Integrally Geared Centrifugal Compressors for High-pressure Process Gas Services

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An integrally geared centrifugal compressor, the High Pressure SUPER TURBO, has been developed for high pressure process gas services. The compressor has a modified design to ensure maximum stability of the rotor for high pressure application, and the analysis results met the requirements of the API standards. A full load running test was performed over 80Bar discharge pressure, which confirmed sufficient rotor stability by verifying low rotor vibration and low temperature rise in the bearing pads. This result has expanded the high pressure application coverage of integrally geared centrifugal compressors. The compressors could be applied to the main market of process gas services.

Introduction

Kobe Steel's integrally geared centrifugal compressors, hereafter simply referred to as compressor(s), are widely recognized for their energy-saving performance and small footprint. Kobe Steel has delivered many of these compressors with discharge pressures up to about 50Bar. In recent years, their range of application has been expanding to both larger and smaller sizes, as well as to higher pressures.

In response to this trend, Kobe Steel has developed a compressor, the High Pressure SUPER TURBO, to cultivate the new market. This compressor matches market needs with its high discharge pressure of up to 80Bar. Kobe Steel has conducted an operational demonstration for the specified pressure, using a test machine. This paper outlines the demonstration test and its results.

Kobe Steel has chosen the name Super Turbo, which has wide market recognition, for a series of geared centrifugal compressors for process gas services.

1. Brief description of test compressor

Fig. 1 shows the outside view of the test compressor, while **Table 1** summarizes its specifications. An integrally geared centrifugal compressor generally includes a gear-up unit with a pinion shaft. At least one impeller is attached to and overhangs at one or both ends of the pinion shaft, configuring a rotor. The rotor design is the most important factor in securing the mechanical stability against destabilizing fluid force, a force which increases as the required pressure increases.

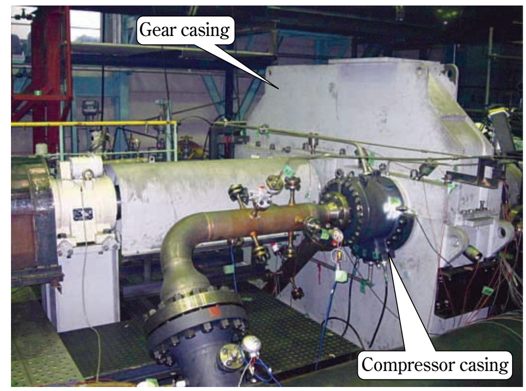

Fig. 1 Outside view of test machine

Table 1 Specifications of test machine

Type	Integrally geared centrifugal compressor (Model : VGP150H)
Application	Dry air booster
Gas	Dry air (MW=28.96)
Number of stage	2
Suc. pres.	LP: 28.1 / HP: 52.1 (barA)
Dis. pres.	LP: 52.2 / HP: 81.0 (barA)
Suc. temp.	40/40 (℃)
Speed	27,900 (rpm)
Motor output	2,900 (kW)
Shaft seal	Tandem dry gas seal

LP : Low pressure stage, HP : High pressure stage

The test machine has a rotor that has been designed for ideal stability. **Fig. 2** compares the rotor profile of a conventional machine (upper) to that of the test compressor (lower). The new design, in which the clearance has been revised, keeps the overhang at a minimum length for improved stiffness. To decrease the mass, the newly developed rotor adopts a pair of impellers made of titanium alloy. A pair of reference impellers made of stainless steel was also prepared. Load tests were performed on both the impellers.

The test machine is designed to use a tandem dry gas seal for sealing its shaft. This seal requires a longer overhang length, which renders the shaft less stable. However, the shaft designed and verified for the tandem dry gas seal can adopt any type of compressor seal. A newly developed casing allows the mounting of the shaft seal.

The bearings significantly affect rotor stability

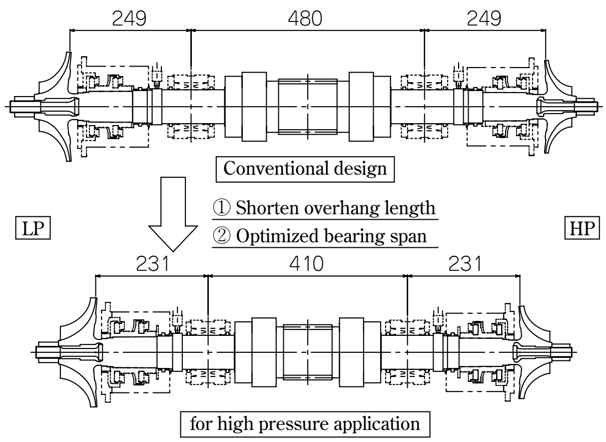

Fig. 2 Comparison of rotor design

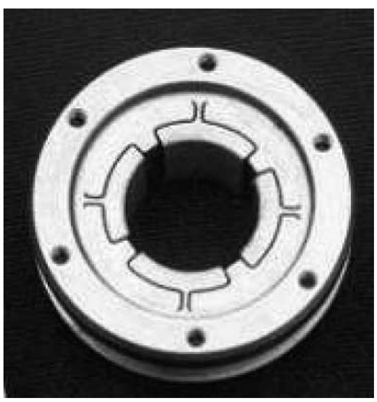

Fig. 3 Tilting pad journal bearing

and high performance is required of them in order to endure high revolutions under high load. With this in mind, two bearings were prepared for the load test, namely,

- a tilting pad journal bearing (**Fig. 3**), and
- a tilting pad journal bearing with a squeeze film damper (SFD).

Bearing specifications (common)

Bearing diameter: ϕ70 (mm)

Revolutions: 27,900 (rpm) [465(Hz)]

Circumferential velocity at the bearing surface: 102 (m/s)

Bearing projection pressure: 213 (N/cm^2)

After careful consideration, it was concluded that conventional gear and casing designs could be adapted to the newly developed machine. Since the compressor is intended to be used for a high pressure, another test was conducted to quantitatively analyze the gas thrust force to meet the high pressure requirement.

2. Rotor stability analysis

The American Petroleum Institute (API) standard [1)], which is most widely known and highly esteemed, defines the anticipated destabilizing fluid force as the

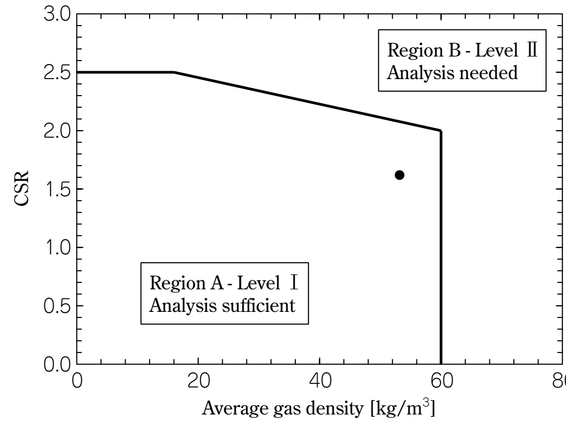

Fig. 4 Screening criteria

cross couple spring stiffness, Q_A, working on the impellers as follows.

$$Q_A=\frac{HP\times B_c\times C}{D_c\times H_c\times N}\times\frac{\rho_d}{\rho_s}\ \text{(kN/mm)} \cdots\cdots\cdots \quad (1)$$

where

HP : gas power (N·m/s = W)

B_c = 3

C = 9.55

D_c : impeller outer diameter (mm)

H_c : impeller outlet width (mm)

N : rotational speed (rpm)

ρ_d : discharge gas density (kg/m^3)

ρ_s : suction gas density (kg/m^3)

To judge the stability of an impeller, an analysis should be conducted assuming the destabilizing force expressed by Equation (1). If the result does not satisfy any of the conditions i), ii), or iii), then the impeller is regarded as stable. If it satisfies any one of them, a more detailed Level Ⅱ analysis is required.

i) $Q_0/Q_A<2.0$

ii) $\delta_A<0.1$

iii) $2.0<Q_0/Q_A<10$ and the critical speed ratio (CSR) falls in region B of **Fig. 4**,

where

Q_0 is the destabilization spring constant which yields zero logarithmic decrement, δ ,

δ_A is the logarithmic decrement rate with assumed destabilization spring constant of Q_A, and

CSR is the maximum continuous speed/first undamped critical speed on rigid support.

Fig. 5 shows the results of the stability analysis conducted on the present test machine. The destabilization spring constant and the logarithmic decrement are given as follows and satisfy neither of the conditions i) nor ii).

- Q_0/Q_A= 2.8 (>2.0)
- δ_A= 0.28 (>0.1)

As indicated by the dot in Fig. 4, the relation between the CSR value, an indicator of rotor stiffness, and the average gas density falls in region A and

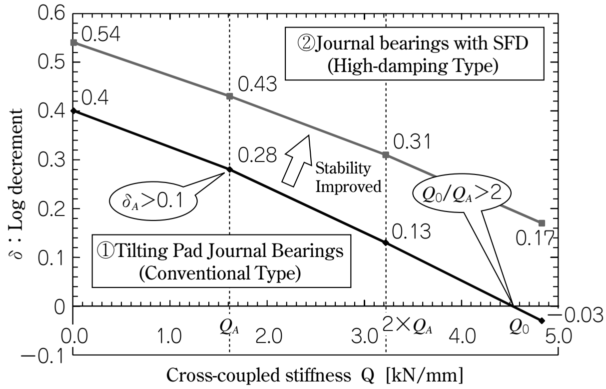

Fig. 5 Rotor stability analysis

Table 2 Rotor hammering test

	Eigenvalue (Hz) (free-free)	
	Measured value	Calculated value
1st mode	343	333 (97.1%)
2nd mode	690	692 (100.3%)
3rd mode	1,339	1,343 (100.3%)

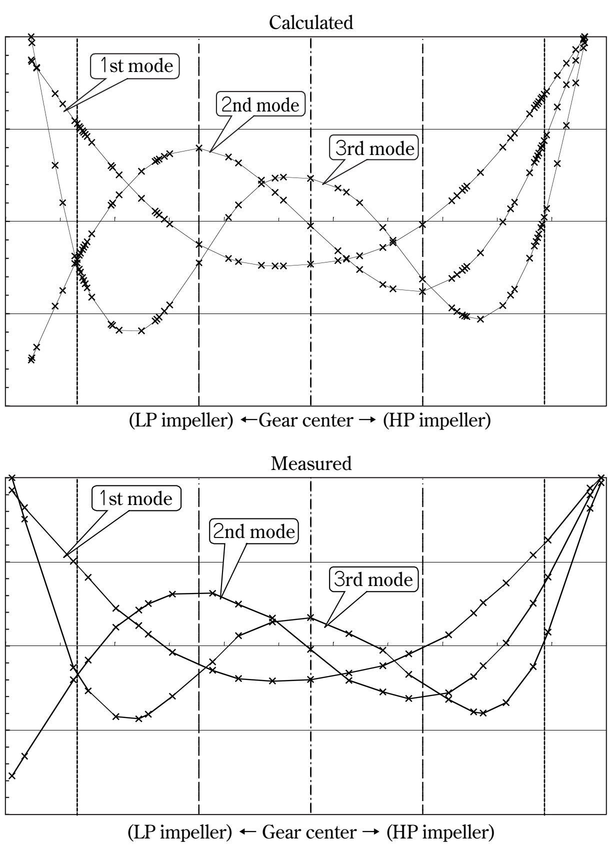

Fig. 6 Comparison of mode shape

does not satisfy condition iii).

Thus, the rotor of the test machine does not satisfy any one of the conditions i) through iii), demonstrating sufficient stability under the API standard. The analysis results show that the machine with the squeeze film damper performs better than the one with only the tilting pad journal bearing.

A rotor hammering test was conducted in free-free mode to verify the appropriateness of the rotor model used for the analysis. As shown in **Table 2**, all the results from the first to the third mode indicate that the differences between the calculated values and measured values are within 3%, verifying sufficient prediction accuracy. As shown in **Fig. 6**, the calculated characteristic vibration modes match well with the measured values, verifying the appropriateness of the calculation model.

3. Actual load running test

A closed loop actual load test was conducted on the compressor using a test bench made by Kobe Steel. Nitrogen, pressurized up to about 80Bar, was used as the test gas. Measured items include shaft vibration and the temperature of the bearing pad (**Table 3**).

The API standard specifies the allowance for shaft vibration during in-house mechanical running to be about 17μm. The test run results were mostly within this allowance even at full load. **Fig. 7** is the result of the Fast Fourier Transform (FFT) analysis of the shaft vibration. As shown in the figure, the predominant vibration appears at 465Hz, the primary frequency component of the rotational speed. No significant asynchronous vibration, characteristic of rotor destabilization, was observed. Thus it is confirmed

Table 3 Records of running test

Item	Unit	① Rated	② Near surge	③ Middle point	④ Max. flow
Motor input	(kW)	2852.8	2387.7	2619.5	3077.6
Shaft vibration LP-H	(μ)	13.4	10.8	11.7	16.7
Shaft vibration LP-V	(μ)	12.5	10.2	11.0	13.9
Shaft vibration HP-H	(μ)	15.8	16.8	15.9	16.0
Shaft vibration HP-V	(μ)	14.2	15.3	14.1	14.2
Supply oil temp. at comp	(℃)	41.6	41.5	41.5	41.5
J-bearing temp. LP	(℃)	63.2	62.5	62.8	63.2
J-bearing temp. HP	(℃)	68.7	65.9	67.3	69.9
LP stage suct. press.	(MPa)	2.883	2.690	2.747	3.016
LP stage disch. press.	(MPa)	5.400	5.211	5.277	5.459
HP stage suct. press.	(MPa)	5.379	5.199	5.262	5.432
HP stage disch. press.	(MPa)	8.010	8.081	8.055	7.790

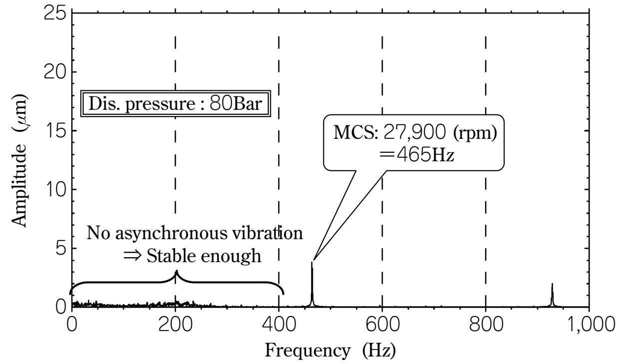

Fig. 7 FFT analysis of shaft vibration

that the test compressor can safely be run at the specified pressure.

In addition, temperatures were measured for the entire journal bearing pad. The measured maximum of 70℃ is well within the standard values of Kobe Steel.

4. Rotor destabilization test

To define the limit at which the rotor destabilizes, a test was conducted to determine the vibration behavior under decreased rotor stability and increased destabilizating fluid force. An impeller made of stainless steel was adopted to increase the added mass. A load test was conducted using argon

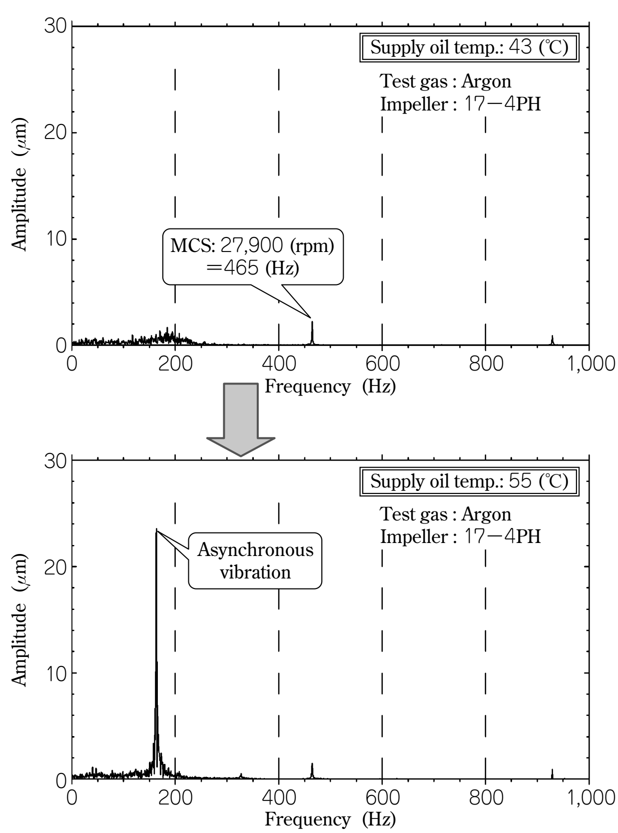

Fig. 8 Results of destabilization test

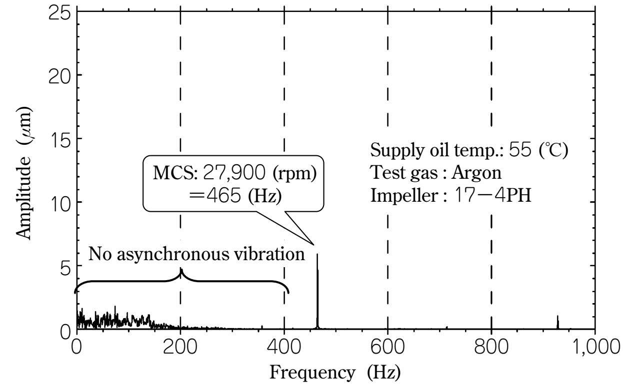

Fig. 9 Test result of SFD bearing

(molecular mass = 39.948) for the test gas and the vibration behavior was observed. The evaluated destabilizing fluid force at 80Bar of discharge pressure using argon gas corresponds to an approximately 13% increase in destabilizing force as against the same discharge pressure predicted by Equation (1) (a 15% average increase in gas density) when the original gas (nitrogen) was used.

The frequency analysis of shaft vibration revealed no significant asynchronous vibration at a supply oil temperature of 43℃(design temperature) and no impairment of rotor stability. Raising the supply oil temperature to 55℃, however, causes an asynchronous vibration as shown in **Fig. 8**.

On the other hand, the substitution of an SFD for the bearing caused no significant asynchronous vibration even in the case where the bearing caused an asynchronous vibration (**Fig. 9**), verifying the improvement in stability due to the SFD.

5. Measuring the gas thrust force

The gas thrust force was measured during the load test to collect the data useful for high load design. The force was measured by the load cells embedded behind the four pads of the thrust surfaces on the loaded side and unloaded side of the thrust bearing for the low-speed shaft. The force is transmitted from the pinion shaft to the low-speed shaft via a rider ring (thrust collar). Verification tests were conducted before the actual load test. The thrust force was evaluated by taking the average of the values measured at the four points. **Fig. 10** shows the bearing and load cells used for the test.

Table 4 summarizes the average values of the measured loads on the four pads which represent the trust force at 80Bar which is in the vicinity of the maximum load. The measured values fall within 10% of the thrust force (17.1kN) calculated on the basis of Kobe Steel's design method. The results verify the appropriateness of the calculated value.

Fig. 10 Thrust bearing with load cell (for measurement of gas thrust force)

Table 4 Measurement of gas thrust force

	Measured value (kN)	Differential (*)
1st trial	17.4	+2.2%
2nd trial	18.0	+5.5%
3rd trial	17.8	+4.2%

(*) Caluculated thrust force : 17.1 (kN)

Conclusions

This development has established a method for designing an integrally geared compressor which can handle up to about 80Bar of process gas. The new design has expanded the applicability of integrally geared compressors characterized by a small footprint and energy saving.

Kobe Steel will continue to develop technologies for further expanding the applications of integrally geared compressors, with an eye to the upsizing and high pressure applications such as carbon dioxide capture and storage.

References

1) API STANDARD 617 SEVENTH EDITION, JULY 2002.