

Whirl of Crankshaft Rear End, Part1: an L6-Cylinder Diesel Engine

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ABSTRACT

As the issue of global warming has become more serious, needs for downsizing or weight saving of an engine has been getting stronger, and forces exerted on engine parts, especially force on a crankshaft, have been getting larger and larger. In addition, since a crankshaft is a heavy engine part, needs for saving weight have been getting stronger and stronger. Therefore, determining the mechanism of high stress generation in a crankshaft system is urgently needed.

This paper describes the characteristics and mechanism of a severe bending stress caused by the whirl of crankshaft rear end of an inline 6-cylinder medium-duty diesel engine. The authors measured bending stress on the fillets of the crankshaft, and found severe levels of sharp peaks in the stress curves for the crankshaft rear. To figure out why the severe levels of sharp peaks appear, they analyzed and studied the measured data. As a result, they have revealed that the crankshaft rear end whirled, and have determined the mechanism of the crankshaft rear end bending resonance characteristic of the shaft whirling motion.

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INTRODUCTION

There are not so many papers which describe the relationship between bending vibration and whirl motion for automotive engine crankshafts. Kimura described the whirl of a torsional damper and the bending vibration at crankshaft front end [1]. Some other papers describe the bending vibration at the crankshaft rear end [2][3][4][5][6]. These papers refer to whirl motion, but they do not describe in detail the relationship between the bending vibration and whirl of the crankshaft rear end.

Novelties of this paper are the following two points:

1. The authors have explained theoretically that the resonance frequency of bending vibration of the crankshaft rear is changed as engine speed is changed if the crankshaft whirled on its rear end. Then, they have revealed in the measurement that the resonance frequency of bending stress on the crankshaft rear is changed according to engine speed because the crankshaft whirled on its rear end.
2. They have also identified the features of crankshaft rear end bending resonance characteristic of the shaft whirling motion, and considering these features, they have theoretically revealed the resonance mechanism.

The characteristics and generating mechanism of severe bending stress caused in the crankshaft rear are described in the following steps:

First of all, this paper describes the theoretical nature of the gyroscopic moment caused by whirl.

Second, this paper explains that, since severe levels of stress were generated in the crankshaft rear, order tracking analysis was performed on the measured bending stresses on the crankshaft rear to find out the cause of severe levels of stress. The analysis results indicate that many resonances of different frequencies and amplitudes occur, and that the curves of amplitudes of individual order bending stresses have sharp peaks at around the resonant engine speeds.

Third, the results of examining the relationship between the resonant frequencies and resonant engine speeds for the measured bending stresses are shown. In the results, such a phenomenon characteristic of the occurrence of whirl motion of the crankshaft rear end is found. Therefore, the occurrence of whirl of crankshaft rear end is confirmed.

Fourth, the results of examining the relationship between the amplitudes of bending stress and exciting force at the resonant engine speeds are reported. In the results, it is found that the stress amplitude is influenced not only by the exciting force, but also by other factors, and that the ratio of the bending stress amplitude to the exciting force amplitude at the resonance engine speed in the case of occurrence of integer-order forward whirl resonance, is approximately 3 times as large as that in the case of occurrence of reverse whirl resonance.

Fifth, this paper reports that the results of examining the relationship between the bending stress amplitude and the flywheel speed fluctuation at the resonant engine speed, indicates that the flywheel speed fluctuation does not influence the bending stress amplitude significantly.

Sixth, the results of examining the relationship between the phases of exciting force and bending stress are described. In the results, it is found that the stress phase delays to the exciting force one in the case of the integer-order forward whirl, while it advances in the case of integer-order reverse whirl.

Seventh, the mechanism of stress phase delay to the exciting force in the case of the integer-order forward whirl and stress phase advances in the case of integer-order reverse whirl is described. Based on the mechanism, the reason why the ratio of the amplitude of bending stress to that of exciting force in the case of integer-order forward whirl resonance is approximately 3 times as large as that in the case of reverse whirl resonance, and the reason why the resonant peaks are sharp are discussed.

Finally, the results of using the three dimensional calculation for the crankshaft rear are shown. In the results, it is confirmed that reducing the crankshaft rear end mass, the distance of its center of gravity center in the crankshaft axis direction, and the inertia moment about the crankshaft rotating axis is effective to reduce vibration in the case of whirl occurrence.

The authors decided that the subject of this research is the severe stress for an engine with power take-off, considering severe levels of sharp stress peaks were found in the curves of measured bending stress on the crankshaft rear in both cases of engines with and without power take-off, and greater stress was observed for an engine with power take-off than for one without it.

GYROSCOPIC MOMENT CAUSED BY WHIRL MOTION

Gyroscopic Moment on Spinning Top

First of all, gyroscopic moment acting on a spinning top is explained.

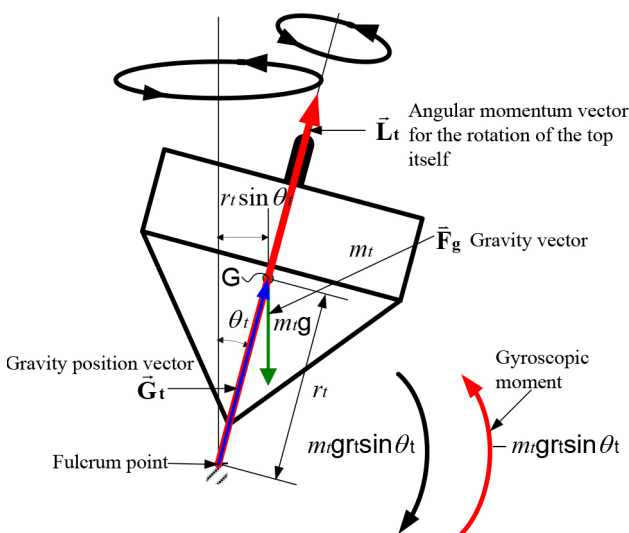


Figure 1. Forward-whirling top

Figure 1 shows a top rotating around its own axis, counterclockwise when viewed from the top of the top, with its axis inclined, where m_t is the mass of the top, G is the center of gravity of the top, θ_t is the

inclination angle, g is acceleration due to gravity, r_t is distance from fulcrum point to the center of gravity of the top, $\vec{L}_t$ is angular momentum vector for the rotation of the top around its own axis, $\vec{G}_t$ is gravity position vector, $\vec{F}_g$ is gravity vector. Angular momentum vector $\vec{L}_t$ is generated in a direction shown in Figure 1 according to the right-hand rule.

With the top inclined, gravity causes moment in the clockwise direction when viewed from the reader, and the moment acts on the top to make it fall. In terms of vectors, the gravity moment vector which acts on the top due to gravity is an outer product of gravity position vector $\vec{G}_t$ and gravity vector $\vec{F}_g$, and thus the magnitude of the gravity moment vector to make the top fall is $m_t g r_t \sin \theta_t$, and the direction of the vector is perpendicular to both $\vec{G}_t$ and $\vec{F}_g$, that is, a direction perpendicular to the paper into the paper, according to the right-hand rule. Since this gravity moment vector is a vector of change in angular momentum for the rotation of the top around its own axis, $\vec{L}_t$, the angular momentum vector $\vec{L}_t$ is changed to the sum of gravity moment vector and angular momentum vector (i.e., the head of angular momentum vector $\vec{L}_t$ is changed in the direction perpendicular to the paper into the paper according to the gravity moment vector). Therefore, the angular momentum vector $\vec{L}_t$ is changed so that the vector whirls in the same direction as the rotation of the top around its own axis, along the $2\theta_t$ angle conic surface, which means the top whirls in the same direction as that of the rotation of the top around its own axis. As a result, the reaction gyroscopic moment $-m_t g r_t \sin \theta_t$ is generated, which is of the same magnitude but in the direction reverse to the gravity moment. The gyroscopic moment balances with the gravity moment, and thus the top keeps whirling as shown in Figure 1.

Whirling in the same direction as that of the rotation of the top around its own axis is called 'forward whirl'. The gyroscopic moment works to reduce θ_t while the top is forward-whirling.

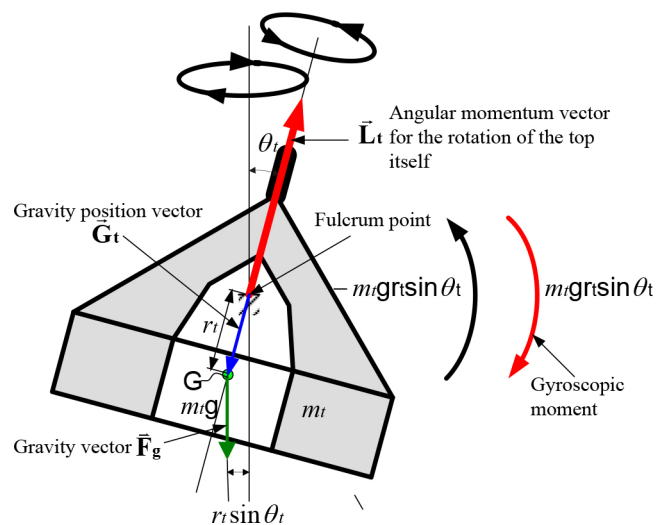


Figure 2. Reverse-whirling top

Figure 2 shows a top rotating around its own axis, counterclockwise when viewed from the top of the top, with its axis inclined, for the top of its center of gravity center located below the fulcrum point. Angular momentum vector $\vec{L}_t$ is generated in a direction shown in Figure 2 according to the right-hand rule.

With the top inclined, gravity causes counterclockwise moment, and the moment acts on the top to make it rise. In terms of vectors, the gravity moment vector which acts on the top due to gravity is an outer product of position vector $\vec{G}_t$ and gravity vector $\vec{F}_g$, and thus the magnitude of the gravity moment vector to make the top rise is $-m_t g r_t \sin \theta_t$, and the direction of the vector is perpendicular to both $\vec{G}_t$ and $\vec{F}_g$, that is, a direction perpendicular to the paper toward the reader, according to the right-hand rule. Since this gravity moment vector is a vector of change in angular momentum for the rotation of the top around its own axis, $\vec{L}_t$, the angular momentum vector $\vec{L}_t$ is changed to the sum of gravity moment vector and angular momentum vector (i.e., the head of angular momentum vector $\vec{L}_t$ is changed in the direction perpendicular to the paper toward the reader according to the gravity moment vector). Therefore, the angular momentum vector whirls in the reverse direction to that of the rotation of the top around its own axis, along the $2\theta_t$ angle conic surface, and the top whirls in the reverse direction to the rotation of the top around its own axis. As a result, the reaction gyroscopic moment $m_t g r_t \sin \theta_t$ is generated, which is of the same magnitude but in the direction reverse to the gravity moment. The gyroscopic moment balances with the gravity moment, and thus the top keeps whirling in the reverse direction to its own rotation as shown in Figure 2.

Whirling in the reverse direction to that of the rotation of the top around its own axis is called 'reverse whirl'. The gyroscopic moment works to increase θ_t while the top is reverse-whirling.

Gyroscopic Moment on Shaft with Disk

With reference to the above example of a spinning top, a model of a whirl-rotating shaft with a disk at its rear end is described next. Figure 3 shows a model of a forward-whirling shaft with a disk, where ω is a whirl angular velocity, θ is an inclination angle, Ω is an angular velocity of the shaft, r is a radius of whirl, M_{gw} is a gyroscopic moment of whirl, F_{cw} is a centrifugal force of whirl. In this figure, gyroscopic moment works to make θ smaller, i.e. make the shaft difficult to bend, while the shaft is forward-whirling.

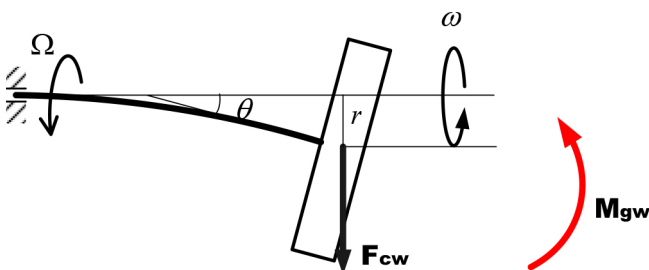


Figure 3. Forward-whirling shaft with disk

On the other hand, Figure 4 shows the occurrence of reverse whirl, where gyroscopic moment works to make θ_t larger, i.e. make the shaft easy to bend, while the shaft is reverse-whirling.

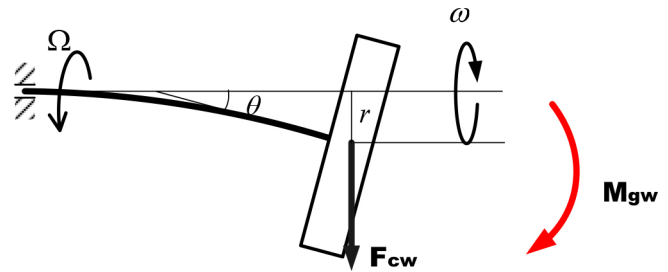


Figure 4. Reverse-whirling shaft with disk

While the shaft is whirling, a gyroscopic moment M_{gw} and a centrifugal force F_{cw} act from the disk on the shaft. In the case of the shaft angular velocity Ω , the whirl harmonic order n , and angular velocity of whirl $\omega = n\Omega$, M_{gw} and F_{cw} are expressed by the following Equations (1) and (2) respectively [7].

$$M_{gw} = n\theta(nI_d - I_p)\Omega^2 \quad (1)$$

$$F_{cw} = mr(n\Omega)^2 \quad (2)$$

Where,

n = whirl order

I_d = inertia moment of the disk about the axis perpendicular to shaft rotating axis

I_p = inertia moment of the disk about the shaft rotating axis

m = mass of the disk

Equations (1) and (2) indicate that gyroscopic moment is proportional to inclination angle θ of the shaft rear end, and centrifugal force is proportional to displacement r of the shaft rear end, that is, the moment and force have the same characteristics as those of elastic bending force on a shaft. Therefore, both of the moment and force influence the bending stiffness of the shaft with a disk.

Moreover, both of them are proportional to the square of disk angular velocity Ω as shown in Equations (1) and (2).

Therefore, in the case of forward whirl, the gyroscopic moment acts to increase bending stiffness, and thus the whirl resonant frequency increases as the disk angular velocity increases. On the other hand, in the case of reverse whirl, the gyroscopic moment acts to decrease bending stiffness, and thus the whirl resonant frequency decreases as the disk angular velocity increases.

Figure 5 shows the whirl motion of crankshaft rear end at intervals of a quarter of the whirl period.

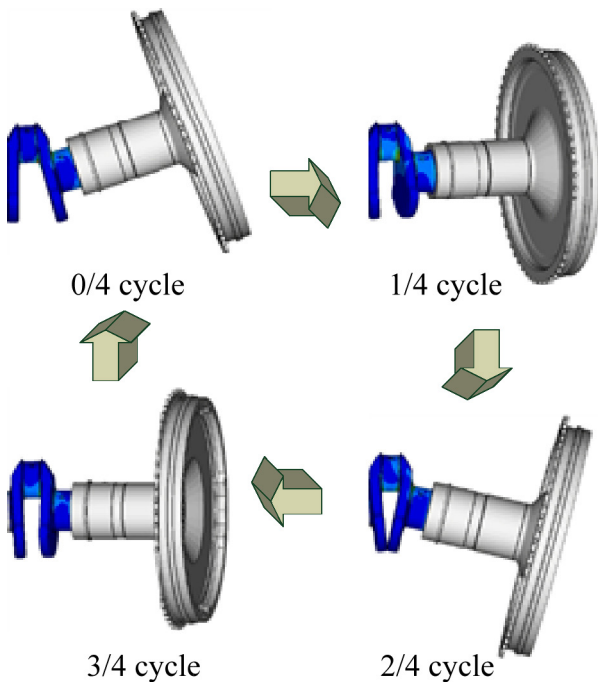


Figure 5. Whirl motion of crankshaft rear end

EXPERIMENT RESULTS AND ANALYSIS

Engine Specifications

Table 1 shows the specifications of an in-line 6-cylinder medium-duty diesel engine used for this study

Table 1. Engine specifications

Engine type	4-stroke, turbo-charged diesel
Cylinder layout	6 cylinders in line
Bore x Stroke	$\phi 115 \times 125$ mm
Displacement	7.790 ℓ
Firing order	1-5-3-6-2-4
Maximum power	221 kW at 2400 rpm
Power take-off (PTO)	With PTO

Experiment Setup and Procedure

Measurement Devices

Strain gauges were attached at the locations of fillets of individual crank pins and journals in a plane including the respective center axes of crank pins and journals, in such a direction (hereinafter referred to as 'pin direction') that stress caused by bending vibration in the plane, can be measured. The gauge locations in the crankshaft rear are shown in Figure 6

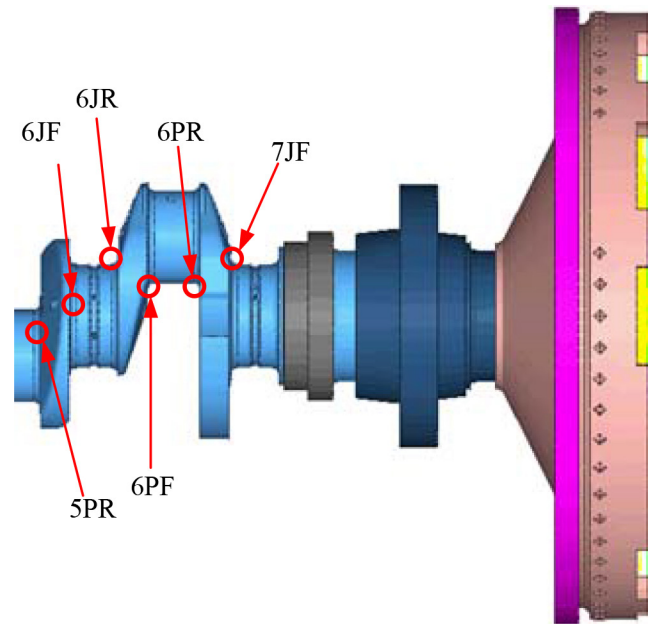


Figure 6. Measurement locations in crankshaft rear

Lead wires were threaded through the crankshaft, and the gauge signals were picked up through a slip ring mounted at the crankshaft front end. The measured strains were converted to stresses.

As shown in this figure, the gauge location of front fillet of No.7 journal is hereinafter referred to as '7JF', the gauge location of rear fillet of No.6 pin as '6PR'. In the same way, the other gauge locations are hereinafter referred to as '6PF', '5JR', '5JF', and '5PR'.

Piezo type pressure sensors were installed in the cylinder head of the test engine to measure cylinder pressure.

Test Conditions

The engine was mounted on a test bench, and stress measurement was carried out with engine speed changed at 25 rpm intervals from 800 rpm to 2450 rpm under full load condition.

Measurement Results

The bending stress amplitudes measured at the locations 5PR, 6PF and 6PR under full load condition are shown in Figure 7. In this figure, no significant peak of bending stress appeared for 5PR, but small stress peaks appeared for 6PF. For 6PR, severe levels of many sharp peaks appeared. This indicates that the stress level becomes higher as the measurement location is moved rearward, that is, a bending vibration, whose stress amplitude greatly changes as engine speed changes, is generated at the crankshaft rear end.

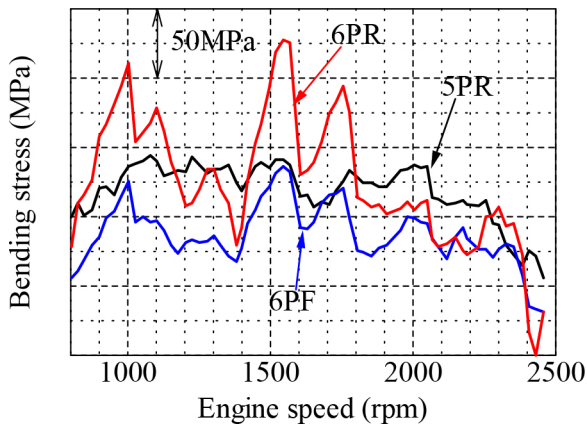


Figure 7. Measured bending stress in crankshaft rear

Analysis Results

Order Tracking Analysis

Considering there is a phenomenon specific to the crankshaft rear, the authors performed order tracking analysis on the 6PR full-load data to analyze the phenomenon. The analysis was performed at 0.5th order intervals, and all the large stress amplitude components were found in the range of the 0.5th to 4th order. The results for the 0.5th to 2nd order are shown in Figure 8, and those for the 2.5th to 4th order are shown in Figure 9.

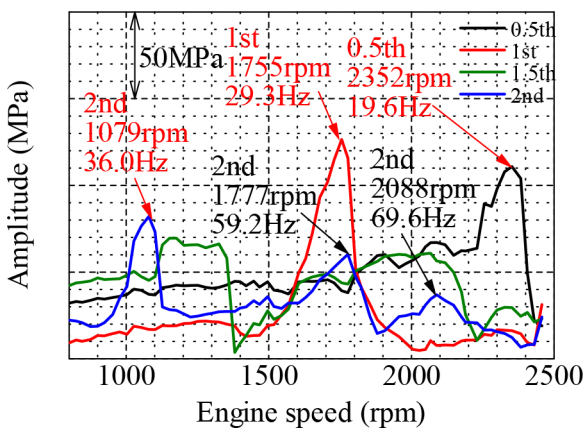


Figure 8. 0.5th to 2nd order bending stress amplitudes for 6PR

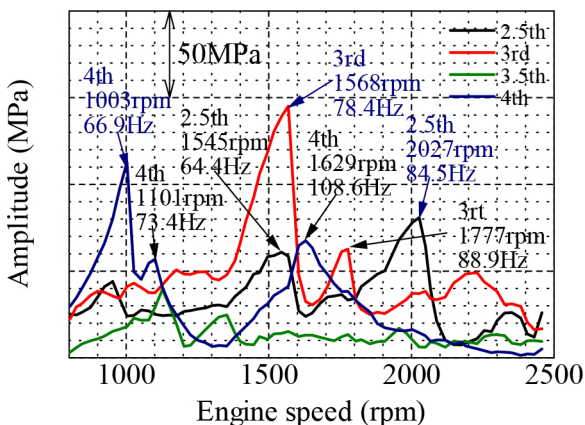


Figure 9. 2.5th to 4th order bending stress amplitude for 6PR

Table 2 summarizes the characteristics of bending stress found in the order tracking analysis results. The authors further analyzed the characteristics to find out the cause of bending stress generation.

Table 2. Characteristics of bending stress found in order tracking analysis

No.1	There are many peaks which are of different resonant frequencies.
No.2	Peak amplitude for the 3rd order is the largest, and that for the 1st order is the second largest.
No.3	The peaks are sharp. For the 2.5th or higher orders, a small peak appeared at a faster engine speed than that for the highest stress peak.

Investigation of Cause of Characteristics Found in Order Tracking Analysis Results

Relationship between 6PR Bending Stress Resonance Frequency and Engine Speed

Figure 10 shows relationship between engine speeds and frequencies for the 6 highest stress amplitude peaks found in Figures 8 and 9

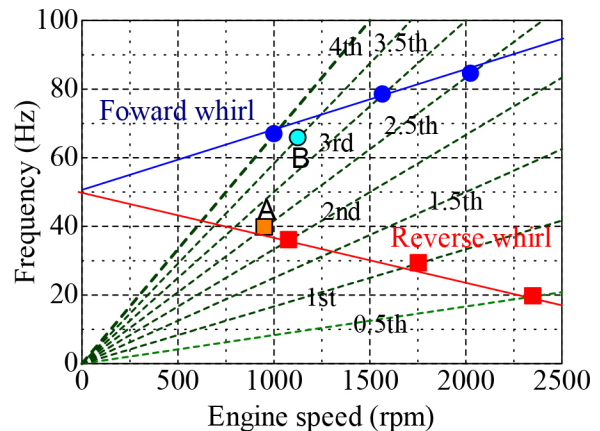


Figure 10. Engine speeds and frequencies for the 6 highest stress amplitude peaks

As seen in this figure, the plots of resonant frequencies for the stress peaks are found almost on the two straight lines. This means that the resonant frequency increases or decreases as resonant engine speed increases, from around 50 Hz at an engine speed of 0 rpm.

As mentioned above, if a crankshaft rear end is whirling, the following phenomena occur:

In the case of forward whirl occurrence, gyroscopic moment acts in such a direction that the moment makes it difficult for the crankshaft rear end to bend, and bending stiffness increases, i.e. the resonant frequencies increases as engine speed increases. In the case of reverse whirl occurrence, gyroscopic moment acts in such a direction that the moment makes it easy for the crankshaft rear end to bend, and bending stiffness decreases, i.e. the resonant frequencies decreases as engine speed increases.

The peak frequencies being on the two straight lines in Figure 10 agrees with the above whirl phenomenon, and this shows that the crankshaft rear end is whirling.

The frequency at the intersection of the two straight lines at around 50 Hz is considered as a natural frequency of bending vibration in the pin direction at the crankshaft rear end.

From this result, the characteristic phenomenon No.1 in Table 2 reveals to be the resonant frequency change caused by whirl.

Next, the authors investigated the cause of No.2 and No.3 characteristic phenomena in Table 2. They focused on the amplitudes of 0.5th to 4th order bending stresses at resonant engine speeds. Figure 10 indicate that forward whirl resonances occur in 2.5th or higher order, and the largest amplitude was that for the 3rd order, the second was that for the 4th order, and the third was that for the 2.5th order. On the other hand, reverse whirl resonance occur in 2.5th or lower order, and the largest amplitude was that for the 1st order, the second was that for the 0.5th order, the third was that for the 2nd order, and the fourth was that for the 2.5th order.

In the investigation, they studied the relationship between the amplitude of 6PR bending stress at resonant engine speed and the radial force which is the exciting force for the bending vibration (the radial force is hereinafter referred to as “exciting force”) because the exciting force is considered the most influential on the bending stress amplitude. They also studied the relationship between the 6PR bending stress and the flywheel speed fluctuation because the fluctuation is considered influential on the magnitudes of the gyroscopic moment and centrifugal force generated by whirl.

Relationship between Amplitudes of 6PR Bending Stress and Exciting Force

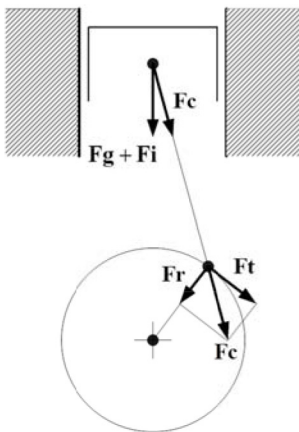


Figure 11. Radial force

Figure 11 shows the dynamics of crank mechanism, where F_g is a force generated by cylinder pressure, F_i is an inertia force, F_c is a connecting rod force, F_t is a tangential force acting on the crank pin, and F_r is a radial force acting on the crank pin.

To determine the amplitudes of individual order exciting forces, the authors first of all calculated the radial force F_r shown in Figure 11 by using the Ricardo/VALDYN software model with the measured cylinder pressures, and then order tracking analysis was performed on the calculated F_r .

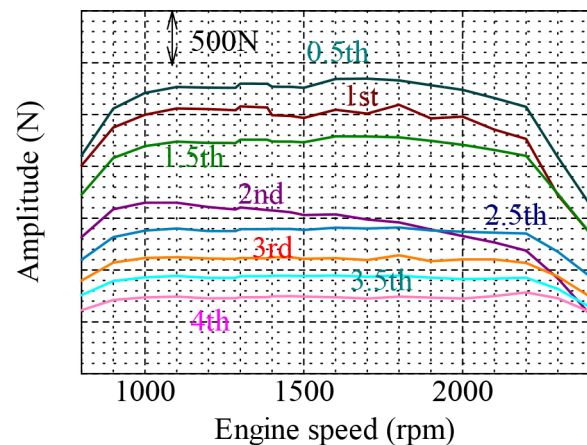


Figure 12. Radial force order tracking analysis

VALDYN calculation model was refined by determining the crank damper spring constant, its damping coefficient, and engine viscous damping coefficient in such a way that the measured torsional torques on individual crank journals (measured at the same time as the bending stress) accurately agree with those calculated by using the VALDYN model.

Figure 12 shows the results of order tracking analysis for the 0.5th to 4th orders. This figure indicates that the amplitude of the 0.5th-order exciting force is the largest, and the amplitude generally becomes smaller as the order increases.

To know whether 6PR stress amplitude at the resonant engine speed is significantly influenced by the exciting force or not, the responsive ratios of bending stress amplitude to exciting force amplitude (hereinafter referred to as “S/E ratio”) were calculated.

S/E ratios for the six largest components of bending stress at resonant engine speeds were calculated. In addition, S/E ratios were calculated also for the plots A and B in Figure 10.

Figure 13 shows the relative S/E ratios with reference to that for the 3rd order, where those for forward whirl is shown in blue, and those for reverse whirl in red.

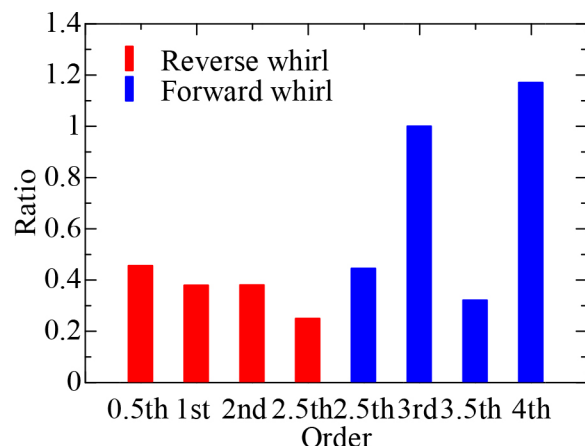


Figure 13. Relative S/E ratios (S/E ratio for 3rd order = 1)

As seen in Figure 13, the responsive S/E ratios for integer-order forward whirl are larger than those for the others, and the resonances can be separated into 2 groups, which are the integer-order forward whirl resonances and the other resonances.

For the integer-order forward whirl resonance group, the responsive ratio ranges from 1 to 1.2, and, for the other resonance group, the ratio ranges from 0.2 to 0.5. The ratios have some variations, but in a small range for both groups respectively. Therefore, the exciting force seems to influence the bending stress amplitude.

Figure 14 shows the ratio of exciting force amplitude to the 3rd-order one in the horizontal axis, and the ratio of bending stress amplitude to the 3rd order one in the vertical axis, and ratios for forward whirl are shown by the blue circle plots, and those for reverse whirl by the red square plots.

As can be seen in Figure 14, the plots for the two groups are almost on the respective straight lines inclined at almost the same angle, and it seems that some other unknown force or moment than the exciting force works positively in the case of the integer-order forward whirl, and negatively in the other cases.

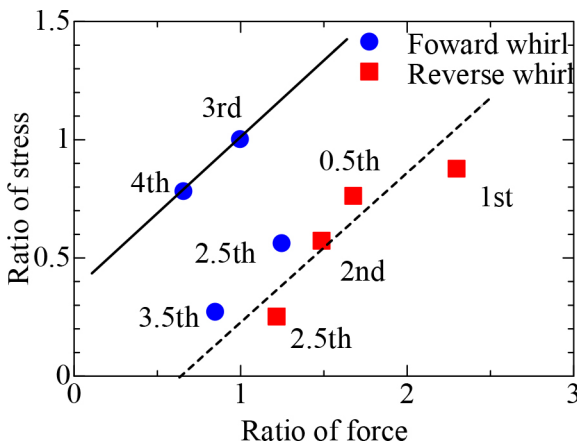


Figure 14. Bending stress ratios versus exciting force ratios

From these results, the authors thought that cylinder pressure significantly influences the amplitudes of 6PR individual order bending stresses at the resonant engine speeds, and that some other force or moment may exist to influence 6PR bending stress.

Relationship between Crankshaft Rear End Rotating Speed Fluctuation and 6PR Bending Stress

Equations (1) and (2) indicate that the magnitudes of gyroscopic moments and centrifugal forces caused by individual order whirl components are proportional to the square of engine speed. So the authors thought the 6PR bending stress amplitude might be affected by the crankshaft rear end rotating speed fluctuation, that is, the flywheel rotating speed fluctuation, and thus studied the relationship between the flywheel speed fluctuation and the 6PR bending stress amplitude.

Flywheel rotating speed fluctuation at each engine speed was calculated by using the Ricardo/VALDYN software model with the measured cylinder pressures, and then order tracking analysis was performed. Figure 15 shows the results of order tracking analysis.

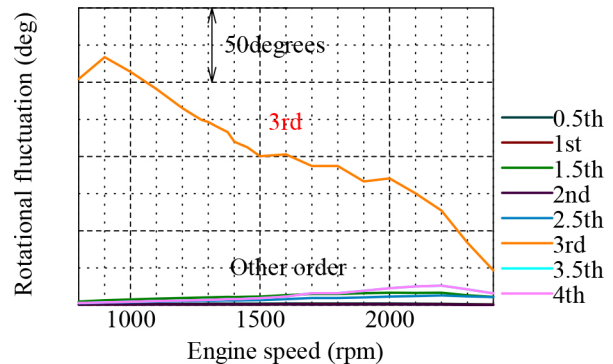


Figure 15. Flywheel rotating speed fluctuation order tracking analysis

This figure indicates that, in comparison with the 3rd-order flywheel rotational fluctuation amplitude, the other order amplitudes are very small. Considering the large amplitudes of bending stress components other than that of the 3rd order one, there is no significant correlation between the individual order bending stress amplitudes and the respective order flywheel speed fluctuation amplitudes.

Relation of 6PR Bending Stress Phase to Exciting Force Phase

The phase of exciting force and that of 6PR bending stress were studied to find the other factors that influence 6PR bending stress.

Phase difference of 6PR bending stress to exciting force was calculated from the results of order tracking analysis on the respective measured and calculated data. Positive phase difference means that 6PR bending stress phase advances from the exciting force, and thus 6PR bending stress increases before exciting force is exerted. On the other hand, negative phase difference means that 6PR bending stress phase delays from the exciting force, and thus 6PR bending stress increases after exciting force is exerted.

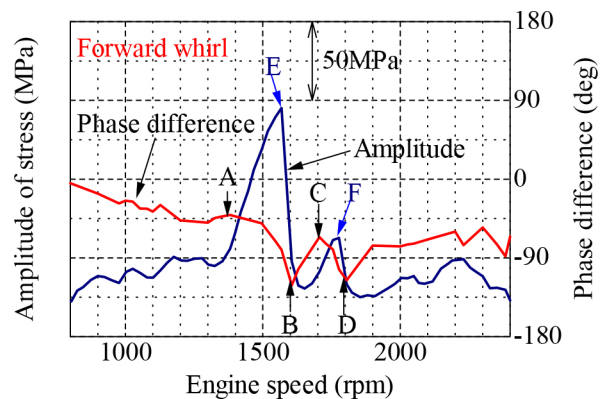


Figure 16. 6PR 3rd-order stress amplitude and its phase difference to exciting force

Figure 16 shows the 3rd-order bending stress amplitude, which represents integer-order forward-whirl bending stresses, and its phase difference to the exciting force.