Development of a torque converter for the Jatco CVT-X mated to engines with fewer cylinders

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Summary

A turbocharged engine with fewer cylinders is effective in improving fuel economy, but idling vibration increases as does the torque difference between the turbocharged and naturally aspirated operating regions. In addition, torque fluctuations increase when accelerating from a low engine speed. Consequently, it was necessary to apply measures to improve the interior noise level obtained with the Jatco CVT-X. This article describes the adoption of a pendulum-type damper for the torque converter of the Jatco CVT-X and the improvements made to hydrodynamic performance characteristics. As a result, the torque converter satisfies both the fuel economy and power performance requirements while resolving noise and vibration issues.

1. Introduction

Turbocharged engines with fewer cylinders have emerged in recent years with the aim of further improving fuel economy. The Jatco CVT-X is our first continuously variable transmission (CVT) to be applied to a 3-cylinder turbocharged engine.

Reducing the number of cylinders causes larger torque fluctuations compared with conventional engines especially in the low vehicle speed range. Locking up the torque converter from the low vehicle speed range is effective in improving fuel economy, but stronger vibration control than before is necessary. Therefore, it was decided to adopt a pendulum-type damper for the torque converter of the Jatco CVT-X in order to resolve this issue.

Applying a pendulum-type damper in an effort to attenuate large torque fluctuations requires more vehicle mounting space. Installing the damper in a narrow space would cause concerns about declines in damping performance and durability. Therefore, it was necessary to consider such concerns when selecting the mass and spring specifications of the pendulum-type damper.

A torque converter serves to amplify engine torque. Large torque amplification is advantageous for improving vehicle launch performance, but there is concern that large engine torque fluctuations during idling would promote vehicle body vibration.

Therefore, the hydraulic performance was tuned to satisfy the requirements for power performance and fuel economy while contributing to the suppression of vehicle body vibration. This article presents examples of the measures that were adopted to resolve these issues.

2. Pendulum-type damper issues and solutions

A 3-cylinder turbocharged engine can produce large torque from a low engine speed, but torque fluctuations also increase (Fig. 1). To suppress interior noise to the previous

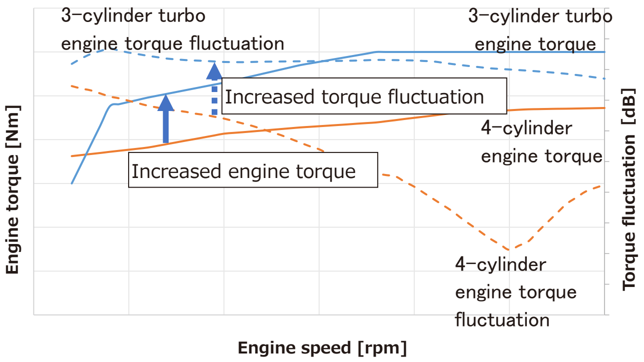

Fig. 1 Engine torque and torque fluctuation

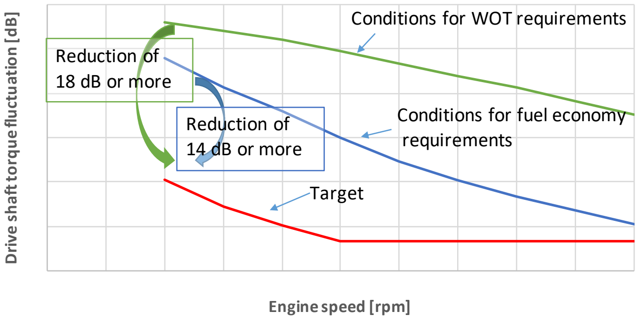

Fig. 2 Targets for reducing drive shaft torque fluctuation

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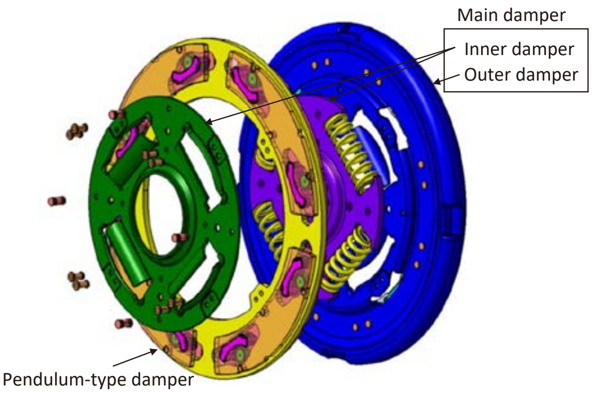

Fig. 3 Structure of pendulum-type damper

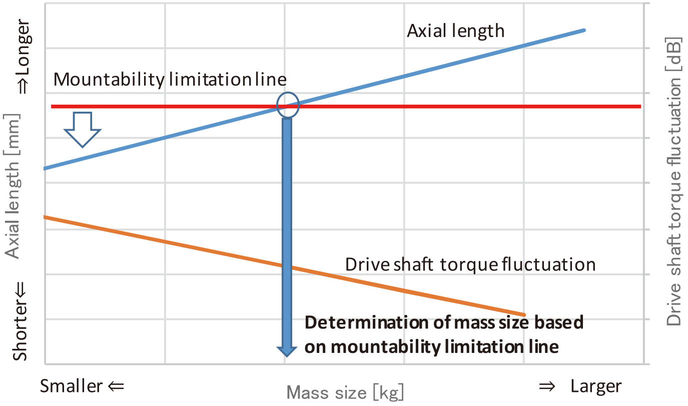

Fig. 5 Sensitivity of axial length and drive shaft torque fluctuation to mass size

level in a locked-up state, drive shaft torque fluctuation must be reduced by 14 dB or more from that obtained with the previous damper under operation that satisfies the fuel economy requirement and by 18 dB or more under wide-open throttle (WOT) operation (Fig. 2). A pendulum-type damper was adopted to attain these reduction targets (Fig. 3).

A pendulum-type damper is built to provide damping force by using the principle of a pendulum to actuate the mass in the opposite phase according to the torque fluctuation frequency. If the pendulum mass moves too much, it will interfere with other parts. This phenomenon is referred to as overshoot.

If pendulum mass overshoot occurs, it gives rise to issues such as noise, vibration and a decline in durability. Therefore, the pendulum mass must be designed so that overshoot does not occur in the region of normal use.

2.1 Study methodology

In general, if the mass stroke is set so that pendulum mass overshoot does not occur in relation to a large torque fluctuation input, a large mass is necessary, which requires sufficient mounting space. However, downsizing is also necessary because of the layout requirements. Accordingly, the torsional stiffness of the damper and the size of the pendulum mass were designed and the specifications were selected so as to achieve a balance between the desired damping effect and the issue of mass overshoot.

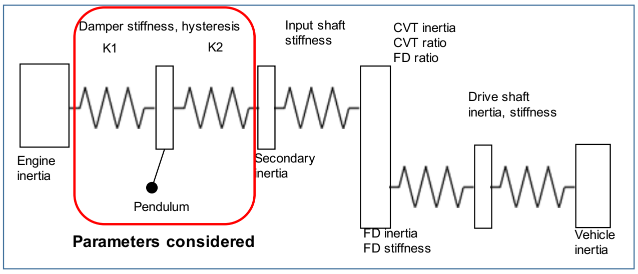

Fig. 4 Analysis model

The model shown in Fig. 4 was used to investigate drive shaft torque fluctuation. A study was made of the feasibility of satisfying the conditions required for fuel economy in relation to the size of the pendulum mass and the spring stiffness of the damper (K1, K2) as parameters. The limits of the mass overshoot region were also simultaneously confirmed.

Increasing the size of the mass has a large effect on reducing torque fluctuation, but the upper size limit is determined by the limitation due to the mountability requirement. Therefore, the value of the mountability limitation was taken as the upper limit of the axial length and the mass size was determined after securing the existing space for the main damper (Fig. 5).

Meanwhile, reducing the spring stiffness of the main damper allows reduction of the engine speed at which pendulum mass overshoot occurs. However, reducing the stiffness more than necessary would require a larger spring size to ensure sufficient strength. That would increase the size of the main damper, making it impossible to satisfy the layout limitation.

Spring stiffness was varied within the existing damper space in order to confirm the drive shaft torque fluctuation level for satisfying the conditions of the fuel economy requirement and the engine speed at which pendulum mass overshoot would occur under the WOT condition. The spring stiffness was then determined so as to satisfy both conditions.

Figure 6 presents the results of an analysis of booming noise when the spring stiffness of the damper was varied. Specification A was for high spring stiffness and specification C was for low stiffness.

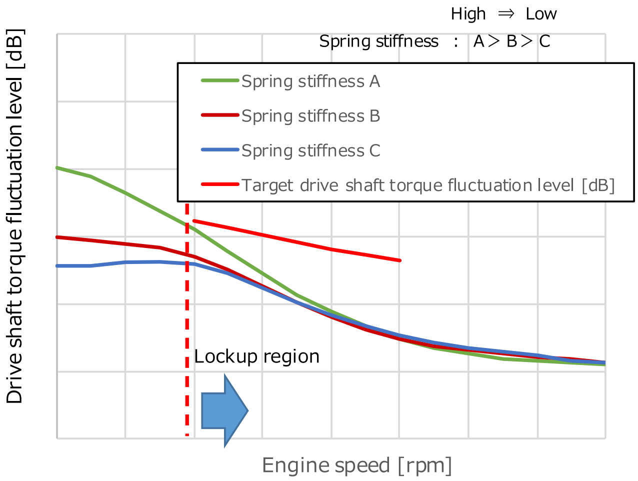

Fig. 6 Booming noise simulation results for various spring stiffness values

The lowest drive shaft torque vibration level was obtained with the low spring stiffness of specification C (blue line), but a drive shaft torque fluctuation value for satisfying the fuel economy target was achievable up to the high spring stiffness of specification A (green line).

Figure 7 presents the results of an analysis of the engine speed for the occurrence of pendulum mass overshoot when the engine torque and resultant drive shaft torque fluctuation were input for each level of damper spring stiffness. The results indicate that lowering the damper spring stiffness makes it possible to reduce the engine speed for the occurrence of pendulum mass overshoot. The spring stiffness of specification C allows an engine speed for satisfying the driveability requirement.

Within the limited space available, a spring stiffness specification was determined for satisfying both the booming noise level relative to the fuel economy requirement and overshoot under the WOT condition.

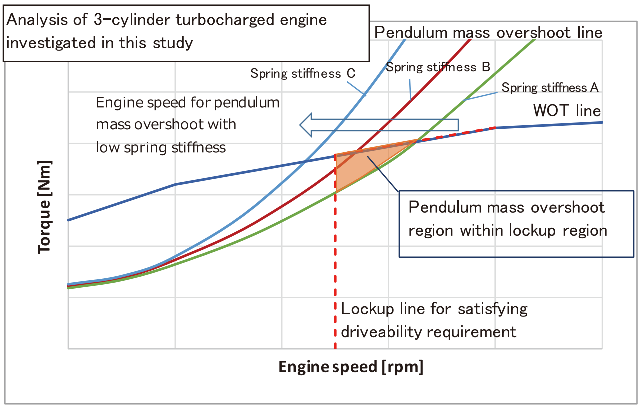

Fig. 7 Engine speed for occurrence of mass overshoot at each stiffness level

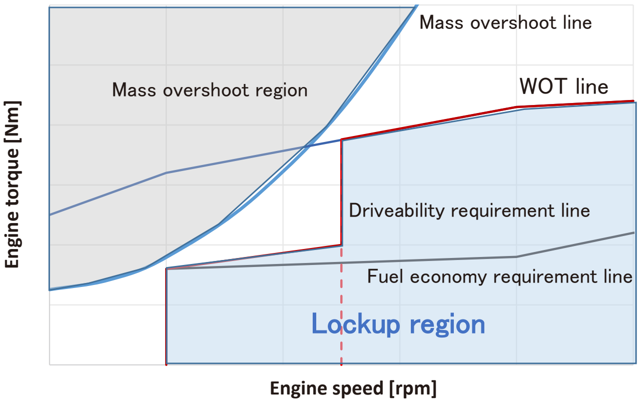

Fig. 8 Lockup region and mass overshoot region

2.2 Level of achievement

Figure 8 is a map of the region where lockup operation is possible in relation to engine speed and engine torque. A lockup region was defined so as to satisfy the fuel economy requirement in the D-range relative to a balance among the pendulum mass size, damper spring stiffness and layout.

The region was also defined so that lockup operation would be possible up to the driveability requirement line without any occurrence of overshoot under the WOT condition that emphasizes power output.

In this way, the lockup region was defined so as to contribute to both power performance and practical fuel economy.

3. Hydrodynamic performance issue and solution

This section presents examples of the improvement of hydrodynamic performance for satisfying the desired fuel economy and power performance while avoiding vibration issues during D-range idling.

Large torque fluctuation occurs during idling of a 3-cylinder turbocharged engine so there is concern that it might promote vehicle body vibration. As a measure against vibration, improvement can be expected by reducing the torque input to the drive shaft. One conceivable approach here is to reduce the output torque of the torque converter.

The output torque of the torque converter (To) is expressed by the following equation in relation to the torque capacity coefficient (τ), engine speed (Ne) and torque ratio (t).

$$To = \tau \text{ x } Ne^2 \text{ x } t$$

Accordingly, reducing the torque capacity coefficient and the torque ratio can be expected to lower the output torque.

However, reducing the torque capacity coefficient of the torque converter overall would cause driveability and fuel economy to deteriorate during acceleration because of engine speed flare. For that reason, it is necessary to keep the torque capacity coefficient at a high level on the high speed ratio side.

Therefore, an effort was made to improve hydrodynamic performance for satisfying fuel economy and power performance requirements while avoiding vibration during idling. That was done by reducing the torque capacity coefficient at the low speed ratio used during idling, including the stall phase, and the torque ratio.

The target for improving hydrodynamic performance so as to achieve the requirements mentioned here was to reduce the torque capacity coefficient and torque ratio relative to the base performance at a low speed ratio below 0.4 and to maintain the same respective values in the region of a speed ratio of 0.4 or higher. Figure 9 shows the improvement targets defined for hydrodynamic performance.

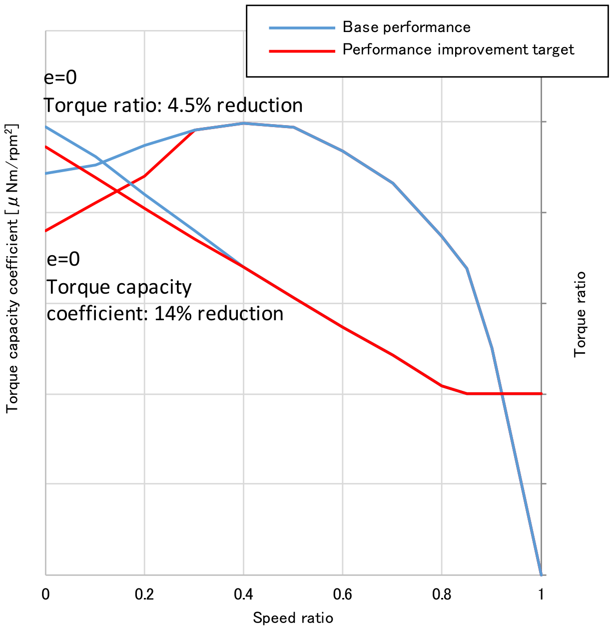

Fig. 9 Targets for improving hydrodynamic performance

3.1 Method for examining hydrodynamic performance

Using the stator blade geometry to increase the amount of flow separation is an effective way to reduce the torque capacity coefficient and torque ratio at low speed ratios. Therefore, hydrodynamic performance was tuned by varying the stator tip geometry so as to ascertain the sensitivity of hydrodynamic performance to the variation.

Figure 10 shows a flow velocity distribution around the stator obtained by simulation at the time of stall. At stall, transmission fluid flows in from the direction of the arrow, striking the stator blade, and the flow branches to the ventral and dorsal sides of the blade. The area of separation at that time is determined by the stator tip geometry. The torque capacity coefficient and torque ratio can be reduced by increasing the amount of flow separation.

Ways of increasing the amount of separation that can be cited include changing the amount of chamfering at the stator blade tip on the inflow side and adjusting the blade thickness at the blade inlet.

The blade geometry was determined by ascertaining the sensitivity of hydrodynamic performance to adjustments made to the amount of stator tip chamfering and blade thickness in relation to the base stator geometry (Fig. 11).

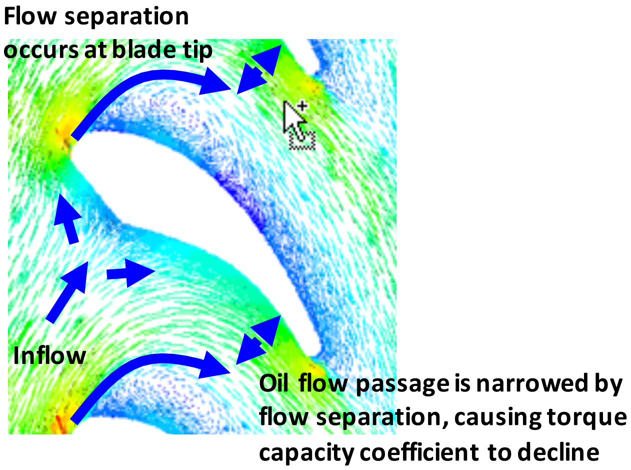

Fig. 10 Flow velocity distribution around the stator

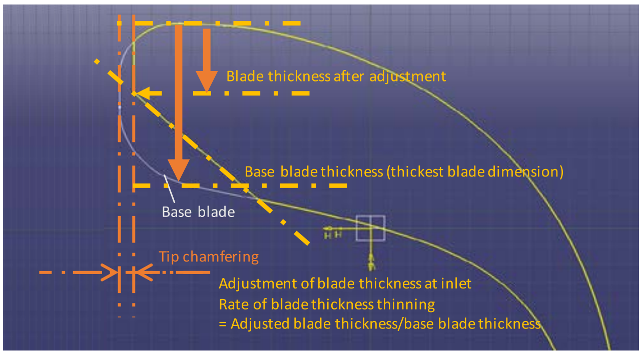

Fig. 11 Cross-sectional view of stator blade: Schematic diagram of tip chamfering

3.2 Performance sensitivity results and level of accomplishment

Figure 12 is a graph showing the rate of change in hydrodynamic performance of the torque capacity coefficient as a function of the amount of stator tip chamfering. The sensitivity of the torque capacity coefficient to the amount of tip chamfering was found experimentally. As a result, it was confirmed that there was a large reduction effect in relation to the amount of chamfering (red line).

However, considering practical productivity, it would be necessary to apply a round shape to the blade tip edge (Fig. 13). Performance sensitivity was obtained taking that factor into account. As a result, it was found that the effect on reducing performance became smaller. The reduction effect decreased from 11% to around 7% for the same amount of chamfering.

The sensitivity of performance to the adjustment of the blade thickness was then investigated (Fig. 14). The results confirmed that thinning the blade thickness was effective in reducing the stall torque capacity coefficient and the torque ratio.

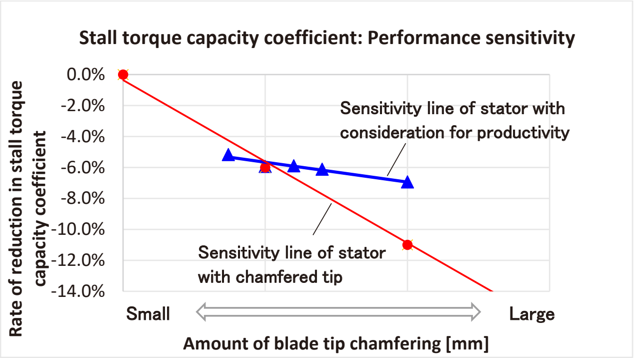

Fig. 12 Stall performance sensitivity to blade tip chamfering

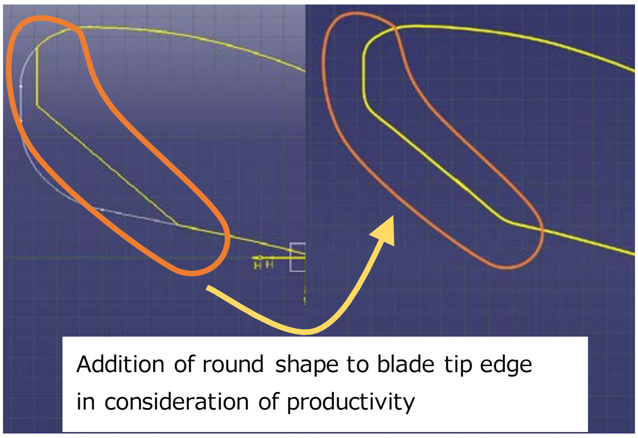

Fig. 13 Stator geometry considering productivity

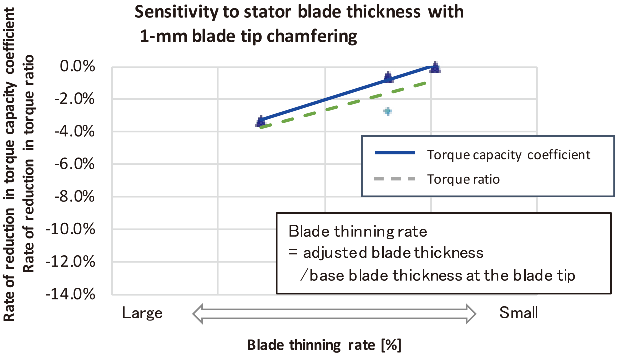

Fig. 14 Sensitivity of torque capacity integer to blade thickness

Stator tip chamfering and blade thickness thinning also reduce the torque capacity coefficient at medium to high speed ratios in addition to low speed ratios (Fig. 15). Therefore, the specifications were determined taking into account the contribution against idling vibration and the effect on power output and fuel economy due to the reduced hydrodynamic performance at medium to high speed ratios.

In terms of the level of accomplishment, the stall torque capacity coefficient was reduced by 8.3% compared with a target of 14% and the torque ratio was reduced by 4.5% compared with a target of 4.5% (Fig. 16).

At speed ratios of 0.6 or higher, both the torque capacity coefficient and torque ratio were maintained at the base performance levels, thereby suppressing the effect on power output and fuel economy.

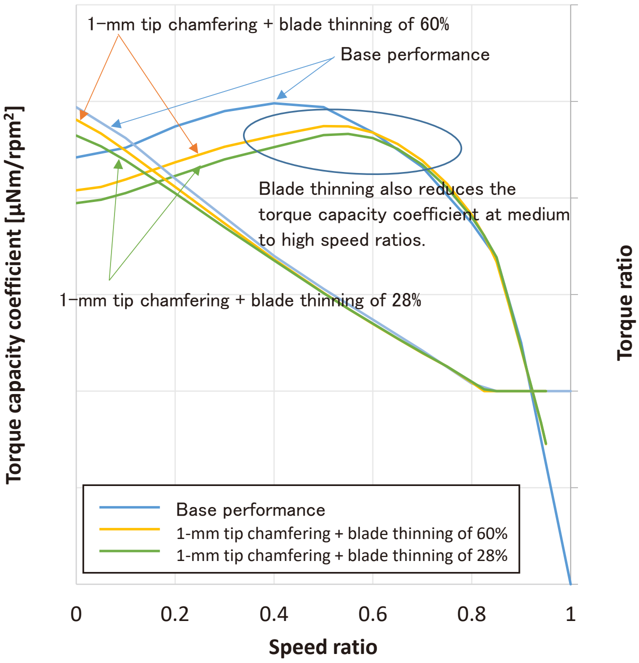

Fig. 15 Overall performance results for blade geometry variation

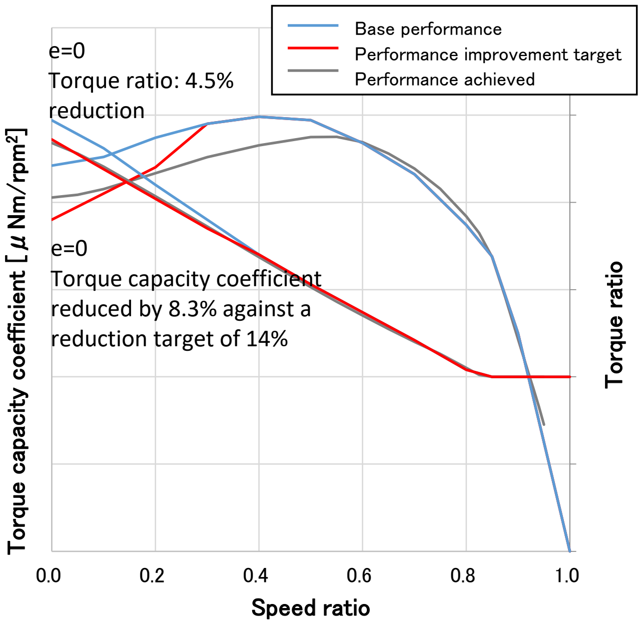

Fig. 16 Level of hydrodynamic performance achieved

4. Conclusion

This article has described the determination of torque converter specifications for JATCO's first application to a 3-cylinder turbocharged engine. Within the limited space allowed, the pendulum mass overshoot issue of the pendulum-type damper and the issue of idling vibration were both satisfactorily resolved.

1) A lockup region was achieved that satisfies the fuel economy requirement and also allows lockup operation for satisfactory driveability without any pendulum mass overshoot as a result of adjusting the damper spring stiffness.
2) Torque converter performance was tuned as a measure against idling vibration. The torque capacity coefficient was reduced by 8.3% and the torque ratio by 4.5%. This tuning provides hydrodynamic performance that satisfies power output and fuel economy requirements while contributing to the suppression of vibration.

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