vibration. In **6 12/14** diesel generator, the engine is mounted on the flexible supports and the generator is mounted on rigid foundation. The whirling vibration can be evident when the generator 2<sup>nd</sup> order resonance structural vibration combined with hull foundation accords with synchronized speed and rotor dynamics misalignment.

In order to reduce shaft whirling vibration of diesel generators, it is necessary to select suitable pressure for lubricating oil film in shaft bearing, prevent it from the critical stiffness. In the case of **6 12/14** diesel generator, the critical pressure of lubricating oil film is identified, about 0.5 MPa. By avoiding operation system in this critical oil film pressure condition, the structure vibration is reduced especially 2<sup>nd</sup> order of whirling vibration at synchronized speed.

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