Development of CO_2 Compressor for Fertilizer Plant

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This paper describes the development of new CO_2 compressor for a fertilizer plant, features of which are high efficiency and compactness as well as high reliability. Since compressed gas at the final stage of a CO_2 compressor has high density, various compressor behaviours including rotating stall, impeller vibration, rotor stability, and leak loss must be fully taken into consideration. In order to fully understand and to solve these subjects, various analytical and experimental studies have been carried out, and a state-of-the-art compressor was developed. Full load and full pressure performance tests have been performed, and satisfactory results have been obtained.

1. Introduction

Because of the high pressure ratio, the CO_2 compressor train for a fertilizer plant is composed of two units : a low-pressure compressor and a high-pressure compressor.

The low-pressure compressor has a discharge pressure of about 45 kgf/cm²abs, the compression range being the same as in compressors for normal process application. However, the discharge pressure of the high-pressure compressor is 150−200 kgf/cm²abs, which causes an extremely high density 150−200 kgf/cm²abs and molecular weight is so high, hence, the following technical problems must be solved.

(1) Since the gas state in the final section is in the proximity of its critical point, the evaluation of gas properties in this zone is considered to be difficult for the accurate evaluation of gas properties in this zone.
(2) The absolute value of pressure fluctuation becomes large when the rotating stall occurs because of the high pressure and high density of the gas, causing the impeller likely to be damaged when resonating at its natural frequency. Hence, it is important to predict the intencity as well as the onset point of the rotating stall, and to carry out tuning of the impeller's natural frequency.
(3) Because of the high density of the gas, the gas exciting force generated in the impeller and labyrinth is larger than in compressors for other process application, and is likely to cause unstable shaft vibration.
(4) When a labyrinth seal is adopted, the internal leakage loss is large at high pressure, and depending on the selection of the sealing pressure and temperature, there is a case that ice is formed owing to the expansion of gas when it passes through the sealing section.

Mitsubishi Heavy Industries, Ltd. (MHI) has solved these technical problems, has manufactured a test machine for high-pressure compressors and has successfully developed a highly efficient and highly reliable CO_2 compressor for fertilizer plants. The outline of test results as well as the technical studies made during the process of development, are described as follows :

2. Specifications and features of the compressor

2.1 Specifications of the compressor

The specifications are for the CO_2 compressor for a 1 100 metric tons per day fertilizer plant with a comparatively high discharge pressure as shown in **Table 1.**

2.2 Compressor features

The cross section of the compressor train is shown in **Fig. 1**. The new compressor is composed of a low-pressure

Table 1 Compressor specification

Item	Unit	Value
Suction flow rate	m³/h	12 700
Suction press.	kgf/cm²abs	1.83
Suction temp.	°C	38
Discharge press.	kgf/cm²abs	186
Max. continuous speed	rpm	13 490

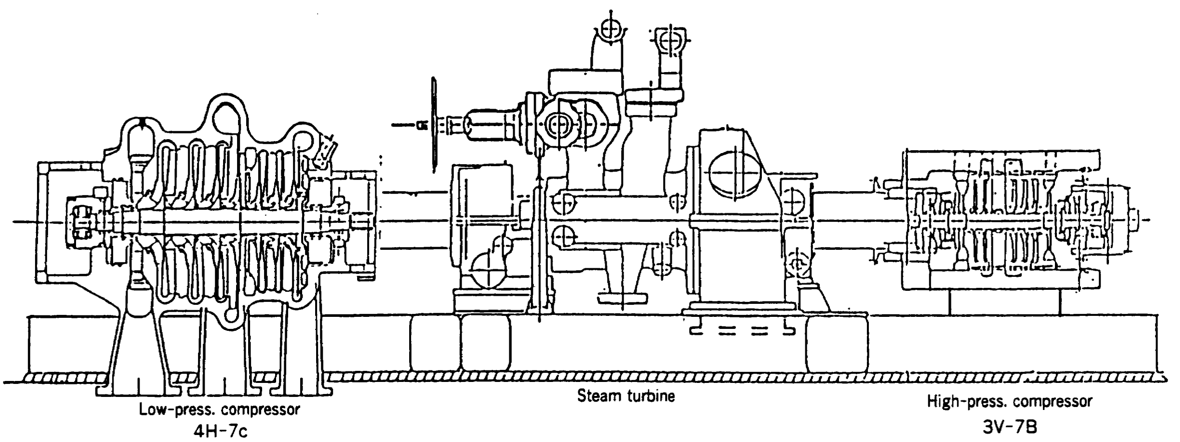

Fig. 1 Cross section of compressor train
An extraction condensing steam turbine is installed in the center, a low-pressure compressor at the left and a high-pressure compressor at the right.

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compressor equipped with a horizontal split type casing(combined type) and a high-pressure compressor equipped with a vertical split type casing (back-to-back type), with the double shaft extended extraction condensing steam turbine, produced by MHI, installed in the center. The compressor has the following features.

(1) High efficiency

The efficiency of the newly developed compressor is compared with the efficiency of a conventional compressor used for a 1 100 metric tons per day plant in **Table 2**. The efficiency has been drastically improved owing to the application of the high-efficiency, three-dimensional impeller and the adoption of the gas seal[2], produced by MHI, to reduce the internal leakage.

(2) Small and compact

Instead of the speed increasing gear unit adopted for the high-pressure compressor in conventional compressors, the newly developed compressor uses a high-specific-speed impeller for the low-pressure compressor to increase the speed, leading to a deletion of speed increasing gear for high pressure compressor and compactness of the equipment.

3. Design of the compressor

3.1 Physical properties of CO_2 gas

It is extremely important to estimate accurately the physical properties of the gas handled as well as the impeller performance when selecting the compressor for given specifications. In particular in the case of a CO_2 compressor for a fertilizer plant, the change in the gas condition is great as shown in **Fig. 2**, and the pressure rise is observed to be in the proximity of the critical point where it is difficult to correctly estimate the physical properties of the gas. Thus, at worst, mismatching between impellers occurs, and the required performance would not be satisfactory.

Hence, data regarding the physical properties of several gases were compared and studied thoroughly, and the performance was predicted using the IUPAC (International Union of Pure and Applied Chemistry) Gas Table which is considered to have a high accuracy of the physical properties of gases even in the proximity of the critical point.

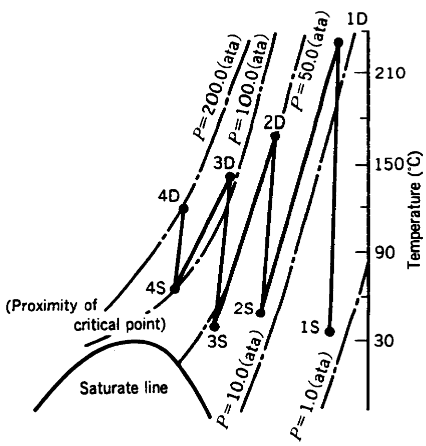


Fig. 2 Change of press. and temp. for CO_2
Changes of gas state at pressure rise is shown on Mollier chart.

Table 2 Merit of new compressor

Shaft power of compressor		Power reduction
New compressor	Conventional compressor made by another co.	
4 391 kW	4 644 kW	253 kW(6%)

3.2 Rotating stall of impeller

As the suction flow rate is gradually decreased, the gas inlet angle from impeller to the diffuser (the angle from tangential direction) become small, leading to a local reverse flow and causing rotating stall. In particular,in the case of the high-pressure, high-density gas in a CO_2 compressor, the rotating stall brings about strong pressure fluctuation when it occurs, causing the shaft vibration to increase to an extent which results in the failure of operation. In worst case, the impeller is likely to resonate and be damaged.

Hence, it is important to keep the rotating stall onset point out of the operating range, and the rotating stall onset point has been designed to be more than 10% away from the minimum flow operating point toward the low-flow side by reducing the diffuser width and enlarging the gas inlet angle.

3.3 Impeller

Since the gas becomes high density at the final discharge section with its specific gravity being one-third that of water, it is necessary to take into account the effect of inertia and damping effect of the fluid just the same as pump impeller design or water turbine rummer design to avoid large errors in estimating the natural frequency, etc. Hence, it is important to determine the impeller structural dimensions by evaluating the vibration characteristics taking due account of the CO_2 gas density, the stiffness of stationary components such as the diaphragm, etc. when designing the CO_2 compressor impeller. In this development the impeller natural frequency in high-density CO_2 gas was obtained by applying the natural frequency analysis method[3]employed for water turbine design. The analysis result showed that the natural frequency in CO_2 was lowered to about 64% of that in a normal atmosphere, and the impeller was designed taking account of this reduction ratio.

4. Verification test

A high-pressure compressor of CO_2 compressor train was manufactured and subjected to various tests and inspections, with the following two items being the main purposes of the verification test.

(1) To confirm that the elements installed in the actual machine show the performance as expected

(2) To verify the performance and reliability of the compressor under actual load operations

4.1 Running test at the shop

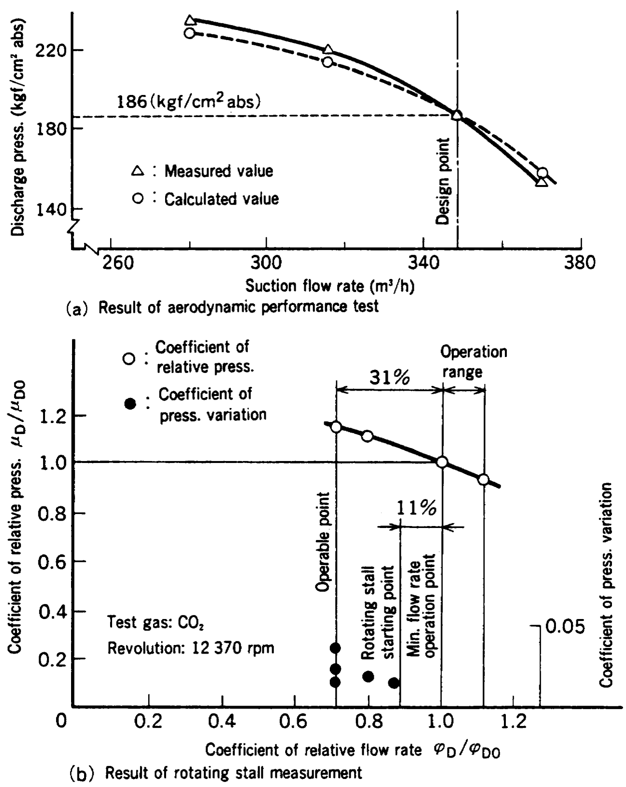
The test compressor was made to have two sections internally to enable the gas loop to be set either independently in series for each of the two sections, and the following tests were made in stages.

(1) High-speed mechanical running test

Confirmation of mechanical stability at low load operation, and gas seal characteristics at high-speed rotation

(2) Low-pressure performance test

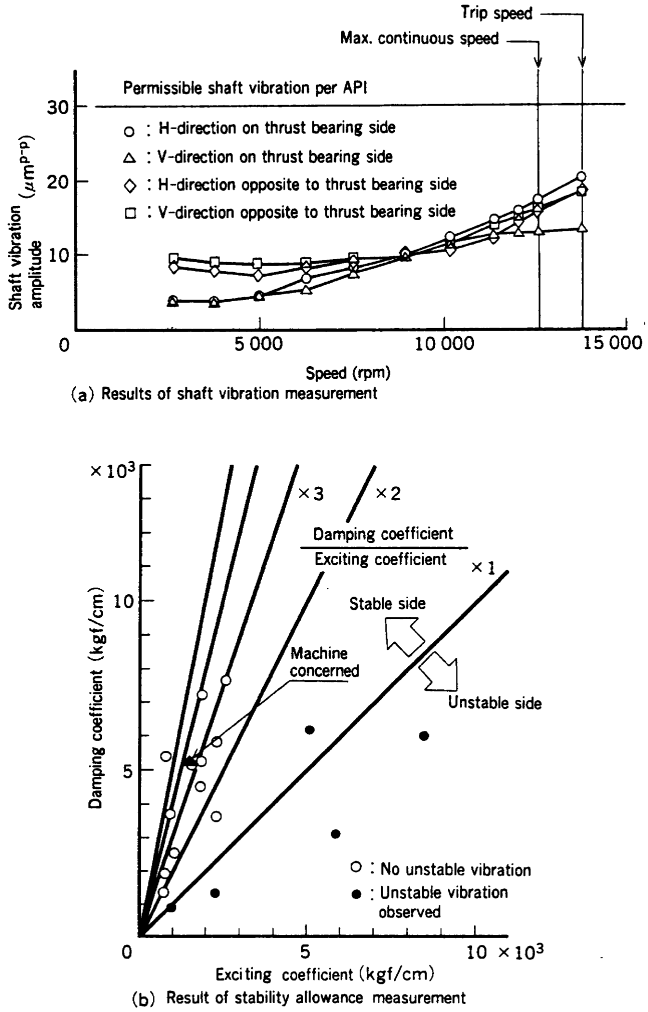
Confirmation of aerodynamic performance and shaft power



(a) Result of aerodynamic performance test

(b) Result of rotating stall measurement

Fig. 3 Test result at full load condition
Measured results of aerodynamic characteristics under actual load testing (flow rate—pressure characteristics) and rotating stall pressure change at final stage.



(a) Results of shaft vibration measurement

(b) Result of stability allowance measurement

Fig. 4 Measured shaft vibration and measured stability for non-synchronous excitation
Measurement results of shaft vibration under actual load testing and those of shaft stability allowance obtained through non-synchronous sweep excitation of the chamber under actual load operation.

(3) Actual load test
Confirmation and verification of aerodynamical performance, mechanical stability and reliability under actual operating conditions

All tests were made according to API 617 and ASME PTC-10, and confirmation tests were further added under special conditions. Furthermore,stability of the rotor, the pressure fluctuation and displacement during possible transient-state or unstable operation, the thrust force, the temperature of the stationary ring of the gas seal, etc. were specially measured in order to obtain all the data needed for evaluating the test compressor.

5. Test results

The test results showed that the test compressor conformed to all its anticipated characteristics and functions, and verified the performance and reliability of the compressor to treat high pressure and high density CO_2 gas. The results are outlined as follows :

5.1 Aerodynamic performance

The aerodynamic performance of the flow and pressure characteristics was obtained as expected as shown in **Fig. 3** (a). The shaft power of the compressor, shown in Table 2, was obtained through these measured values, verifying the high efficiency of the developed compressor, which is the greatest feature of the machine.

5.2 Rotating stall

The pressure fluctuation at the impeller outlet was measured to check its behavior under rotating stall, with the results shown in Fig. 3 (b). The rotating stall onset point was 11% away from the minimum flow operation point toward the low-flow side, which was better than the design target of 10% ensuring enough stability within the operation range. Even if rotating stall occured, the impeller did not resonate, enabling operation up to 31% from the minimum flow operation point towards the low-flow side which ensured a wider operation range than conventional compressors.

5.3 Rotor vibration characteristics

The shaft vibration obtained had good margines compared to the permissible value of shaft vibration as specified in API standards, similar to the Mitsubishi centrifugal compressor up to now manufactured by MHI [**Fig. 4** (a)]. Furthermore, the shaft stability obtained through non-synchronous sweep excitation[4]of the casing at full load operation was confirmed to be within the degree of allowance as shown in Fig. 4 (b).

5.4 Impeller vibration measurement

The final stage impeller vibration for each section was measured using a non-contact displacement sensor. The result, as shown in **Fig. 5** (a), shows the presence of only rotational

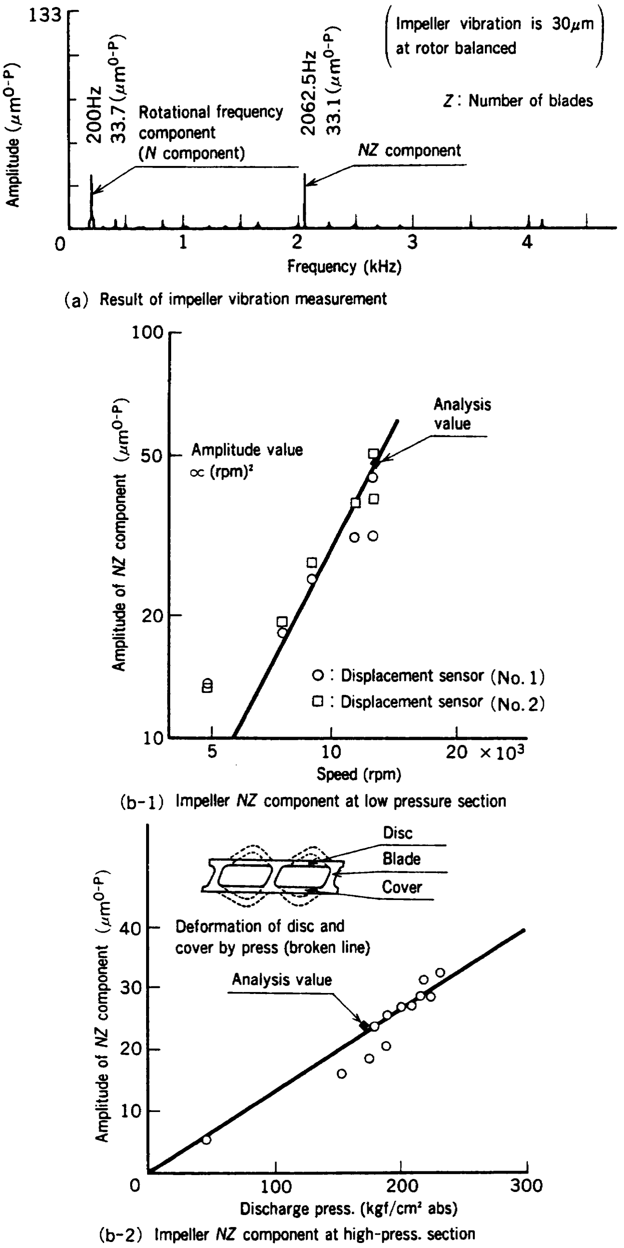


Fig. 5 Impeller vibration characteristics
Measurement results of the impeller vibration at final stage under actual load operation and impeller NZ component vibration (analytical drawing of the component multiplied by the number of blades).

Table 3 Impeller natural frequency

Mode	Final-stage impeller		
	Measured value in air (A)	Measured value in CO_2 gas (B)	Reduction ratio B/A
0 ND	2 312 Hz	1 492 Hz	0.65
1 ND	1 960 Hz	1 250 Hz	0.64
2 ND	2 246 Hz	1 375 Hz	0.61

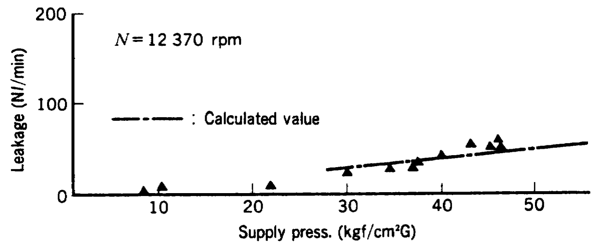


Fig. 6 Measured gas seal leak
Measurement results of gas seal leak under actual load testing.

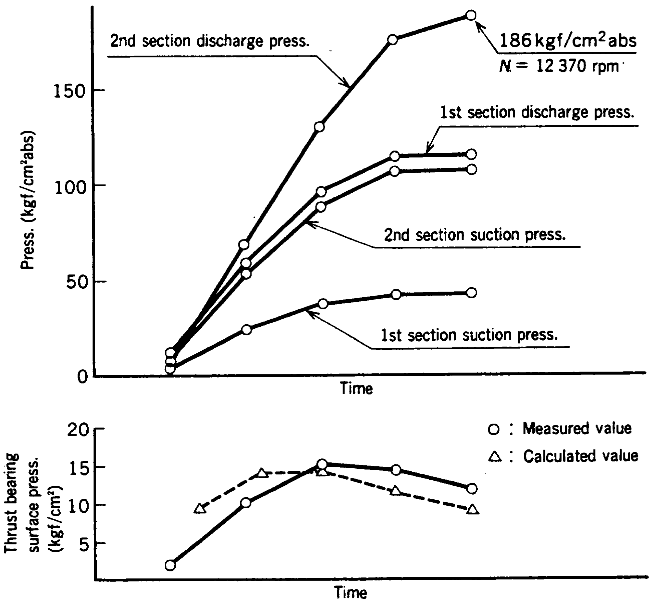


Fig. 7 Measured thrust load (start up)
Measured results of thrust bearing load at start up.

frequency component (N component) and rotational frequency component multiplied by the number of blades (NZ component) even under full load operation. The N component was measured as the impeller angular displacement resulted by assembly, while the NZ component was measured as the deformation of the impeller owing to centrifugal force in the low-pressure section and as the deformation of the impeller owing to internal gas pressure in the high-pressure section as shown in Fig. 5 (b), thereby verified no sign of abnormal vibration. Furthermore, the natural frequency estimated from random vibration is shown in **Table 3**, with the reduction ratio of natural frequency found to be as in the design target.

5.5 Gas seal characteristics

As shown in **Fig. 6**, the gas seal leak under full load testing was found to be about 50 N*l/min* at a seal supply pressure of 45−50 kgf/cm²G, nearly conforming to the estimated value by calculation and much lower than the target value (220 N*l/min*).

5.6 Reliability at start and stop

The CO_2 compressor has a large thrust force generated at each impeller, so that the balancing of thrust forces is a big concern at design. It is verified,as shown in **Fig. 7**, that the surface pressure exerted on the thrust bearing at start up is substantially lower than the permissible surface pressure of 30 kgf/cm².

Furthermore, the shut-down test, made under full load operation as a simulation of an emergency stop, verified that each part starts the speed down operation and comes to a safe stop without any abnormality.

5.7 Overhaul inspection

The overhaul inspection was carried out after completion of all tests, and no abnormalities were confirmed.

6. Conclusion

MHI has developed a highly efficient, highly reliable, small, and compact compressor by solving a number of the technical problems regarding CO_2 compressors for fertilizer plants through integration of our independently developed fundamental technology.

The gas physical properties computing method (IUPAC), the prediction of rotating stall, and analytical technology for an impeller's natural frequency in high-pressure, high-density gas, obtained through the process of development, have been verified the expected level of accuracy through the verification test, and it is considered that they will contribute to the development and improvements in quality of the compressor in the future.

Hopefully, this newly verified technology will be helpful in improving the efficiency and reliability of plants in the future.

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