

Study of Noise Reduction for an Overdriven Compact Oil Pump

Koji HIRANO*

Yasunori ONO**

Hisayuki HASHIMOTO*

Yoshiaki SENGÅ**

Masakazu KINOSHITA*

ABSTRACT

An issue involving noise reduction due to an abnormal noise from the oil filter arose during the development of an overdriven compact oil pump. This issue was approached by focusing on the amplitude of the oil pressure fluctuation at the pump discharge side (hereafter oil pressure fluctuation amplitude) which was the excitation force. Noise reduction was implemented using technology to reduce the oil pressure fluctuation amplitude. This involved revising the number of teeth in the rotor and changing the characteristics of the opening of the relief valve.

A small-diameter, multi-toothed compact rotor was developed to take full advantage of compactness and low friction for the 2003 Accord. The number of teeth in the inner rotor was increased from four to eight, thus reducing the oil pressure fluctuation force. The relief valve was also given additional pre-relief holes in order to improve the response at the valve opening timing, with the objective of improving the product's marketability with respect to noise and vibration.

1. Introduction

The emphasis on environmental issues in recent years has prompted efforts to greatly improve fuel consumption in automobile engines. Weight reduction, lean-burn combustion and the reduction of mechanical friction in respective parts are sited in support of technologies for improving fuel consumption.

In seeking ways to build more compact engines, the reduction of the overall length has contributed to the improved degree of freedom in vehicle installations.

Overdriven compact oil pumps are an effective technology for realizing reduction of mechanical friction and compact design. However, there are many issues with noise stemming from high speed driving and the small number of teeth on the rotor.

In this development project, abnormal noise radiating from the oil filter occurred at a certain level of engine speed after the engine was completely warmed up. This abnormal noise was transmitted to the vehicle cabin and significantly reduced the marketability of the vehicle.

In seeking countermeasures to resolve this issue, the mechanisms generating the abnormal noise were clarified and a technology was developed to reduce the noise.

2. Aim of Research

To highlight the main topics, Fig. 1 illustrates a block diagram of the oil circuit and Fig. 2 lists the assumptions regarding the mechanisms that generate abnormal noise around the filter.

Since the abnormal noise is a phenomenon unique to the overdriven compact oil pump and it was observed at a particular speed range, it was believed that associated responses such as the oil pressure fluctuation amplitude as excitation forces and resonance mechanism generated at a particular speed range contributed to the abnormal noise.

Regarding excitation forces, in comparison with conventional crankshaft direct drive type uniform speed pumps (hereafter, called coaxial drive pumps), the rotor size for overdriven compact oil pumps is smaller and the changes

* Tochigi R&D Center

** Yamada Seisakusho Co., Ltd.

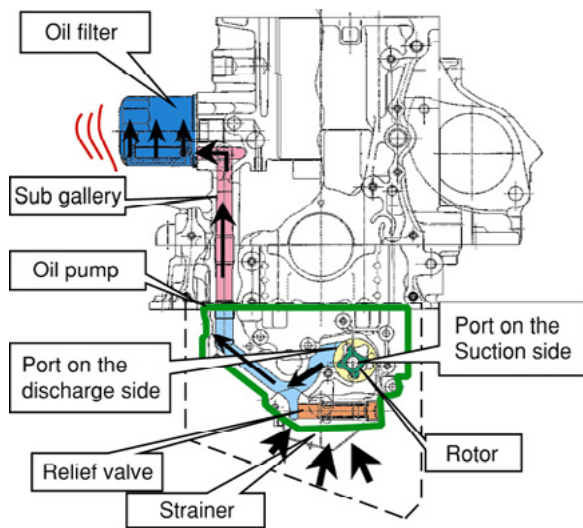


Fig. 1 Block diagram of the oil circuit

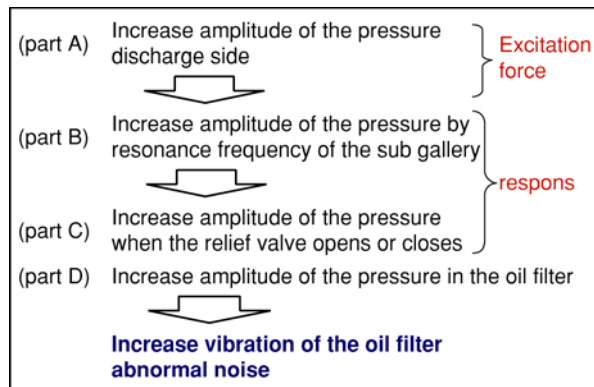


Fig. 2 Assumptions regarding the mechanism of abnormal noise from the oil filter

in pressure are bigger for each unit time. In effect, this means that the pressure fluctuation amplitude is larger than on conventional configurations.

The next issue regarding such influential responses drew attention to the oil pressure resonance and response characteristics of the relief valve. An oil pressure resonance occurred inside the flow passage from the oil pump to the oil filter and caused the oil pressure fluctuation amplitude to increase. When the peak pressure in the flow passage reaches the opening valve pressure, the relief valve begins to stroke. The relief valve is a valve for adjusting pressure. Therefore, the valve opens and closes repeatedly to adjust the pressure inside the flow passage in close proximity to the relief pressure. Particularly, when the valve closes, an oil hammer occurs due to the sudden stop of oil flow. This, it is believed, leads to a further increase in pressure fluctuation amplitude inside the flow passage and excites the filter and thus leads to the generation of abnormal noise.

This topic seeks to verify these described assumptions and develop a technology for reducing noise by taking advantage of the low friction merits embodied in the overdriven compact oil pump.

3. Basic Structure

The developed overdriven compact oil pump, which is attached to the lower engine block, uses an internal gear

trochoid type rotor similar to that used in typical coaxial type drive pumps. Whereas the outer dimensions of the coaxial outer drive rotor is approximately 80 mm, the dimensions of the rotor for the overdriven compact oil pump have been reduced to 40 mm to achieve compactness.

The crankshaft coaxial type drive rotor is driven at the same speed as the engine. However, because the theoretical oil discharge per single rotor turn for overdriven compact oil pumps is small, the theoretical discharge per engine rotation has been adjusted to approximate that of the crankshaft coaxial pump by increasing the rotor speed. This results in a rotor speed increase of 1.6 to 2 times over the crankshaft by silent chain.

Figure 3 shows a comparative view of the described compact overdriven pump and a conventional pump. Table 1 lists the comparative oil pump specifications. Figure 4 illustrates the oil circuit system including the overdriven compact oil pump.

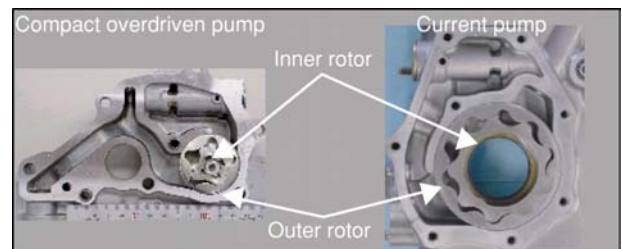


Fig. 3 Comparison of the structures

Table 1 Specification of the oil pump

	Overdriven compact oil pump	Current oil pump
Over driven ratio	Double speed	Single speed
Number of the teeth	Inner 4 / outer 5	Inner 9 / outer 10
Rotor size	$\Phi 40 \times 15.0t$	$\Phi 80 \times 6.0t$
Delivery oil (cc) (at round pump)	6.12	12.2
Delivery oil (cc) (at round crankshaft)	12.2	12.2

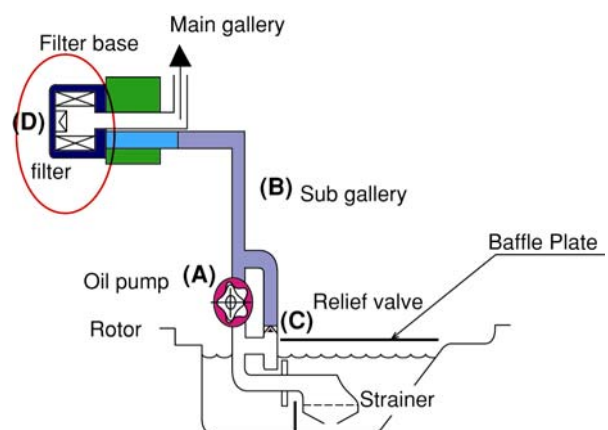


Fig. 4 Oil circuit system of the engine

This oil pump is attached to the lower engine block and is situated in the oil pan. The chain driven rotor conveys the oil to the sub gallery from the strainer via the discharge side port. The oil, which has passed through the sub gallery, is filtered through the oil filter and is then supplied to the journal and cylinder head via the main gallery. When the pressure on the discharge side, including the sub gallery, exceeds the set pressure value, the relief valve begins to operate and the excess oil is returned to the intake controller port via the relief valve passage.

4. Verification of Mechanisms Causing Abnormal Noise

4.1 Understanding the Issue

The abnormal noise occurs at particular engine speeds once the engine has been warmed up and the oil temperature exceeds 90 °C. The abnormal noise radiates mostly from the filter.

Figure 5 shows the measured results of the abnormal noise level measured near the filter.

From the measured results, it was learned that the main cause of the noise was attributed to the oil pump meshing order components and the total order frequency unexpectedly increased. Figure 6 shows the measured results of the pressure fluctuation amplitude on the discharge side of the oil

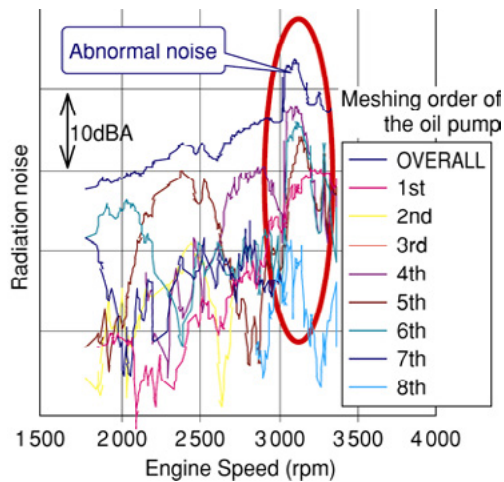


Fig. 5 Sound pressure level nearby filter

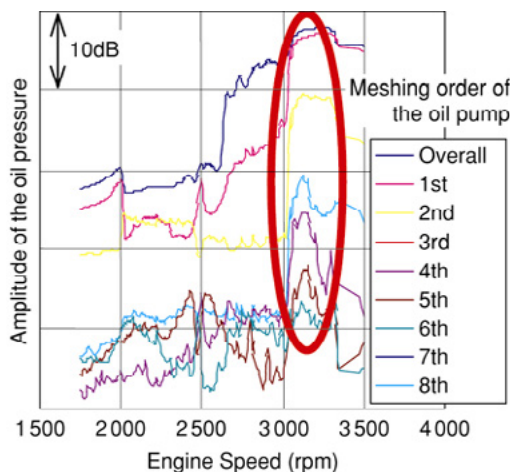


Fig. 6 Oil pressure fluctuation on the discharge side

pump, which was thought to be the source of the excitation force.

This also showed the same results as the filter radiation noise. The noise on the filter occurs at particular engine speeds and oil temperature. Therefore, the noise can be caused not only by the excitation force due to oil pressure amplitude on the discharge side of the oil pump but also by the total response system which creates resonance. In verifying the mechanisms that cause the noise, the following points needed to be investigated:

- Excitation forces effect – oil pressure fluctuation amplitude caused by the discharged oil from the rotor
- Responses system effect – effect by resonance that occurs inside the sub gallery flow passage and effect by the relief valve operation.

4.2 Introducing Unit Test Bench Analysis

Once it was known that the abnormal noise radiated only from the filter surface, the next step was to verify the mechanism. However, since it would be difficult to conduct testing by unit engine test to evaluate only the mechanisms at issue, shown in Section 2, a new unit test bench was devised to satisfy the following three conditions while monitoring the noise that occurred near the filter.

- (1) To verify resonance inside the oil flow passage, the passage shape can be changed.
- (2) Relief valve motion can be verified.
- (3) Pressure fluctuation amplitude on the oil circuit can be easily measured.

Figure 7 outlines the unit test bench. The unit test bench consists of an oil pump attached to a jig which is configured to the sub gallery and filter base shapes. The motor then drives the pump and the pressure within the flow passage is set with the regulating valve configured to the filter after flow.

To measure the excitation force caused by pressure fluctuation, and the resonance inside the oil flow passage, a pressure sensor was fitted to the oil pump and inside the flow passage while a gap sensor was fitted to measure the motion of the relief valve. The results gained from the unit bench test verified the same abnormal noise occurrence as the test on the actual vehicle.

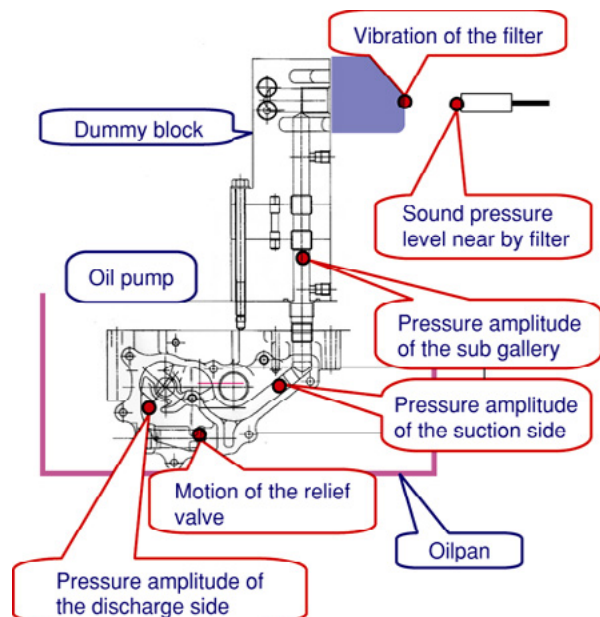


Fig. 7 Test bench

4.3 Effects of Excitation Force

The first thing to be thought of as the excitation force is the oil pressure fluctuation amplitude brought on by the oil discharged from the rotor. This is based on the supposition that this is affected by the theoretical discharge volume per rotor tooth. Note here that this value is the theoretical discharge volume per engine rotation divided by the number of teeth on the oil pump rotor that mesh in a single engine rotation.

Specifically, to diminish the oil pressure fluctuation amplitude, it was believed necessary to increase the number of rotor teeth on the pump producing a large volume of discharge flow.

The unit analysis bench test shown in Fig. 7 was applied to measure the excitation force first. Figure 8 shows the comparative results of respective oil pressure fluctuation amplitudes on the overdriven compact oil pump and the coaxial drive pump. The oil pressure fluctuation amplitude on the overdriven compact oil pump generates what is thought to be a peak resonance level at around 3,000 rpm. However, the amplitude on overdriven compact oil pumps is approximately twice as large over that on conventional coaxial drive pumps in the speed range below 2,000 rpm which was not included in the range affected by resonance. As shown in Table 2, the overdriven compact oil pump generates 13% more oil discharge volume per rotor tooth in comparison with that generated by the conventional coaxial drive pump which naturally leads to an estimated increase in pressure fluctuation amplitude. However, it was difficult to predict that the pressure fluctuation amplitude would actually be twice that of the test results indicated.

Based on these test results, the pump's drive speed, in addition to the theoretical oil discharge volume per rotor tooth, is also believed to cause the increase in pressure amplitude which is the excitation force on the pump.

To verify the actual measurement results, the pressure fluctuation amplitude of the oil discharged from the rotor was estimated by calculation.

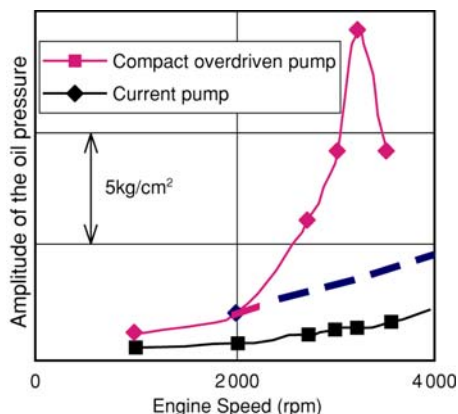


Fig. 8 Result of the pressure fluctuation amplitude measured on the discharge side

Table 2 Delivery oil per rotor tooth

	Overdriven compact oil pump	Current oil pump
Over driven ratio	Double speed	Single speed
Number of the teeth	Inner 4 / outer 5	Inner 9 / outer 10
Delivery oil (cc) (at round crankshaft)	12.2	12.2
Delivery oil (cc) (at rotor tooth)	1.53	1.35

As shown in Fig. 9, the cell located between the inner rotor and outer rotor is filled with oil. As the rotor turns, the cell volume changes which leads to oil being discharged from the cell. To simplify the calculation, it was supposed that the cell was filled to capacity with oil all the time. Also, the oil was treated as a fluid devoid of any pressure influences.

The cell volume is highlighted by the grey area shown in Fig. 9 and is indicated by V . If the rotor's maximum volume rotation angle indicated by θ is zero, the cell's volume changes in accordance with the rotation angle as shown in Fig. 10. The amount of oil flow per rotor tooth discharged from the rotor is expressed by Q , the volume of the rotor cell by V and the rotor rotation angle by θ ,

$$Q(\theta) = V_{\max} - V(\theta) \quad (1)$$

The amount of oil flow per rotor tooth discharged at the moment from the rotor where $\theta = \omega t$ is expressed by the following formula.

$$\frac{dQ}{dt} = -\frac{dV}{dt} = -\omega \frac{dV}{d\theta} \quad (2)$$

The total volume of oil flow was calculated according to the total number of rotor teeth and then that value is substituted in the formula^{(1), (2)} for pulsation ratio ε .

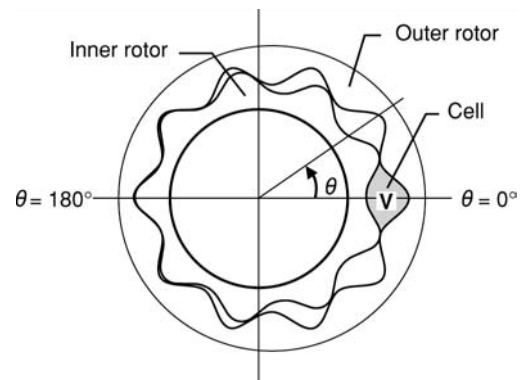


Fig. 9 Model of the oil pump rotor

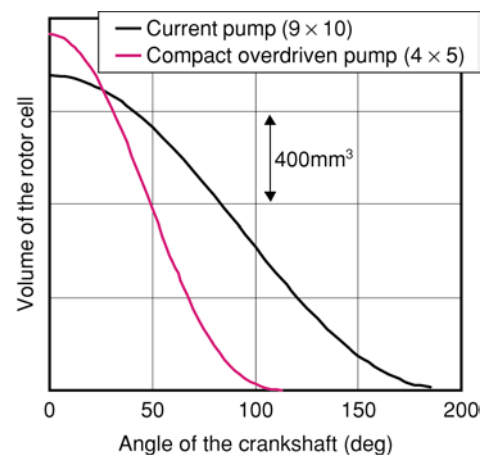


Fig. 10 Rotor cell volume (calculation)

$$\varepsilon = \left\{ \left(\frac{dQ}{dt} \right)_{\max} - \left(\frac{dQ}{dt} \right)_{\min} \right\} / \left(\frac{dQ}{dt} \right)_{\text{mean}}$$

$$= \sum_{i=1}^N \left[\left\{ \left(\frac{dV_i}{d\theta} \right)_{\max} - \left(\frac{dV_i}{d\theta} \right)_{\min} \right\} / \left(\frac{dV_i}{d\theta} \right)_{\text{mean}} \right] \quad (3)$$

Q : amount of oil flow
 V : cell volume
 $V_{\max}$: maximum value for cell volume
 θ : angle of rotor rotation
 ω : rotor angular velocity
 N : the number of meshed sprocket teeth

The pulsation rate is expressed in formula (3) as the change in cell volume. Figure 11 shows a comparative evaluation of rotor cell volume change per rotor tooth for an overdriven compact oil pump and coaxial drive pump. Figure 12 shows the cell volume change according to crankshaft rotation angle.

In comparing the changes in cell volume, it was shown that the change in cell volume for overdriven compact oil pumps was virtually double. These results also matched the results yielded in the unit analysis bench test. The pressure pulsation ratio, expressed by ε , is calculated in formulas^{(1), (2)} using the number of rotor teeth as a parameter. Figure 13 shows the calculation results of the pressure pulsation.

From these calculation results, it was learned that in order to make the pressure pulsation ratio in overdriven compact oil pumps equal to that in coaxial drive pumps, the number of rotor teeth needed to be double that of the current rotor configuration, that is, 8 on the inner rotor and 9 on the outer rotor.

However, with the current method of designing a rotor,

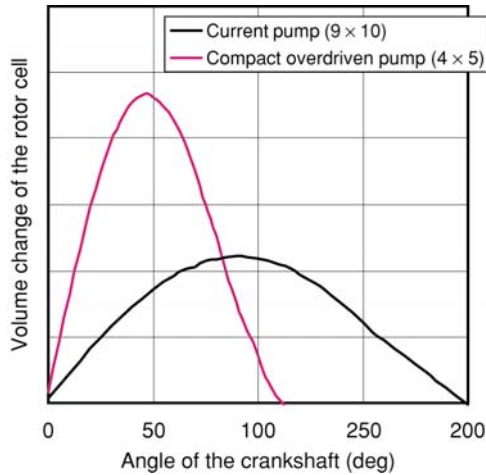


Fig. 11 Volume change of the rotor cell (calculation)

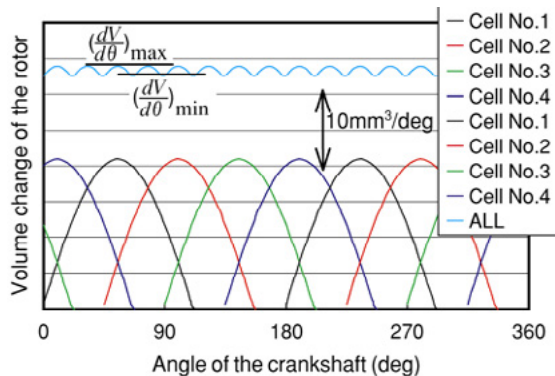


Fig. 12 Volume change of the rotor (calculation)

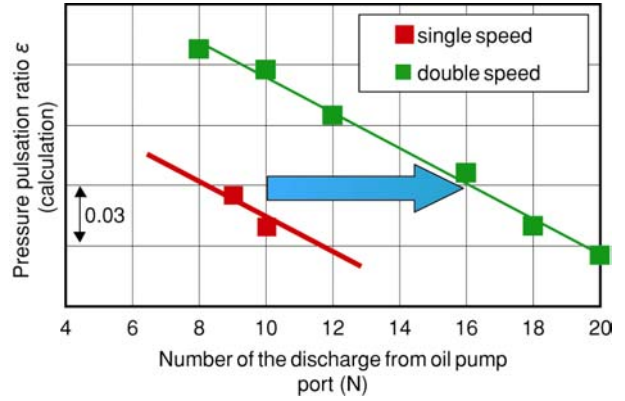


Fig. 13 Result of calculation of the pressure pulsation

which is based on the trochoid curve, the increase in the number of rotor teeth also increased the degree of friction linked to the increase in the rotor's outer dimension.

Figure 14 shows the relationship between the friction of the oil pump and the pressure amplitude. To significantly diminish the friction, a small diameter multi toothed rotor was developed. The developed rotor measures 45 mm in diameter and has 8 teeth on the inner rotor and 9 teeth on the outer rotor. The new rotor was designed to reduce the contact area on the sliding rotor parts to achieve a reduced level of friction. Figure 15 illustrates the structural design for reducing the contact area. Figure 16 shows the effect of the reduced friction on the oil pump.

This design modification has taken advantage of the merits of the low friction characteristics associated with compact rotors while achieving a 60% reduction in pressure fluctuation amplitude.

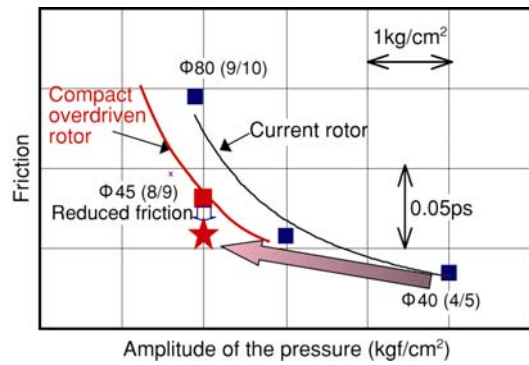


Fig. 14 The relation between pressure fluctuation amplitude and friction of the oil pump

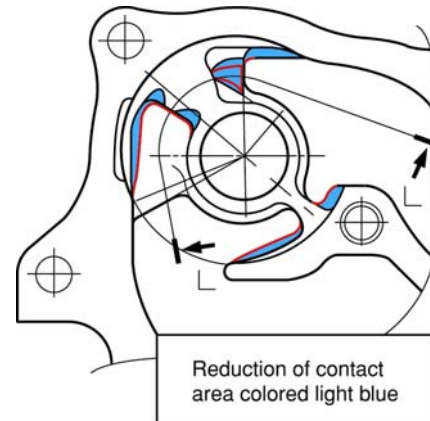


Fig. 15 Effect of reduced friction of the oil pump

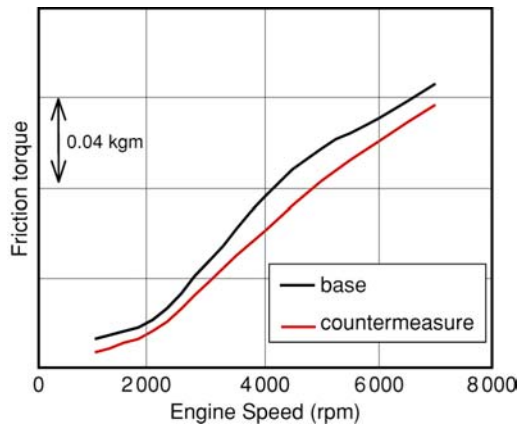


Fig. 16 Effect of reduced friction of the oil pump

4.4 Effects of the Response System

The effect of the response system was investigated by focusing on the resonance occurrences inside the sub gallery flow passage and the influences from the relief valve.

The occurrence of abnormal noise drew attention to the occurrence of resonance on the sub gallery flow passage generated at a certain level of motor speed and at certain oil temperature levels. The sub gallery flow passage length was used as parameter in the unit test bench. Figure 17 shows the pressure amplitude at the sub gallery.

These results showed that by changing the sub gallery passage length, indicated by L, the corresponding change in peak of pressure fluctuation amplitude inside the flow passage also changed and was affected by the resonance on the flow passage.

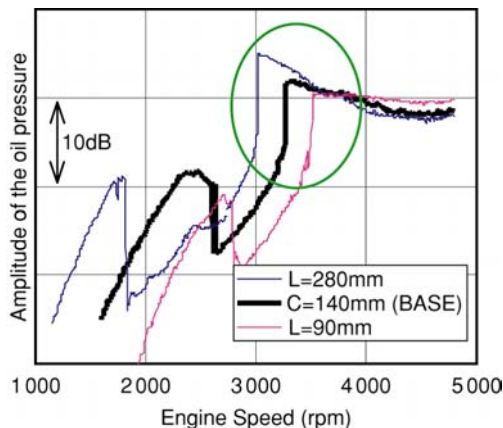


Fig. 17 Amplitude of the pressure at sub gallery

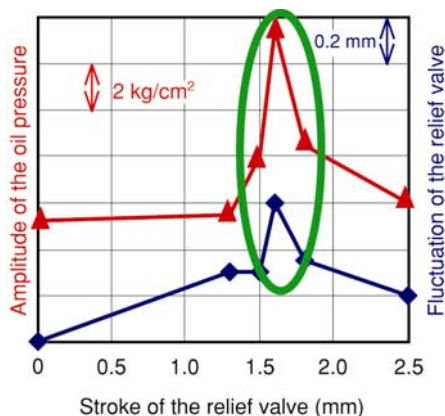


Fig. 18 The relation between amplitude of the pressure and relief valve location

As the graph in Fig. 17 illustrates, the oil pressure amplitude generated as the engine speed approaches 3,000 rpm shows a sudden exaggerated increase. The point at which this exaggerated increase occurs was in proximity to the opening pressure of the relief valve and it was estimated that the oil hammer occurred when the relief valve opened and closed. To verify the correlation between the increase in oil pressure amplitude and the opening and closing of the relief valve, a gap sensor was used to measure the relief valve motion. Figure 18 shows the results of the relationship between the oil pressure amplitude and the relief valve location.

As estimated, the relief valve motion amplitude and oil pressure fluctuation amplitude showed a sharp increase at the point where the relief valve opened. The oil pressure amplitude was greater for the overdriven compact oil pump than that for the conventional coaxial pump. Furthermore, the amplitude on the relief valve motion tended to increase along with the oil pressure amplitude increases brought on by the resonance inside the flow passage.

When the maximum value of oil pressure inside the flow passage exceeds the pressure level of the opening valve, the relief valve, being a pressure controller, vibrates in response to increased oil pressure. The valve maintains the pressure level from the pump to the filter by releasing oil excess to the suction port. In this respect, the positioning of the relief valve has presented a concern. In particular, the oil hammer tended to occur as a result of the valve passage movement associated with the relief valve closing. Figure 19 shows the relationship between the relief valve stroke and the open area of the relief passage. The highlighted area indicates the location of the relief valve when the oil pressure fluctuation amplitude measured by unit test bench increased sharply. As can be seen in Fig. 19, the relief valve vibrates at the point where the opening area of the relief passage changes sharply. These results can be summed up as follows.

- (1) The relief valve operates in response to the pressure amplitude inside the flow passage near the relief opening.
- (2) When the relief valve opening position exceeds 1.5 mm, the oil begins to flow and the pressure inside the flow passage is reduced.
- (3) The relief valve closes to maintain the set pressure.
- (4) For the relief valve to stop the sudden flow of oil, an oil hammer occurs and the pressure inside the passage is temporarily increased.

As the processes described in the previous four steps are repeated, the pressure fluctuation amplitude inside the flow

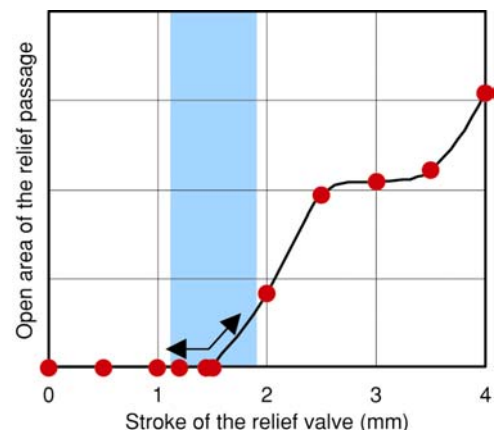


Fig. 19 The relation between relief valve stroke and open area of the relief passage

passage sharply increases causing the oil filter to resonate, which is thought to lead to the abnormal noise. To countermeasure this condition, the relationship between the relief passage opening area and the stroke volume were modified to improve the opening characteristics of the relief valve. Figure 20 shows this relationship. As shown in Fig. 21, the countermeasure was achieved by adding two relief holes to change the original relief open area configuration. This made it possible to diminish the sudden change of cross section area.

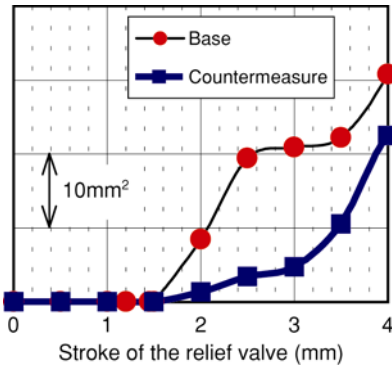


Fig. 20 The relation between area of the passage and stroke of the relief valve

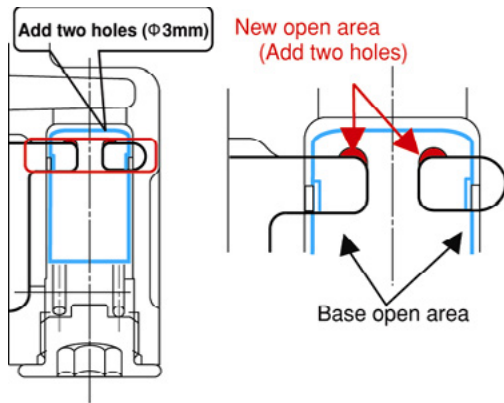


Fig. 21 Remove open area of the relief passage

5. Countermeasure Details

In consideration of the mechanisms outlined in Section 2, the following were discovered to be the cause of the abnormal noise. The size and number of rotor teeth was a contributing factor to the increase in the excitation force compared with the conventional pump. Also, resonance inside the sub gallery flow passage and the shape of the relief valve passage contributed to the sharp increase in oil pressure fluctuation amplitude brought on by relief valve response characteristics.

As shown in Table 3, the oil pump specifications were modified to reduce the excitation force as a countermeasure to the abnormal noise. The number of rotor teeth was increased, from 4 teeth on the inner rotor and 5 teeth on the outer rotor, to 8 teeth on the inner rotor and 9 teeth on the outer rotor. Also, two pre relief holes were added to change the relief valve opening characteristics. Figure 22 shows the results of the countermeasures in reducing oil pump noise and the levels at which the abnormal noise ceased to occur.

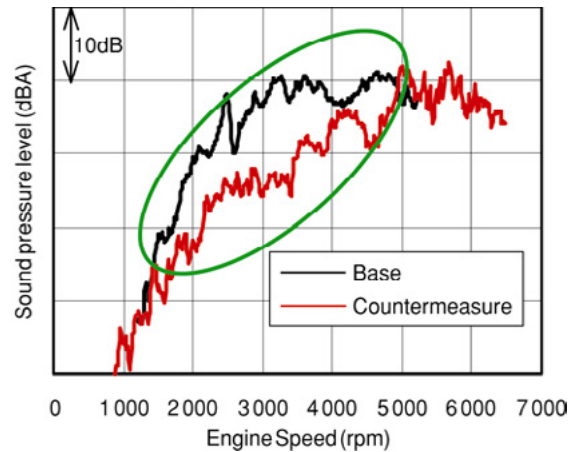


Fig. 22 Results of the countermeasures to reduce oil pump noise

Table 3 Oil pump specification comparison

	Base	Countermeasure
Over driven ratio	Double speed	Double speed
Number of the teeth	Inner 4 / outer 5	Inner 8 / outer 9
Rotor size	Φ40 × 15.0t	Φ45 × 14.3t
Delivery oil (cc) (at round pump)	6.12	5.94
Delivery oil (cc) (at round crankshaft)	12.2	11.9

6. Conclusion

A unit test bench was devised to determine the mechanisms that led to the occurrence of abnormal noise. The unit test bench enabled adjustments to the oil passage shape and the pressure inside the flow passage to significantly reduce noise occurrences in overdriven compact oil pumps. By applying the unit test bench and the effective countermeasures described below in the development of the new oil pump design realized the reduction of oil pump noise.

- (1) To reduce noise in overdriven compact oil pumps, it was learned that the number of rotor teeth needed to be set in proportion to the drive speed to diminish the oil pressure amplitude. The optimal number of teeth for a compact multi toothed inner rotor was verified and set to 8.
- (2) To diminish sudden changes to the relief hole opening surface area, the hole configuration was changed.

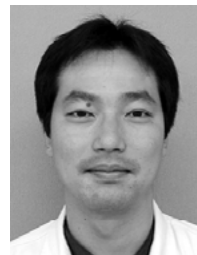
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■ Authors ■



Koji HIRANO



Hisayuki HASHIMOTO



Masakazu KINOSHITA



Yasunori ONO



Yoshiaki SENGÅ