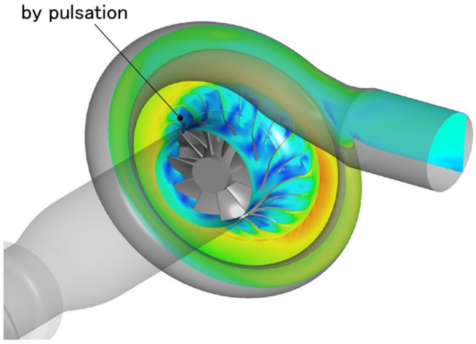
Flow Field Analysis of a Turbocharger Centrifugal Compressor Under Pulsating Conditions



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There is growing awareness of global environmental conservation and the regulations on fuel consumption and exhaust emissions of automobile engines have been strengthened. In recent years, especially, due to the spread of fuel efficiency improving technologies such as downsizing, miller cycle and super lean burn, a wider operating range centrifugal compressor for turbochargers which achieves a high-pressure ratio in a wide operating range has been in high demand. On the other hand, since a compressor is operated under pressure pulsation caused by air intake of an engine, its internal flow exhibits a complicated unsteadiness, which has made it difficult to grasp the flow phenomena toward enhancement of the operating range. Therefore, we conducted a flow measurement using a pulsation test device which replicates an actual engine, and by applying an unsteady numerical analysis with the test result as the boundary condition, we conducted the analysis of the detailed flow structure in the pulsating flow which had not been revealed so far. The result showed that in the pulsating flow, the Helmholtz resonance frequency of a pipe had a large effect on the operating range and the change of the flow pattern due to the appearance or disappearance of full stall at the impeller tip was predominant.

1. Introduction

In recent years, there has been growing awareness of global environmental conservation, and the regulations on exhaust emissions and fuel consumption of automobile engines have been strengthened. For turbochargers, improvement of the performance toward achievement of lower fuel consumption of an engine has been strongly demanded. **Figure 1** shows the system layout of an automobile turbocharger. The engine exhaust gas energy rotates the turbine, which drives the centrifugal compressor on the same shaft, and the compressed air is fed into the engine via the downstream intercooler. At this time, the flow field of the centrifugal compressor and the turbine is exposed to pressure pulsation caused by air intake of the engine shown in **Figure2**. Therefore, for improvement of the performance of a turbocharger, it is important to consider the pulsating flow during development.

In the past development of a turbocharger considering the pulsating flow, MHI focused attention on the flow under pulsation of turbine scroll to improve the efficiency, which was described in the MHI Technical Review Vol. 50 No. 1. This report introduces an example of the development focusing attention on the performance of a compressor under pulsation.

In the recent development trend for centrifugal compressors, fuel efficiency improving technologies such as downsizing, miller cycle and super lean burn have widely spread and a wider operating range of centrifugal compressors which achieve a stable supercharging pressure in a wide operating range has been essential. **Figure 3** shows the typical operating points of a centrifugal compressor for a turbocharger. A compressor is used at various operating points according to the

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vehicle operation state, and the operating range is restricted to the range from the operation limit on the small flow rate side due to surging to the operation limit on the large flow rate side due to choking. Surging, in particular, which restricts the operating range in the vicinity of the maximum torque point of an engine, is an unstable flow phenomenon in which a reverse flow due to the reverse pressure gradient of a compressor is involved. In order to achieve a wider operating range, it is necessary to accurately grasp the flow structure of surging in an actual engine.

As previously described, however, a turbocharger is exposed to pressure pulsation associated with air intake of an engine. Therefore, the internal flow is a complicated unsteady flow which largely deviates from a steady flow without pulsation. Concerning surging of centrifugal compressors, many studies [1)2)], irrespective of experimental or numerical analysis, have been conducted conventionally, and a part of the surging is being revealed. But there are few examples of detailed analysis of surging under the operating conditions involving pressure pulsation as described above, and no sufficient findings have been obtained yet.

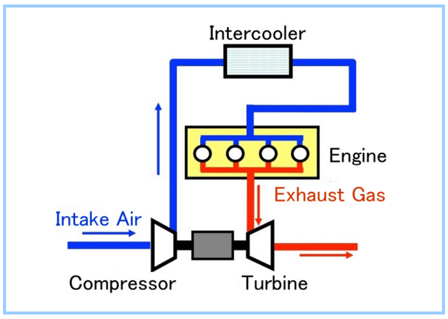

Figure 1 System layout of turbocharger

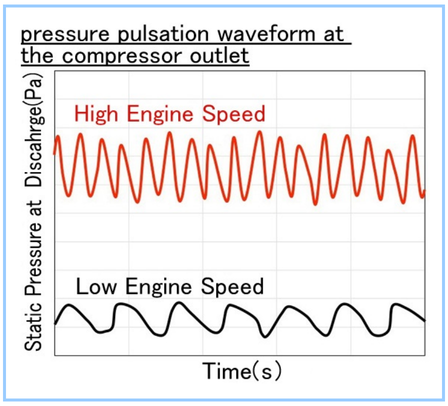

Figure 2 Pressure pulsation of the turbocharger engine

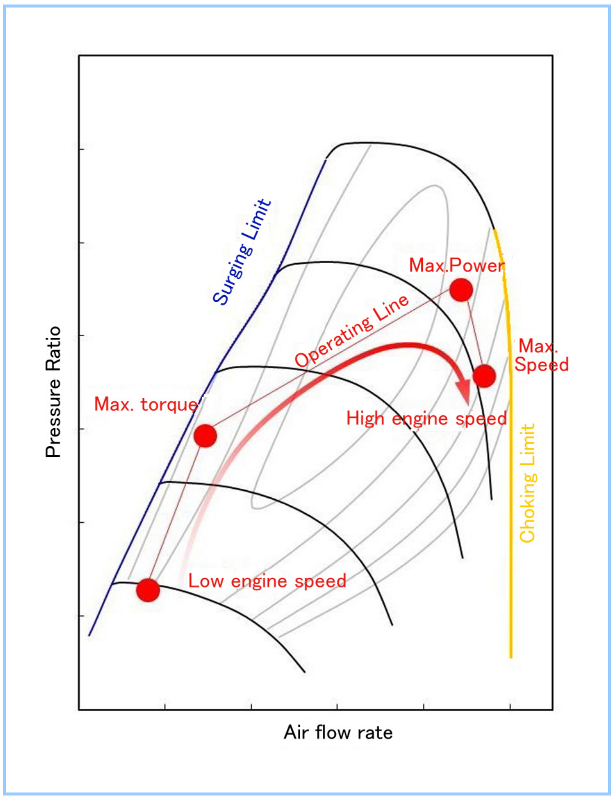

Figure 3 Operating points of turbocharger compressor for automobile

2. Experiment and flow analysis methods

2.1 Compressor under study

Figure 4 shows the turbocharger compressor used in the current study. This compressor is a centrifugal compressor used for a turbocharger and uses an open-type backswept impeller which has 10 full blades and 10 splitter blades s. After air is given kinetic energy by the impeller, its speed is reduced and its pressure is increased by the downstream vaneless diffuser, and the air is supplied to the engine through the scroll connecting to the air supply manifold. This compressor has no additional device such as casing treatment[*i]. **Figure 5** shows the relationship between the flow rate and the pressure ratio for each rotational speed of the test compressor (performance characteristic diagram). In this study, the flow near surge point at a pressure ratio of 1.7 is analyzed. This operating point corresponds to the engine low end torque region where the turbocharger operates at low flows, and thus, compressor surge margin plays a key role.

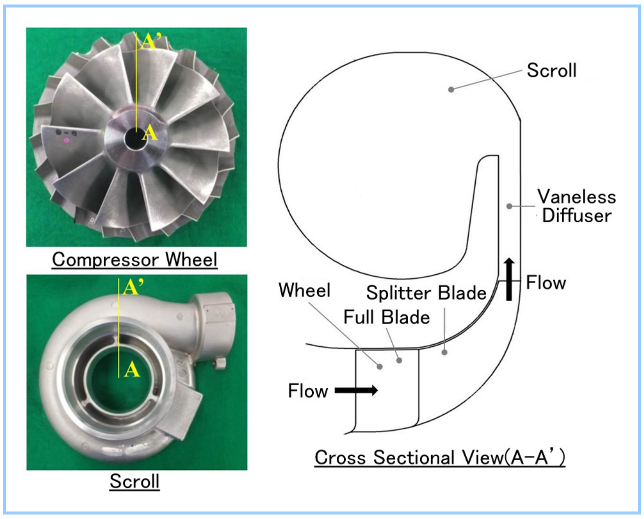

Figure 4 Appearance and schematic diagram of test compressor

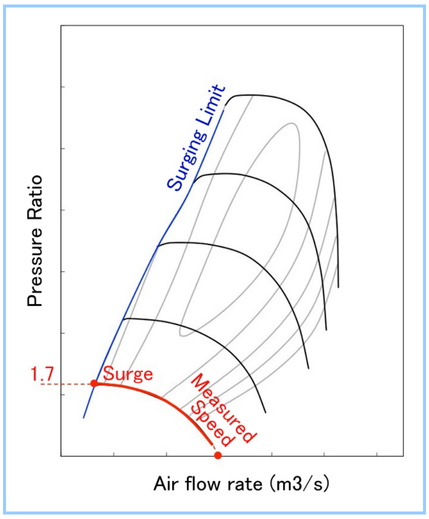

Figure 5 Pressure and flow rate characteristic diagram

2.2 Measurement method for surge under pressure pulsation[3)]

To analyze the effect of the pressure pulsation waveform on the surge characteristic, MHI has conducted a joint research with Imperial College London, being this institution one of the most experienced in measuring turbocharger performance. Imperial College team has developed a pulsating flow device able to reproduce a wide variety of pulsating conditions similar to those encountered in engine operation. [3)] **Figure 6** shows the compressor facility including the bespoke pulsation flow device. This pulsating flow device is composed of a rotary valve located downstream of the compressor outlet, which generates different pulsing conditions – amplitude and frequency of the pulse can be set in the pulse generator by changing the rotating speed and/or the disc size/shape, respectively.

Table 1 shows a summary of the test conditions evaluated during the experimental campaign carries out at Imperial College London with the above-described pulsation flow device. In this test, the pressure waveform at the engine torque point of a general passenger car was replicated by the pulsating flow device. The selected engine condition used as reference in this campaign corresponds to a sinusoidal wave form and pulse frequency of 66.7 Hz (equivalent to an engine speed of 2,000 rpm) in the pulsating flow device. The experimental campaign considered the compressor performance measurement at different outlet pulsating frequencies and amplitudes. The surge flow rate at each test condition was measured and the effect of the pressure pulsation waveform on the surge characteristic was analyzed.

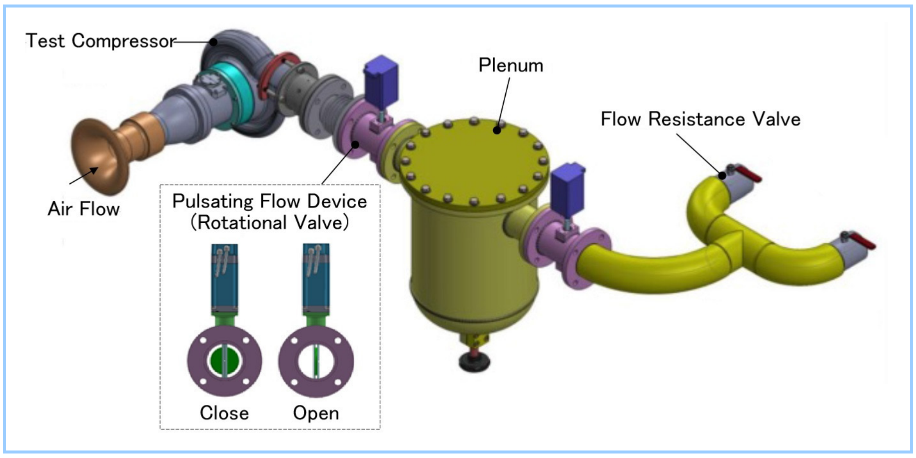

Figure 6 Pressure pulsation test device

Table 1 Pressure pulsation test condition

Rotational Speed of Wheel	40000rpm
Frequency of Pulsation	20～100Hz 66.7Hz for reference
Amplitude of Pulsation	Approx. 5% from mean pressure (Near surge point)
Downstream Pipe Volume	$0.1661m^3$

2.3 Unsteady flow analysis method

In order to analyze the detailed internal flow in the above test result, unsteady flow field analysis was conducted using Computational Fluid Dynamics (CFD). The numerical analysis model and the analysis condition used in this study are shown in **Figure 7** and **Table 2**. The ANSYS-CFX analysis code was used and the SST--kω turbulence model was used. The analytical area imitated the test device in the foregoing chapter, the hemispherical domain which corresponds to the open field upstream of the impeller inlet pipe was provided, the outlet boundary was set at the position corresponding to the rotational valve of the test device, and analysis was conducted at the rotational speed equivalent to the test condition. For the computational grid, impeller and diffuser are created by a structural grid and an upstream/downstream pipe including scroll is created by an unstructured grid. The total number of cells is about 13 million. The static pressure fluctuation obtained by the measurement result in the foregoing chapter was added as the outlet boundary condition, and the time history data of unsteady flow for four cycles of pressure pulsation waveform was obtained.

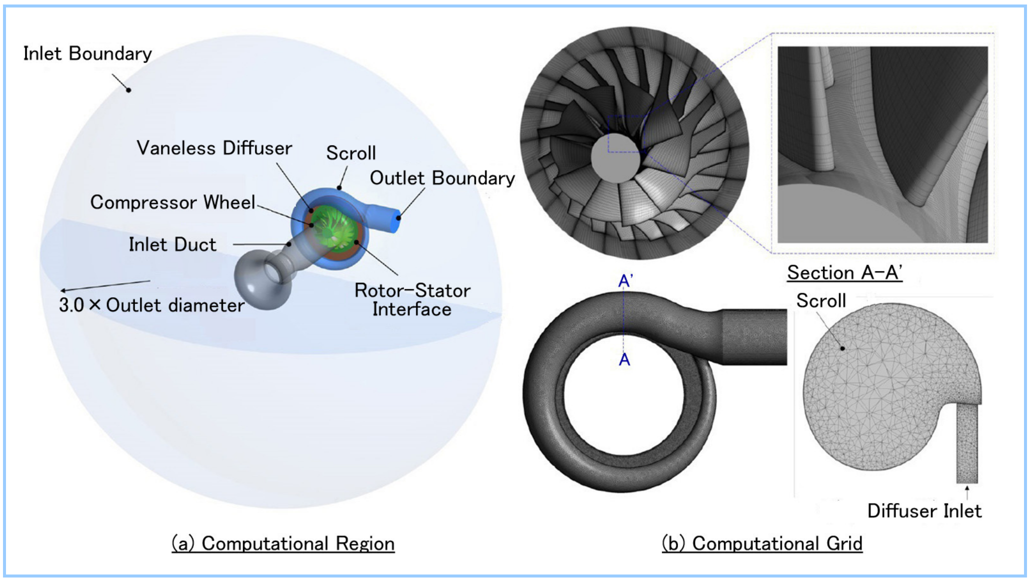

Figure 7 Numerical analysis model

Table 2 Unsteady numerical analysis condition

Flow Solver	ANSYS-CFX 14.5
Governing Equation	3D-RANS
Numerical Scheme	2^{nd}-Order
Turbulence Model	SST k-omega
Inlet Boundary Condition	Total Pressure 1atm Total Temperature 293.15k
Outlet Boundary Condition	Static Pressure (Ref. Fig.8)

3. Clarification and improvement of the flow phenomenon under discharging pulsation

3.1 Measurement result for surge characteristic under pulsation[4) 5)]

In order to analyze the effect of the pressure pulsation waveform, the surge flow rate was measured with different pulsation frequencies and pulsation amplitudes. In this study, the pulsation for the frequency of 66.7 Hz (equivalent to 2,000 rpm) corresponding to the maximum torque point of a general 4-cylinder engine was set as the reference waveform (**Figure 8**), the pulsation frequency f was changed in the range of 20 Hz to 100 Hz, and the surge flow rate in each case was obtained.

Figure 9 shows the comparison of the pressure flow characteristic between the steady flow without addition of pressure pulsation and the pulsating flow with the reference pressure waveform added. When the pressure pulsation is added to the compressor, the inertia force associated with increase or decrease of the flow rate acts on the fluid in the flow path, which suppresses a reverse flow associated with surging, and the surge flow rate is reduced compared to the steady flow. [5) 6)] According to the test result with this compressor, the surge flow rate SP2 in the pulsation flow is shifted to the small flow rate side by 8.0% relative to the surge flow rate SP1 in the steady flow, and this indicates that the pressure pulsation suppresses surging in the compressor.

Figure 10 shows the pressure flow characteristic of the test compressor for different pressure pulsation frequencies. The largest surge flow rate is at the pulsation frequency of 20 Hz, where the operating range is narrow. This corresponds to the resonance frequency of the downstream pipe of the test compressor, and the Helmholtz resonance frequency[*2] is calculated from the cross-sectional area and volume/length of the pipe is around 23 Hz. The test result shows that at 20 Hz, which is closest to the Helmholtz resonance frequency, the surge flow rate has an extreme value. As the frequency increases from the above condition, the surge flow rate reduces, and the operating range is the largest at 43.3 Hz close to the frequency which is twice as high as the Helmholtz resonance. Accordingly, the surging characteristic under pulsation has a strong correlation with the resonance frequency of the downstream pipe, and it is considered that an appropriate design of the pipe shape of an actual engine can increase the stable operating range on the small flow rate side.

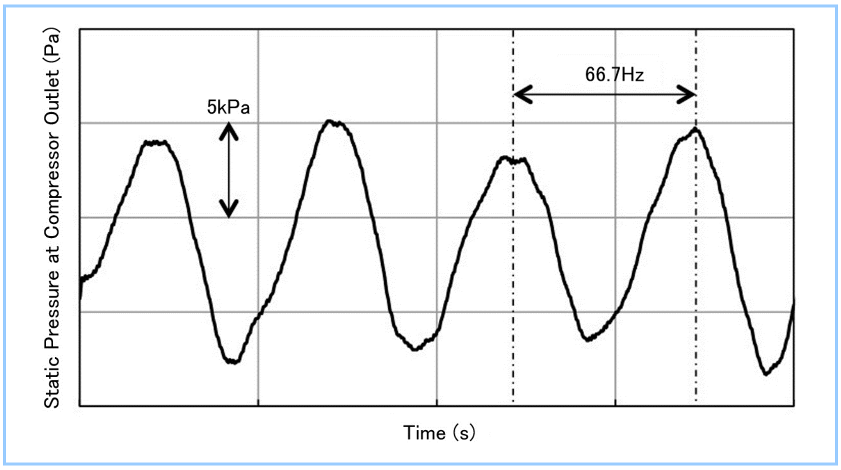

Figure 8　Reference pressure waveform

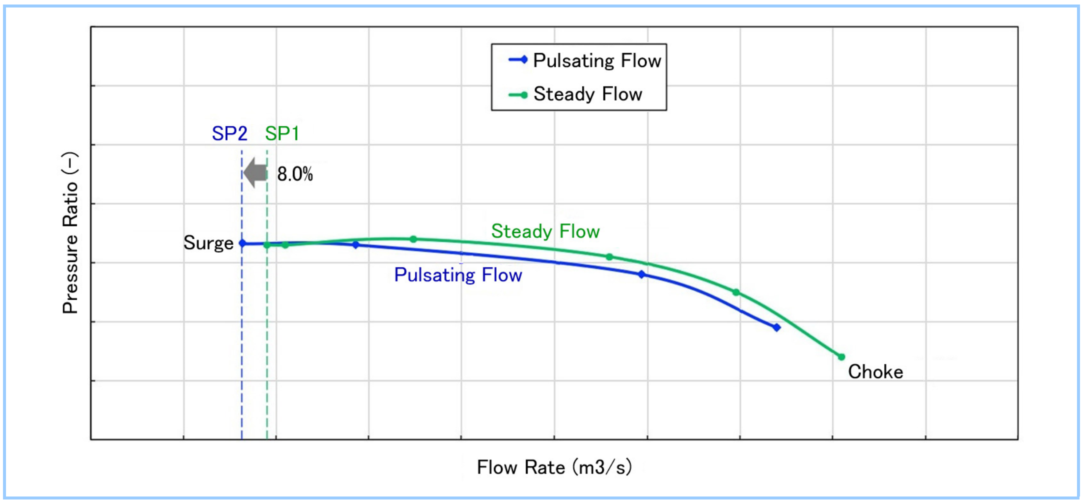

Figure 9　Relationship between presence or absence of pressure pulsation and surge flow rate

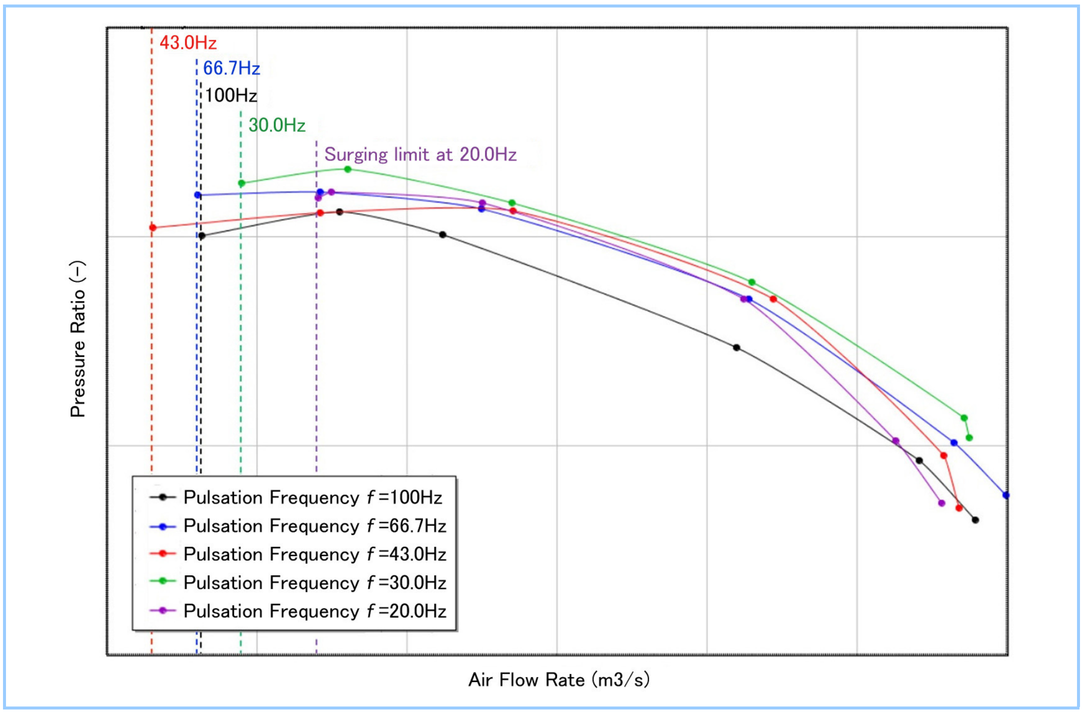

Figure 10 Relationship between pulsation frequency and surge flow rate

3.2 Result of unsteady flow analysis

In order to analyze the detailed internal flow under pressure pulsation, flow analysis was conducted under the conditions equivalent to that of the test result for the above reference waveform. **Figure 11** shows the comparison of the pressure flow characteristic between the numerical analysis result and the experimental result. In Figure 10, the characteristic curve shows good agreement, and it is considered that the analysis result adequately reproduced an actual flow. **Figure 12**(a) and (b) show the fluctuations of the velocity vector angle (flow angle) over time at the compressor impeller inlet and the diffuser outlet. In Figure 11, at the impeller inlet (a), the flow angle suddenly increases and decreases at several points corresponding to the pressure pulsation cycle, while at the diffuser outlet (b), the fluctuations synchronized with pulsation are small. This measurement was conducted at the operating point of a low-pressure ratio of around 1.7, where the speed reduction rate of the diffuser is smaller compared to that at other operating points. Accordingly, it is considered that the fluctuations synchronized with pulsation at the impeller rather than at the diffuser were shown.

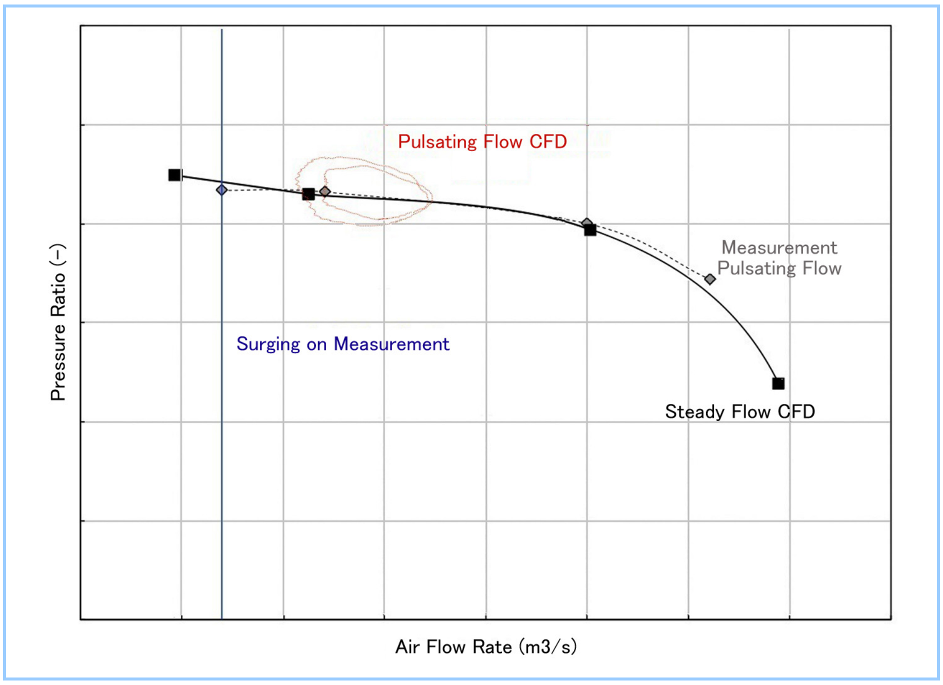

Figure 11 Pressure and flow rate characteristic (numerical analysis result and experimental result)

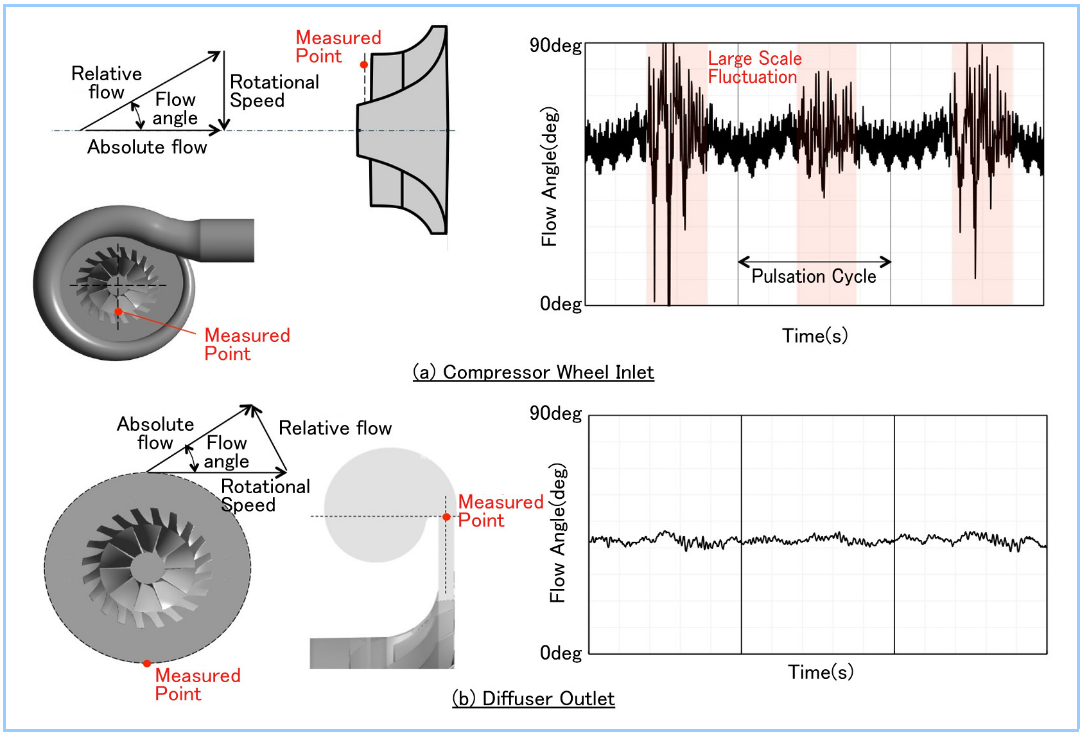

Figure 12　Fluctuations of flow angle at impeller inlet and outlet

In order to visualize the change in the internal flow of the impeller mentioned above, the velocity contour in the reverse flow region in the vicinity of the impeller tip at each unsteady analysis time (designated as Points A to D) is shown in **Figure 13**. The size of the reverse flow range at each of Points A to D corresponds to the intake air flow rate shown in the same figure and the reverse flow develops in the upstream side from the impeller inlet. So, it is considered that the above-mentioned fluctuations of the flow angle at the impeller inlet indicate the development and reduction of the reverse flow range. Especially, at Point C, which is the minimum flow point, a full-stall flow field is observed where the reverse flow region covers the area all around the impeller inlet and the pulsation causes a strong unsteady flow in which appearance and disappearance of full stall are repeated.

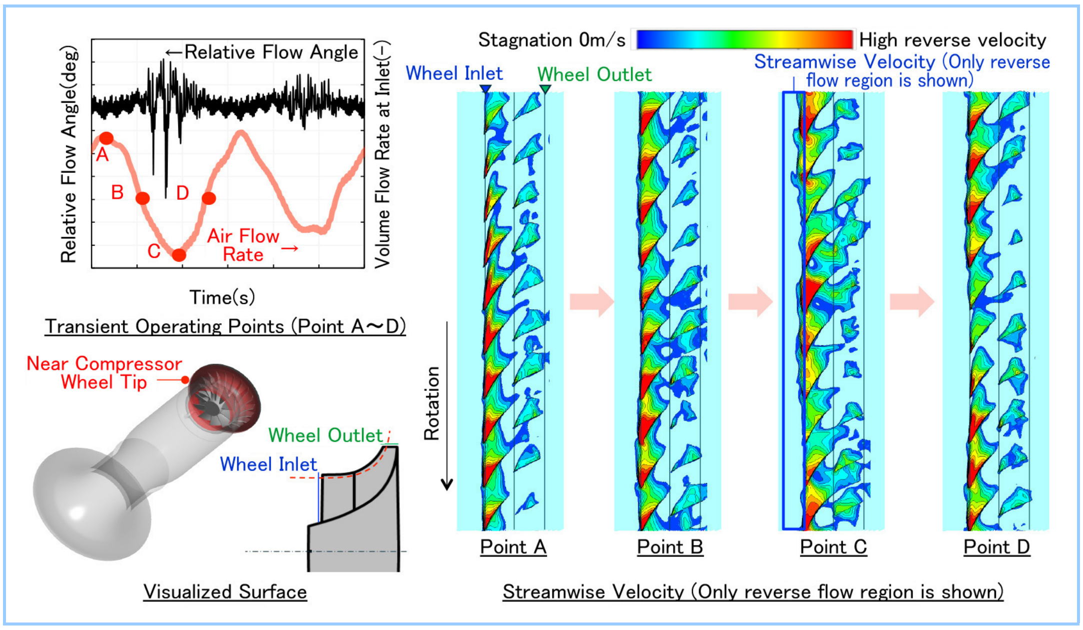

Figure 13　Distribution of reverse flow region inside impeller

In order to analyze the three-dimensional flow structure for the impeller internal unsteady flow described above, a visualization process for the vortex structure based on the singular point theory 7) was performed. **Figure 14** shows the three-dimensional vortex structure at each of Points A to D. The vortex structure is colored by nondimensional helicity (cosine of the velocity vector and the vortex vector), and +1.0 (red) in the figure shows a reverse flow and -1.0 (blue) shows a vertical vortex in the forward flow direction. In the same figure, the typical three-dimensional vortex structure patterns for a centrifugal compressor impeller, such as leakage vortex at the impeller tip and secondary vortex and horseshoe vortex at the impeller hub, can be observed. At each operating point of Points A to D, the leakage flow vortex at the tip exhibits significant changes such as inversed helicity by reverse flow, while the vortex structure at the hub maintains almost the same pattern. Correspondingly, according to the impeller inlet velocity distribution in the same figure, a significant change in the flow velocity occurs in the reverse flow region at the tip, but due to this effect, the fluctuations in the flow rate are cancelled at the tip, where the flow is almost steady. Accordingly, in the pulsating flow, the flow change at the impeller tip is predominant, and it is suggested that the surge margin can be further expanded through positive control of the flow at the tip .

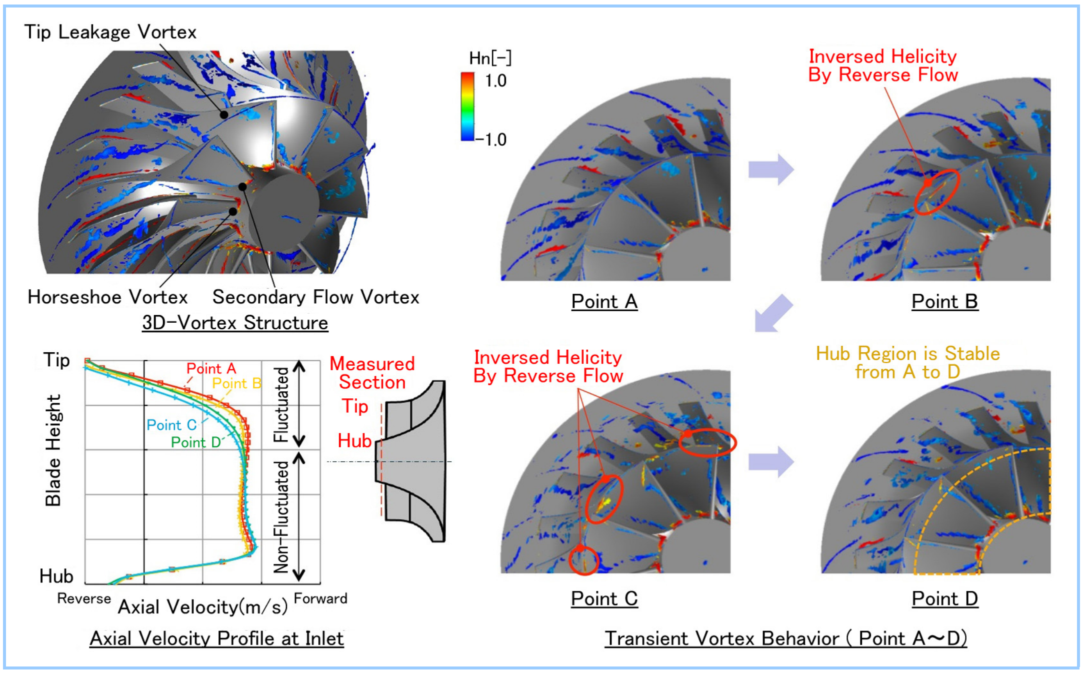

Figure 14 Three-dimensional vortex flow structure inside impeller

4. Challenges and future prospects

With this technology, the flow structure in a compressor under pressure pulsation was elucidated. The future challenges are further detailed measurement for verifying the analysis result and improvement of the precision of numerical analysis by a more advanced numerical analysis scheme/turbulence model. In the future, we would like to promote development of an aerodynamic device adaptable to pulsating flow by reflecting the findings obtained by this technology in the design.

5. Conclusion

In order to make detailed flow analysis in the vicinity of the surge point in a centrifugal compressor for a turbocharger, we developed a pressure pulsation test device which imitated an actual engine, conducted unsteady measurement for the surge characteristic under pulsation, which was conventionally difficult, and clarified unsteady internal flow. This technology facilitates extraction of the flow that relates to expansion of the surge margin and will contribute to expansion of the operating range of a compressor. In the future, we will continue to contribute to improving engine performance and environmental conservation through improving the performance of turbochargers.

The test device used in this study was developed in joint research with Imperial College London (U.K.), and we are grateful to Professor Ricardo Martinez Botas and Dr. Maria Esperanza Medrano of the same college.

*i Recirculating casing treatment: Device for expanding the operating range by connecting the impeller internal wall (casing) and the inlet pipe and forming a recirculation flow

*ii Helmholtz resonance frequency: Calculated by the frequency, sound velocity, cross-sectional area of opening, pipe length and pipe volume when resonance is produced in a pipe line having an opening

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