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EXPERIMENTAL EVALUATION OF MECHANICAL RELIABILITY OF THE IMPELLER BLADE FOR LARGE INTEGRALLY GEARED COMPRESSORS

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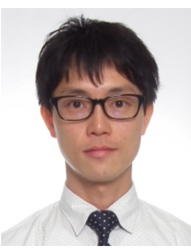

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ABSTRACT

Main air compressors used for air separation applications are important components that account for the majority of total power used in a plant. As a result, highly efficient integrally geared compressors (IGCs) are increasingly being used for these applications. Recently, plant sizes have increased to improve operation efficiency, therefore larger IGC and high flow coefficient impellers with high reliability are required. In terms of mechanical reliability, the evaluation of the blade's strength against the centrifugal force and the external excitation force is very important when using unshrouded impellers for IGC because of its high blade height.

In this paper, a non-contact blade vibration measurement that needs less effort to implement than strain gauges is employed. The confirmation of the accuracy of the dynamic blade stress evaluation method, and mechanical reliability evaluation of large IGCs' impellers are described. The blade vibration measurement system measures the blade passing time via non-contact sensors such as optical or eddy current probes. Synchronous and asynchronous vibrations of each blade can be analyzed on the basis of the deviation of the blade passing time in each rotation.

Firstly, the authors simultaneously measured the blade vibration and the strain on the blade surface with strain gauges on a scaled-down model compressor of a 1st stage actual IGC. The vibration displacement at the blade tip and the dynamic stress on the blade surface were evaluated under various operating conditions. The dynamic stress at the strain measurement points was estimated by measuring vibration displacement along with the FEM modal analysis results. The estimated value was closely in agreement with the measurement result from the strain gauges. Thus an evaluation method of the dynamic blade stress using blade vibration measurements was successfully established.

Finally, the authors measured the blade vibration for two types of actual IGCs with suction capacity of 313,000 Nm^3/h and 417,334 Nm^3/h, and evaluated their dynamic stress on the 1st and 3rd stage impellers. IGCs were operated at various flow rates, inlet guide vane angles, and speeds including under full load conditions, choke, and surge. The estimated dynamic stresses in the various operating points including the blade resonance and the surge were at sufficiently low levels when considering the material strength, confirming the mechanical reliability of the impellers.

INTRODUCTION

Centrifugal compressors are used in various plants such as chemical, petrochemical and oil refineries, most of which are operated over a period of more than 20 years with periodic maintenances. Since such industrial centrifugal compressors have high power consumption, highly efficient compressors are required so high mechanical reliability of the compressors is absolutely imperative to realize stable long time operation.

The impellers which are main parts of the centrifugal compressor are exposed not only to the mean stress due to the centrifugal force, but also to dynamic stress caused by resonance with engine order interactive excitation and unsteady flow including surge and choke. For this reason, the evaluation of mechanical reliability with respect to the fatigue strength of the impeller blade should be necessary. However, since it is difficult to predict dynamic stress preciously at the resonance, surge and the other conditions by simulation, experimental evaluation is often used. A typical technique for measuring dynamic blade stress during operation is a method directly measuring strain on the blade surface with strain gauges, and some measurement results are reported for industrial centrifugal compressors by Higashio et al. (2010), Wagner et al. (2011) and Jenny et al. (2016). Although this method can measure the strain directly related to the stress, there is a disadvantage in that signal transmission and reception between rotating and stationary parts become complicated. Conversely, there is a blade vibration measurement (BVM) method using non-contact displacement sensors as a technique to easily measure blade vibration behavior. In this case, it is necessary to estimate the dynamic stress from measured vibration amplitude. However, there are advantages in that there are no sensors to install on a rotating part. The vibration of all blades can be measured by the sensors mounted on a stationary part. The blade vibration measurement method is being applied to axial flow type large gas turbines and steam turbines; it is described by Tamura (2014), but there are no reports about measurement results for industrial centrifugal compressors.

A compressor used for an air separation application is required to have a large capacity, so un-shrouded (open) impellers with high flow coefficient are used. The high flow coefficient impeller has high blade height, and tends to have relatively low natural frequencies and stiffness. Therefore, we evaluated the dynamic stress of the blades for the validation of the blade strength design. For this purpose, the method for estimating the dynamic blade stress from the BVM result with the FEM modal analysis was introduced and it was experimentally validated by test operation of a scaled-down model compressor. This method was then also applied to the evaluation of the actual integrally geared compressors.

OVERVIEW OF TESTED COMPRESSORS

Main Air Compressor for ASU Applications

An air separation unit (ASU) is a plant that compresses air and separates it into oxygen, nitrogen, argon, etc. by cooling, liquefying and distillation. Industrial gas produced by ASU has been widely used for metal refining processes in steel manufacturing and inert gases in the process industry. Recently, it has become increasingly used for Integrated coal Gasification Combined Cycle (IGCC), Gas To Liquid plants (GTL), etc., aiming at effective utilization of coal and natural gas of which production areas are not limited compared to petroleum.

The main air compressors used for air separation applications are important components that account for the majority of total power used in a plant. As a result, highly efficient integrally geared compressors are widely used for these applications. The integrally geared compressor is a multistage compressor, and it is a great advantage in that it is highly efficient because it is possible to design the optimum rotational speed for each pinion shaft and to easily install the intercooler between stages. Recently, plants are becoming larger and more intensive in order to improve plant operation efficiency, so a main air compressor is required to be large and an increase in capacity and high reliability is also required.

Tested Integrally Geared Compressors

The tested large IGCs are referred to as model VG683 and VG691. The specification of the compressors is shown in Table 1. The impellers for all stages were high flow coefficient un-shrouded impellers. The VG683 is a three stage compressor, and the VG691 is single stage and it is larger than VG683. The VG691 was modified from the VG683 and uses the same gear set as the VG683. The specification of the scaled-down model compressor is also shown in Table1. It is a single stage compressor designed based on 1st stages of the VG683 and VG691. All the impellers are the same blading (i.e. same model).The blade vibration and the strain on the blade surface were simultaneously measured on the scaled-down model compressor, and the blade vibration was measured on the VG683 and VG691. The photograph and sectional drawing of the VG683 are shown in Figure 1 and 2 and the scaled-down model compressor is shown in Figure 3.

Table 1: Specification of Tested Large IGCs and Scaled-down Model Compressor

Model		VG683			VG691	Scaled-down model
Gas		Air			Air	Air
# of Stages		3			1	1
Suction Capacity	Nm^3/h	313000			417334	16430
Suction Pressure	MPaA	0.10			0.10	0.10
Suction Temperature	degC	16.0			16.0	30.0
Rated Speed	rpm	1st	2nd	3rd	4357	20353
		4979	4979	5688		
Impeller Tip Diameter	mm	1400	1120	900	1600	340
Impeller Tip Speed	m/s	365	292	268	365	362

Scaling down

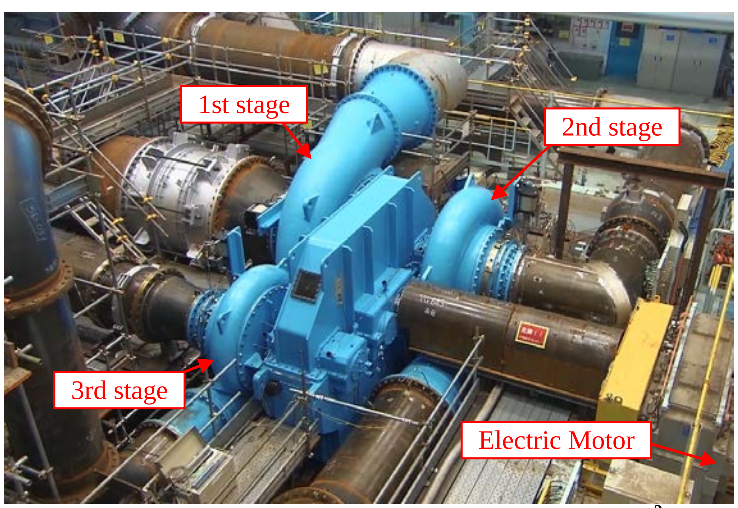


Figure 1: Tested Large IGC (VG683; Qs=313,000 Nm^3/h)

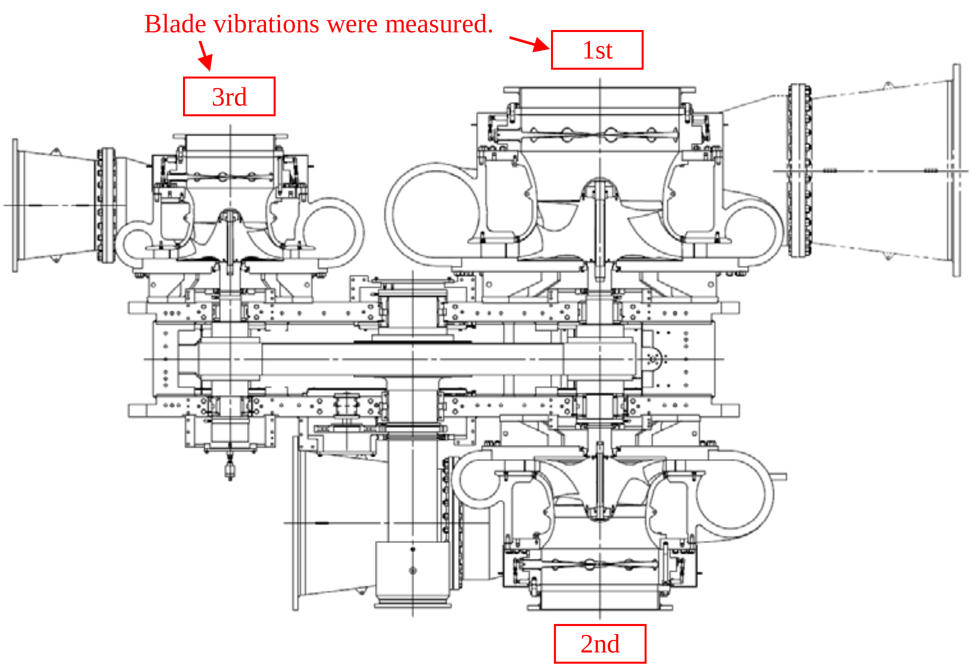


Figure 2: Sectional Drawing of Tested IGC (VG683)

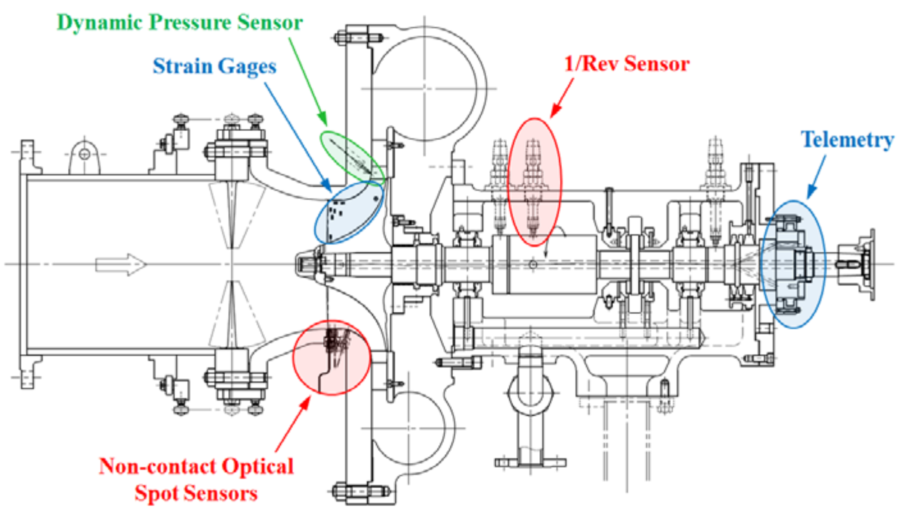


Figure 3: Sectional drawing for the scaled-down model compressor
(Scale factor = 1/4.12 of VG683 1st stage, 1/4.71 of VG691 1st stage)

Resonance Avoidance Design for Impellers

The resonance conditions of an impeller are determined by two factors, (1) interactive excitation between impeller blades and vanes on a stationary part, for example inlet guide vanes (IGVs) and diffuser vanes (DVs), and (2) vibration mode shapes including nodal diameter (ND) of impeller. Figure 4 shows the Campbell diagram of the scaled-down model impeller that indicates resonance condition (1).

Figure 5 shows the interference diagram of the scaled-down model impeller. The interference diagram (or Safe diagram, see Singh et al. (1988)) shows both resonance conditions (1) and (2) schematically. The horizontal axis is the nodal diameter of the impeller vibration mode. The vertical axis is the frequency. Resonance conditions are shown as intersections of the zigzag line with specified excitation lines such as rotational speed and multiples of it including stator vane numbers (i.e. 9X, 12X, 15X, 18X, 24X, …). If natural frequency is coincident with these intersections, resonance will occur. The operating conditions and specifications of the impellers are shown in Table 2. For the scaled-down model impeller, we investigated in detail for 12 and 15 diffuser vanes conditions. Regarding the 12 diffuser vanes, our standard vane position and the position closer to the impeller than our standard were investigated.

In the design phase of an impeller, resonance avoidance design is carried out based on natural frequencies calculated by FEM modal analysis. In order to validate the calculated result, identification of natural frequencies and vibration modes of the impeller for the scaled-down model compressor was conducted by experimental modal analysis. In the experimental modal analysis, the scaled-down model impeller was mounted on a base plate with a similar fixing to the rotor. The hammering test was performed and the frequency-response functions at each hammering point were measured and then the natural frequencies and mode shapes were identified. Natural frequencies and their nodal diameters calculated by FEM analysis and identified by experimental modal analysis were plotted in Figure 5, and the example of vibration mode shape was shown in Figure 6. The deviations of natural frequencies between measurement and FEM modal analysis were less than 5%, and the mode shapes by FEM analysis and measurements were in good agreement. Thus the validity of the FEM modal analysis was confirmed.

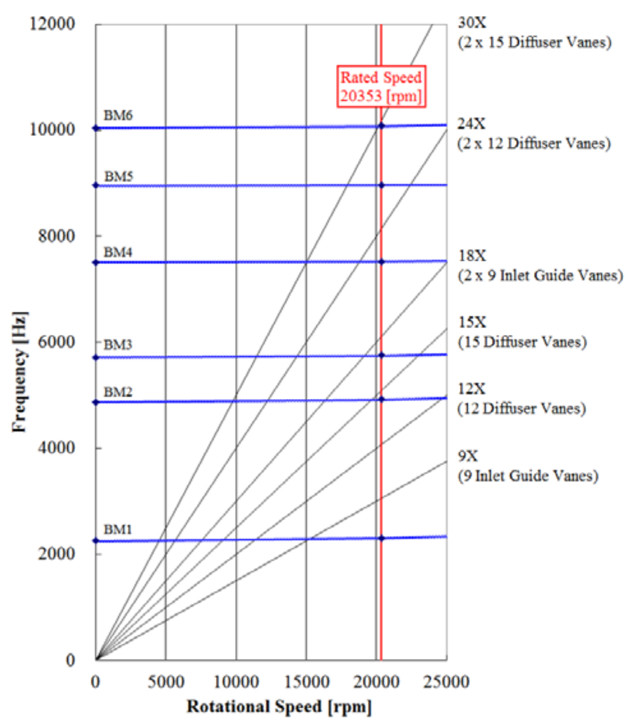


Figure 4: Campbell Diagram of the Scaled-down Model Impeller (FEM Result)

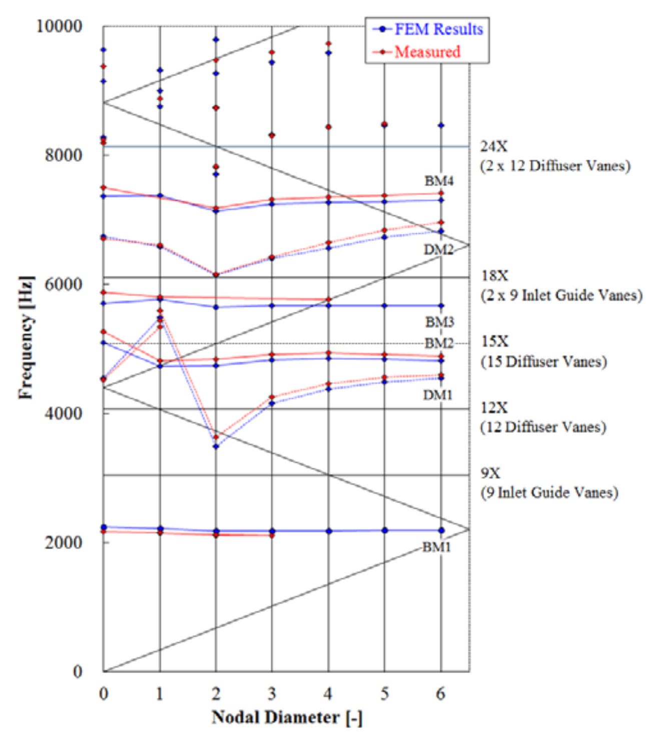
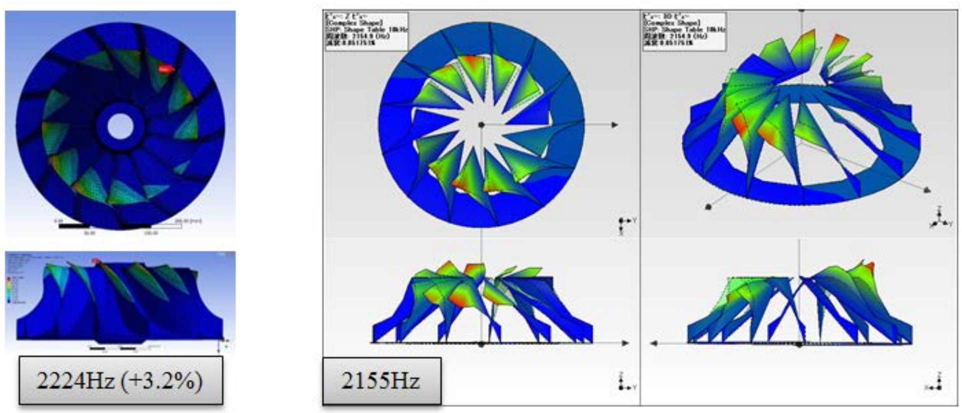


Figure 5: Interference Diagram of the Scaled-down Model Impeller

Table 2: Operating Conditions and Specifications of the Investigated Impellers

Model		VG683		VG691	Scaled-down model	
Stage		1st	3rd	1st	1st	
Rated Speed	rpm	4979	5688	4357	20353	
Impeller Tip Diameter	mm	1400	900	1600	340	
Impeller Tip Speed	m/s	365	268	365	362	
# of Impeller Blades		13	13	13	13	
# of Inlet Guide Vanes		11	9	n/a	9	
# of Diffuser Vanes		12	12	12	12	15
Position of Diffuser Vanes		Standard	Standard	Standard	Standard or 8% Closer Position	Standard



(a) FEM Modal Analysis Result **(b) Experimental Modal Analysis Result**

Figure 6: Typical Natural Vibration Mode Shape of the Scaled-down Model Impeller (BM1, 1ND)

EXPERIMENTAL METHODOLOGY

Direct Measurement Method Using Strain Gauges

A direct measurement method using strain gauges has been commonly used for a long time as a dynamic blade stress evaluation method while operating. This method needs a slip ring or telemetry system to transfer strain signals from a rotating part to a stationary part.

In the testing of the scaled-down model compressor, strain gauges were attached on the surface of the blades, as shown in Figure 7. Strain signals were transferred to stationary parts by the digital telemetry system with 19 kHz output (See Figure 3). In order to measure peak amplitude of blade dynamic stress, single axis strain gauges were attached at a high stress position in the direction of maximum principal stress in each vibration mode identified by the FEM modal analysis.

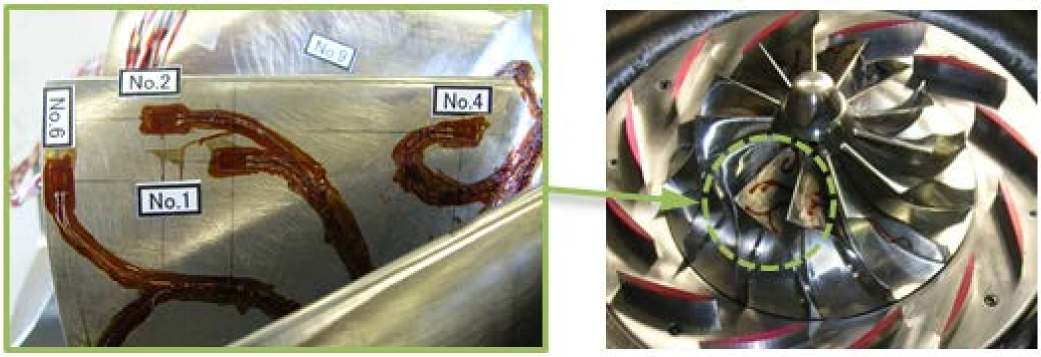


Figure 7: Strain Gauges on the Scaled-down Model Compressor

Dynamic Blade Stress Estimation Method Using Blade Vibration Measurement and FEM Analysis

Measurement Principle of Blade Vibration Measurement

The blade vibration measurement (BVM, so-called blade tip-timing measurement) system measures the blade passing time via non-contact sensors such as optical probes shown in Figure 8 or eddy current probes. Blade vibration amplitudes and frequencies are analyzed based on a deviation in blade passing time per rotation. The non-contact sensors are installed in the stationary casing, and their output analog signals of blade tip passing are triggered to generate pulses as shown in Figure 9. When the blade tip oscillates, the time difference between the tip passing timing and non-oscillating pulse signal for shaft rotational speed detection changes, and the blade vibration is analyzed based on the deviation of the passage timing for each rotation. Here, the blade vibration to be measured is a circumferential component against the actual out-of-plane vibration of the blade. For this measurement, the blade tip passing speed was about 270 m/s at maximum, and the high-frequency 80 MHz clock was used for precise timing of the blade passing pulse (i.e. vibration measurement resolution of about 3 um).

Since non-contact sensors are mounted on the stationary part, the vibration response of the blade can be measured more easily than strain gauge measurements attaching the strain gauge to the rotating part (blade surface), and further it is possible to measure vibration of all blades. In addition, by installing a number of sensors in the circumferential location, the vibration waveform is produced from the pulse signals passing through the blade, thereby improving the accuracy in analyzing the amplitude and the frequency. In these measurements, we installed 5 or 6 sensors in the circumferential direction. See Appendix A for blade vibration data analysis method.

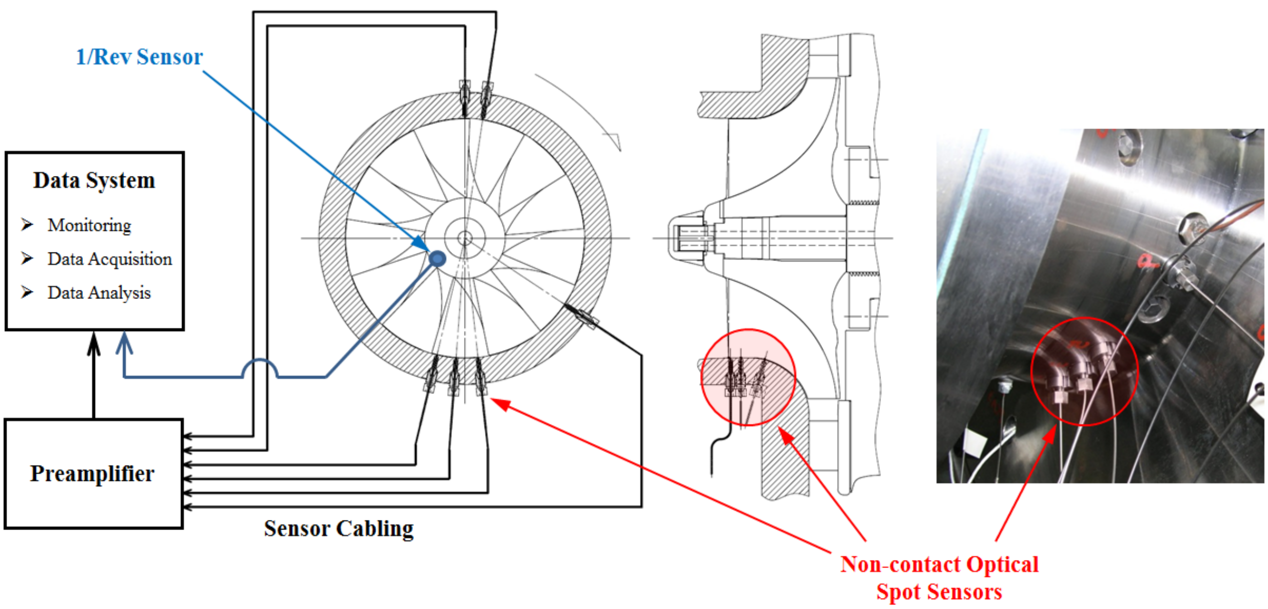


Figure 8: Optical Spot Sensors for Blade Vibration Measurement for the Scaled-down Model Compressor

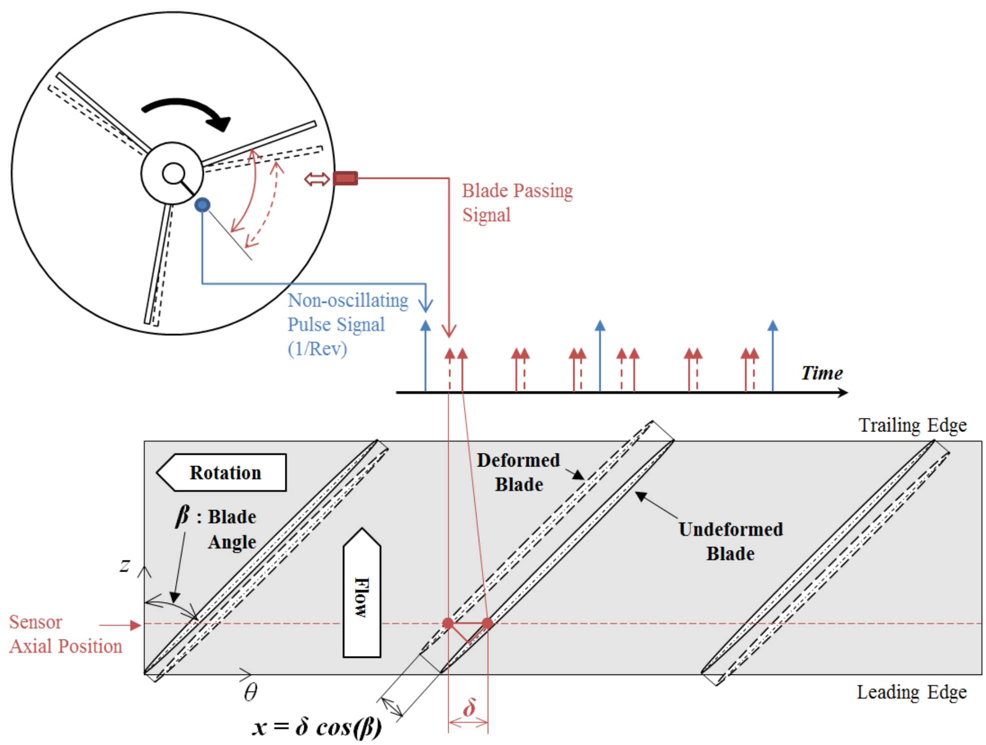


Figure 9: Measurement Principle of Blade Vibration Measurements

Dynamic Blade Stress Estimation Method

When the mode shape of the impeller changes, the peak amplitude of the dynamic stress and the location where it occurs are different even if the vibration displacement of the blade tip is the same. Therefore, it is important not only to measure tip vibration amplitude, but also specify the vibration mode. The displacement-stress relationship in a specific vibration mode can be obtained by FEM modal analysis as shown in Figure 10. When the vibration mode shape and the blade tip amplitude can be specified by BVM, it is possible to estimate the dynamic stress using the displacement-stress relationship in the corresponding vibration mode. At this time, circumferential direction component which is the direction of BVM is used for the stress estimation. In order to reduce the estimation error, it is good to measure the tip vibration at the largest positions in each vibration mode.

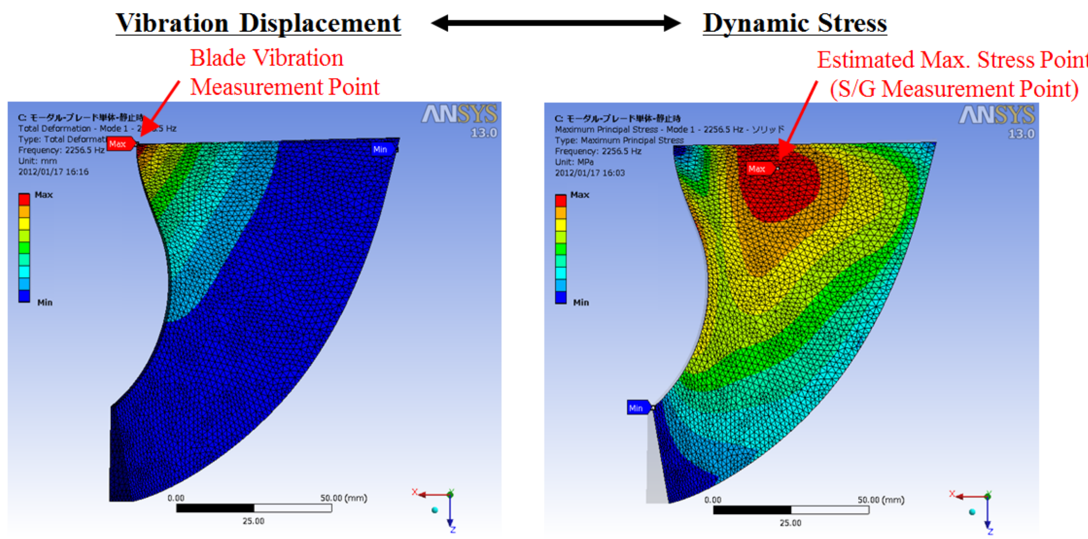


Figure 10: Typical Result of FEM Modal Analysis (Scaled-down Model Impeller, BM1)

RESULTS ON THE SCALED-DOWN MODEL COMPRESSOR
STEP 1: DYNAMIC BLADE STRESS EVALUATION USING STRAIN GAUGES

Strain Gauge Measurement Results at Blade Resonance (Rotational Speed Sweeping)

For blade resonance, strain was measured during start-up and coast-down at IGV angle = 0 degrees (full open) and various flow rates. Figure 11 shows measured dynamic stress more than 0.2% of σ_w on the Campbell diagram in conditions of start-up and number of diffuser vanes = 12. The telemetry system noise was removed in this figure. This dynamic stress comes from measured data of a strain gauge at the peak position of BM1 (Blade Mode 1). Blade resonance due to rotor-stator interference was found when the harmonics of rotational frequency multiplied by the number of stator vanes accord with blade natural frequency (12X from Diffuser Vane, 9X and 9X x 2 = 18X from IGV). It is shown that the level of dynamic stresses is sufficiently low against the material strength.

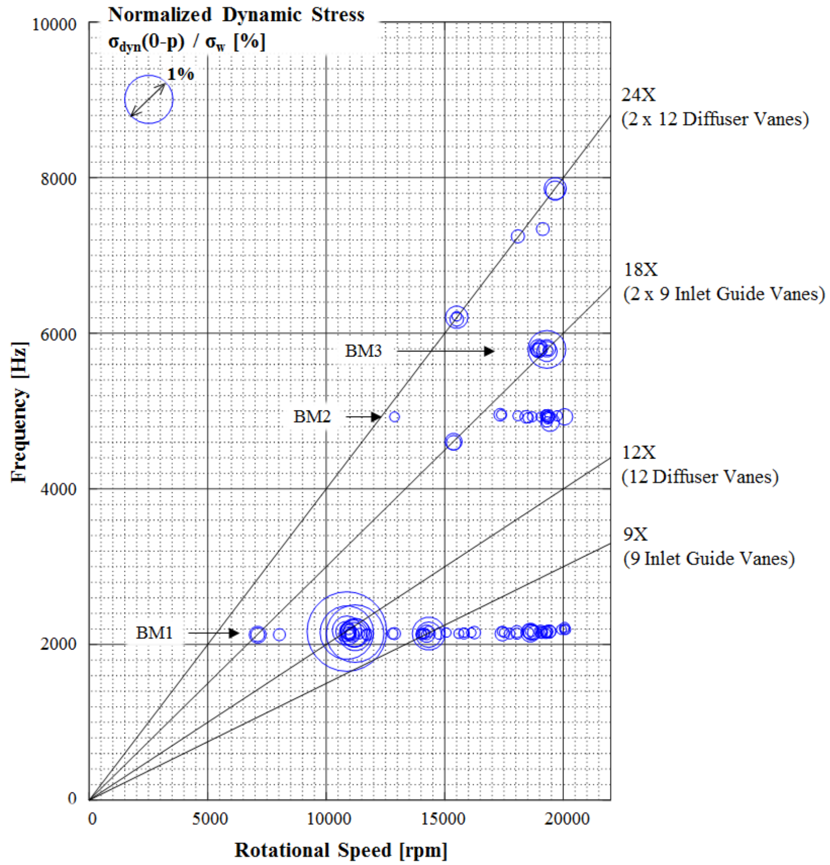


Figure 11: Measured Campbell Diagram of the Scaled-down Model Impeller (12 Diffuser Vanes, 9 Inlet Guide Vanes)

Strain Gauge Measurement Results at Rated Speed

Strain was measured at various IGV angles and flow rates at constant rated speed. Figure 12 shows the peaks of dynamic blade stresses and dynamic pressures on the flow path surface in case of the number of DV 12. The horizontal line shows three flow rate conditions of Max (full opened discharge valve, or choke), Mid (design point at IGV 0degrees) and Min (surge). The highest dynamic blade stress and dynamic pressure at rated speed are found at surge. Surge is the phenomenon of flow pulsation of the whole system including piping, the blades are excited by instantaneous and intermittent reverse flow when surge occurs. Figure 13 shows fatigue limit diagram (modified Goodman diagram) of measured stress at surge. The horizontal axis shows measured mean stress, and the vertical axis shows measured dynamic stress. Dynamic blade stress at surge is higher than the other operating conditions including blade resonance, but still sufficiently low against the material strength.

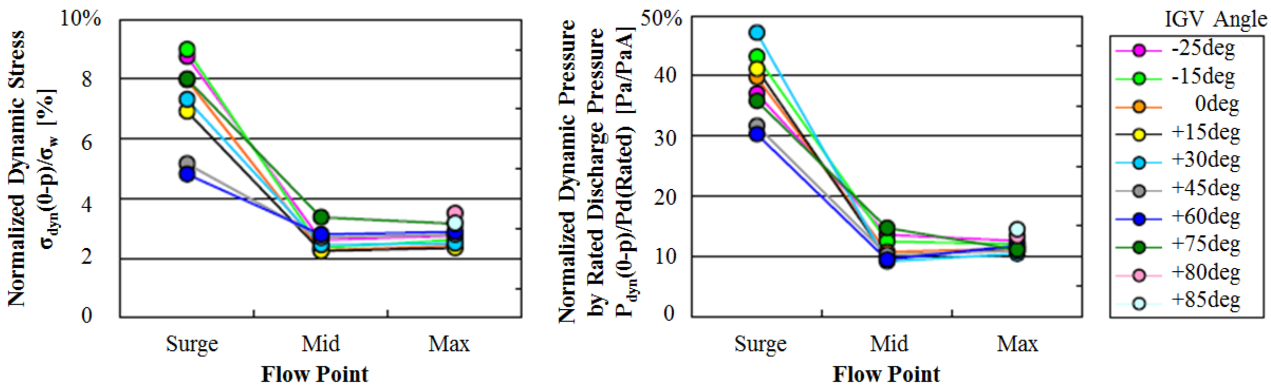


Figure 12: Measured Dynamic Blade Stress and Dynamic Pressure at Impeller Outlet (at Rated Speed of the Scaled-down Model Compressor, 12 Diffuser Vanes)

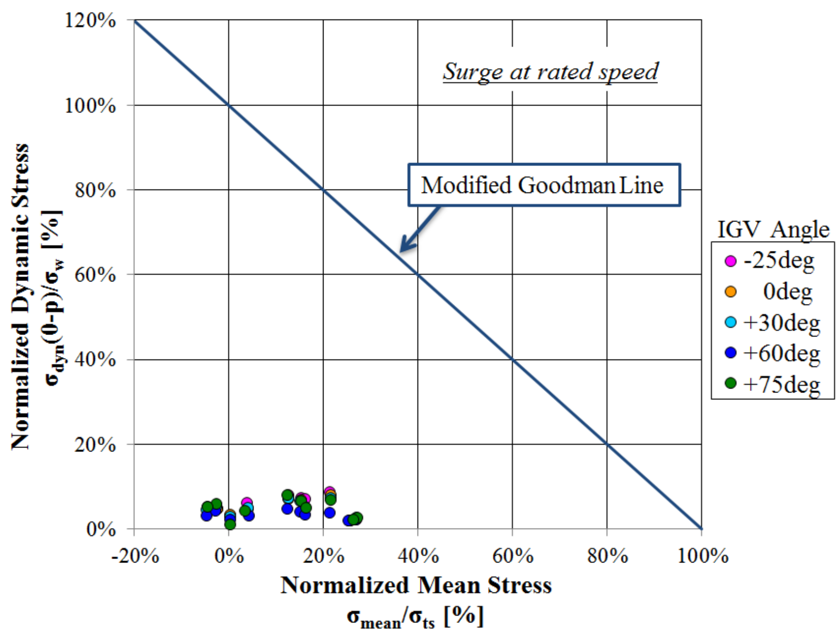


Figure 13: Measured Blade Stress at the Surge (at Rated Speed of the Scaled-down Model Compressor, 12 Diffuser Vanes)

Strain Gauge Measurement Results at Constant Rotational Speed

Figure 14 shows measured results of dynamic blade stress and dynamic pressure at impeller outlet at four constant resonant speeds. From these results at various flow rates, both dynamic stress and dynamic pressure peak at the surge of each speed. However, the peak level is lower than the dynamic stress at surge at rated speed (Compare with Figure 12).

Figure 15 shows measured frequency responses of dynamic blade stress at 94.6% and 96.4% constant resonant speeds. The higher order resonant modes (BM3 and BM4) are mainly excited at middle flow, but at surge, BM1 is a dominant mode of dynamic blade stress even when it is in the blade resonance condition.

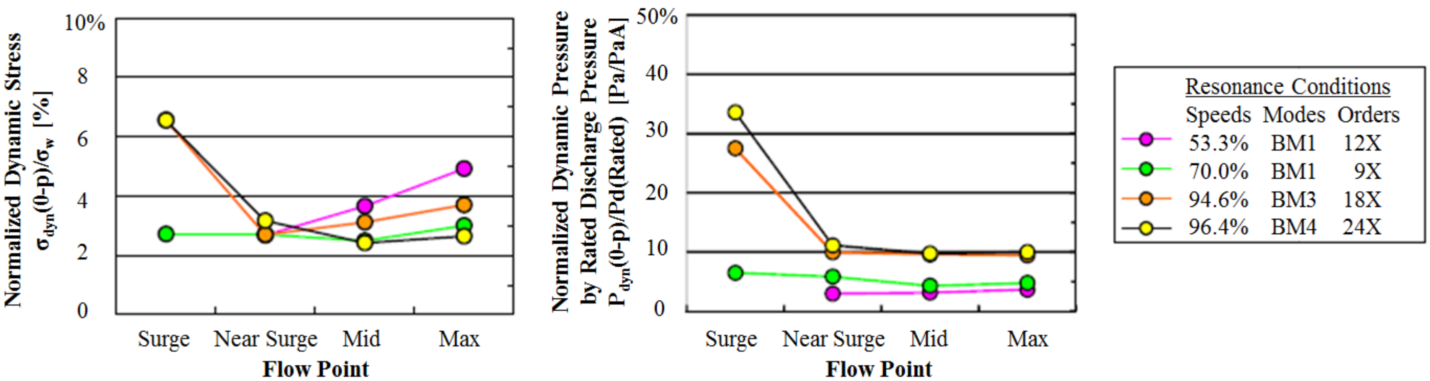


Figure 14: Measured Dynamic Blade Stress and Dynamic Pressure at Impeller Outlet
(at Constant Rotational Speed of the Scaled-down Model Compressor, 12 Diffuser Vanes and 9 Inlet Guide Vanes)

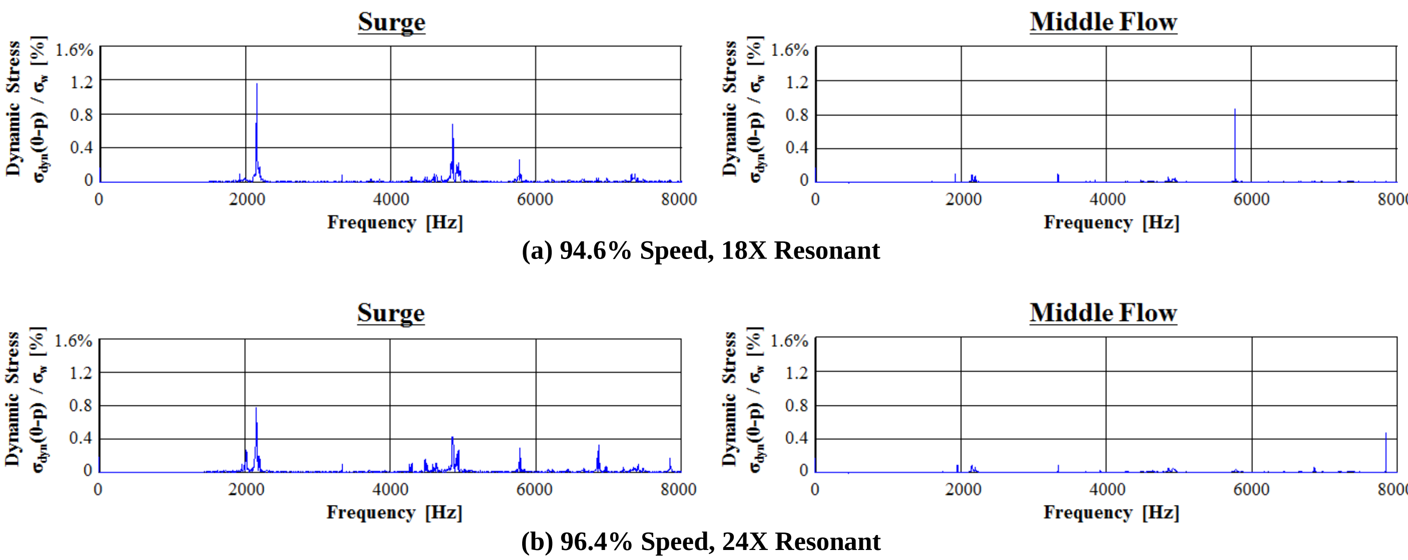


Figure 15: Measured Dynamic Blade Stress of the Scaled-down Model Compressor (12 Diffuser Vanes, 9 Inlet Guide Vanes)

Summary of the Strain Gauge Measurement Results

Dynamic blade stress of the scaled-down model compressor (refer to Table 1 for the specification) was measured by strain gauges at various flow rates, IGV angles, and speeds including blade resonance points, choke, rated point and surge. The measurement results can be summarized as follows.

- The highest dynamic blade stress was measured at surge at rated speed.
- Even when the compressor is in the blade resonance condition, BM1 was a dominant mode of dynamic blade stress at surge.

RESULTS ON THE SCALED-DOWN MODEL COMPRESSOR
STEP 2: VALIDATION OF DYNAMIC STRESS ESTIMATION METHOD USING BLADE VIBRATION MEASUREMENT AND FEM MODAL ANALYSIS

Overview of the Validation of Dynamic Stress Estimation Method

The dynamic blade stress estimation method as describe above, was validated with strain gauges in the scaled-down model compressor. The strain gauge measurement and the blade vibration measurement (BVM) were simultaneously carried out at various operating conditions including blade resonance and surge, and those measured results were compared. Both of those measurement results at surge are higher than the other operating conditions including blade resonance, and the trend of the level relation corresponds well at each operating conditions. The two validation results at blade resonance and surge are described below.

Validation of Dynamic Stress Estimation Method at Blade Resonance

According to the blade resonance response in the condition of coast-down, Figure 16 shows two dynamic blade stress responses in 15X (in the case of number of diffuser vanes = 15), one from the strain gauge measurement and the other from estimation using BVM and FEM. Five measured blade vibrations at different circumferential positions were analyzed by the synchronous analysis to extract amplitudes for each blades (see Appendix A (1) for the synchronous analysis method). Then, the dynamic stress at the same position as strain gauge was estimated using FEM result, and compared with another result simultaneously measured by strain gauge. Both estimated resonance response and resonance frequency are in good agreement with the strain gauge results.

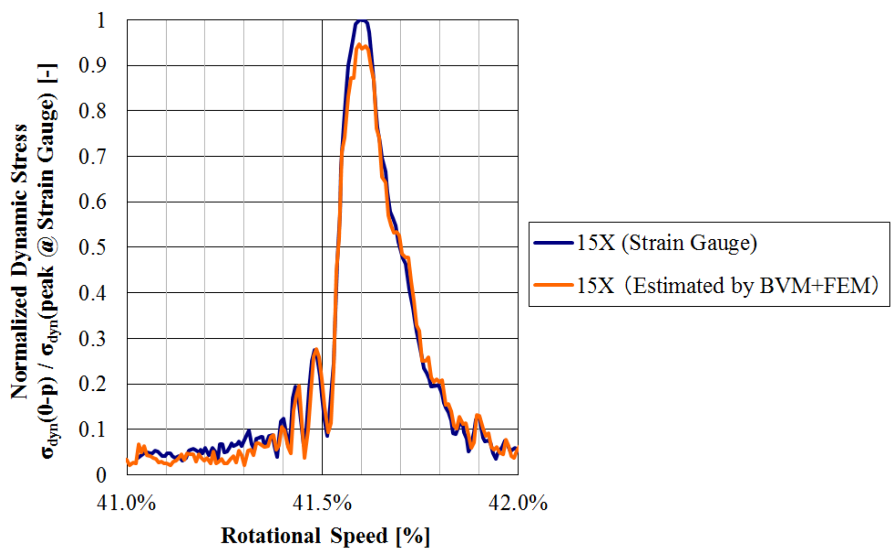


Figure 16: Typical Measured Dynamic Stress at the BM1 Blade Resonance of the Scaled-down Model Compressor (15 Diffuser Vanes, 9 Inlet Guide Vanes)

Validation of Dynamic Stress Estimation Method at Surge

Figure 17 shows the two types of simultaneously measured time waveform and spectrum at the surge of the IGV angle = 0 degrees, one is dynamic stress measured by strain gauge, the other is blade vibration measured by Tip-timing (see Appendix A (2) for the asynchronous analysis method). Blade vibration is the measured results at the inlet, which has the largest amplitude at the tip. From these time waveforms, the blade is excited by flow pulsation in about 4 Hz from the plenum volume (system capacitance including piping), and two measured responses have a good agreement in this low frequency time waveform. From the spectrum of the strain gauge data, dynamic blade stress consists of some vibration modes, but the blade 1st mode (BM1) is a dominant mode for it. This is why BM1 is targeted for the following comparison between two measurements at surge. Figure 18 shows the relationship for BM1 dynamic stresses between strain gauge results and estimated results. These are the results in several measurement conditions of IGV angle and diffuser vane types, and also there are high stress groups in the condition of 8% closer diffuser vane position than our standard position. It is shown that the BM1 dynamic stresses are estimated in the error range of -30% to +30%. Maximum dynamic stress of closer DV position case is about ten times higher than normal cases, but still it is under the half level of material 10^7 cycle fatigue limit stress.

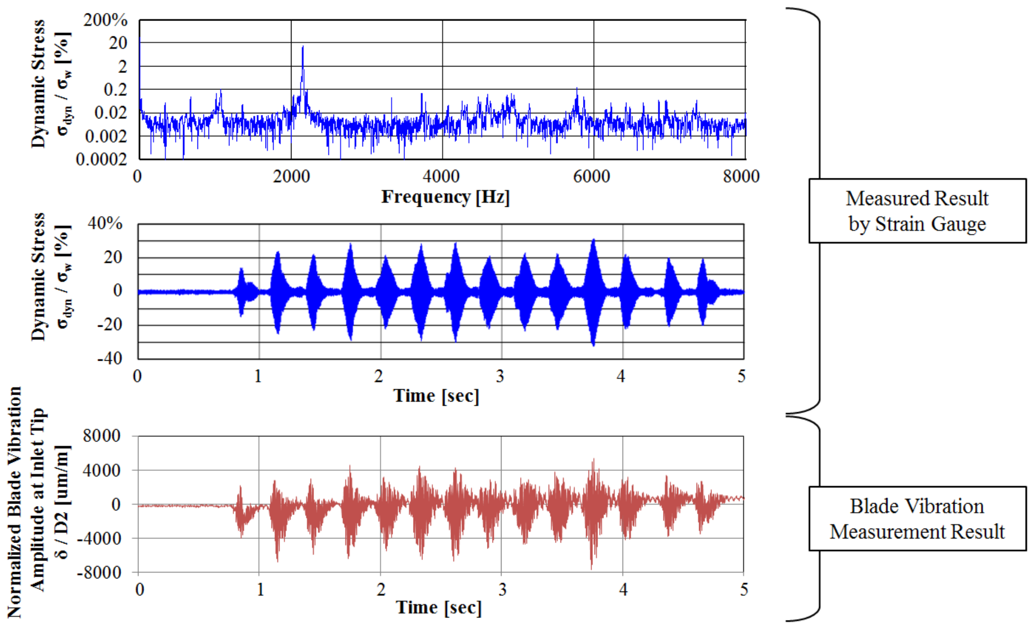


Figure 17: Example of Measured Blade Response Time Waveform and Spectrum at Surge (Scaled-down Model Compressor with 12 Diffuser Vanes at Closer Position)

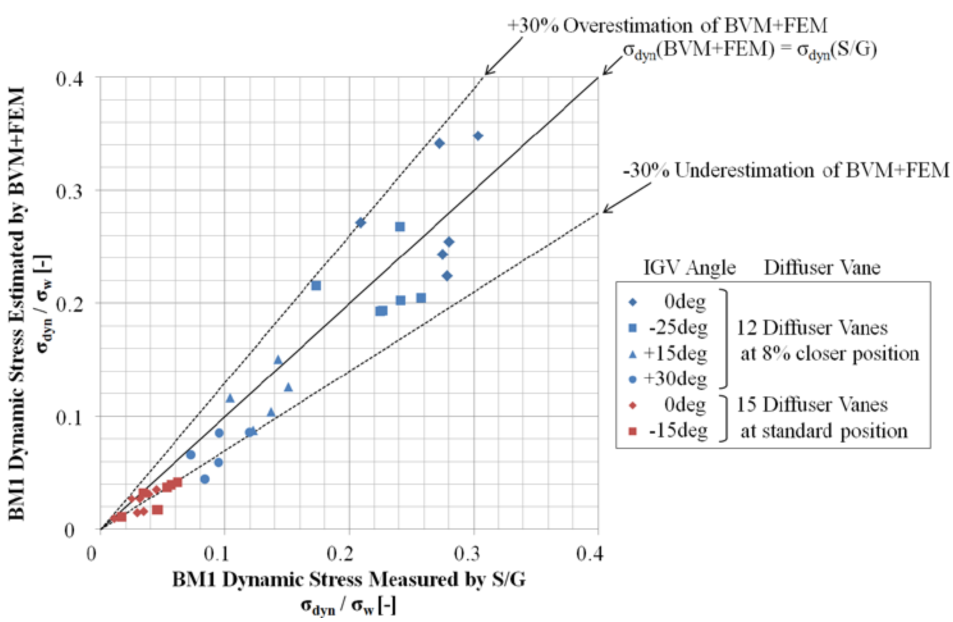


Figure 18: Dynamic Blade Stress of Blade 1st Mode at the Surge of the Scaled-down Model Compressor

Summary of the Validation Results of Dynamic Stress Estimation Method

The dynamic blade stress estimation method using blade vibration measurement and FEM modal analysis was validated with strain gauge results at various operating conditions including blade resonance and surge in the scaled-down model compressor. The validation results can be summarized as follows.

- At blade resonance, both estimated dynamic blade stress and frequency were in good agreement with the strain gauge results.
- At surge, estimated dynamic stresses of the dominant BM1 were in the range of -30% to +30% of the strain gauge results.

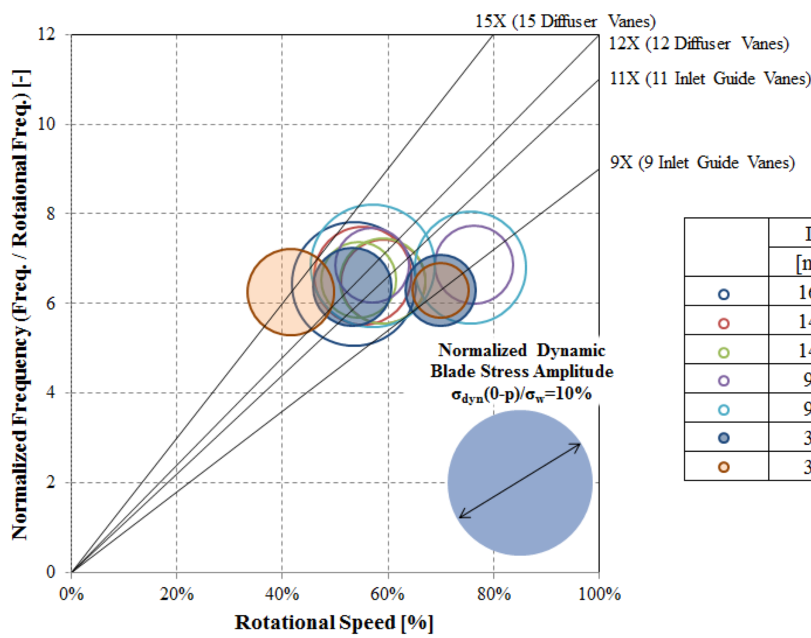
EVALUATION OF LARGE IMPELLERS' DYNAMIC BLADE STRESS

Measured Impellers of Large IGCs

A blade design for large IGCs was validated using established estimation method for dynamic blade stress. Measured impellers are 1st and 3rd stage impellers of three stages IGC 'VG683' with a suction capacity of 313,000 Nm^3/h (D2 = 1400 and 900 mm), and 1st stage impeller of single stage IGC 'VG691' with a suction capacity of 417,334 Nm^3/h (D2 = 1600 mm). All three impellers are the same type as the scaled-down model impeller (D2 = 340 mm, similarity shapes). See Table 2 for these specifications.

Evaluation of Dynamic Blade Stress at Blade Resonance

For each impeller of large IGCs, dynamic blade stresses at blade resonance were estimated by following the same process as the scaled-down model impeller. Figure 19 shows a Campbell diagram with each BM1 resonance stress. The horizontal axis shows rotational speed ratio normalized by rated speed, and the vertical axis shows frequency normalized by rated rotational frequency. The plotted data includes the difference of impeller size, and also harmonic index of resonance condition (9X and 11X for IGV, 12X for Diffuse Vane) and IGV angle. Regardless of size, dynamic blade stresses at BM1 resonance are almost the same level as the scaled-down model impeller, and they are sufficiently low against the material strength.



	D2	IGV		DV
	[mm]	Angle [deg]	# of Vanes	# of Vanes
○	1600	n/a	n/a	12
○	1400	+85	11	12
○	1400	+90	11	12
○	900	0	9	12
○	900	+75	9	12
●	340	0	9	12
○	340	0	9	15

Figure 19: Estimated Dynamic Blade Stress at Blade Resonance (BM1)

Evaluation of Dynamic Blade Stress at Rated Speed

For each impeller of large IGCs, blade vibration data was acquired at various operating points including design point and surge. Figure 20 shows each compressor performance curves at IGV angle = 0 degrees with circles, those diameter indicate blade vibration overall amplitudes at impeller inlet tip normalized by impeller tip diameter, D2. Both the scaled-down model compressor and the large IGCs have the maximum blade vibration amplitude at surge. From this and strain gauge results for the scaled-down compressor, it is suggested that dynamic blade stress at surge is the highest of the other operating points.

Figure 21 shows each time waveform of the measured blade vibration at surge. All impellers are excited by low frequency flow fluctuations like impact force of hammering and the blades response instantaneously. Figure 22 shows maximum BM1 dynamic stresses; estimated values in red and blue, measured values by strain gauge in green. All blades average and standard deviation of estimated values are indicated in the red bar and error bar. High stress bars are the data for the scaled-down model compressor in the condition of closer diffuser vanes position than our standard position. According to the data of scaled-down model compressor, the estimated BM1 dynamic stresses are in good agreement with the strain gauge results. The estimated BM1 dynamic stresses of each large impeller are almost the same as the scaled-down model impeller with our standard diffuser vane setting, and much lower than the high stress cases of nearer diffuser vanes. Thus it is evaluated that the dynamic stress of the large IGC impellers is sufficiently low against the material strength.

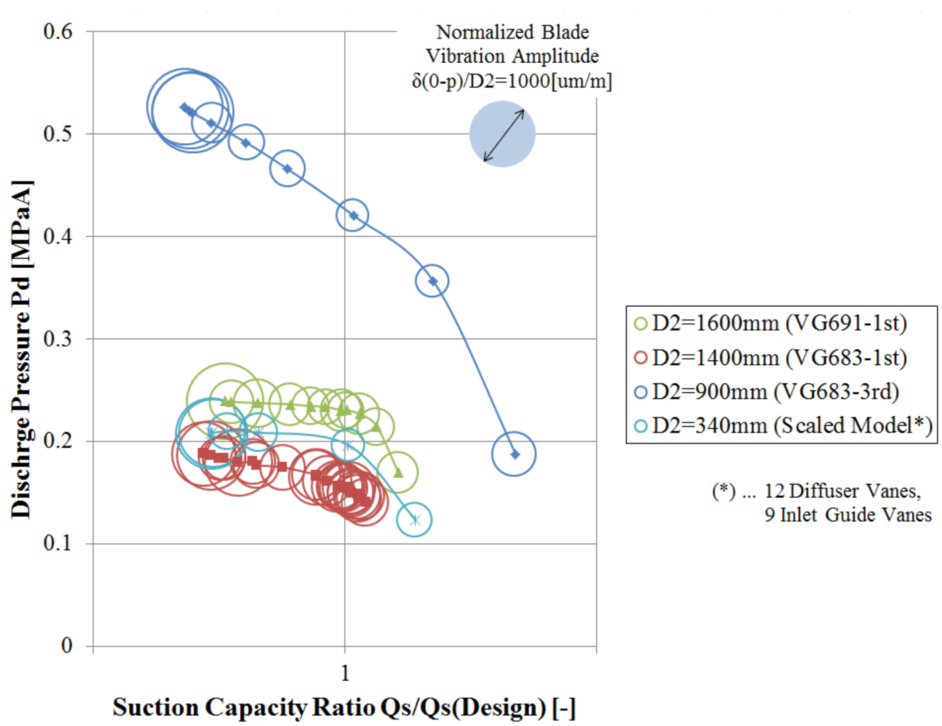


Figure 20: Operating Conditions and Measured Blade Vibration Amplitudes at Inlet Tip (IGV Angle=0deg, Rated Speed)

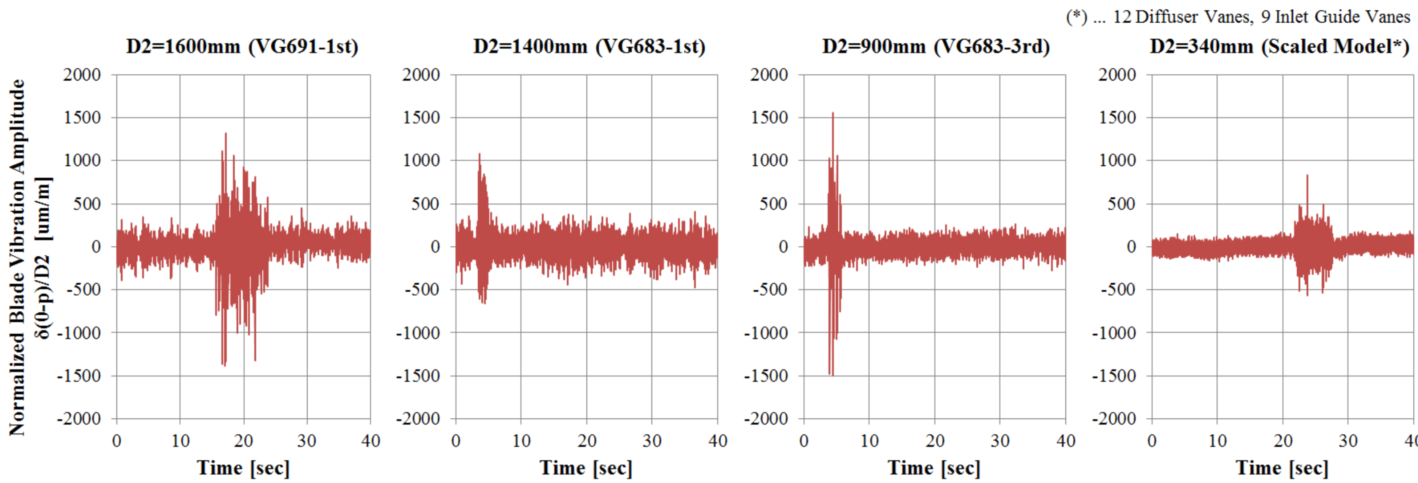
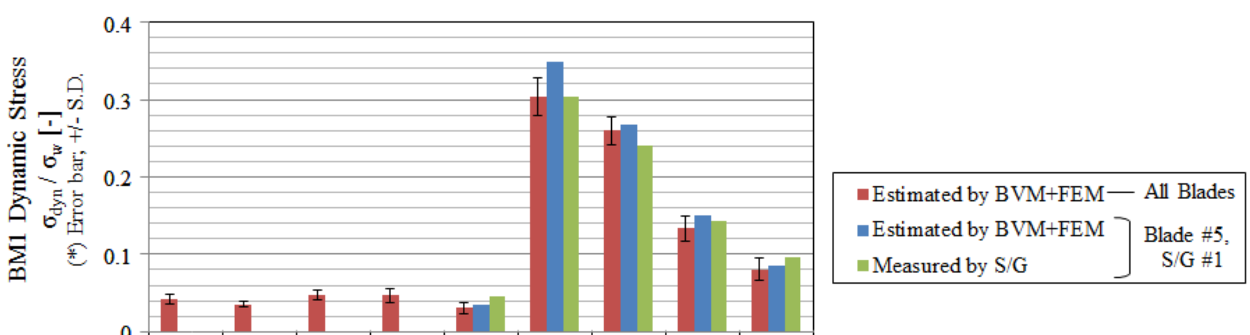


Figure 21: Measured Blade Vibration at Inlet Tip at Surge (at Rated Speed)



D2	[mm]	1600	1400	1400	900	340	340	340	340	340
IGV Angle	[deg]	n/a	0	+85	0	0	0	-25	+15	+30
Diffuser Vane	# of Vanes	12					12			
	Position	Standard Position					8% Closer Position			

Figure 22: Blade 1st Mode's Dynamic Stress at the Surge

CONCLUSIONS

High flow coefficient un-shrouded impellers for air separation applications tend to have lower stiffness than impellers with a lower blade height (i.e. lower flow coefficient). For this reason, a blade design validation based on measured dynamic stress was needed. In this paper, the evaluation method and its application examples using blade vibration measurement were introduced. Blade vibration measurement used non-contact sensor on the stationary part, so it needs less effort to implement than strain gauges.

The dynamic blade stress estimation method using blade vibration measurement and FEM modal analysis was validated with the data from the scaled-down model compressor that was simultaneously measured by strain gauges. Following this validation, dynamic blade stress of large integrally geared compressors at various operating conditions including blade resonance and surge was evaluated by the estimation method, and the mechanical reliability of the large impellers was confirmed. The tested large integrally geared compressor (VG683) was shipped to a customer and it has been running successfully.

NOMENCLATURE

D2	= Impeller tip diameter
Pd	= Discharge pressure
P_{dyn}	= Dynamic pressure
Qs	= Suction capacity
σ_{mean}	= Mean stress
σ_{dyn}	= Dynamic stress
σ_{ts}	= Tensile stress
σ_w	= 10^7 cycle fatigue limit stress
δ	= Blade vibration amplitude at the blade tip (Circumferential component)

Acronyms

ASU	= Air Separation Unit
BM	= Blade Mode
BVM	= Blade Vibration Measurement
DM	= Disc Mode
DV	= Diffuser Vane
FEM	= Finite Element Method
IGC	= Integrally Geared Compressor
IGV	= Inlet Guide Vane
ND	= Nodal Diameter
S/G	= Strain Gauge
0-p	= zero to peak

APPENDIX A

Tip-timing Data Analysis Method for Blade Vibration

Tip-timing Data Analysis Method is divided broadly into two categories; (1) Synchronous analysis method, and (2) Asynchronous analysis method. Each has some methods, and the methods adopted in this paper are described below (refer to Heath et al. (1998)).

(1) Synchronous analysis method

Synchronous analysis is mainly performed for blade resonance caused by rotor-stator interference. The analysis is using Circumferential Fourier Fit (i.e. Direct Analysis Methods) that is a least-squares sine fitting technique assuming an integral engine order sinusoidal wave. The fitting is performed based on circumferential position of sensors and blade passing timing at each sensor, then it can give the vibration amplitudes and frequency. In order to reconstruct the signal, twice the frequency of the original signal is needed for sampling based on Nyquist sampling theorem. But in case of blade vibration measurement for turbomachinery, the number of sensors located circumferentially is normally fewer than this sampling criterion for rotor-stator interference, so the frequency cannot be underspecified. Therefore the frequency and amplitude is determined by fitting error and the resonance condition (i.e. number of stator vanes).

(2) Asynchronous analysis method

The following two analysis methods are used for asynchronous response at rated speed in this paper. Firstly, the ‘Overall Response Amplitude’ is an analysis method used for rough estimation of peak to peak vibration amplitude that is extruded from the maximum difference of each blade passing timing. Secondly, the ‘All-Blade Spectrum Analysis’ is a Fourier analysis method used mainly for the details of transient vibration at surge or flutter, etc. It assumes that the impeller vibrates in a nodal diameter mode and specific frequency, and constructs forward travelling waves under the effect of rotation. The nodal diameter, frequency, and amplitude are determined from the all-blades FFT spectrum and the circumferential position of sensors.

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