Proceedings of the International Gas Turbine Congress 2003 Tokyo
November 2-7, 2003

Blade-Strength Assessment of a Marine Turbocharger under Development

Fuminori IWAKI[1], Ken MITSUBORI[1], Hidetoshi TAGUCHI[1],
Masakazu OBATA[2] and Andrew R. MECH[3]

[1] Department of General Machinery Engineering
Ishikawajima-Harima Industries Co., Ltd.
3934, Tatsuno, Kamiina, Nagano, 399-0492, JAPAN
Phone: +81-266-41-2990, FAX: +81-266-41-1963, E-mail: fuminori_iwaki@ihi.co.jp
[2]Kanazawa Institute of Technology, Ishikawa, Japan
[3]Rose-Hulman Institute of Technology, Indiana, U.S.A

ABSTRACT

In response to the requirement for higher output of diesel engines in recent years, IHI has recognized that a turbocharger with higher pressure-ratio and volume-flow rate will be required, and has commenced the development of such a turbocharger. An important consideration in the design is the relatively high failure rate of blades presently used in the market. The new turbocharger will need to rotate at a faster speed than similarly-sized turbochargers of the past. Therefore the turbine and compressor blades will be subjected to excessive centrifugal force and it is forecast that strength problems will become increasingly severe. Under such severe operating conditions, damage to the turbine blades and compressor blades is possible as a result of prolonged resonance due to the impact of nozzle wake and pressure distribution.

The design life of a loaded blade may be predicted from the centrifugal force and resonance stress to which it is subjected. Centrifugal force may be calculated using a finite element model. However, the prediction of resonance is unreliable as the value of the stimulus (refer to Eq. (1)) of the blade at resonance is not necessarily clear. The stimulus value becomes more difficult to measure as the rotor speed becomes higher due to damage to the sensor attached to the rotating body and problems related to axial vibration. Therefore, only a small amount of data has been obtained in the past.

In this report, the turbocharger under development was tested to determine the blade vibration and logarithmic decrement during resonance of the turbine and compressor blades, as well as, to estimate the order of the stimulus value and to subject these values to review.

DISCRIPTION OF THE TURBOCHARGER

Structure of the Turbocharger

Figure 1 shows the cross-section of the turbocharger that is under development. The rotor is comprised of a turbine and a compressor. The load in the axial direction is supported by a thrust bearing and the load in the radial direction is supported by two journal bearings (floating bush). The distance between the journal bearings is 65 mm and the length of the shaft is 294.5 mm. The shape of the floating bush is formed from triple arcs on the inner circumference. A feature of this structure is greater stability in terms of vibration compared to a structure employing a true circle.

The compressor wheel is a blisk type and is comprised of 8 main blades and 8 splitter blades for a total of 16 blades that have been formed by cutting aluminum.

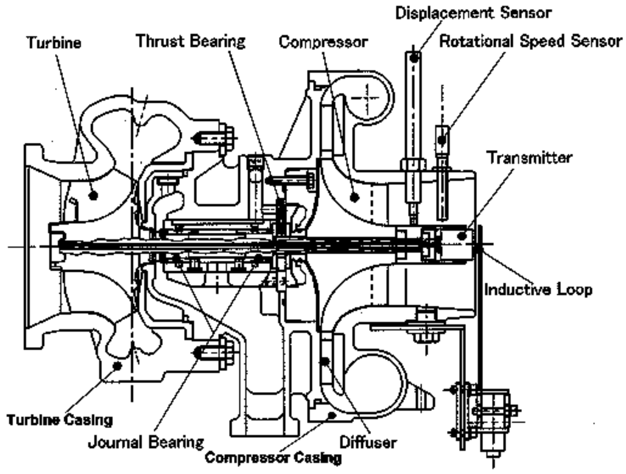

Figure 1 Cross-Section of Test Machine

The maximum diameter of the wheel is 130.8 mm. Moreover, 17 diffusers are equipped at the discharge of the compressor wheel.

The maximum design pressure ratio of the compressor is 4.0 and the maximum rotational speed of the turbocharger is 540 m/s.

On the other hand, the turbine wheel is also a blisk type with 12 blades and is formed through precision casting of a nickel alloy. The maximum diameter of the intake is 124.8 mm. No nozzle blade has been installed.

Analysis of the Natural Vibration Frequency for the Compressor and Turbine Blades

In the design of compressor and turbine blades, the vibration mode analysis is performed for each blade by applying a finite element method to avoid the excitation at maximum working rotation. In the market, it is well known that the failure of compressor and turbine blades is not occurred in the higher modes more than 2nd mode of the natural frequency of the blade. Therefore, the analysis was performed at the 1st mode of the natural frequency for the both blades.

Figure 2 shows the results of analysis of the 1st mode of natural frequency of the main compressor blade. In this case, the natural vibration frequency is 5715 Hz and the design avoids the 4th and lower order of excitation at maximum rotation of the turbocharger.

For the turbine blades as compressor blades, the 1st mode of natural frequency was also considered.

Manuscript Received on April 13, 2003

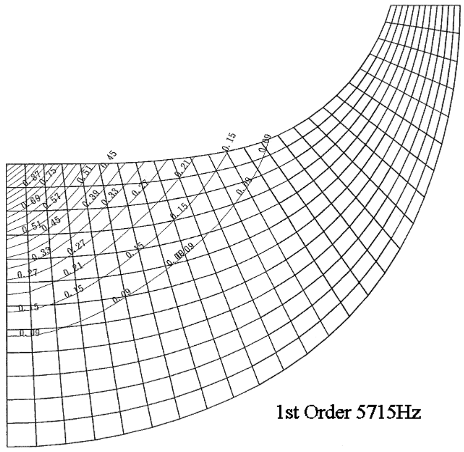

Figure 2 Resonance Frequency Analysis for the Compressor Blade TYPE-C1

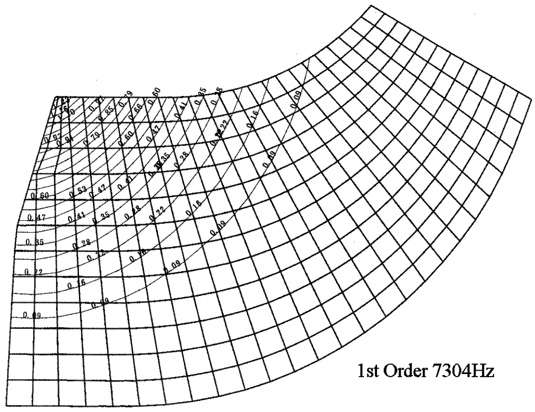

Figure 3 Resonance Frequency Analysis for the Turbine Blade TYPE-T1

Figure 3 shows the results of analysis of the 1st mode of natural frequency of the turbine blade. In this case, the natural vibration frequency is 7304 Hz and the design perspective avoids the 5th order and lower order of excitation at maximum rotation of the turbocharger.

BLADE VIBRATION TESTS

Compressor Blade Vibration Test

Tests were performed to measure the resonance stress of a selected blade at the exciting operating points.

Test Piece and Method Since the blades are manufactured in the same manufacturing process, it was judged that the lower the natural vibration frequency, the lower the stiffness and thus strength. To select the blade on which the strain gauge was to be affixed, the natural vibration frequency of each of the 8 main blades was determined through hammering and the blade with the lowest natural vibration frequency was selected. The blade on which the strain gauge was affixed shall be called TYPE-C1.

The location of affixing the strain gauge was, based on the results of the analysis shown in Fig. 2, the location and direction in which the absolute value of the maximum of the principle component of stress is maximized. The strain gauge used was a 350 Ω gauge that is approximately 3 mm long and designed for aluminum.

Table 1 Operating Point of the Compressor Blade TYPE-C1

Engine Order	Compressor shroud speed (m/s)	Operating point: Surge		Middle		Choke	
		Q (m^3/s)	πc (—)	Q (m^3/s)	πc (—)	Q (m^3/s)	πc (—)
17th	131	0.06	1.1	0.08	1.1	0.18	1.1
8 th	283	0.19	1.6	0.31	1.6	0.42	1.2
7 th	317	0.26	1.8	0.42	1.9	0.47	1.3
6 th	372	0.40	2.1	0.52	2.0	0.57	1.5
5 th	451	0.59	2.8	0.69	2.6	0.72	1.9

The lead wire of the strain gauge was connected to the transmitter of the telemetry system that was attached to the rotor end on the compressor side (see Fig. 1). Vibration with respect to compressor blade TYPE-C1 was measured using a transmitter with a 25 mm major diameter and 17 g weight, in which the maximum responsibility of the transmitter is 30kHz. Measurements of rotation were taken using an optic sensor attached in the radial direction.

A decision was made to take measurements of the resonance stress at the 8th, 7th, 6th, and 5th orders of excitation in ascending order of rotation and considering the resonance phenomenon generated by the 17 diffusers at the 17th order of excitation. At each of these orders of excitation, measurements were taken at three operating conditions based on the design compressor performance curve. The three operating points were: near surge, near choke and in the middle between surge and choke. Table 1 indicates the selected operating points.

Results of Measurement of the Vibration of the Compressor Blade Figure 4 shows the results of measurement of the vibration of compressor blade TYPE-C1 on the Cambell diagram. It is seen that for the n-th order of excitation, the generated resonance stress increases as the mode of vibration decreases (i.e., as the tip speed increases). Despite its low rotation, the fact that the measured rotation is low, the stress generated by the 17th order of excitation is higher than that of the 7th or 8th order of excitation.

In Fig. 4, three circles may be identified for each given order of excitation. These correspond to the resonance stress measured at the three operating points: near surge, near the middle and near choke. Three distinct measured values of the resonance stress can be clearly determined from Fig. 4 for the 5th and 6th orders of excitation.

Table 2 indicates the operating point at which the measured resonance stress is largest for each of the orders of excitation and the measured value at that point. The measured values are presented as ratios to the maximum value which was found at the 5th order of excitation with the compressor operating point near the middle.

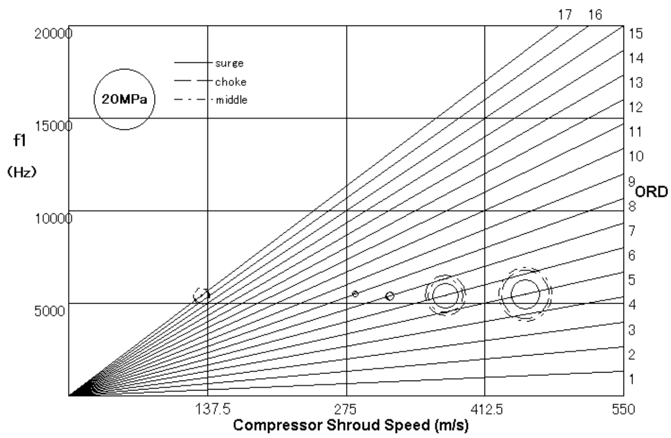

Figure 4 Campbell Diagram of the Compressor Blade TYPE-C1

Table 2 Maximum Resonance Stress of the Compressor Blade TYPE-C1

Engine Order	Operating point	Resonance stress ratio
17th	Middle	0.28
8 th	Surge	0.12
7 th	Surge	0.16
6 th	Choke	0.75
5 th	Middle	1.00

Turbine Blade Vibration Test

Two tests were made on a turbine blade: a test for confirming the source of exciting force and a test for measuring vibration in the turbine blade.

Test Piece and Method The test for confirming the source of exciting force was carried out after considering the durability of the strain gauge and air was selected as the working fluid of the turbine and the rotational speed was limited to only the 7th order of excitation. The selection of the blade on which the strain gauge was to be affixed and the location of that strain gauge was, as in the case of the compressor blade, determined from the results of the analysis shown in Fig. 3. The blade to which the strain gauge was affixed shall be called TYPE-T1. The strain gauge used was a 120Ω gauge for high temperature that is approximately 3 mm long.

Since the strain gauge used in the measurement of vibration in the turbine blade is subjected to a large centrifugal force at high temperature, it was affixed using a thermal spray coating and was covered with a metal foil membrane in order to prevent sensor damage. The lead wire from the strain gauge was passed through a 2.5 mm diameter hole in the rotational body and connected to the transmitter for the telemetry system attached at the rotor end on the compressor side. Moreover, the rotational speed sensor was attached to the compressor casing was in the same circumferential direction as the only position on the turbine casing where there is a tongue of the casing. Through this scheme, it was possible to determine the vibration of the turbine blade to which the strain gauge was attached as it passed the location of the tongue. The strain and rotation measuring system was the same as in the vibration tests of the compressor blade.

Considering the durability of the strain gauge, measurement of the resonance stress was initially performed with respect to the 7th order of excitation for which the rotation speed is low. Next, the test was undertaken for the 6th order of excitation at which the maximum rotation speed was not exceeded. The measurement conditions involved testing near the middle operating point of the compressor at a constant turbine inlet temperature (TIT) of 450°C. For the 6th order of excitation, measurements were also taken for TIT of 525°C and 375°C.

Confirmation of the Source of Exciting Force Figure 5 shows the results of the 7th order excitation test for confirming the source of the exciting force. The horizontal axis is time and the vertical axis is a dimensionless quantity for the output signal assuming the maximum output of the rotation signal to be "1". The rectangular wave of the figure represents the rotation signal and the smaller amplitude wave represents the blade vibration.

This figure indicates that the turbine blade on which the strain gauge has been attached passes the location of the tongue of the casing at 0.5 ms, 1.8 ms and 3.0 ms. That is the time when the rotation signal changes from minus to plus. Accordingly, the turbine blade TYPE-T1 vibrates seven times in the time from when the blade first passes the tongue of the casing to when it passes the tongue of the casing a second time. This means the blade vibrates seven times in one rotation of the rotor by the only one exiting force.

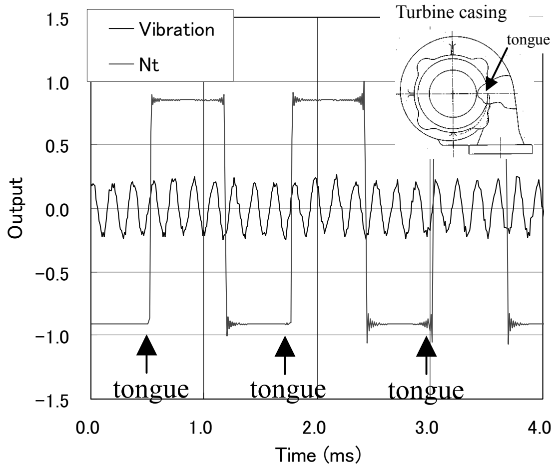

Figure 5 Rotational Speed and Blade Vibration by 7th Engine Order Excitation Force

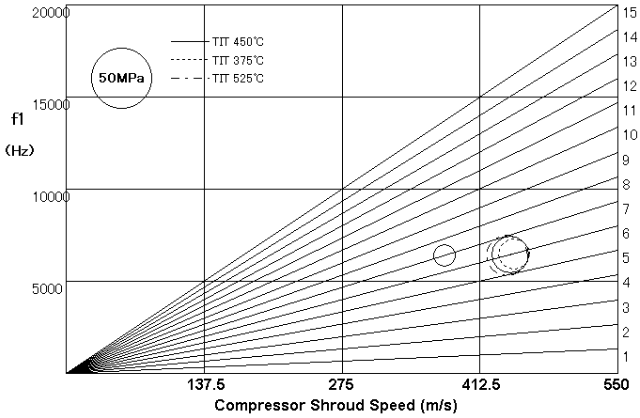

Figure 6 Campbell Diagram of the Turbine Blade TYPE-T1

Table 3 Maximum Resonance Stress of the Turbine Blade TYPE-T1

Engine Order	TIT(℃)	Resonance stress ratio
7 th	450	1.03
6 th	375	1.45
6 th	450	1.74
6 th	525	1.92

Therefore, it is believed that the main exciting force for the turbine blade TYPE-T1 comes from the pressure difference caused by the presence of the tongue of the casing.

Results of the Turbine Blade Vibration Test Figure 6 shows the results of the measurement of blade vibration for the turbine blade TYPE-T1 on the Cambell diagram

As in the case of the compressor blade, at n-th order of excitation, the generated resonance stress increases as the mode of vibration decreases. For the 6th order of excitation, Fig. 6 shows the effect of changing TIT. The trend identified is that as the TIT increases, the resonance stress also becomes higher.

Table 3 shows the measured values of the resonance stress. Once again, the measured values are presented as ratios to the maximum compressor value which was found at the 5th order of excitation with the compressor operating point near the middle.

CALCULATION OF THE STIMULUS VALUE

Basic Equations

In general, the resonance stress of the blade may be obtained from the following relation.

$$\sigma_v = \frac{\pi}{\delta} \cdot H_n \cdot S_n \cdot \sigma_{bs} \qquad (1)$$

Here, σ_v, δ, H_n, S_n and σ_{bs} are resonance stress, logarithmic decrement, resonant response factor, stimulus value and average bending stress, respectively. It was decided to determine the logarithmic decrement δ and to use the results of the measurement of resonance stress to calculate the stimulus value S_n from Eq. (1).

The logarithmic decrement δ may be obtained as the sum of structural damping, material damping and air damping. As the present TYPE-C1 compressor and the TYPE-T1 turbine are blisks the structural damping is null. Then, the logarithmic decrement δ may be obtained as the sum of the material damping and air damping and the halfpower method of the following formula was used.

$$\delta = \pi \cdot \frac{\Delta f}{f} \qquad (2)$$

Where f is the frequency at the peak amplitude (Apeak) of the resonance point of the blade and Δf is the span of the frequency (Hz) at which the relative amplitude, A/Apeak, is "$1/\sqrt{2}$."

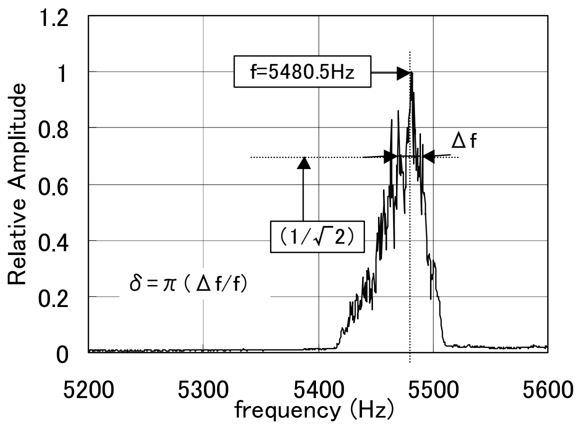

Figure 7 5th E. O. Logarithmic Decrement TYPE-C1 Compressor Blade

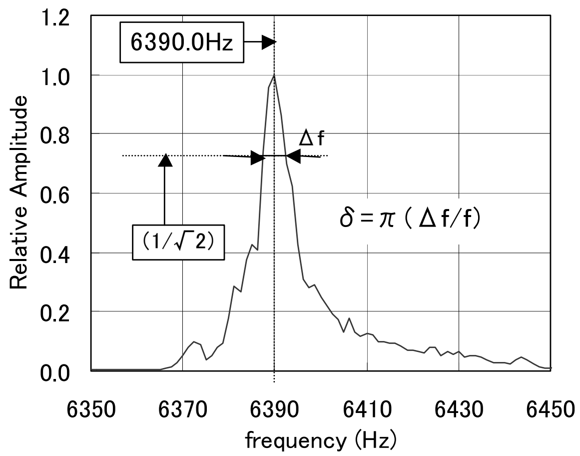

Figure 8 6th E. O. Logarithmic Decrement Turbine Blade TYPE-T1 (TIT 525℃)

Calculation of the Logarithmic Decrement

The logarithmic decrements for the tested compressor and turbine will be shown. Compressor blade data were taken only during rotation, while the logarithmic decrement for the turbine blade was obtained using the halfpower method shown in Eq. 2 for both the static and rotation states.

Figures 7 and 8 show an example of the wave form used to obtain the logarithmic decrement for the compressor blade and turbine blade, respectively. The measurement values of the natural frequency are different a little from those of analysis values shown in Fig. 2 and Fig. 3. This is because the analysis does not consider the change of Young modulus depending on the centrifugal force and the material temperature.

The logarithmic decrement of the turbine blade in static state was calculated from data from a hammering test where load was varied. Figure 9 shows the logarithmic decrements of the turbine blade in the static state test condition. The logarithmic decrement of the static state corresponds to concentrated load of the exciting force.

The logarithmic decrement of the compressor blade and turbine blade in the rotation state was calculated using data as compressor tip speed was changed. Figures 10 and 11 show the results of calculations of logarithmic decrement for each type of blade TYPE-C1 and TYPE-T1 in rotating test conditions. The logarithmic decrement of the rotation state corresponds to the distributed load of the exciting force.

Generally, in the static state, air damping is null and since the logarithmic decrement δ during rotation is the sum of material damping and air damping, the value of δ in the rotation state should be larger than the value of δ in the static state. However, when the results of the turbine blade Type-T1 in Figs. 9 and 11 are compared, the order of δ values for the static state is larger by a factor of about ten.

The cause for this difference was discussed as follows. First, the impact of temperature will be reviewed as a cause.

For the turbine blades, the temperature of the blade rises during rotation due to heat transfer from the combustion gas. However, the results shown in Fig. 11 were taken while keeping the rotation constant and changing TIT by 150 °C and shows that a change of 150 °C in the TIT has very little impact on the logarithmic decrement value. Next, from Fig. 10, it is clear that the exciting force also has very little impact on the logarithmic decrement value.

Moreover, when looking at the impact of the centrifugal force, Fig. 10 and Fig. 11 show that the difference in tip speed does not cause a regular change in the logarithmic decrement.

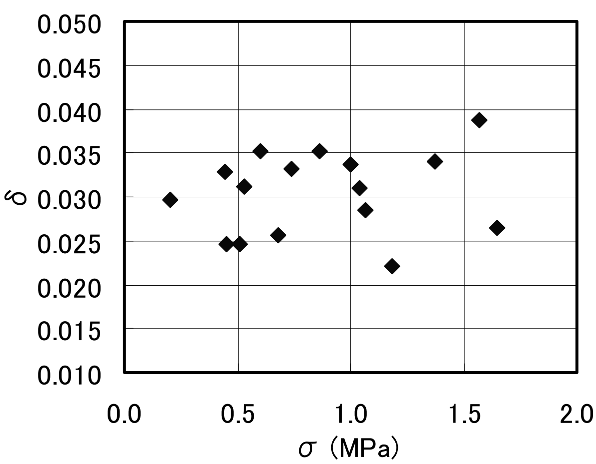

Figure 9 Logarithmic Decrement by Hammering Test of the Turbine Blade TYPE-T1

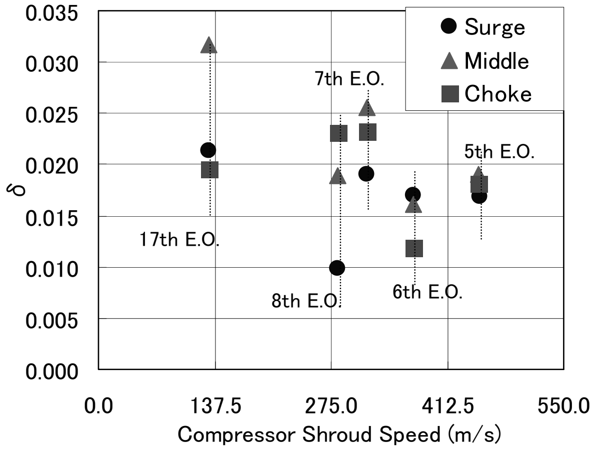

Figure 10 Logarithmic Decrement of the Compressor Blade TYPE-C1

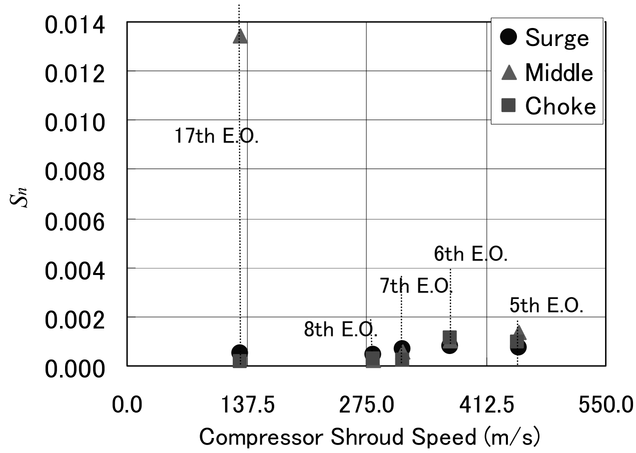

Figure 12 Stimulus Value of TYPE-C1 Compressor Blade

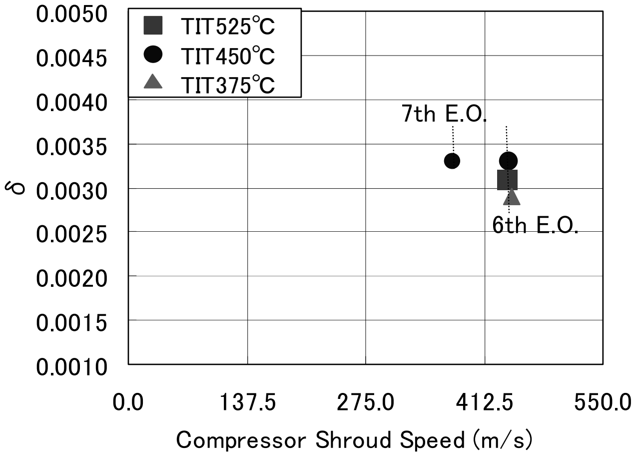

Figure 11 Logarithmic Decrement of the Turbine Blade TYPE-T1

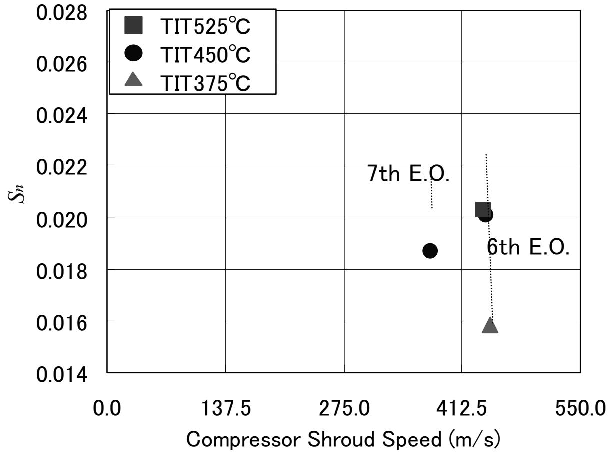

Figure 13 Stimulus Value of the Turbine Blade TYPE-T1

The remaining factor that could have a significant impact on logarithmic decrement is the method of application of the excitation (concentrated load or distributed load). The possibility that logarithmic decrement differences are due to the method of application is high compared with other factors.

Calculation of the Stimulus Value

When obtaining the stimulus value S_n from Eq. (1), the resonance response factor H_n may be calculated as the ratio of the dynamic bending moment and the static bending moment. The calculated value for the 1st mode of vibration was 0.891. With respect to the bending stress, the average stress σ_{bs} for a single blade was calculated by obtaining the torque to which the entirety of the turbine blades and compressor blades were subjected.

Figures 12 and 13 show the results of calculation of the stimulus value S_n for the compressor blade TYPE-C1 and the turbine blade TYPE-T1.

From Fig. 12, it is seen that the value of S_n for the compressor blade TYPE-C1 is large (about 0.013) only for one operating point of the 17th order of excitation. This occurs because the resonance stress value for the 17th order of excitation is large near the middle operating point compared to the resonance stress values near the surge and choke points.

Meanwhile, it is seen from Fig. 13 that the S_n values of the turbine blade TYPE-T1 are larger by about ten fold than those of the compressor blade and rise with an increase of the TIT.

The difference in S_n values results from the difference of the exciting force between the compressor and turbine blades. As the exciting force of the compressor blade comes from the presence of diffuser and tongue of the casing, the strongest impact on the compressor blade is at the blade outlet, but it is small, whereas, the impact for the turbine is strong at the blade inlet by the presence of the tongue of the casing. Therefore the latter exhibits a stronger exciting force and the S_n values become larger than the former.

CONCLUSIONS

The vibration of the turbine blade and compressor blade upon resonance and logarithmic decrement were determined and the relationship between the tip speed and the logarithmic decrement was studied for the high pressure, large volume flow turbocharger that was tested.

(1) It is believed that the source of exciting force for the blade vibration at the n-th order of excitation is the tongue of the casing.

(2) The logarithmic decrement δ was between 0.008 to 0.024 for the compressor blade (excluding the middle operating point for the 17th order of excitation) and between 0.0029 and 0.0033 for the turbine blade. No correlation between δ and either the rotation speed or temperature was apparent. Moreover, it is believed that the factor that impacts logarithmic decrement is the method of application of the excitation (concentrated load or distributed load).

(3) The stimulus value S_n was between 0.0005 and 0.0014 for the compressor blade (excluding the middle operating point for the 17th order of excitation) and between 0.0159 and 0.0203 for the turbine blade. The S_n values became larger as the rotation became higher. However, this trend is limited to cases of constant TIT for the turbine.

(4) By accumulating logarithmic decrement and stimulus value data for various types of turbochargers, the prediction of resonance stress may be made more precise in the future. The resonance response factor and average bending stress in Eq. (1) are relatively simple to obtain. Therefore, it is sufficient to improve the accuracy of the logarithmic decrement and stimulus value to improve resonance stress predictions.

(5) For each order of excitation, it was possible to calculate the values for the logarithmic decrement and stimulus value. As a result, it has become possible to improve forecasting of resonance stress using Eq. (1) by determining the operating conditions of a turbocharger.

REFERENCES

(1) Prohl, M. A., 1958, “A Method for Calculating Vibration Frequency and Stress of a Banded Group of Turbine Buckets,” Trans. ASME, Vol. 80, pp. 169-180.

(2) Weaver, F. L. and Prohl, M. A., 1958, “High-Frequency Vibration of Steam-Turbine Buckets,” Trans. ASME, Vol. 80, pp. 181-194.

(3) Naguib, M. M., 1965, “Theoretical Estimation of Dynamic Forces and Vibratory Stresses for Turbine Blade,” Report No. 9, Eidgenoessische Technische Hochschule, Zurich.

(4) Iwaki, F., Mitsubori, K. and Obata, M., 2003, “Effect of Turbine Inlet Pressure Fluctuation on Blade Vibration for a Marine Turbocharger,” Mechanical Engineering Congress 2003 in Japan, Vol. 7, No. 2413, pp. 269-270.

(5) Tanaka, S. and Okada, Y., 1978, “Vibrational Characteristics of a Rotating Blade with Mechanical Damper,” Trans. Jpn. Soc. Mech. Eng., (in Japanese), Vol. 44, No. 381, pp. 1522-1533.

(6) Hanson, M. P., Meyer, A. J., and Manson, S. S., 1950 ,“A Method of Evaluating Loose-Blade Mounting Compressor Blade Vibration,” Annual Meeting of the Society for Experimental Stress Analysis in New York, NY, Vol. 10, No. 2, pp. 103-115.