

Development of Light Weight and Lower Cost Aluminum VTEC Rocker Arm

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ABSTRACT

A VTEC rocker arm has been produced from aluminum die-cast materials to reduce weight and lower cost. The new arm, like the majority of rocker arms in use, employs a roller follower to reduce drive friction loss in the valve system. The VTEC connecting holes, in which strength must be guaranteed, were a primary focus to enable the shift to aluminum for the new VTEC system. A coaxial roller structure with a hollow roller bearing shaft was adopted, allowing the roller shaft to be press-fit into place and ensuring sufficient strength and accuracy in the new rocker arm. As a result, both the weight and the cost of the new rocker arm were reduced by more than 20% compared to the forged steel arm. The new arm was first mass-produced for use in the Fit.

1. Introduction

The use of VTEC engines has grown in recent years because of the improved output and fuel economy of these engines⁽¹⁾. They occupy a 60% share of the entire range of engines in use today.

The rocker arm of the conventional VTEC engine has been made of forged steel with the target of providing high strength. However, as a large number of rocker arms were used in the engines, lighter weight and cost reduction became serious issues.

In view of this background, the need arose to strike a balance between weight reduction and the use of existing facilities, as well as curtailing the processing time and reducing costs by cutting down the number of processes. Aluminum was proposed as the material to resolve these issues, but its use in the VTEC rocker arm did not allow the proper balance between strength and precision.

An aluminum VTEC rocker arm has been developed. The methods used for resolving issues related to the structural development of the rocker arm and their effects are reported here.

2. Development Objectives

Figure 1 shows the structure of the newly developed aluminum VTEC rocker arm and the conventional VTEC rocker arm.

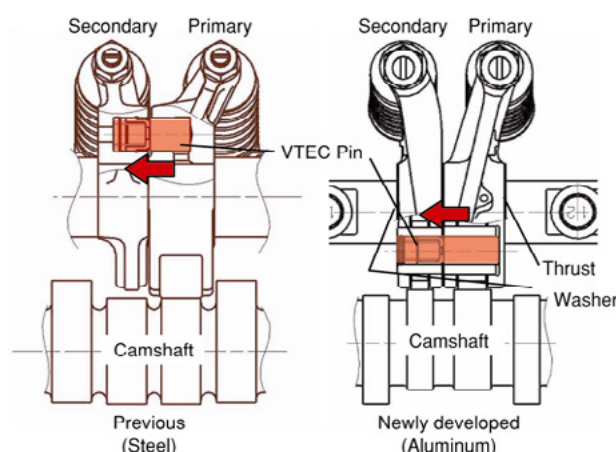


Fig.1 Comparison of VTEC rocker arms structure

The basic rocker arm structure consists of the rocker shaft hole, the rocker bearing hole and the VTEC connecting hole. The connecting hole in particular requires extremely high positioning accuracy because not only high strength but proper function and accuracy should be maintained during VTEC changeover. Furthermore, the connecting hole needs to possess high strength and wear resistance since durability for the synchronizing piston pin has to be ensured⁽²⁾. The conventional VTEC rocker arm was made of steel, and was then carburized or nitrided to satisfy the required

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performance, however a large number of machining processes were required for the materials used in the rocker arm as shown in Fig. 2, and thus cost became a challenge.

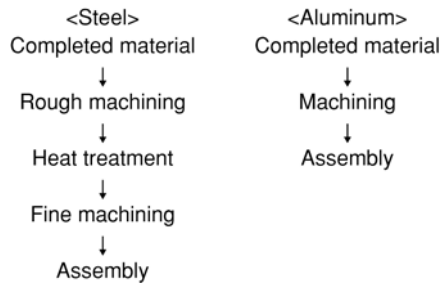


Fig.2 Comparison of machining process

The objectives of the present investigation were to develop a rocker arm that satisfies the requirements of the VTEC system mentioned above, based on conventionally used aluminum die-cast material in order to achieve weight and cost improvements, and to ensure valve system motion and reliability in the same space as forged steel VTEC rocker arms so that only the rocker arm can be replaced without making any changes to the valve system layout for engines in which existing forged steel rocker arms are being used.

3. Basic Structure and Technical Concerns

The issues that relate to the manufacture of aluminum VTEC rocker arms include strength of each part of the rocker arm; rigidity affecting the valve system motion; and strength, wear resistance, and accuracy of the components around the connecting hole. The newly developed rocker arm (Fig. 1) was designed with a shape that satisfies the required rigidity and strength using FEM analysis.

The thickness of the roller bearing arm, which is the weakest part in the rocker arm structure, was increased to enhance the strength. As a result, the width of the roller had to be reduced. Since high stresses also occur between the roller hole and the shaft hole, the thickness between these holes was optimized. Stresses also occur on the inside diameter of the VTEC oil supply hole because of deformation on the internal diameter of the rocker shaft hole. Consequently, the oil supply hole was moved in the upward direction away from the center of the rocker shaft hole and positioned at a location where compressive strain occurs on the inside diameter of the rocker shaft, thereby reducing the internal stress in the oil supply hole (Fig. 3).

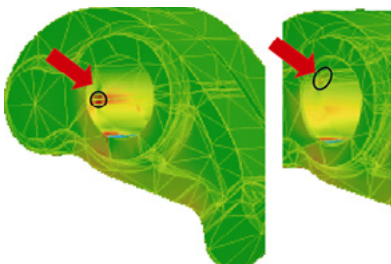


Fig.3 FEM analysis (oil supply hole location)

A rigidity greater than that of steel was obtained by adopting a structure in which the roller shaft and connecting shaft were made coaxial. The results of adopting such a structure are shown in Fig. 4.

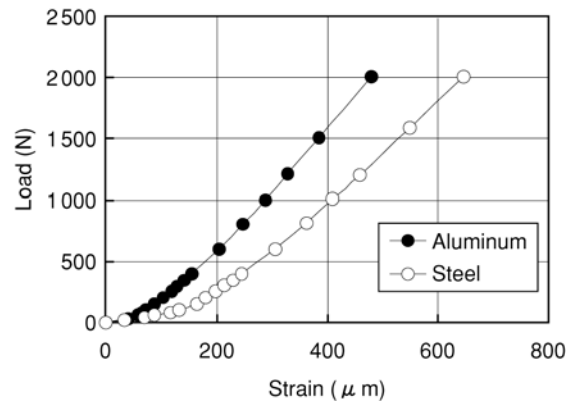


Fig.4 Rigidity comparison

By adopting a coaxial structure, the connecting hole does not engage the synchronizing piston with aluminum but with the roller shaft material, thus ensuring a good balance between strength and wear resistance.

The method of attaching the roller shaft by press fitting rather than by staking was selected because of the accuracy requirement of the connecting hole diameter. Consequently, the deformation of the inside diameter could be restrained and adequate accuracy of the connecting hole could be maintained.

There were also some concerns about wear on the end faces of the aluminum rocker arm caused by rubbing with adjacent rocker arms in the VTEC structure so steel washers were inserted as a measure to resolve this issue.

4. Fixing of Press Fit Assembly Method and Setting of Interference Fit

Since the method of attaching the roller shaft by press fitting was uniquely adopted at this time, the loads occurring during actual engine operation, the attachment method, the fatigue strength of the press-fit hole in the roller arm, the method of selecting the amount of interference for the press fit, and other factors affecting the part strength due to the press-fitting process were clarified, and important control items were set.

4.1. Drive Torque Generated in the Engine

Drive torque and thrust load are stress inducing inputs to the roller shaft. Figure 5 shows the method to measure those inputs.

The roller shaft was allowed to rotate freely and a load cell that could measure forces in the rotational and thrust directions was connected. Under this condition, the engine was operated and measurements were made. Figure 6 shows the measured results at various engine rpms.

The results show that a maximum drive torque of 1.11 Nm and a thrust load of about 3.31 kN were acting on the roller shaft.

It was observed that the load increased in the high-speed region where the valve motion became unstable. Load also increased during mis-engagement when the rocker arm

dropped onto the low lift cam because of an error when the VTEC was switched over. To account for the differences in the behavior during actual measurements, such as use of a rocker arm with load cell, variations in the spring load and so on, the requirements for target strength were taken as a drive torque greater than 2.94 Nm and a thrust load greater than 6.87 kN.

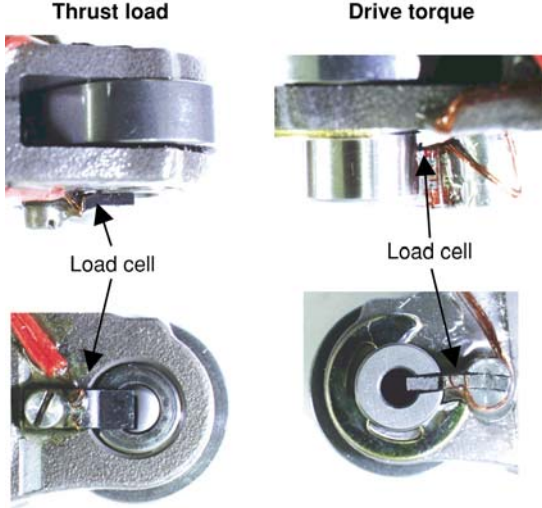


Fig.5 Load cells for measuring thrust load and drive torque

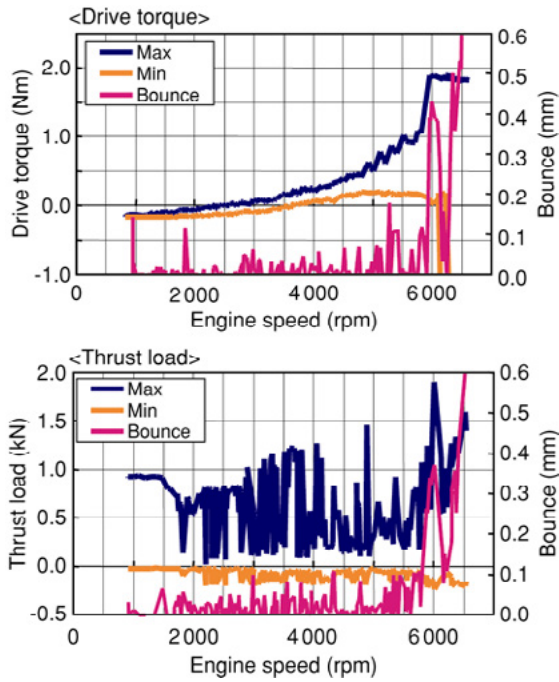


Fig.6 Drive torque and thrust load measurement results

4.2. Press Fit Tolerance and Part Strength

Figure 7 shows the part strength (drive torque and thrust load) for a variety of press fit tolerances, while Fig. 8 shows the remaining press fit. The remaining press fit is reduced as temperature is increased. The oil temperature during operation was taken as 150 °C, and the part strength was measured.

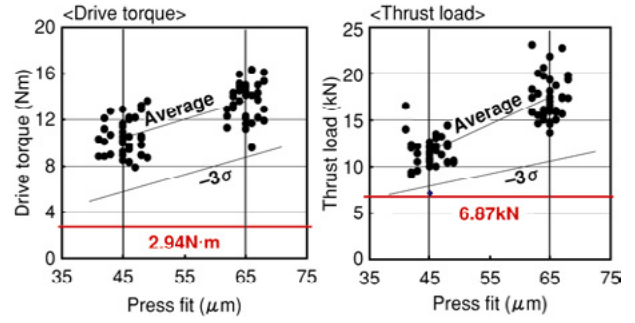


Fig.7 Drive torque and thrust load

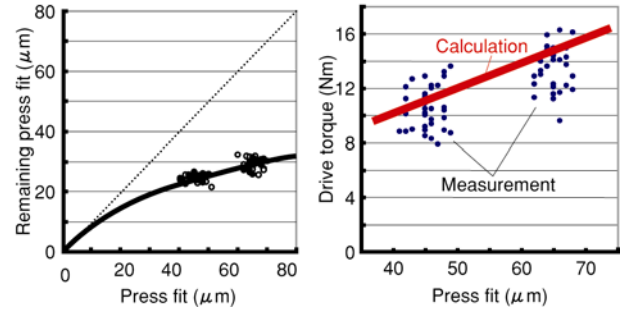


Fig.8 Remaining press fit and comparison between calculated and measured drive torque

The measured values were compared with the calculated values using the measurement of the remaining press fit and the coefficient of friction between the roller shaft and the aluminum die cast rocker arm. The double pipe equation shown below was used in the calculations.

- A : Contact area between rocker arm and roller shaft
- r_1 : Inner diameter of roller shaft
- r_2 : Outer diameter of roller shaft
- r_3 : Outer diameter of rocker arm
- E_A : Young's modules of rocker shaft
- ν_A : Poisson's ratio of rocker arm
- E_B : Young's modules of rocker shaft
- ν_B : Poisson's ratio of rocker arm
- δ : Strain
- P : Surface pressure
- μ : Coefficient of friction between rocker arm and roller shaft

$$\cdot \text{Press fit force} \quad F = \mu \cdot P \cdot A \quad (1)$$

$$P = \frac{\delta}{r_2} \frac{1}{\frac{1}{E_B} \left(\frac{r_3^2 + r_2^2}{r_3^2 - r_2^2} + \nu_B \right) + \frac{1}{E_A} \left(\frac{r_2^2 + r_1^2}{r_2^2 - r_1^2} - \nu_A \right)} \quad (2)$$

· Stress at each location on rocker arm

$$\text{Outer diameter } (\sigma)_{r_3} = P \cdot \frac{2 \cdot r_2^2}{r_3^2 - r_2^2} \quad (3)$$

$$\text{Inner diameter } (\sigma)_{r_2} = P \cdot \frac{r_3^2 + r_2^2}{r_3^2 - r_2^2} \quad (4)$$

The remaining press fit is only about 50% of the initial press fit because of plastic deformation during the press fitting process. Under such conditions, it was observed that the calculated values agreed well with the test results, and the part strength can be expressed by a relationship between the press fit and the coefficient of friction.

In view of the above, it can be said that the press fit tolerance is an important item for controlling the strength of the roller shaft. Furthermore, it can be concluded from the results above that a press fit greater than 45 μm is necessary for the rocker arm to satisfy the part strength during actual operation.

4.3. Strain and Fatigue Strength of the Rocker Arm

The fatigue strength of the aluminum parts that underwent plastic deformation after press fitting is described here. Figure 9 shows the press-fit amount and the resulting fatigue strength for the rocker arm.

With an increase in the press fit amount, the fatigue strength is observed to improve. By preparing test specimens of the same material and measuring the relationship between plastic deformation and fatigue strength, it was confirmed that the fatigue strength increase was due to the increase in plastic deformation of the aluminum as a result of the increase in the press fit.

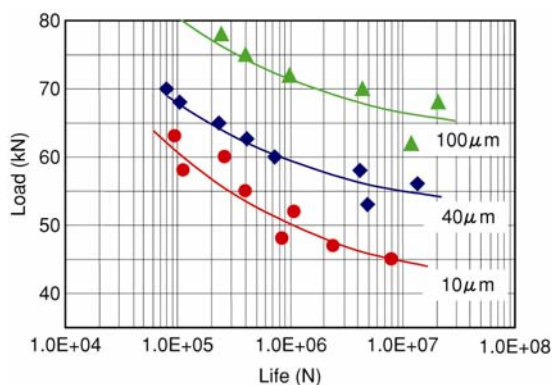


Fig.9 Fatigue strength

These results were evaluated using the fatigue limit curves. It was found that the residual stress increases on account of the increase in the press fit, and the fatigue strength improves. Therefore, evaluation can be performed using the fatigue limit as is done conventionally (Fig.10).

It was also observed that confirmation of the toughness of the rocker arm by unit and durability tests of actual engines at the lower limit of part strength may be accomplished using components with the lower limit of press fit.

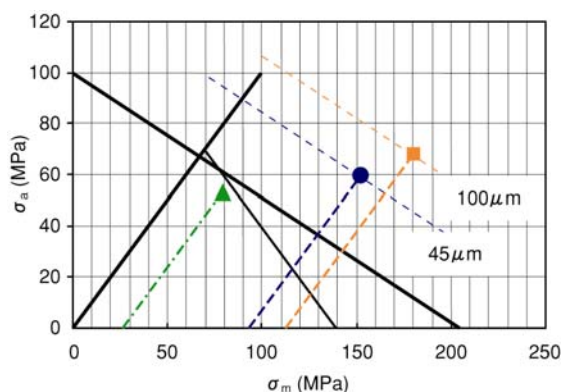


Fig.10 Fatigue strength limit

4.4. Summary of Press Fit Settings

Press fitting causes deformation in various parts and leads to dimensional changes. Accordingly, the upper limit of the press fit needs to be set in order to ensure dimensional accuracy.

Figure 11 shows scoring of the aluminum material of the rocker arm by the shaft during press fitting and the limit of occurrence of peeling. When peeling occurs, not only does the problem of grinding of the roller by dust particles occur, but the part strength is also affected. The upper limit of the press fit is also determined by the peeling limit.

Consequently, the procedure for determining the press fit is decided by setting the lower limit to ensure adequate part strength against engine load and by setting the upper limit based on dimensional accuracy and peeling limit.

The lower limit of the press fit for the newly developed rocker arm was set at 45 μm considering part strength. Considering dimensional accuracy, the upper limit was set at 65 μm since a 20 μm tolerance is considered to be necessary for good manufacturability.

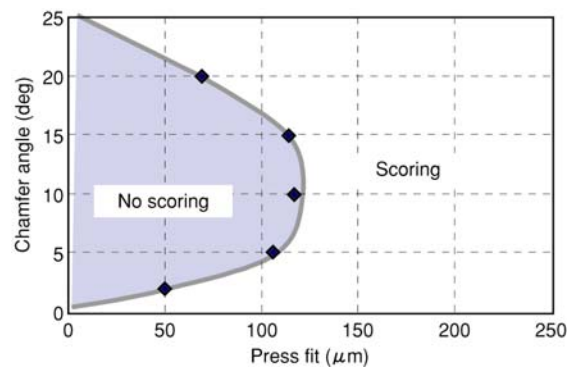


Fig. 11 Scoring limit

5. Reduced Roller Width

With the use of aluminum as the material for the rocker arm, the thickness of the roller support arm had to be increased to ensure its strength and rigidity. As a result, the width of the roller had to be reduced. The size of the roller is determined from the roller life according to the rated capacity and the stress that occurs in the outer race. The rated load of the roller can be lowered by reducing the weight of the rocker arm, and consequently, the roller width can also be reduced. However, the lower limit is considered to be two times the needle bearing diameter considering the shape and function of the needle bearing.

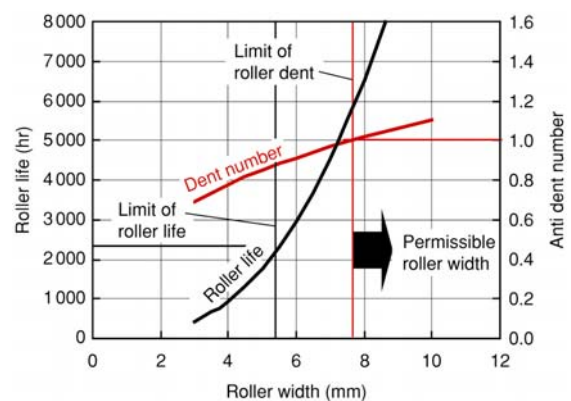


Fig.12 Optimization of roller width

If used in the VTEC rocker arm, dents from the needle bearing occur in the inner race of the roller on account of impact loads when VTEC mis-engagement occurs. It was concluded that the lower limit size of the roller had to be set so that this phenomenon could be avoided. Figure 12 shows the roller dimensions determined from the necessary rated loads, the weight reduction of the rocker arm, and from the number of dents allowed.

6. Confirmation of Durability

6.1. Valve System Motion

Factors that impact valve motion include rigidity and inertial weight. The comparison between the steel rocker arm and the aluminum rocker arm mentioned earlier showed that the rigidity of the aluminum rocker arm was greater than that of the steel rocker arm. Additionally, the inertial weight was reduced by 30%.

Figure 13 shows the measurement results of valve bounce relative to engine speed for each rocker arm design.

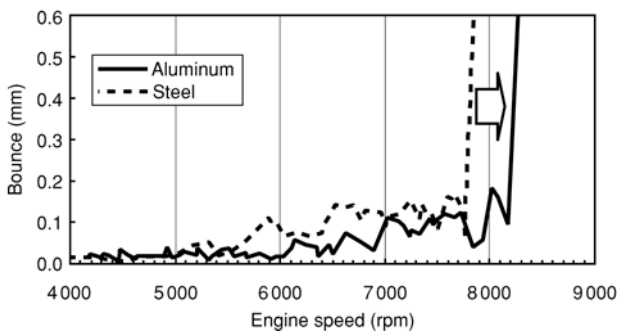


Fig.13 Valve bounce comparison between steel and aluminum rocker arm

An improvement in the engine speed by 500 rpm was confirmed due to the weight reduction. When this impact is translated to valve spring load, the effect is a 12% reduction. This caused a reduction of about 1% in friction in the roller rocker arm system, thus reducing fuel consumption.

6.2. Rocker Arm Stresses

The stresses that occurred in various parts of the rocker arm during operation were measured. The measurement points were the high stress locations according to FEM analysis. Figure 14 shows the measurement results at typical points.

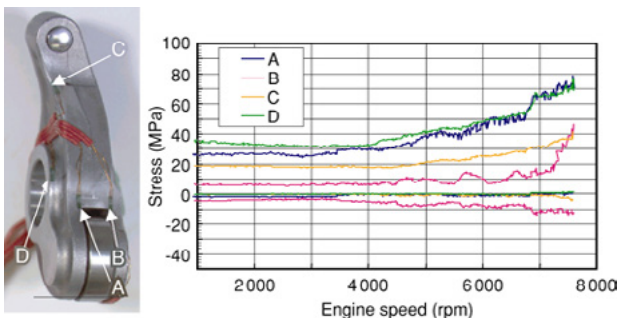


Fig. 14 Rocker arm stress under VTEC-engaged condition

It was confirmed that the locations of maximum stress were on the end faces of the rocker shaft hole and the outer surface of the roller support arm, but the stresses were within the fatigue strength limits of the material confirming sufficient durability.

6.3. Change in the Tappet Clearance

Figure 15 shows the change in the tappet clearance during the running of endurance tests.

The results of the mass-produced steel rocker arm and the newly developed aluminum rocker arm were compared. The performance of the newly developed rocker arm was confirmed as equivalent to that of the steel rocker arm. From the results above, it was concluded that the valve lift curve of the camshaft can be made the same as that of the steel rocker arm, and when used in the engine, only the rocker arms need to be replaced.

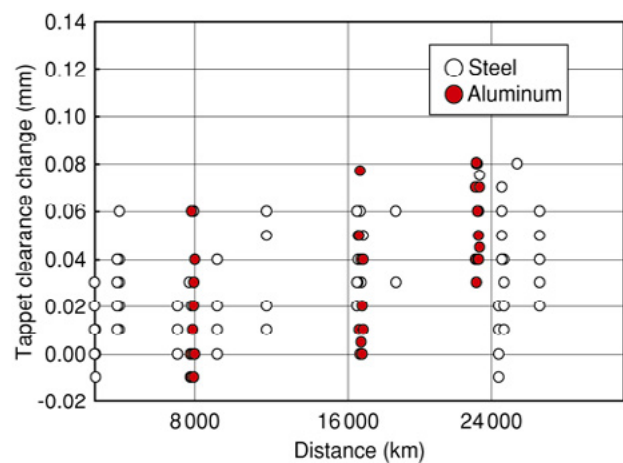


Fig.15 Tappet clearance change variation

6.4. Confirmation of Reduction in Fixing Strength of Roller Shaft after Durability Test

The durability test was conducted in the specified evaluation mode. It was confirmed that no damage, wear or excessive changes in tappet clearance occurred. Figure 16 shows the part strength confirmation results after the durability test.

It was observed that the initial part strength had been retained. Moreover, there was no shift of the roller shaft in the radial or the thrust direction, and adequate reliability of the part strength was confirmed.

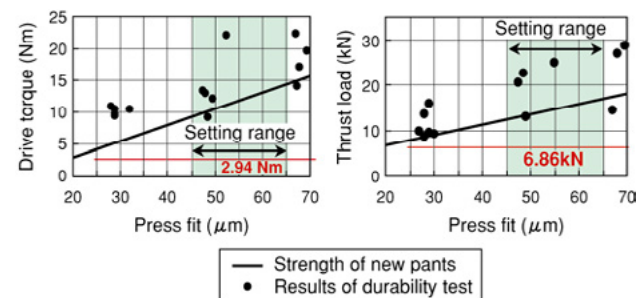


Fig.16 Fixing strength after durability test

6.5. End Face Wear of the Rocker Shaft

Wear of the end faces due to rubbing between adjacent aluminum rocker arms is a problem in the structure of the

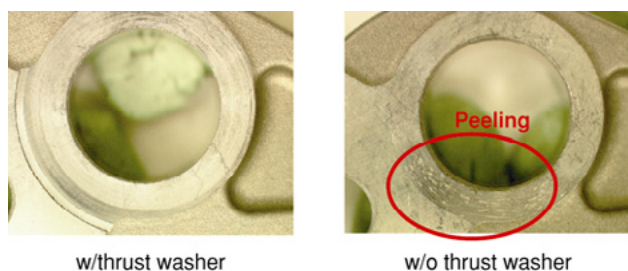


Fig.17 Effect of thrust washer against peeling of rocker arm end face

VTEC system. Figure 17 shows the status of wear in the end faces.

Excessive wear occurs if no washer is used. There was concern that the switching response would deteriorate due to the increase in the clearance between the VTEC switching holes. As a countermeasure, steel washers were inserted between the rocker arms, the rubbing between the aluminum members was avoided, and a reduction in wear was confirmed.

7. Productivity and Costs

7.1. Concept of Cost Reduction using Aluminum

Figure 18 shows a typical example of cost breakdown of the VTEC rocker arm. With the improvement in machinability due to the use of aluminum, both labor and equipment costs of machining could be reduced. Theoretically, this cost reduction amounted to about 50%. This means that a 20% reduction in the total cost can be achieved while maintaining the level of performance.

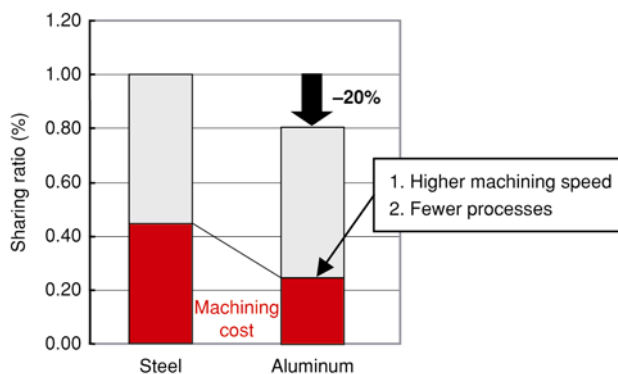


Fig.18 Impact of cost reduction

8. Conclusion

An aluminum die-cast VTEC rocker arm was developed achieving a proper balance between weight and cost reduction relative to conventional VTEC rocker arms made of steel.

- (1) The load occurring in the roller shaft was successfully measured, and the required part strength for the roller shaft was clarified.
- (2) Reliable durability around the shaft of the rocker arm with the press-fitted roller shaft structure was established by obtaining a good balance between the part strength and the fatigue strength.

- (3) The dimensional limits that affect the layout of the roller VTEC rocker arm were established.

Acknowledgment

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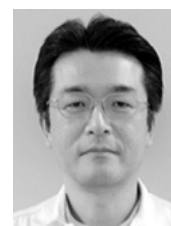
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