N89-22917

HIGH STABILITY DESIGN FOR NEW CENTRIFUGAL COMPRESSOR

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It is essential that high-performance centrifugal compressors be free of subsynchronous vibrations.
    A new high-performance centrifugal compressor has been developed by applying the latest rotordynamics knowledge and design techniques.

(1) To improve the system damping, a specially designed oil film seal was developed. This seal attained a damping ratio three times that of the conventional design. The oil film seal contains a special damper ring in the seal cartridge.
(2) To reduce the destabilizing effect of the labyrinth seal, a special swirl canceler (anti-swirl nozzle) was applied to the balance piston seal.
(3) To confirm the system damping margin, the dynamic simulation rotor model test and the full load test applied the vibration exciting test in actual load conditions.

INTRODUCTION

    Performance improvement of centrifugal compressors for chemical plants has been a very important subject because these compressors substantially dominate the cost effectiveness of the plants.

    A new high-performance centrifugal compressor has been developed by applying up-to-date techniques in response to these needs [1], and we have succeeded in developing an innovative new type of high-performance compressor and have demonstrated its high performance and reliability in shop tests under actual load conditions.

    This paper describes the rotordynamic research undertaken in developing the new compressor.

DESIGN SPECIFICATIONS AND CHARACTERISTICS OF THE NEW COMPRESSOR [1]

    The design specifications of the new compressor are shown in Table 1. This compressor was designed for the Ammonia & Methanol Synthesis Process. The specifications are typical for the 1000 T/day ammonia plant where the process gas is a mix of nitrogen and hydrogen.

    The total cross section of the compressor is shown in Fig. 1. The low-pressure compressor is a back-to-back type and the high-pressure compressor contains the gas recycling stage.

    The distinguishing features of the compressor are as follows, and all these merits are realized by applying the high-speed rotor system.

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E-4227-26

(1) High performance: A comparison of the required power for the conventional and new compressors for the 1000 T/day ammonia synthesis plant is shown in Table 2. The efficiency is significantly improved by applying the 3-dimensional high flow coefficient stages (high-speed impellers) shown in Fig. 2.

(2) Compactness: A comparison of the new compressor with the conventional type in a similar plant is shown in Fig. 3. The compressor train is simplified and reduced to half size.

(3) Easy maintenance: As shown in Fig. 3., both the end drive turbine and the cartridge-type internal compressor are applied to the new compressor train. These enable the compressor to be maintained without removing its casing.

The up-to-date techniques shown in Fig. 4 have been applied to the new compressor to realize the features mentioned above.

DEVELOPMENT OF THE COMPRESSOR FROM THE VIEWPOINT OF ROTORDYNAMICS

The new compressor, having up to nine stages, is operated at the highest speed in the world for this class of compressor. In spite of its compact, light-weight rotor, this compressor absorbs relatively high power and handles high-pressure, high-density gas. In similar high-pressure compressors, troubles caused by nonsynchronous vibrations have been reported [2],[3],[4],[8].

Therefore, countermeasures against this kind of vibration were critical in achieving reliable operation of the new compressor.

The forces acting on the compressor are shown in Fig. 5, and vibration stability of the system depends on the balance of the stabilizing force (damping force) and the destabilizing force. Accordingly, increasing the damping force and decreasing the destabilizing force is an essential solution. For the new compressor, with the help of our comprehensive research in this field, the following new techniques have been put into effect through quantitative evaluation of the damping force and destabilizing force.

A flow diagram of rotordynamic research for this compressor is summarized in Fig. 6.

ROTORDYNAMICS ANALYSIS

Rotordynamic characteristics of the high-pressure compressor are related to many components such as the bearings, oil film seals, labyrinth seals, and impellers. A flow diagram of the applied stability analysis method is shown in Fig. 7[8]. The component values are evaluated by using an accumulated data base and analytical tools.

(1) The dynamic coefficients of the oil film bearings are evaluated by the data base accumulated and qualified by the experimental model tests and numerical analysis.

(2) The dynamic coefficients of the oil film seals are estimated by the analytical method for oil film bearings.

(3) The destabilizing coefficients of the impellers are estimated by the following equation (Wachel equation modified to Alford force) [2]:

$$K_c = \frac{9583.2 (KW)(MW)}{D \cdot H \cdot (RPM)} \cdot \frac{\rho_d}{\rho_s} \qquad (1)$$

where (KW) is the output, (MW) is the molecular weight of the gas, (RPM) is the shaft speed, D is the impeller outside diameter, H is the impeller tip opening at discharge, ρ_d is the density of gas at discharge, and ρ_s is the density of gas in suction. This force estimation must be improved for better accuracy.

(4) The destabilizing coefficients of labyrinth seals are estimated by the theoretical analysis which was recently developed [7] by our research. The stability analysis of the system was done by the modal analysis shown in Fig. 7 or by complex eigenvalue analysis.

DEVELOPMENT OF NEW STABILIZING ELEMENTS

The stability analysis showed that the conventional type rotor bearing system had no margin for unstable vibration when it was applied to this kind of high intensity design. Then the new technology had to be applied to obtain a sufficient margin from the onset of unstable vibration.

The first innovation is a new type of oil film seal with a damper function as shown in Fig. 8. The new seal has a superior damping effect without any sacrifice of space or sealing function.

The mechanism of the damper seal consists of a multilobe bearing and a squeeze film damper. The effectiveness of the new seal system was verified by the model test system shown in Fig. 9.

The newly developed experimental method was used to verify the damping capacity of the rotor-bearing/oil-film-seal system. The method is called "Vibration Exciting Test Method in Working Condition," and it is essential to revealing the phenomena of nonsynchronous vibration.

Fig. 10 is an example of measured frequency response for two typical cases corresponding to the conventional system and the new system with the damper seal.

The speed-dependent vibration characteristics for the first mode are plotted in Fig. 11. As this figure shows, the test method enables the appointed mode to be evaluated for any rotating speed.

A summary of the results of the stability evaluation for the model system is shown in Table 3. The following stability coefficient (modal damping coefficient) is applied to the evaluation.

$$D = 2mw^2 \cdot \zeta \tag{2}$$

where   D: stability coefficient
            m: model mass of the mode
            w: angular frequency of the mode
            ζ: damping ratio of the mode

The results show that the increased rate of the damping capacity is about three times that of the conventional system.

On the other hand, the special swirl canceling mechanism that is illustrated in Fig. 12 is developed and applied to this compressor in order to minimize the destabilizing force generated by the labyrinth seal. The design of the labyrinth seal was based on the recently developed theory and model tests published in a NASA Conference Publication in 1986 [6]. This research confirmed that the gas flow at the labyrinth seal, particularly inlet swirl, is the major cause of the

destabilizing forces. Then, the design of the reliable swirl canceling mechanism was developed. The low destabilizing force with this mechanism was verified by a full load test that is described later.

VERIFICATION TEST IN THE SHOP

To verify the new compressor's performance and mechanical reliability, a four-step testing program has been carried out as shown in Table 4.

In addition to normal tests and inspections for the prototype machine, special rotordynamic tests were performed in the shop to verify vibration stability and smooth operation.

Fig. 13 is a photograph of the shop test setup showing the vibration-exciting test apparatus. The vibration-exciting tests were performed for mechanical run and high-pressure performance conditions. Fig. 14 shows the measured vibration for the high-speed run and actual load tests. Fig. 15 shows the plots of damping vs. load.

These results show a sufficient stability margin even for the full load, full speed condition.

This fact verified the following two points.
(1) The new oil film seal system gives high system damping for any operating condition, especially for high-speed conditions.
(2) The destabilizing effect of the compressor, which has not ever be found directly, is verified by the difference value between the high-speed mechanical run and the actual load test.

The results also verified that there would have been marginal instability if the new stabilizing mechanism had not been applied.

The stability evaluation for the compressor is plotted on Fig. 16 with a comparison to the existing unit. This figure shows a sufficient margin of safety against unstable vibration.

EVALUATION OF VIBRATION SENSITIVITY TO THE DISTURBANCE IN NONSTEADY OPERATING CONDITIONS.

Fig. 17 shows the vibration level for all speed ranges. The shaft vibration did not exceed 10 μm p-p including runout. There was no significant critical speed peak for any speed within maximum speed. These phenomena support the fact that the system has a sufficiently large damping ratio [5].

Fig. 18 shows the example of measured data during surge. This figure shows that the vibration, pressure pulsation, and thrust force are all sufficiently small values even in the abnormal condition.

CONCLUSIONS

Rotordynamic research and results for developing a high-speed, high-performance compressor are described.

The newly developed damper seal and swirl canceling mechanism for the labyrinth seal produces a highly stable compressor. The high stability and low

vibration sensitivity of the prototype compressor was verified by many operating tests and special vibration-exciting tests.

Production compressors based on the design of the prototype have been operating smoothly and with high efficiency.

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(3) Jenny, R., "Labyrinth as a Cause of Self-Excited Rotor Oscillations in Centrifugal Compressors." Sulzer Technical Review 4, 1980, pp. 149-156.

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(7) Kanki, H., Morii, S., & Hizume, A., "Theoretical and Experimental Study on the Destabilizing Force by Labyrinth Seal." IFToMM 1986, to be published.

(8) Morii, S., Nishimoto, K., Kanki, H., & Hirayama, F., On the Sub-Synchronous Whirl in the Centrifugal Compressor, ICVPE Xi'an, 1986-6.

(9) Kanki, H., Fujii, H., Hizume, A., et al., "Solving Nonsynchronous Vibration Problems of Large Rotating Machineries by Exciting Test in Actual Operating Condition." IFToMM Conference, 1986.

(10) Umemura S., Mase, M., & Murai, T., Study on the Vibration Analysis of Radial-Flow Impellers, Mitsubishi Juko Giho, Vol. 14, No. 2, 1977.

Table 1   Table of compressor specifications

Item	Unit	Value
Suction flow rate	Am^3/h	5 500 — 6 500
Suction press.	kgf/cm^{2}G	25
Suction temp.	°C	45
Discharge press.	kgf/cm^{2}G	150 — 220
Max. continuous speed	rpm	15 540

LP compressor (4V -9B type)   Steam turbine driver (5EH - II type)   HP compressor (4V -9C type)
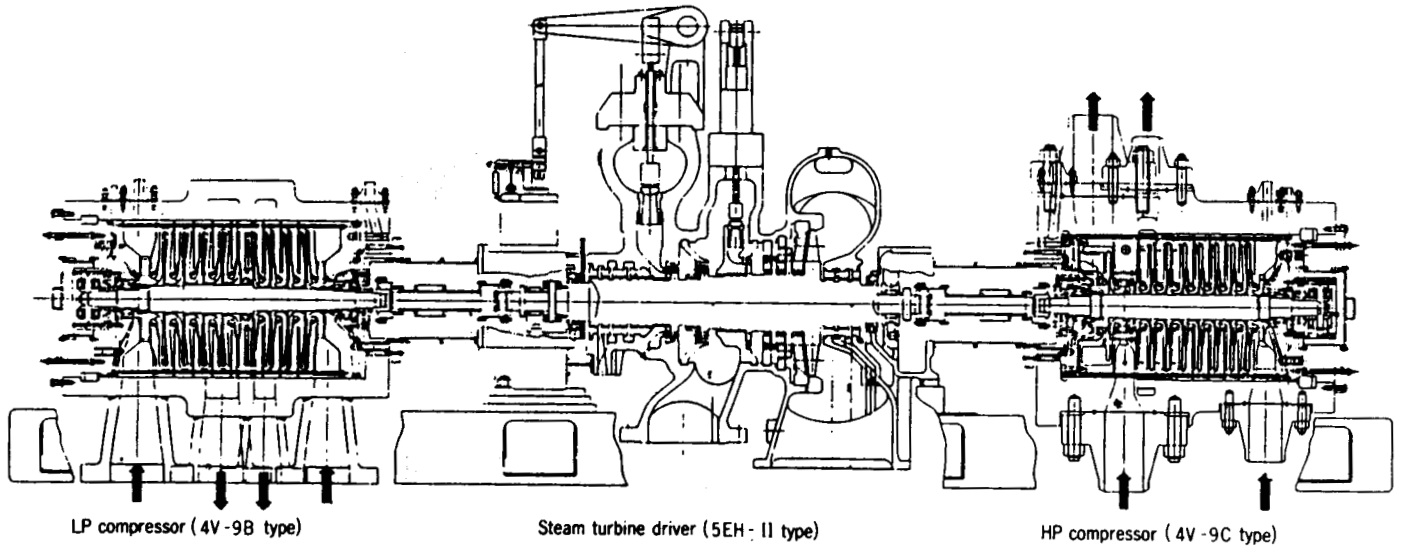
Fig.1   Cross section of the compressor train
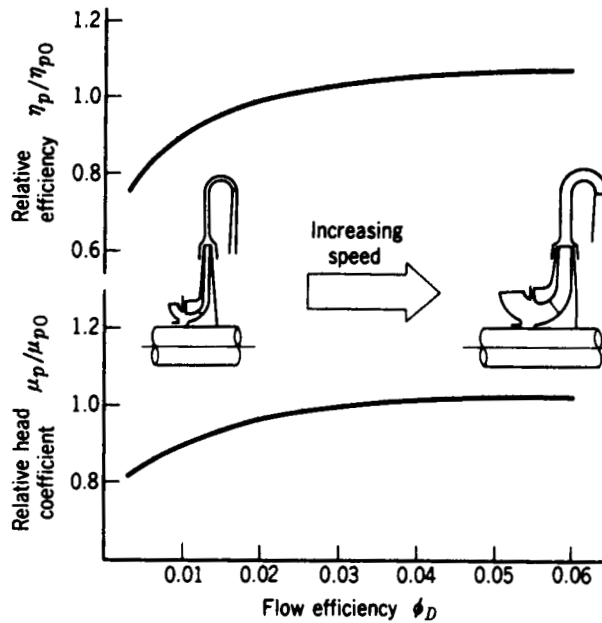
Fig.2   Stage performance vs. flow coefficient

Table 2   Merit of the new compressor (High performance)

Required power of compressor		Reduction in required power (Saving rate)
Conventional unit	New unit	
16 500 kW	14 980 kW	1 520 kW (9%)
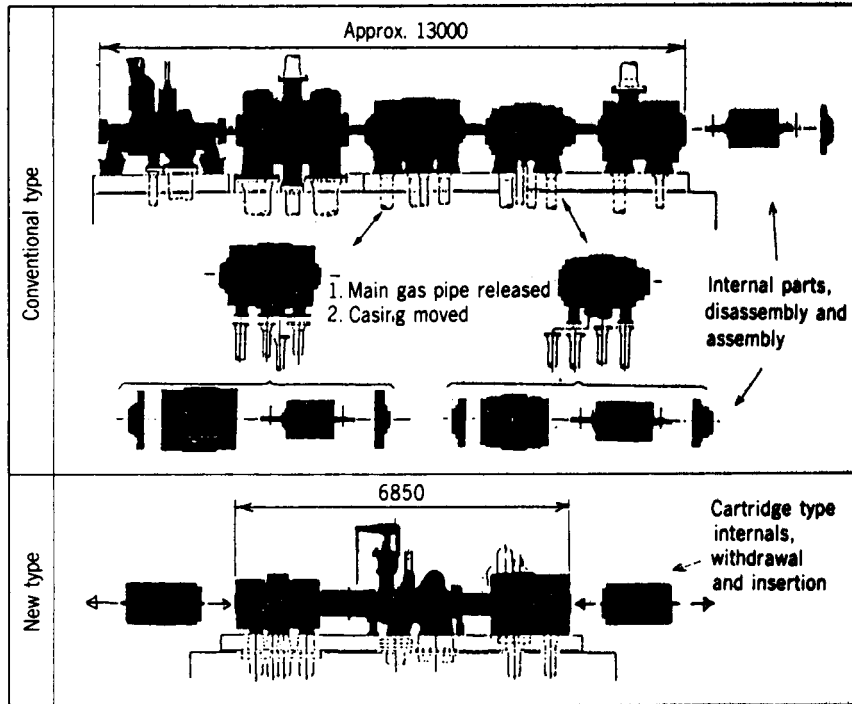
Fig.3   Merit of the new compressor (Compactness, ease of maintenance)
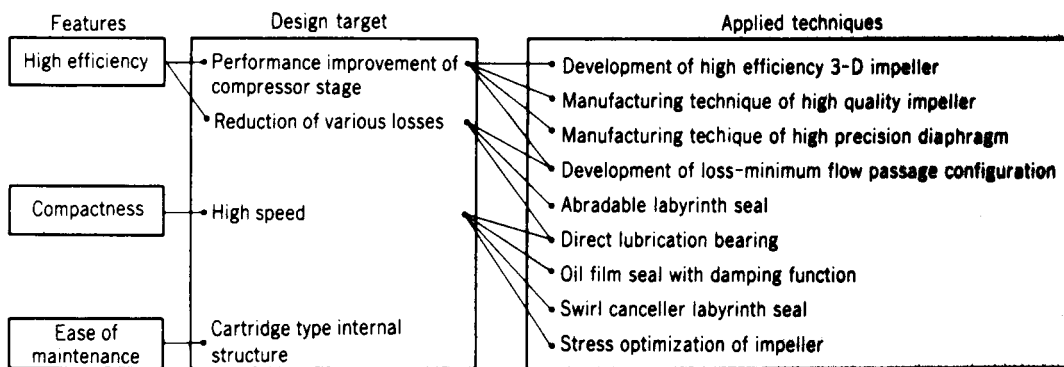
Fig.4   Application of up-to-date techniques
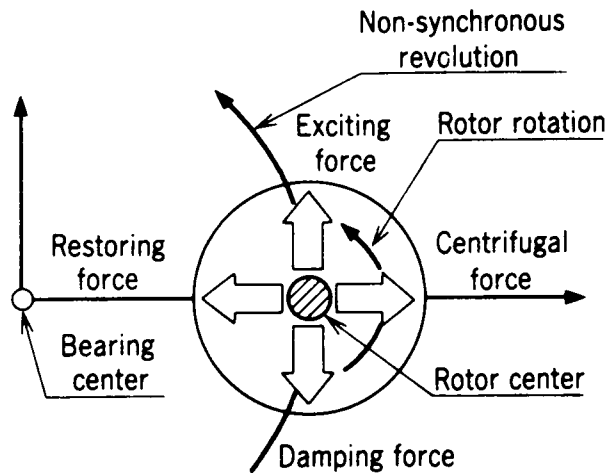
Fig.5   Forces acting on the rotor
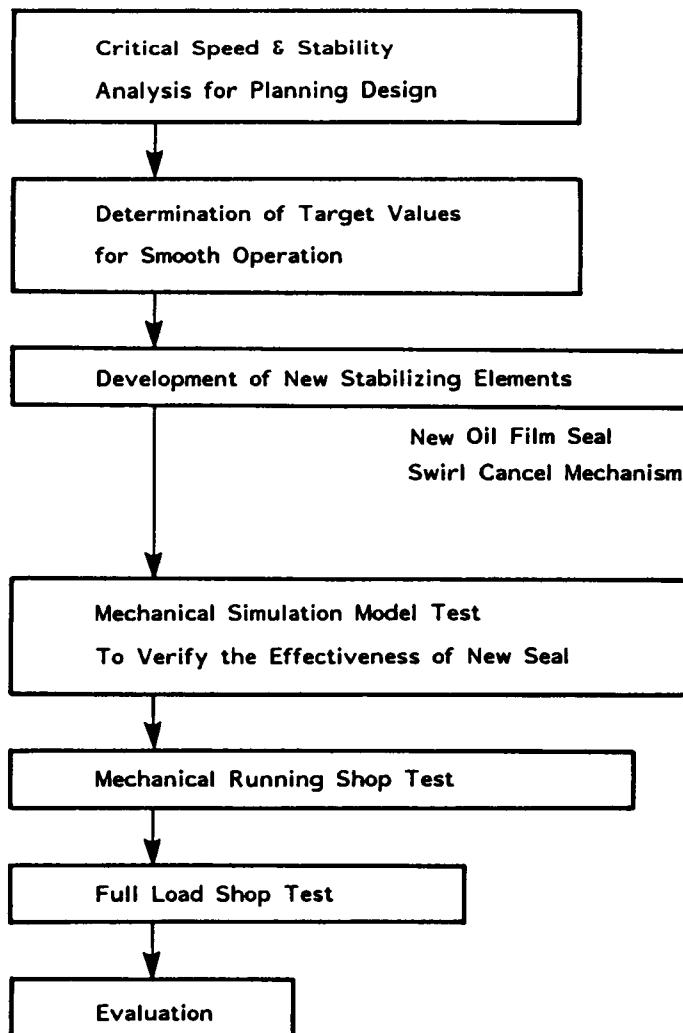
Fig.6   Flow diagram of rotordynamic research for the development of the new compressor
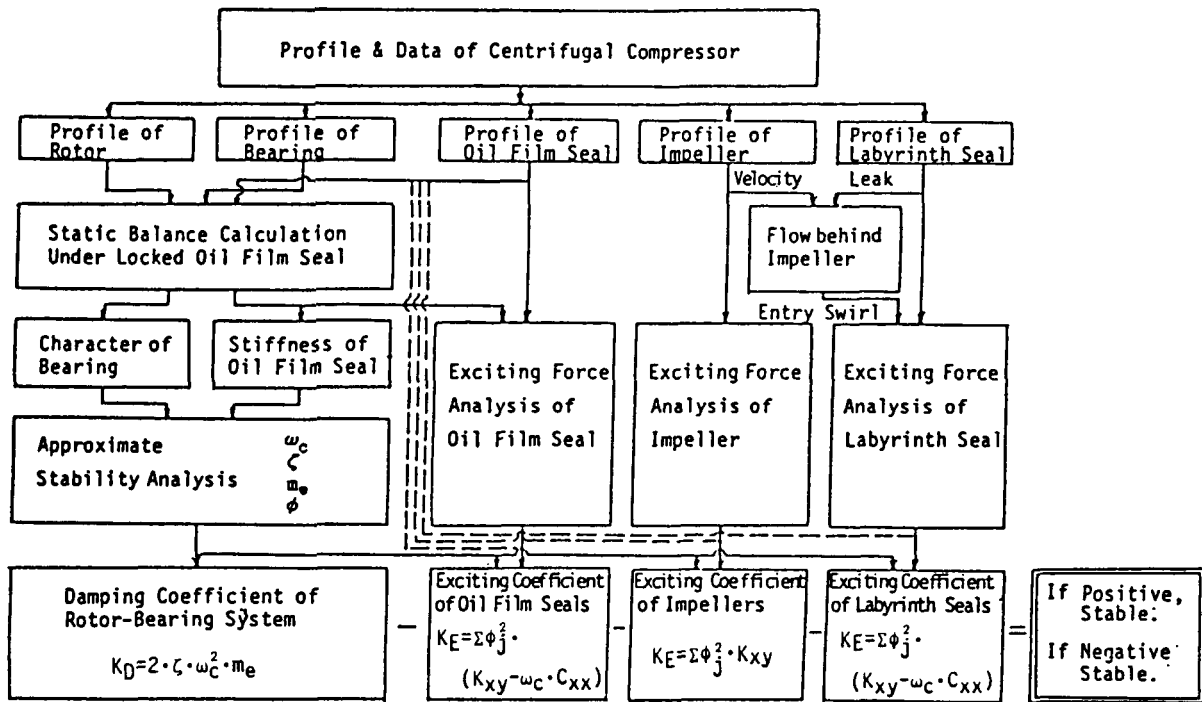
Fig.7   Block diagram of stability analysis method
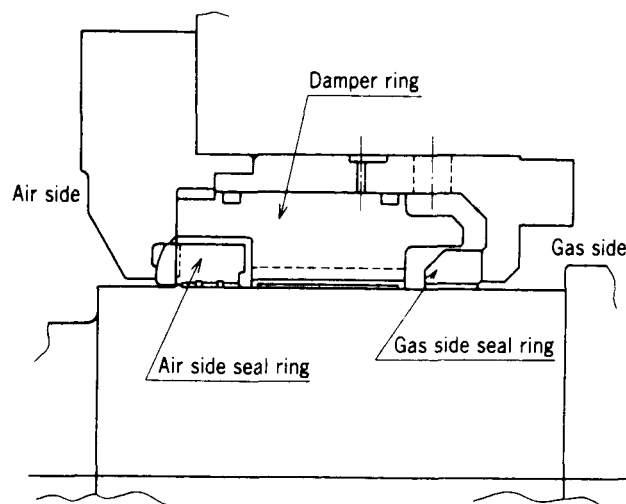
Fig.8   New oil film seal with damping function
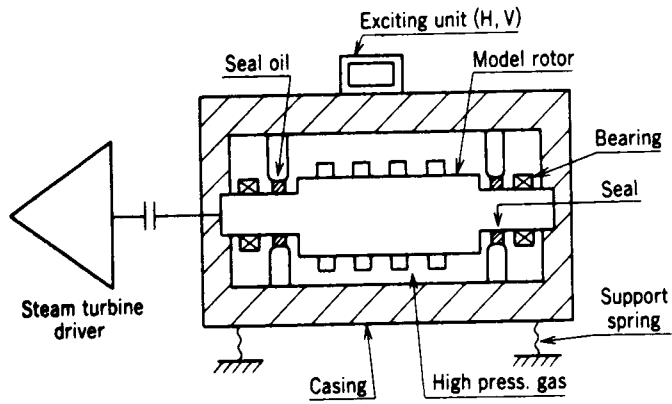
Fig.9   High pressure seal test apparatus
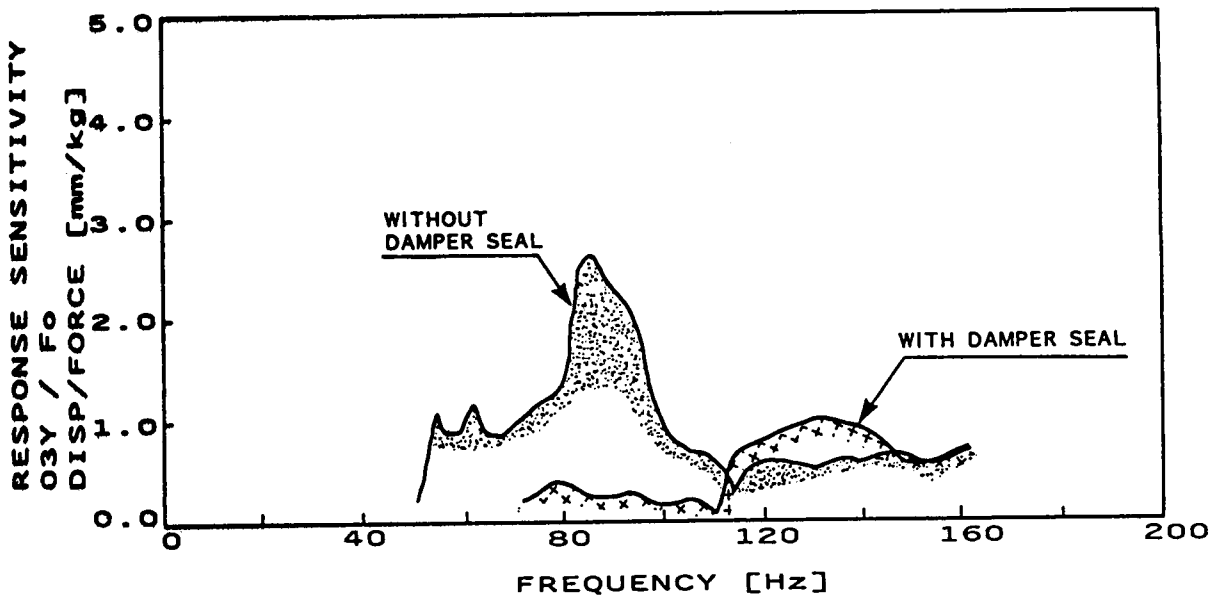
Fig.10   Typical frequency response for seal test
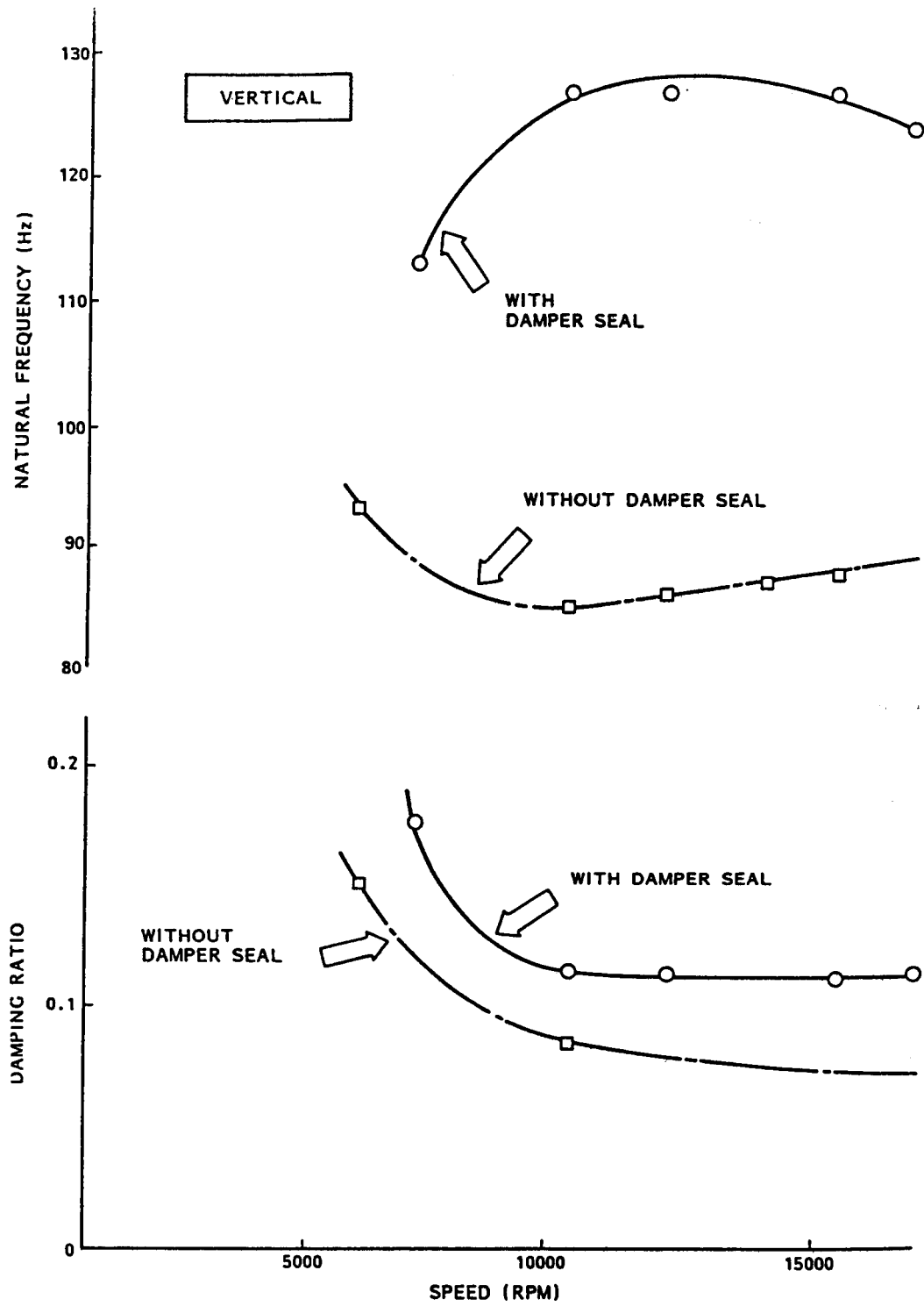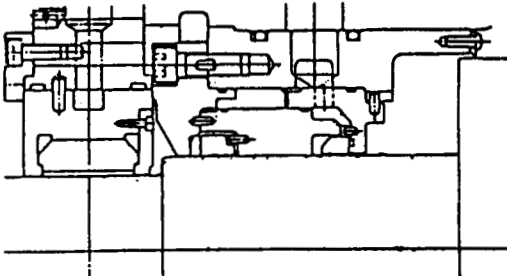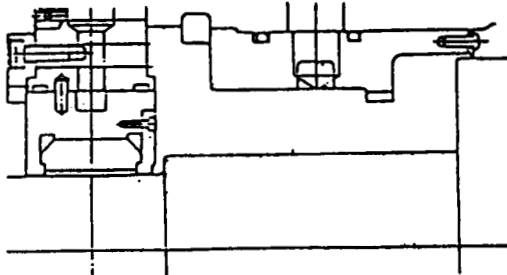
Fig.11　Comparison of vibration characteristics

Table 3 Test results of new oil film seal

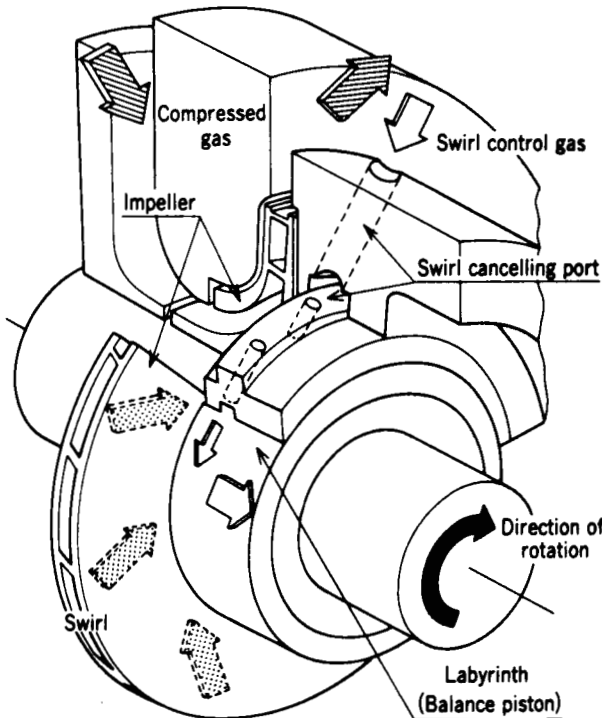|	SEAL TEST MODEL	CRITICAL SPEED Q FACTOR FOR UNBALANCE RESPONSE	1ST MODE		
			FREQUENCY	DAMPING RATIO	EQUIVALENT DAMP. COEF.
WITH DAMPER SEAL		6800 RPM Q = 2.8	127.5Hz	0.118	25700 kgf/cm
WITHOUT DAMPER SEAL		5600 RPM Q = 3.3	87.2Hz	0.075	7650

Fig.12 Labyrinth seal with swirl canceler

Table 4 Table of shop test program

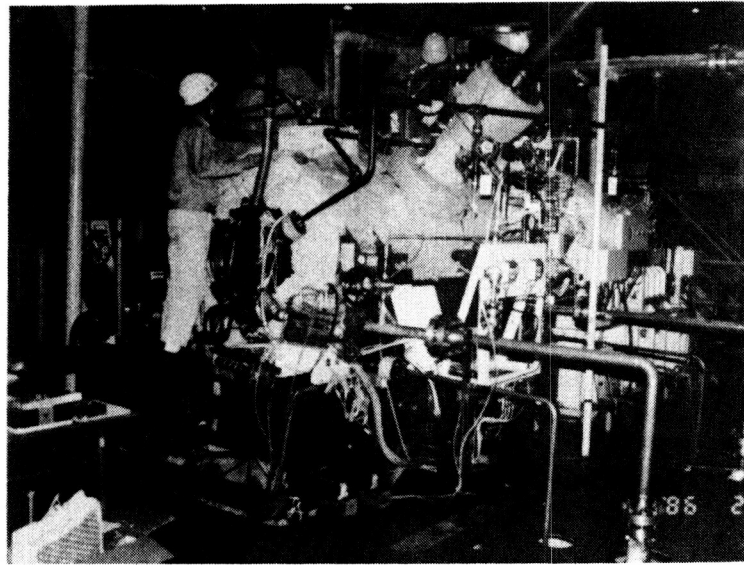| Test step	Object
Static seal test	Verification of dynamic behavior of oil film seal and seal oil supply unit
Aerodynamic performance test	Verification of compressor efficiency and required power
High speed mechanical running test	Verification of mechanical stability during high speed running
High press. performance test	Verification of mechanical stability and aerodynamic performance under actual operation conditions

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Fig.13   Vibration-exciting test apparatus for shop test
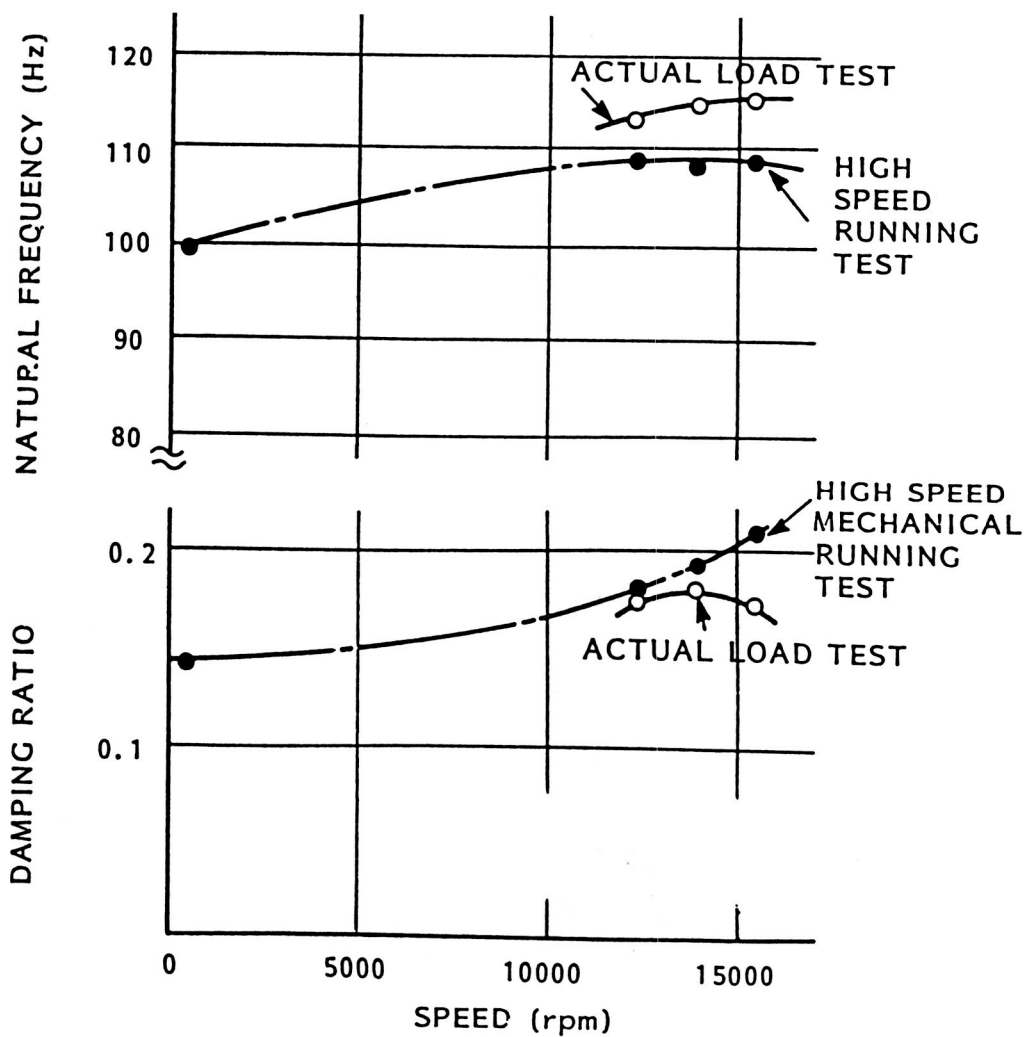

Fig.14   Results of vibration-exciting test
(speed vs. natural frequency and damping)
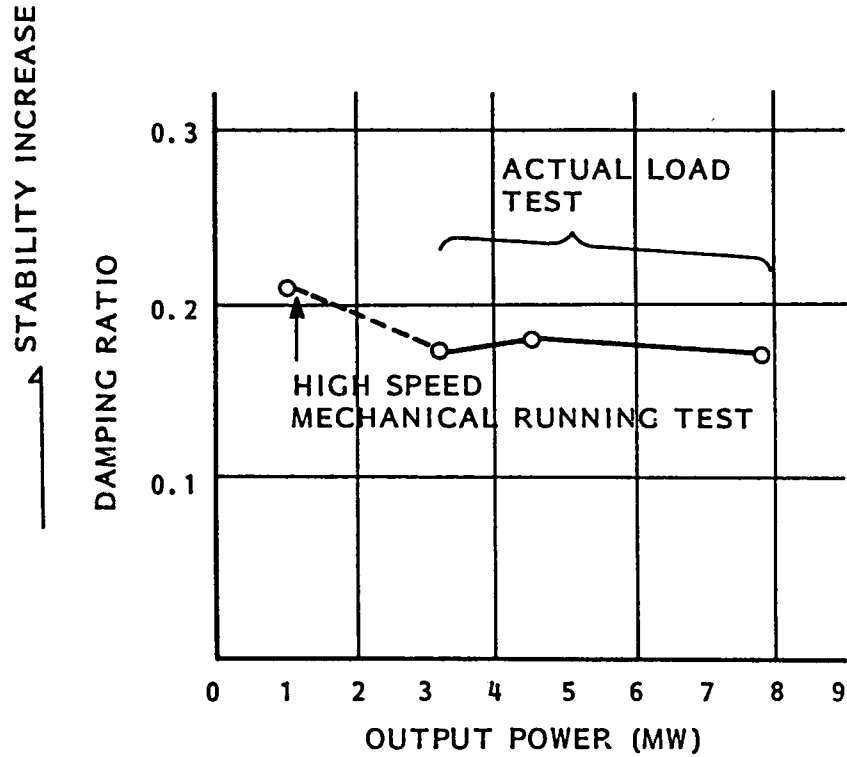
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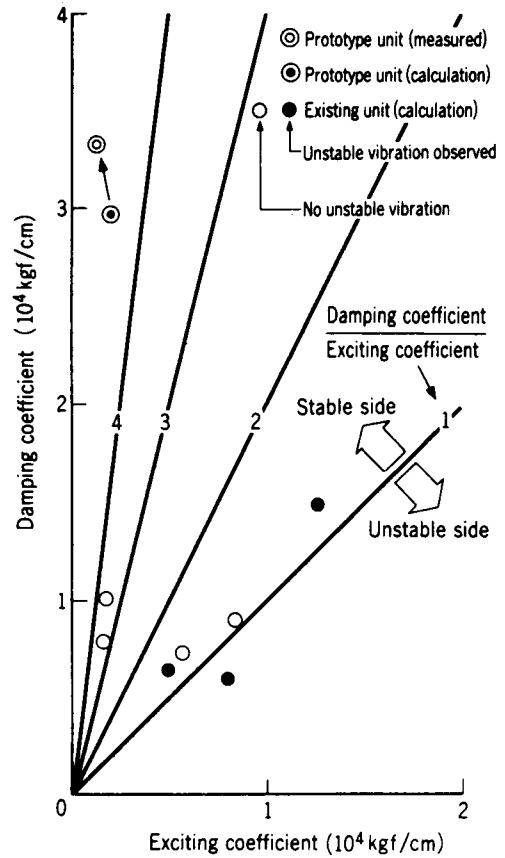Fig.15   Results of vibration-exciting test
          (stability vs. output)

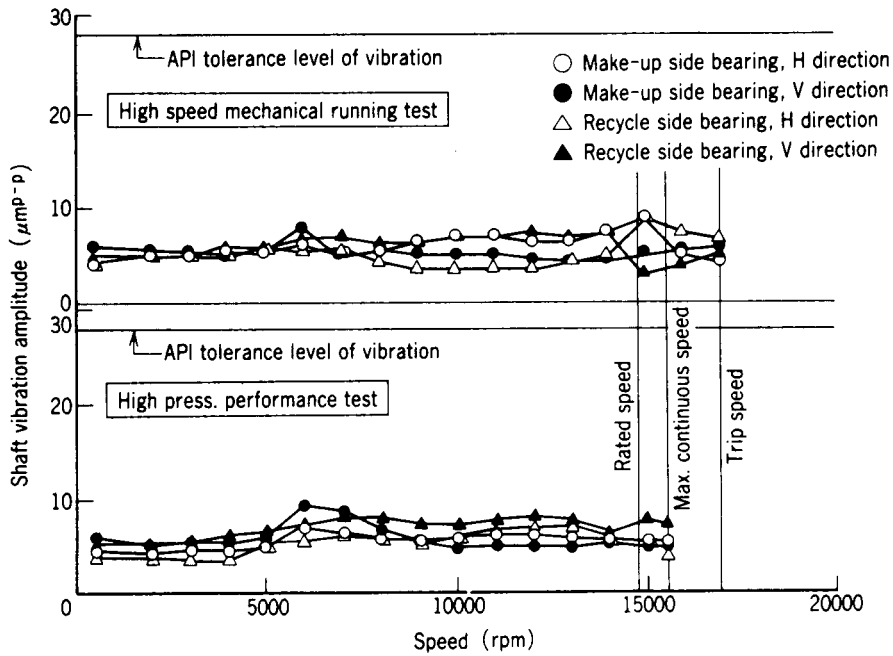Fig.16   Measured stability margin for nonsynchronous vibration
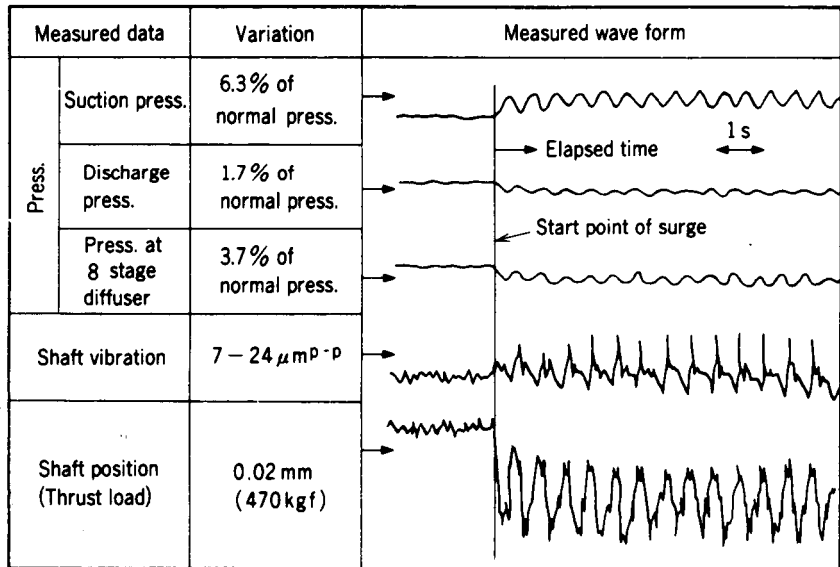
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Fig.17   Measured shaft vibration

Measured data		Variation	Measured wave form
Press.	Suction press.	6.3% of normal press.	
	Discharge press.	1.7% of normal press.	Elapsed time 1 s
	Press. at 8 stage diffuser	3.7% of normal press.	Start point of surge
Shaft vibration		7 − 24 μm p-p	
Shaft position (Thrust load)		0.02 mm (470 kgf)	

Make-up stage surge   Suction flow :   910 Am³/h   Suction press. :   71 kgf/cm²A
Speed : 15540 rpm   Discharge press. : 190 kgf/cm²A

Fig.18   Example of measured data during compressor surge

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