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New Generation of Large Turbochargers

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Abstract: High efficiency and larger capacity are always desired to large turbochargers for diesel engines. High efficiency of turbocharger can decrease thermal load and improve thermal efficiency of the engine. Also, large capacity of turbocharger can minimize the size of turbocharger that is beneficial to cost and space of the engine.

On the other hand, turbocharger designer sometimes is reluctant to change the design because new design may be cause of technical problem while reliability of conventional design have been proven by many service experiences.

However, Mitsubishi Heavy Industries, Ltd. Has been decided to make necessary design modification on their MET type turbochargers, i.e. turbine blades and gas inlet, gas outlet casing are to be changed in order to improve turbocharger efficiency and turbine capacity. Mechanical strength of present design that is proven by excellent service experiences have been verified and compared with new design.

Result of computational analysis and experiments of above new components that shows far higher performance than present design will be presented in the final paper.

Also, in order to reduce both noise level and pressure drop at air intake silencer, the silencer design was modified based on the analysis results. The advantages of the new design has been demonstrated by turbocharger bench test.

Above modifications are not very drastic, but efficient to improve performance with minimum influence to the design compatibility or proven reliability of the turbocharger. The full paper will suggest how next generation Mitsubishi turbocharger will be.

INTRODUCTION

Trends toward higher output for large diesel engines are continuing, and there will soon be engines that feature brake mean effective pressure of over 2MPa. Accordingly, engine scavenging air pressure is approaching 0.4MPa, and high performance is being demanded with across the entire low~high pressure ratio region of turbocharger compressors and turbines. In addition, the higher pressure and speed inside the turbocharger (accompanying rising compressor pressure ratios) are causing greater noise and increased thrust force. As a result, the following technical issues have surfaced:

- Sufficient turbocharger efficiency across the entire region from low pressure ratio to high pressure ratio
- Impact on thrust bearing life from increased thrust force of the rotor
- Noise
- Compressor impeller life time

These issues must be resolved without sacrificing reliability. In the case of rotating components in particular, such as compressor impellers and turbine blades, changes in material properties over time and the influence of corrosion must be considered in the context of any changes in design, and it is often difficult to sufficiently evaluate strength using numerical analysis (simulation) and test rigs. Accordingly, turbocharger designers must approach changes in such components with caution.

Figure-2 summarizes the history of large turbochargers manufactured by MHI. Following development of the Original MET in 1965, there have been three broad classifications in terms of structure. These are:

- Original MET and MET 0 Series
- MET-S, Sa, SB, and SBII
- MET-SC, -SD, -SE, and SEII

The models in each of these categories have the same basic turbocharger structure, as well as sharing major components such as bearings, turbine blades, and compressor impellers.

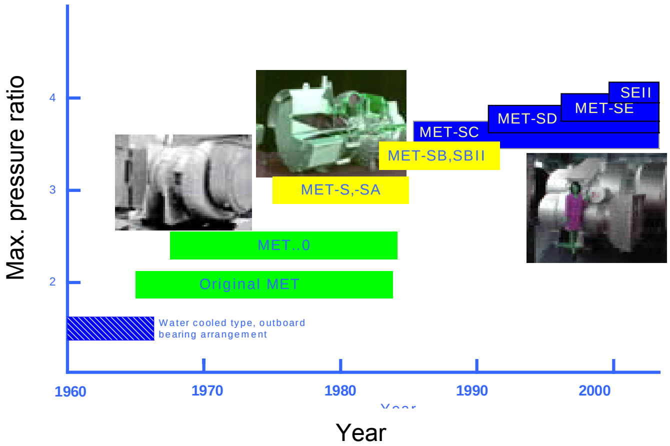


Figure –2 History of MET turbochargers

Figure-3 indicates design evolution in compressor impellers and turbine blades. There have been four basic types of compressor impellers, and three types of turbine blades. In the case of turbine blades, those used for the MET-SB model turbocharger developed in 1983 are currently used in the MET-SE. Blades having the same profile are also used in the MET-SEII, with a change only in the angle of blade placement.

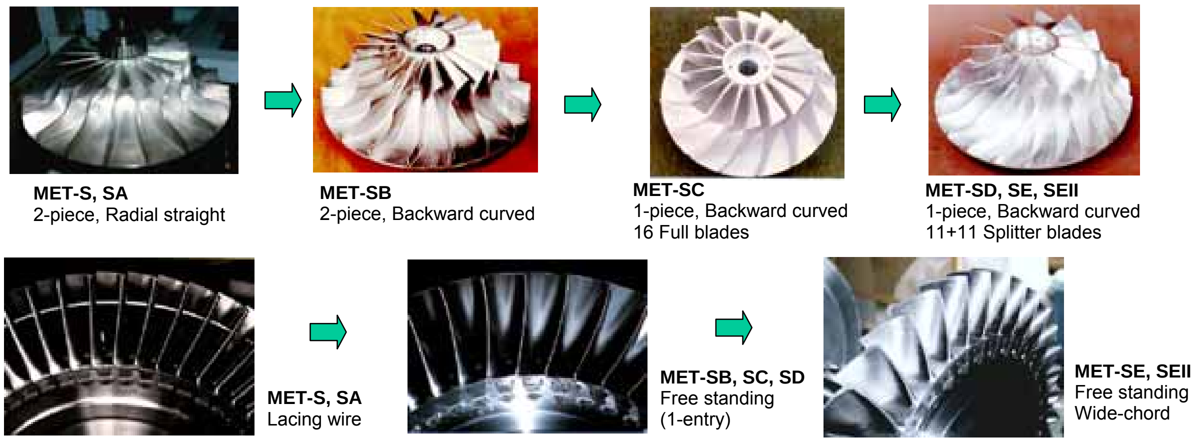


Figure –3 Design evolution of compressor impellers and turbine blades

When creating a new design, the strength of the existing and new designs must be compared using both simulation and tests so as to ensure that the new design is superior to the existing one in terms of the strength safety factor, and that the reliability of the new design is equivalent to that of the existing one as demonstrated by the track record to date. The strength of the most recently developed turbine blade was compared with that used in the MET-SE (which has had no reports of damage since shipments of this turbocharger model began), and its safety was verified.

Simulation was also used to elucidate issues related to the conventional shapes of the gas inlet casing and gas outlet casing flow passages. Optimized shapes were accordingly proposed for the new model turbocharger.

Improvement of the load capability of thrust bearings is an important issue for the expansion of turbocharger capacity and the achievement of higher pressure ratios. The authors of this report produced a design that realizes a high load with stable lubrication, based on the shape of conventional taper-land type thrust bearings. This has resulted in rotor thrust force support capability for sufficient response to turbochargers with increased capacity.

The role of the silencer is important in the reduction of turbocharger noise. On the other hand, a silencer is also an air inlet, and the complex sound-absorbing structure of a silencer can also increase the air flow pressure loss, thus reducing turbocharger performance. The changes in the silencer introduced by the authors served to reduce both pressure loss and the noise level.

Details concerning the above-noted development work are further examined below.

MAIN PART

1. Turbine

1-1 Turbine blade design

Figure-4 presents a comparison of the shape of the newly developed turbine blade and that used for the existing MET-SEII. There are no major differences in terms of blade width or pitch, but the profile from the blade root to the tip has been redesigned so as to deliver better performance across the entire low~high pressure ratio range.

Also, while the blade height is about 7% greater than for the existing version, vibratory stress and centrifugal stress are equivalent or less.

The blade material is the same as conventional blades for MET-SE series or earlier models, consisting of 12%Cr steel alloy.

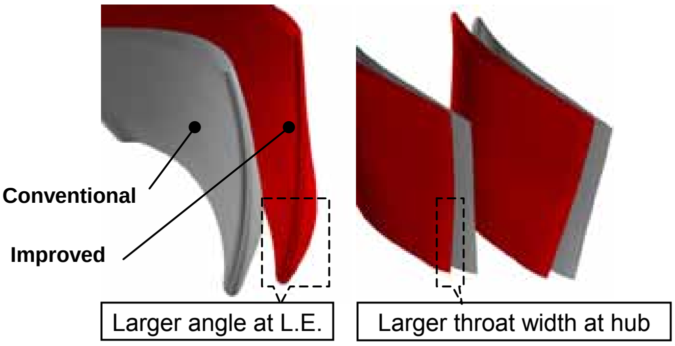


Figure –4 Comparison of new and existing turbine blades

Figure –5 External view of turbine blades

Figure-5 shows photos of the external appearances of the existing and newly developed turbine blades for MET33 size

1-2 Gas outlet casing design

Figure-6 presents numerical analysis results for the flow velocity distributions corresponding to the new and existing gas outlet casings. The gas outlet diffuser has been lengthened in the new gas outlet casing, and the shape of the flow passage has been optimized. Authors' analysis indicates that the flow separation occurring in the existing design at after gas outlet guide is eliminated in the new design, and that smooth gas pressure recovery, e.g., improved turbine efficiency, can be expected.

1-3 Test results

Turbine blades to fit a test rig the size of the MET33 were fabricated in order to conduct laboratory tests for aerodynamic performance with the newly developed gas outlet casing, gas outlet guide and turbine blade. *Figure-7* presents test results for turbine efficiency by changing theoretical velocity ratio. Tests were performed at the three turbine pressure ratios of 1.5, 2.0, and 2.5, and gains in turbine efficiency of 1.5~2 points were obtained for each pressure ratio.

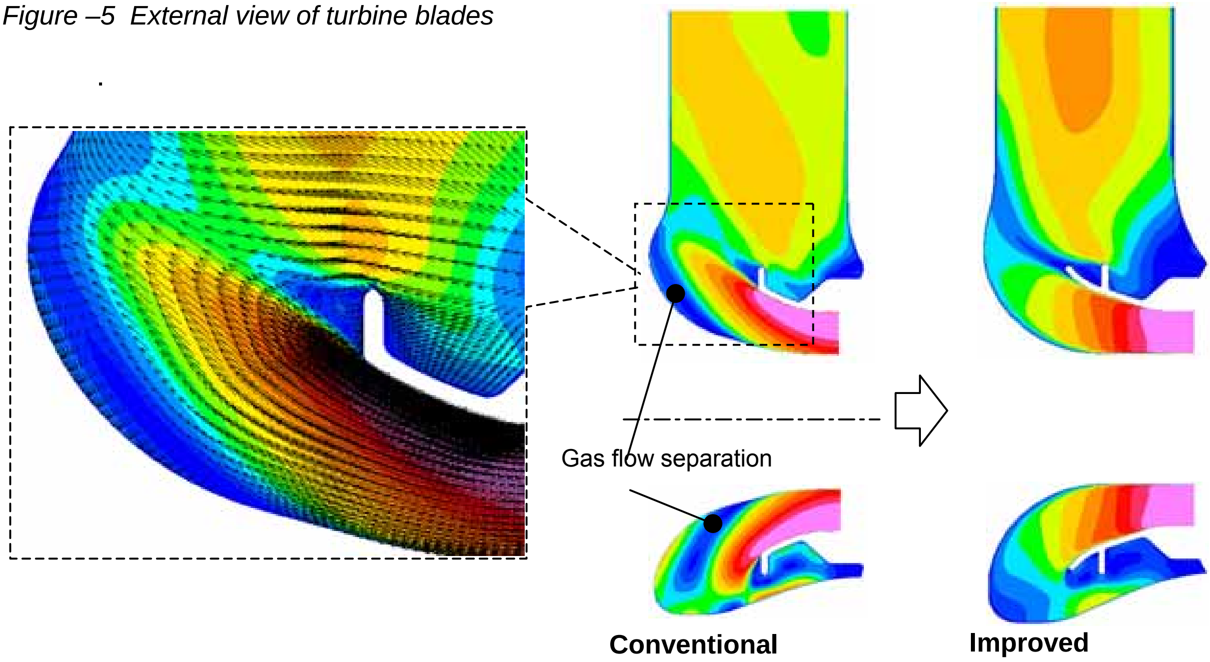


Figure –6 Analysis results for velocity distribution inside the gas outlet casing

Figure -8 presents a comparison between the new and existing turbine blades in terms of the gas flow coefficient value of GRT/P, a parameter indicating turbine capacity. This was confirmed to have increased by approx. 7%, corresponding to the increase in blade height.

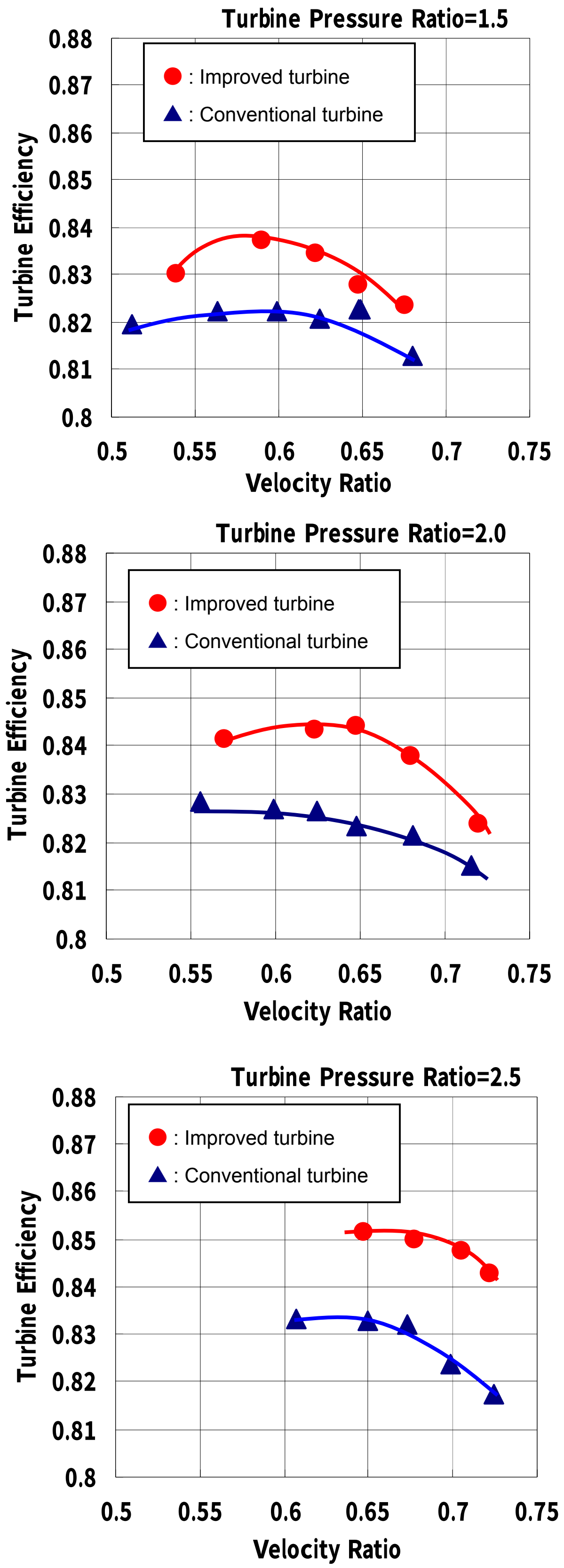
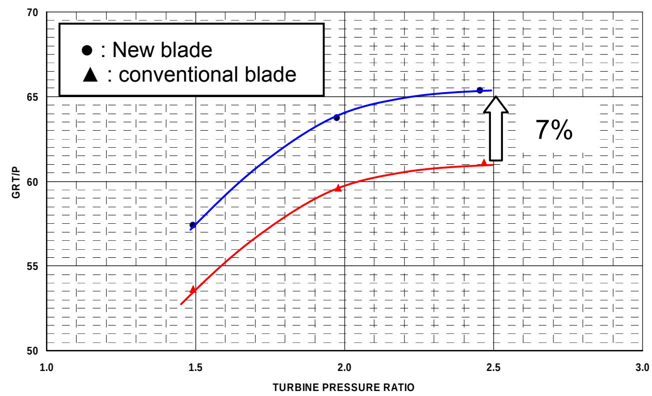


Figure-7 Comparative results for turbine efficiency in test rig



Figure–8 Increase of turbine capacity

Next, blades were fabricated so as to fit the MET83 series, and these were used to replace the existing blades in an MET83SEII model turbocharger. The height of the test blades was made the same as for the MET83SEII, due the requirements of the test model. *Figure-9* indicates overall turbocharger efficiency. Maximum efficiency of 74% was achieved at pressure ratio2.4 – 3.3, representing an improvement of about 2 points over the MET83SEII with conventional blades.

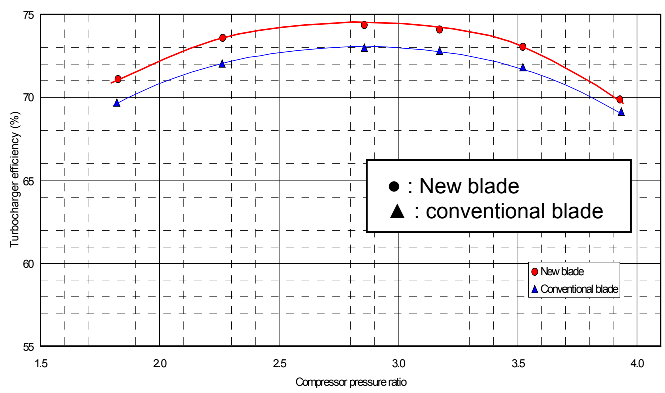
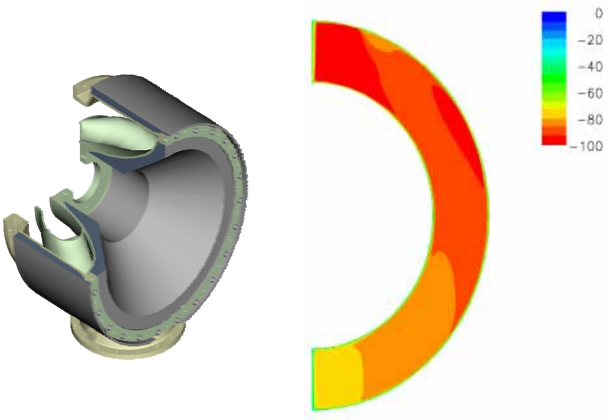


Figure-9 Efficiency of MET83SEII using new turbine blade shape

1-4 Gas inlet casing design

The shape of the MET turbocharger gas inlet casing is similar to that of the Mitsubishi turbocharger MET-S model developed in the 1980s. As shown in *Figure -10*, the shape of the current gas inlet casing has some room to improve flow velocity distribution at the nozzle inlet. Accordingly, improvement so as to achieve uniform flow velocity could serve to raise turbine efficiency. The authors plan to analyze various casing shapes, and to undertake testing to confirm efficiency improvements.



Figure–10 Analysis results of flow velocity distribution at nozzle inlet in existing gas inlet casing

New shapes of gas inlet casing was proposed in order to reduce pressure loss and to have uniform velocity distribution. The shape shown in *Figure-10* is one of the candidate for next Mitsubishi turbochargers. Location and size of gas inlet flange are the same as conventional gas inlet casing. *Figure-12* shows analysis results of flow velocity distribution at nozzle inlet. Almost uniform velocity distribution or better turbine performance can be expected.

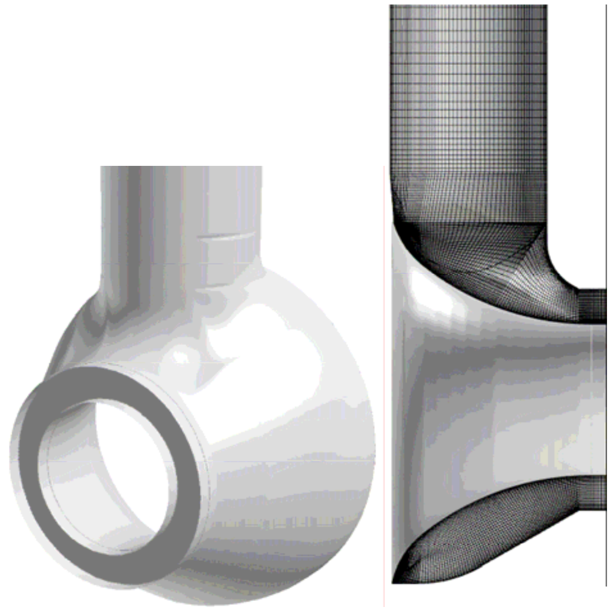


Figure-11 A canditate of new gas inlet casing shape

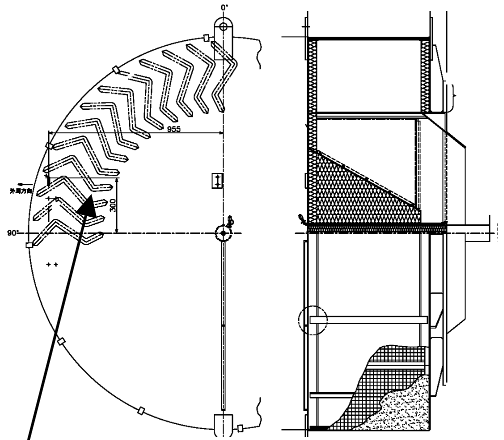
Figure-12 Analysis results of flow velocity distribution at nozzle inlet in new gas inlet casing

2. Air intake silencer

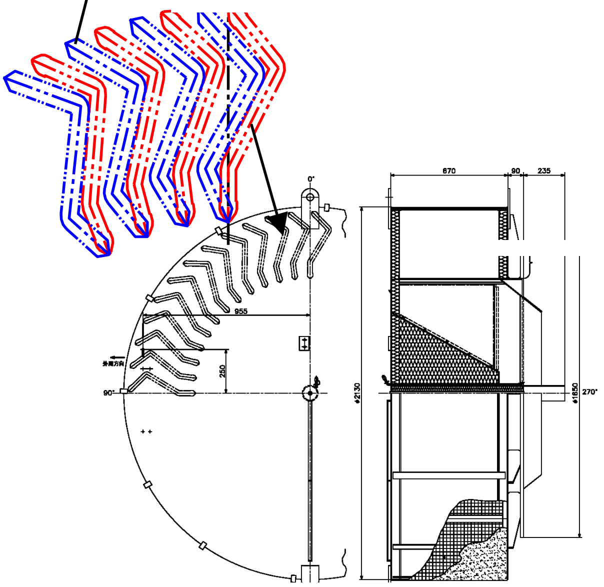
The higher the compressor pressure ratio of turbocharger, the greater the impact of silencer pressure loss. For example, at a compressor pressure ratio of 4.0, a pressure loss of 200mmAq represents a scavenging air pressure differential of 0.008MPa. On the other hand, when the air passage is made larger to as to reduce pressure loss, noise-suppressing effectiveness is reduced.

Zigzag shaped noise absorbing elements made of aluminum frame and glass wool are arranged circularly in the MET turbocharger silencer, and air passes between these elements. Noise emitted by the compressor impeller is lessened when these elements are encountered. While a greater angle of bending allows a greater degree of encounter of noise and therefore a reduced amount of externally perceptible noise, resistance to airflow naturally becomes higher.

Figure-13 presents the existing and improved silencer shapes. The improved silencer features the same width, but with modified element shape. As indicated by the simulation results shown in *Figure-14*, the flow velocity distribution at the element outlet becomes uniform, and it is predicted that the resistance to airflow would decrease.

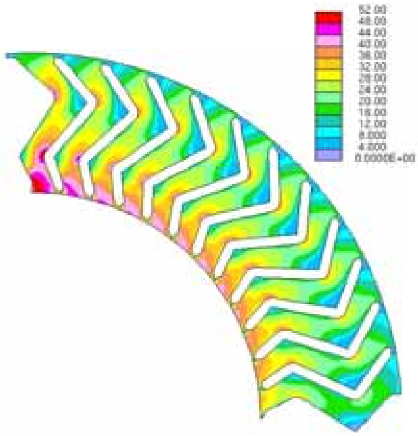


Conventional silencer element

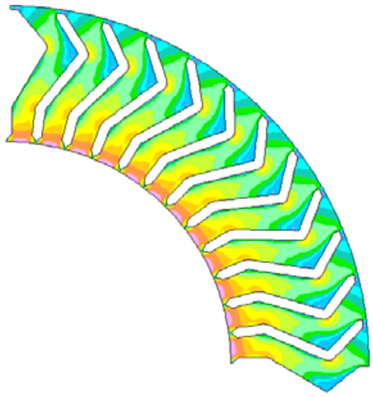


Conventional silencer element

Figure–13 Comparison of new and existing silencer shapes



Conventional silencer element



Improved silencer element

Figure-14 Flow velovity distribution in conventional and improved silencer element

Figure-15 provides a comparison of overall turbocharger efficiency when using the new and existing silencers. The new silencer delivers an improvement in efficiency of about 1.5 points.

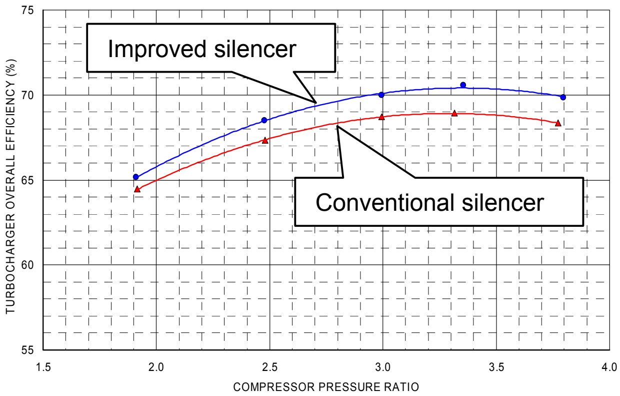


Figure-15 Improvement of turbochager overall efficiency by improved silencer

The uniform flow velocity distribution makes the airflow smoother, and helps to reduce noise generated at leading edge of the silencer elements. *Figure-16* presents a comparison of the noise level at before compressor impeller wheel inlet when using the new and existing silencers, with a far lower level offered by the new silencer in 5 – 7 dB(A).

Conventional silencer element was designed to block and absorb the noise emission by zig-zag shape element while noise is passing through it, and it was considered that smaller angle of the zig-zag results poor noise reduction because noise can also be released smoothly. It is correct and proven by the measurement of *Figure–16* that significant difference of noise level at impeller inlet in conventional and new silencer of *Figure-15* becomes smaller at outside of silencer. However, new silencer is advantageous that noise level at outside is still smaller than conventional silencer, and furthermore, it can improve turbocharger efficiency.

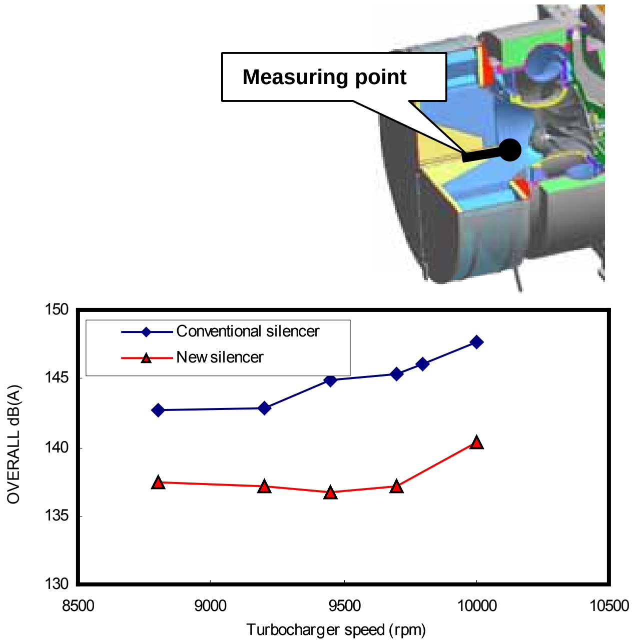


Figure-16 Comparison of noise level at compressor impeller inlet with conventional and improved silencer

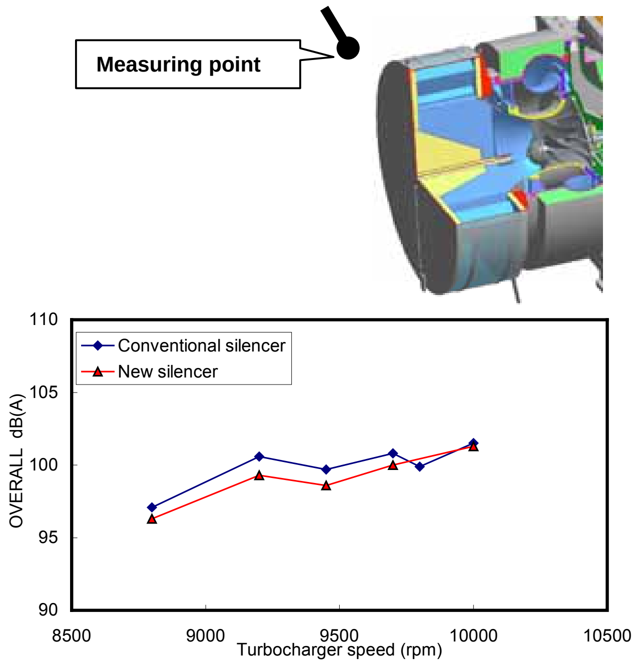


Figure-17 Comparison of noise level at 1.0m outside of silencer with conventional and improved silencer

3. Conceptualization of the next-generation MET turbocharger

Figure-18 shows a cross-sectional diagram of the MET turbocharger in which the improved gas outlet casing discussed above was introduced.

The improvements noted in this report are capable of improving both turbocharger performance and capacity without major changes to the traditionally simple structure of the MET turbocharger. Accordingly, the bearings, impeller, and other components that have a proven track record of reliability can be used without modification. The newly modified turbine blade is confirmed through simulation and testing to deliver reliability that is equivalent to or better than the existing design. Furthermore, ease of breakdown and overhaul continues to be a feature of the new model.

Steps will continue to be taken to provide a superior turbocharger platform based on future results so as to meet ongoing diesel engine requirements.

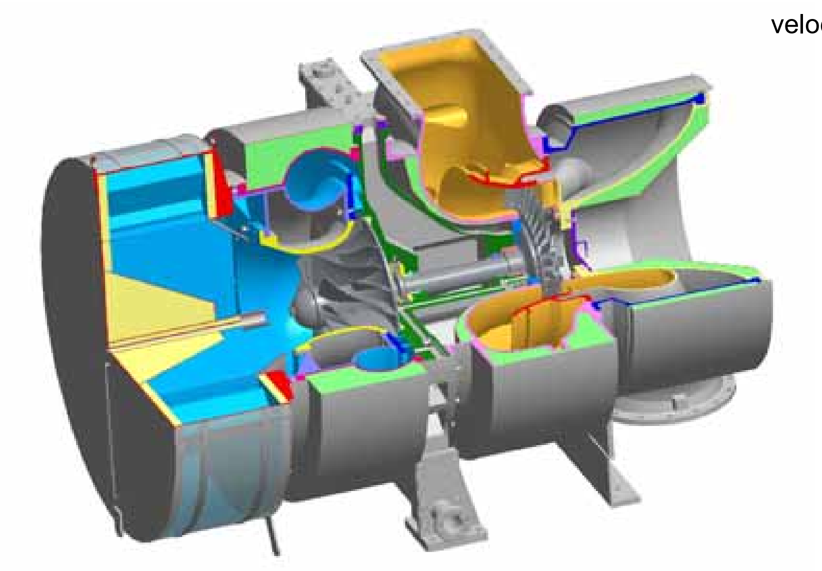


Figure-18 Cross section of Mitsubishi MET turbocharger

CONCLUSIONS

Authors have developed new turbine blades and corresponding gas side casings, and improvement of performance and capacity has been confirmed by experiments.

Small but effective modification on turbocharger silencer was made, and remarkable improvements were confirmed.

Above new designs are worth featuring for Mitsubishi's new turbocharger in future.

NOMENCLATURE

GRT/P : Turbine flow coefficient, a function of gas amount, gas inlet temperature and gas pressure before turbine

velocity ratio : Ratio of turbine blade peripheral speed *U* and theoritical velocity of inlet gas *Co*, i.e. *U/Co*