ASYNCHRONOUS VIBRATION PROBLEM

OF CENTRIFUGAL COMPRESSOR

Takeshi Fujikawa, Naotsugi Ishiguro and Mitsuhiko Ito
Kobe Steel, Ltd.
Kobe, Japan

INTRODUCTION

Unstable asynchronous vibration problem in high pressure centrifugal compressor were reported by J.C. Wachel(ref.1), K.J. Smith(ref.2), D.W. Fowlie (ref.3), etc. This paper describes the similar problem we have experienced and the remedial actions against it.

GENERAL CONFIGURATION

A schematic of compressor train system is shown in Fig.1. 5000kW moter drives L.P. (low pressure) and H.P. (high pressure) compressors through the speed up gear. Asynchronous vibration was observed in H.P. compressor. Fig. 2 shows the cross sectional view of the H.P. compressor. The rotor has 6 shrink fitted impellers and one balance piston which is integrated with the shaft. The rotor is supported by tilting pad journal bearing characteristics of which are described in table 1. The Kingsbury type thrust bearing is used. The compressor has 1703 mm bearing span length with the rigid critical speed of 3894 rpm in the original design. L.P. and H.P. compressor are connected with a diaphragm type flexible coupling.

OBSERVED VIBRATION

Asynchronous vibration of H.P. compressor took place when the discharge pressure Pd was increased by controlling the valve, after the rotor was already at full speed. Fig. 3 shows the typical spectral data of the shaft vibration while Pd is increased. As the pressure Pd increases, pre-unstable vibration appears and becomes larger, and large unstable asynchronous vibration occurs suddenly at Pd = 5.49MPa in this case, as shown in Fig. 3 (d), (e).

Fig. 4 is the spectral time history of the shaft vibration in the condition like Fig. 3 (c). The amplitude of pre-unstable vibration fluctuates at some levels. A typical relationship between vibration amplitude and Pd is shown in Fig. 5. Fig. 6 shows the shaft orbit just before the vibration growing up large.

REMEDIAL ACTIONS AND SOLUTION

In order to prevent the above vibration, various kinds of remedial actions are tried based on the results of the complex-eigenvalue analysis mentioned later. Remedial actions adopted and their results are summerized in table 2

and Fig. 8, where

Case ① : The original design

Case ② : The width of pads of journal bearings is reduced to 2/3 of case ①.

Case ③ : The bearing span is shortened from 1703 mm to 1553 mm to increase the shaft stiffness.
Rigid critical speed increased from 3894 rpm to 4560 rpm.

Case ④ : Bearing width is put back to original one and 0.5 pre-load journal bearings are used.

Case ⑤ : Lubricating oil is changed from #90 (viscosity of 34 cSt at 38 degrees C) to #140 (50 cSt at 38 degrees C) in order to increase the damping effect.

Case ⑥ : 0.1 pre-load journal bearings are used.

After all, the full load operation is successfully carried out in the condition of case ⑥.

ANALYSIS

The computer program is used to analyze the rotor stability problem in order to help the remedial plans. The program calculates the logarithmic decrement and the damped natural frequency of the rotor bearing systems. The procedure of the program is as follows. Using a finite element method, the equation of motion of rotor bearing system is given by :

$$[M]\ddot{x} + [C]\dot{x} + [K]x = 0 \tag{1}$$

where   [M] : mass matrix

[C] : damping matrix

[K] : stiffness matrix

The mass matrix consists of the concentrated mass of the rotor sections. The impeller has the mass effect and the gyroscopic effect. The stiffness matrix of rotor is obtained based on the beam theory. The bearing characteristics and the destabilizing factors are dealt with as the concentrated added damping and spring coefficients in the program. The destabilizing force Q is estimated by eq. (2) based on Lund (ref.4).

$$Q = \beta \frac{T}{2rh} \tag{2}$$

where β : destabilizing force coefficient

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T : torque

r : nominal radius of impeller

h : nominal width of impeller

In order to get the eigenvalues, eq. (1) is transformed into the canonical form and QR method is applied. The example of calculation model and results are shown in Fig. 7(a), (b).

DISCUSSION

Since there are many obscure points in the destabilizing forces (aero cross coupling forces), the six cases of remedial actions were prepared and carried out in order to be able to operate at full pressure load.

Fig. 8 shows the calculated results of log-decrement δ and the maximum pressure Pd attained without large vibration for the six cases in the field test.

Fig. 9 shows the change of stability due to oil temperature. Lower oil temperature gives better δ . In calculation of δ, the nondimensional factor β is estimated to 5 including the labyrinth effect. It is seen that the increasing of the rotor stiffness and the oil viscosity are effective to improve the stability of the system. The high preload bearing is not good. The system became to be operated in almost full load in case ⑤ . However, the pre-unstable vibration level was not small, so the case ⑥ was carried out.

The comparison between the results of calculation and field data is shown in Fig. 10 . The bearing was selected in order that the limit β and δ of case ⑥ be larger than those of case ⑤ . The stability of case ⑥ was improved and the pre-unstable vibration was suppressed sufficiently small in the field test.

CONCLUSION

(1) The high speed and high pressure compressor has inherently the possibility of the occurrence of unstable vibration. The stability analysis of δ is effective in the design stage of the rotor bearing system and in the remedial actions in order to prevent the unstable vibration. Details of the mechanism of destabilizing force is not clarified, so it seems better to consider the cross coupling force and the negative damping into the stability analysis.

(2) As the remedial actions, it is effective to increase the shaft stiffness and to select the appropriate bearings in order to increase the system damping.
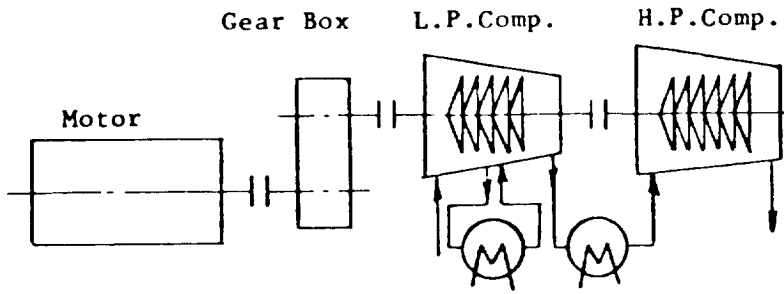
REFERENCES

(1) Wachel, J. C. : Nonsynchronous Instability of Centrifugal Compressors.
   ASME Paper No. 75-Pet-22

(2) Smith, K. J. : An Operation History of Fractional Frequency Whirl.
   Proc. 4th Turbomachinery Symposium, Gas Turbine Laboratories,
   Texas A & M University, Oct. 1975

(3) Fowlie, D. W. and Miles, D. D. : Vibration Problems with High Pressure
   Centrifugal Compressors.
   ASME Paper No. 75-Pet-28

(4) Lund, J. W. : Stability and Damped Critical Speeds of a Flexible Rotor in
   Fluid-Film Bearings.
   Journal of Engineering for Industry, Trans. ASME, May 1974, pp. 509-517

Table 1 Tilting pad journal bearing

Diameter	76.2 mm
Width	31.8 mm
Number of pads	5
Arrangement	Load on pad

Table 2 Remedial actions

	Bearing span length mm	Width of bearing pad mm	Bearing diametral clearance 2 c/D	Bearing preload	Lubricating oil
CASE ①	1703	31.8	2.5/1000	0	#90
②	"	21.2	"	0	"
③	1553	"	"	0	"
④	"	31.8	"	0.5	"
⑤	"	"	"	0.5	#140
⑥	"	"	"	0.1	"

Specifications of H.P. compressor

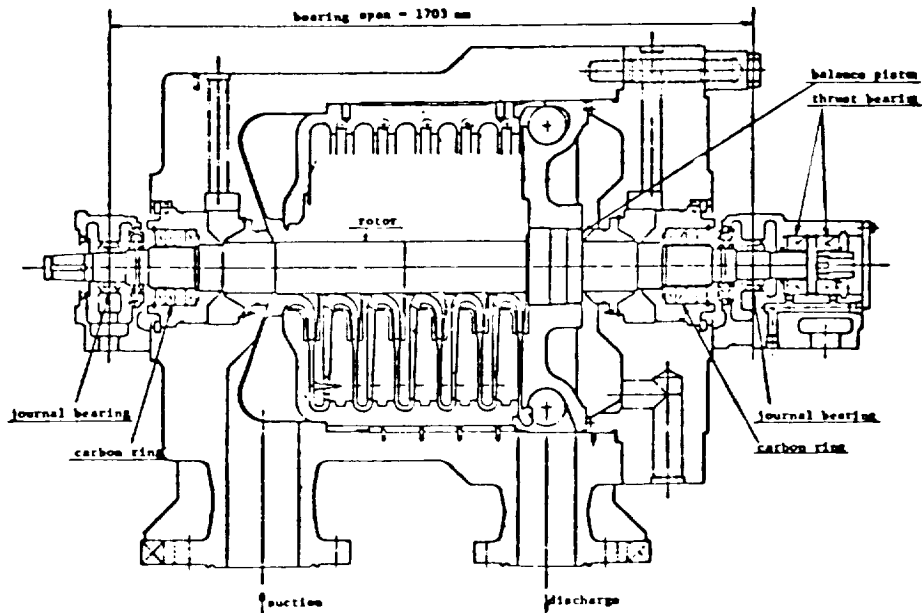| Suction pressure	1.912 MPa
Discharge pressure	6.865 MPa
Shaft horse power	2980 kW
Shaft speed	10400 rpm
Fluid	N_2 gas

Fig.1 Schematic of compressor train

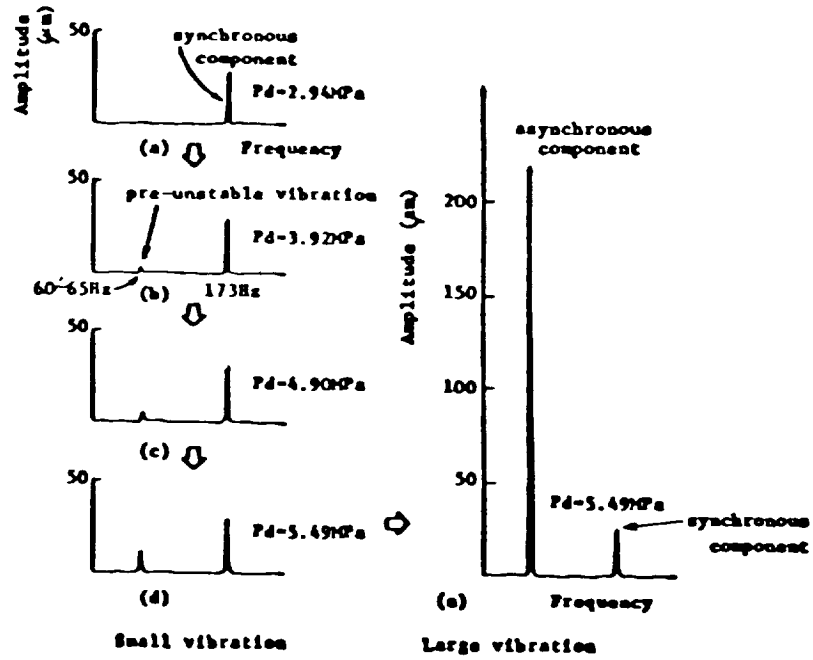Fig.2 Cross-sectional view of H.P. compressor
(Original design)

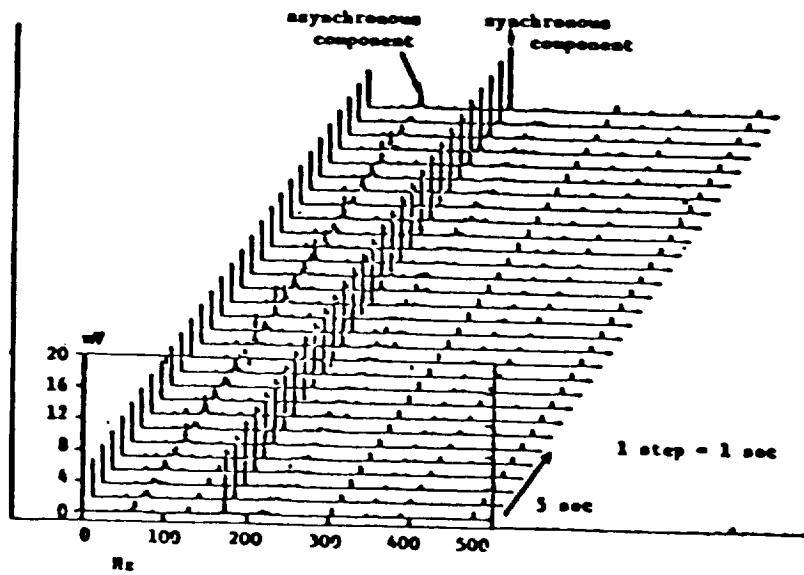Fig.3 Variation of spectral rotor vibration with discharge pressure

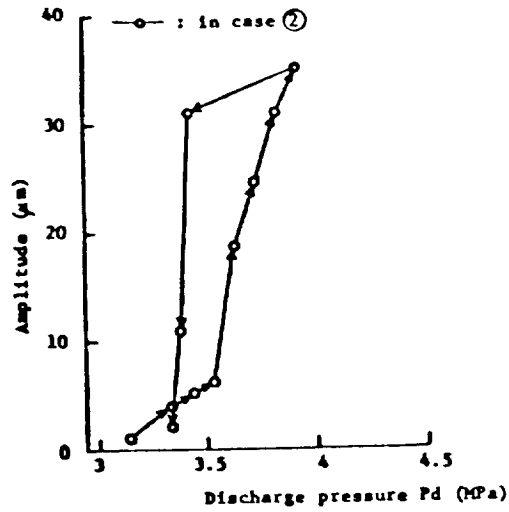Fig.4 Spectral time history of asynchronous small vibration

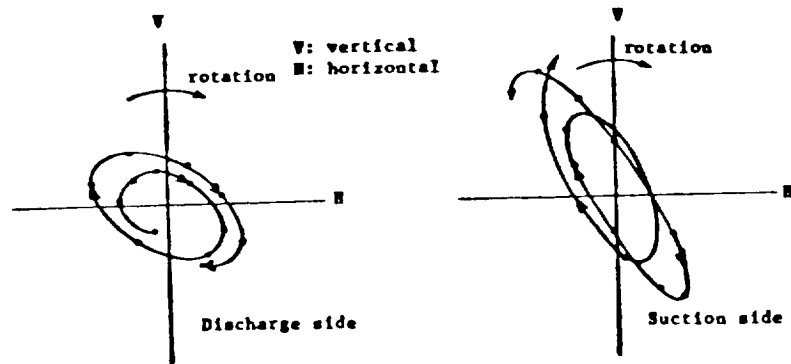**Fig.5 Variation of rotor vibration with discharge pressure Pd**
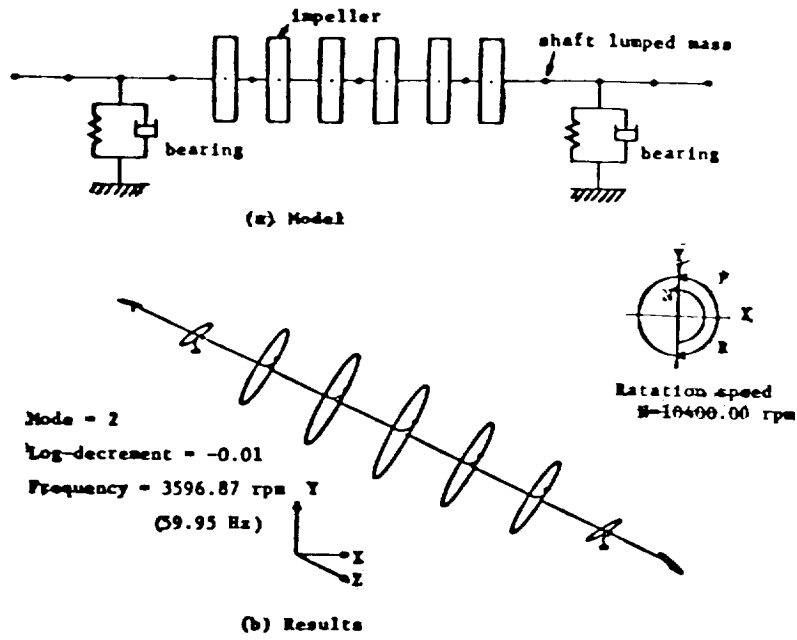

Fig.6 Whirl orbit of asynchronous vibration

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impeller

shaft lumped mass

bearing

bearing

(a) Model

Rotation speed
N=16400.00 rpm

Mode = 2
Log-decrement = -0.01
Frequency = 3596.87 rpm
(59.95 Hz)

(b) Results

Fig.7 Calculation model and results

max. pressure Pd
without vibration
(MPa)

full load

calculated
Log-decrement
δ

original
design

final
condition

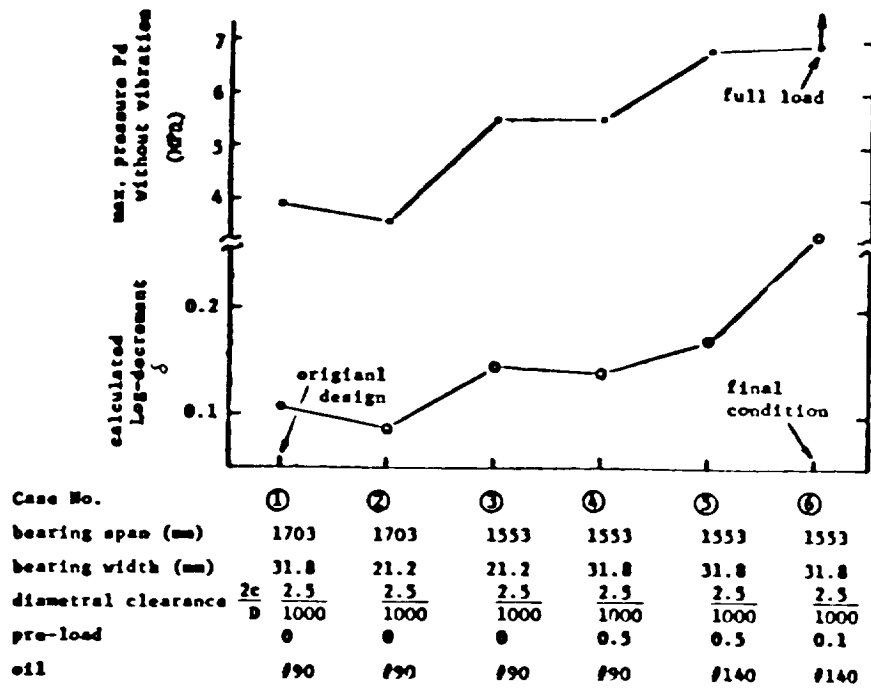| Case No.	①	②	③	④	⑤	⑥
bearing span (mm)	1703	1703	1553	1553	1553	1553
bearing width (mm)	31.8	21.2	21.2	31.8	31.8	31.8
diametral clearance $\frac{2c}{D}$	$\frac{2.5}{1000}$	$\frac{2.5}{1000}$	$\frac{2.5}{1000}$	$\frac{2.5}{1000}$	$\frac{2.5}{1000}$	$\frac{2.5}{1000}$
pre-load	0	0	0	0.5	0.5	0.1
oil	#90	#90	#90	#90	#140	#140

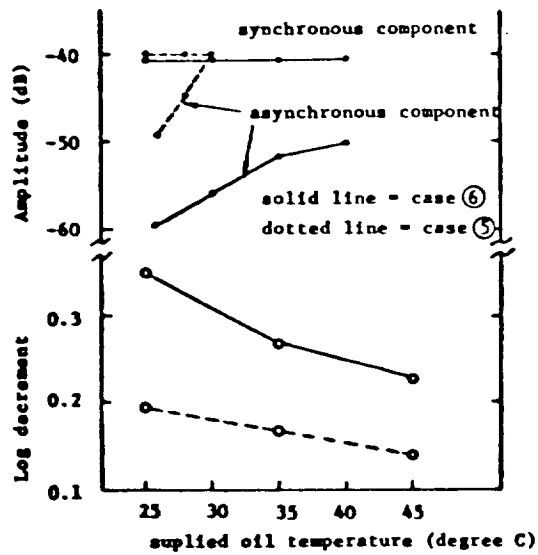Fig. 8 System damping and maximum discharge pressure attained

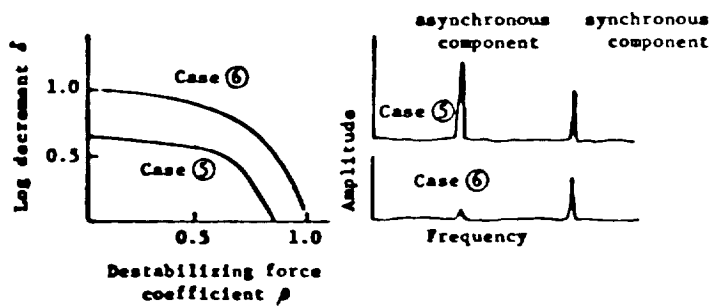**Fig.9 Variation of stability due to oil temperature**

**Fig.10 Comparison between calculated δ and field data
(relative representation)**

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