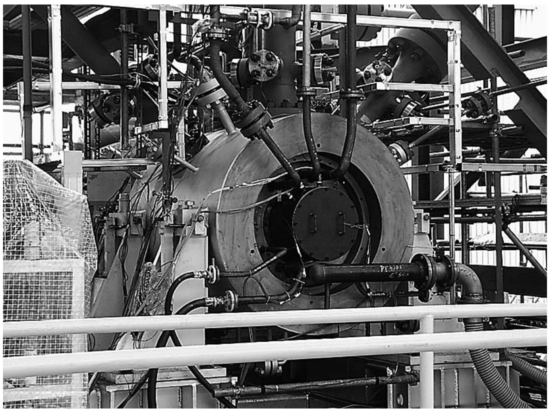

High Pressure 450 bar Natural Gas Injection Compressor

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As the demand for oil and gas increases, the need for enhanced oil recovery, also known as EOR, and the demand of natural gas injection compressors has been on the rise. Mitsubishi Heavy Industries, Ltd. (MHI) has developed an injection compressor placing particular emphasis on reliability for handling high-pressure and high-density gas in addition to high efficiency as essential features for MHI's compressors. MHI carried out shop full load tests using natural gas at a discharge pressure of 450 bar, and demonstrated the high reliability of the compressor under the same operating conditions as those on site.

1. Introduction

Natural gas injection compressors are used for re-injecting gas deep in the ground near oil wells and natural gas wells. Since they handle ultra high-pressure and high-density gas, they have unique difficulties in terms of design and operation. In actual operations on site, a large number of problems with compressors, including the unstable shaft vibration, have also been reported.

MHI has delivered a variety of high-pressure compressors mainly for the petrochemical industry, and has developed a new natural gas injection compressor using technologies developed so far. In the design of the compressor, emphasis was placed mainly on ensuring high reliability demanded under a wide range of use conditions.

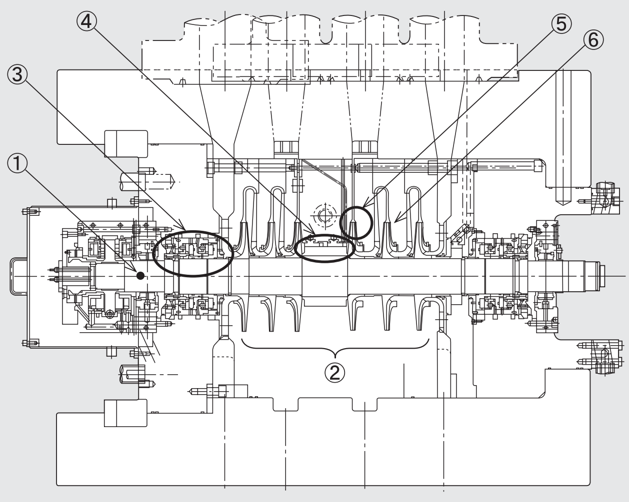
2. Specifications of compressor

In some cases, compressors having a discharge pressure of up to 700 bar are used for natural gas injection. Since the frequency of use of compressors with discharge pressures of 450 bar or less is high, MHI used this pressure as a design parameter. The specifications of the newly developed compressor are shown in **Table 1**, and a cross sectional view is shown in **Fig. 1**.

Table 1 Specifications of injection compressor

Compressor model	3V-6B
Gas handled	Natural gas
Suction pressure	205 bar
Discharge pressure	450 bar
Design speed	11 000 rpm
Required power	8 900 kW

The impeller was divided into high- and low-pressure sections, and arranged as back to back. Thus, losses due to internal circulation flow could be reduced to increase efficiency, and variations in shaft thrust force due to variations in operating conditions could also be reduced. When the impellers are arranged in series, the thrust force may vary significantly. Continuous operation may be obstructed when the compressor is operated at off-design points or the labyrinth seal is worn by long-term operation. However, the thrust forces are set against each other through a back-to-back arrangement so as to reduce variations in the total thrust force acting on the shaft.



① High rigidity rotor
② High bore/tip ratio impeller
③ 350 bar tandem dry gas seal
④ Abradable seal with swirl canceller
⑤ Reduction in flow at point where rotating swirl occurs
⑥ High precision gas passage impeller

Fig. 1 Cross sectional view of injection compressor

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In addition, the shaft diameter and bearing diameter relative to the diameter of the impellers were increased, and the bearing span was minimized. Use of this highly rigid rotor has made it possible to realize an increase in the reliability on high-density gas with respect to instability.

3. Aerodynamic performance

3.1 Design and manufacture

The aerodynamic design and manufacture of the impeller can be briefly summarized as follows.

(1) A high efficiency three-dimensional impeller is applied to all compressors.

(2) All standard impellers are designed using CFD analysis.

(3) Performance tests are performed on all standard impellers, and a database for the results thus obtained is built and maintained.

(4) All impellers are similarly designed based on data on the standard impellers. Verified data is also used for the physical properties of the gas, which is indispensable to proper aerodynamic design.

(5) All impellers are machined using highly accurate, 5-axis control machine.

3.2 Results of performance tests

Shop performance tests were conducted in accordance with the ASME PTC-10 Performance Test Code to verify the validity of the design of the injection compressor in terms of performance. The relation between the flow coefficient obtained from the tests and the polytropic efficiency/work coefficient is shown in **Fig. 2**.

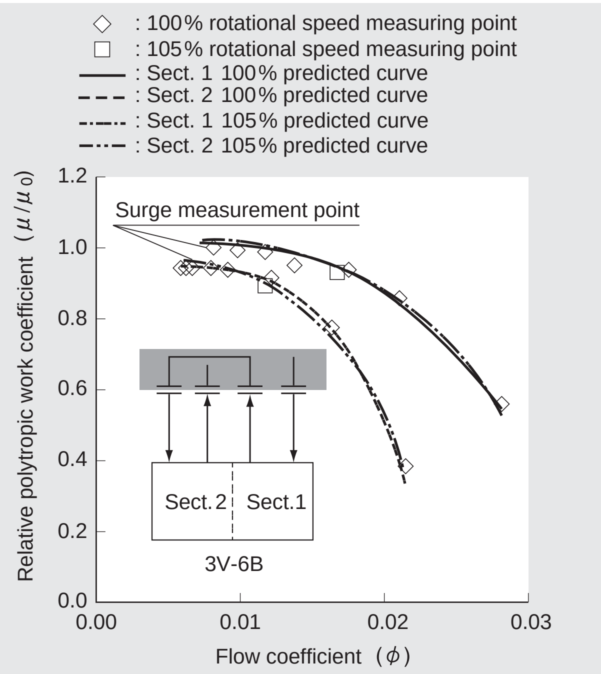


Fig. 2 Results of low pressure performance test
Calculated values indicated by solid and dotted lines match well the measured values, which are indicated by ◇ marks.

The design values were found to match the predicted values quite well. As a result, the techniques used for predicting performance were confirmed to be highly accurate, and the test and evaluation methods based on the law of aerodynamic similarity were also verified to be appropriate for application to validating high-pressure compressor designs, as well.

To confirm the performance of the compressor under actual operating conditions, full load performance tests were performed using natural gas at a discharge pressure of 450 bar. The test conditions, based on ASME PTC-10 type I, are equivalent to the actual operating conditions on site. The relation between the flow coefficient obtained from the tests and the polytropic efficiency/work coefficient is shown in **Fig. 3**.

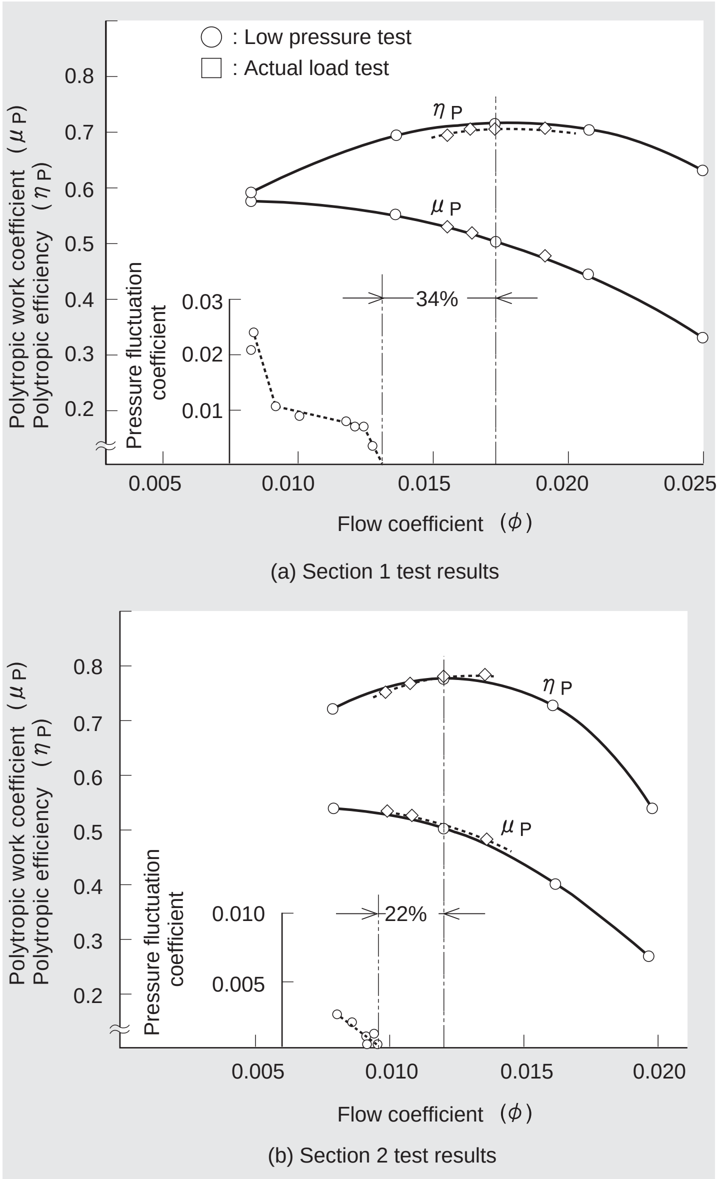


Fig. 3 Comparison between low pressure test results and full load performance test results
Low pressure test results indicated by ○ marks closely match the full load performance test results indicated by ◇ marks.

The results of the full load tests also match the results of the low-pressure tests, thereby confirming the predictive accuracy of performance under high pressure.

3.3 Rotating stall

It is well known that when gas flow is reduced from normal operating conditions, gas flow around the impeller and diffuser exhibits rotating stall before surging. Though this occurs in all compressors, in many cases, it is not detected from outside the compressor and does not cause any problems with operation.

However, since high-density gas is used in an injection compressor, the compressor generates unstable shaft vibrations as a result of the gas force that is produced by rotating stall. Such instability could potentially lead to trouble during continuous operation.

A diffuser with a smaller width than that normally used in a low-pressure compressor was adopted in the injection compressor. As a result, the point at which rotating stall occurs can be extended to a low flow.

Fig. 3 shows the results obtained by measuring pressure fluctuations caused by rotating stall in full load performance tests with a sensor fixed on the diffuser. Rotating stall occurs at low flow point of 34% in the first section from the design point and at a low flow point of 22% in the second section from the design point. This indicates that a safe operating range has been sufficiently secured.

4. Rotor dynamics

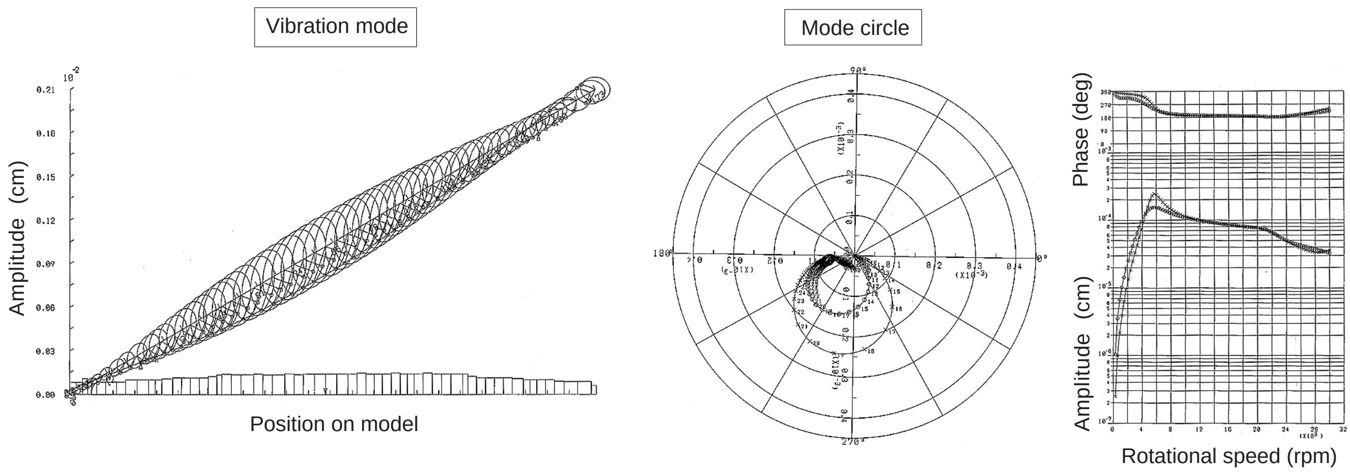
4.1 Critical speed

As can be seen in **Fig. 4**, the calculated critical speed of the compressor sufficiently differ from the operating speed, and the amplification factor is also small. The results obtained from the shop tests match the calculated values quite well, thereby confirming the validity of the model used for the calculations.

4.2 Stability of rotor

When the excitation force acting on the rotor is increased compared with the damping force, an unstable self-exciting vibration occurs in the rotor. In a compressor handling high-pressure gas, the exciting force is produced mainly by gas swirl at the labyrinth seal section. The smaller the labyrinth clearance, the larger the exciting force becomes. On the other hand, a damping force is generated mainly at the bearing section.

An abradable seal is used at the labyrinth section of an injection compressor to increase aerodynamic performance. However, in order to minimize the excitation force, a swirl canceller exclusive to MHI was installed to prevent gas swirl. In addition, an MHI exclusive overhung damper was also applied in order to increase the damping force. This is formed by adding a shaft vibration damping function formed by an oil film to the rotating shaft section.



Results of calculation

Rotational speed (rpm)	Mode	Amplification factor	Rotational speed (rpm)	Mode	Amplification factor
4 650	H-1st	1.5	19 000	H-2st	4.2
5 000	V-1st	3.3	19 000	V-2st	5.1

Measured values in shop test

Rotational speed (rpm)	Mode	Amplification factor
5 100	V-1st	2.6

Fig. 4 Analysis of critical speed
The calculated critical speed match well with the actually measured values, and the amplification factor is also small.

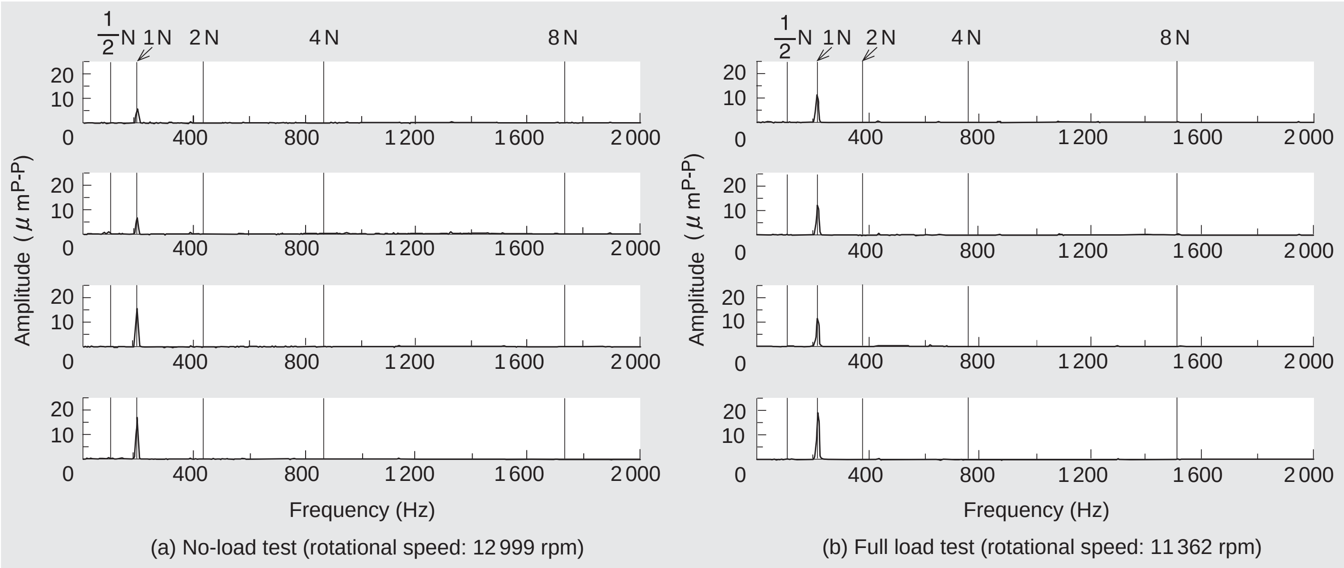


Fig. 5 Analysis of shaft vibration frequency
In both the low-pressure test and full load test, the nonsynchronous components of the operational speed are sufficiently small.

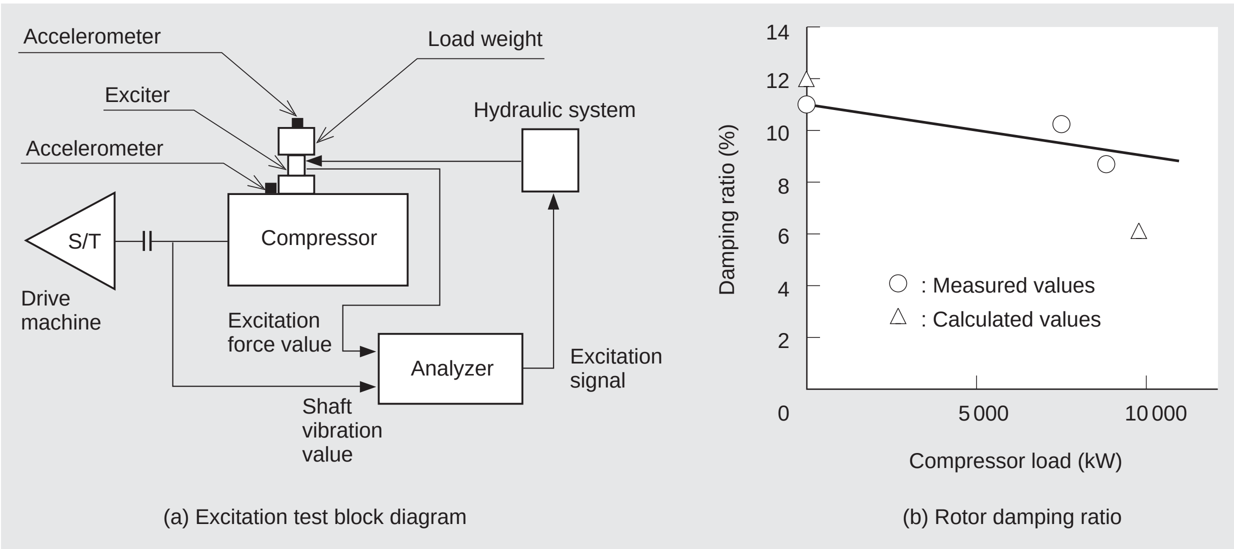


Fig. 6 Results of compressor excitation test
The excitation test was performed as part of the full load test. As a result, the damping ratio was found to be 8.7%. This indicates that the compressor is sufficiently stable.

The results of analysis of the shaft vibration frequency measured during in shop tests are shown in **Fig.5**. It could confirmed that, under full load operating conditions, any vibration of nonsynchronous components that may cause unstable vibration does not occur also as in the no-load operation.

In addition, the compressor excitation test as shown in **Fig. 6** was performed, which confirmed that the rotor has sufficient damping force and that stability has been secured.

5. Mechanical design

5.1 Impeller

The impeller is formed in such a way that deformation due to centrifugal force is optimized, the disk is joined to the cover by welding or diffusion welding, and the impeller is formed so as to be shrinkage-fitted to the shaft.

Since the natural frequency of the impeller is lowered in a high-density gas environment, the design of it was formed so as to avoid it to secure reliability.

5.2 Bearings

Shaft thrust force is reduced by the back-to-back arrangement of the impeller, and a direct lubrication system is applied to the thrust bearings and journal bearings to increase load capacity. The thrust collar is hydraulically fitted to the shaft to increase reliability and facilitate maintenance.

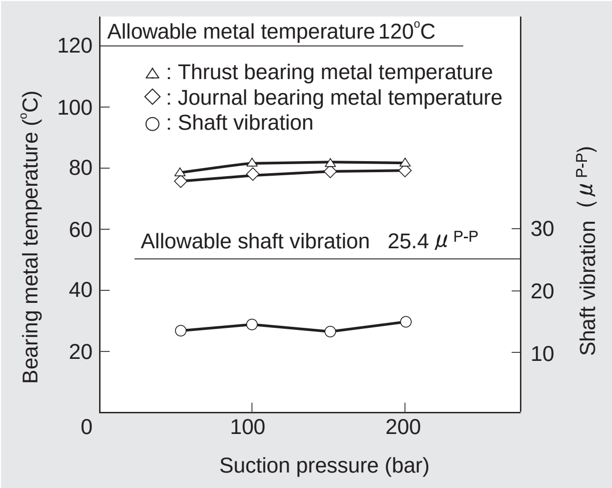


Fig. 7 Bearing metal temperature and shaft vibration amplitude
Both the metal temperature and shaft vibration amplitude are stable.

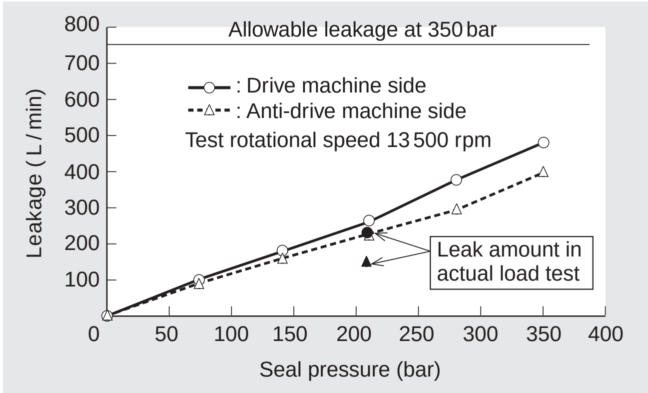


Fig. 8 Results of measurement of dry gas seal leakage

Fig. 7 shows bearing metal temperatures measured during the full load tests together with the shaft vibration amplitude. All values were sufficiently small against the allowable values. In addition, it was also confirmed that variation of these values due to variations in operating pressure are small, as well as stable under high-pressure operation.

5.3 Seals

A tandem dry gas seal with a design pressure of 350 bar was applied to the shaft seal. Compressor discharge gas is supplied as seal gas through a filter. **Fig. 8** shows that the amount of leakage from the seal during the full load tests matches well the results of the bench test. This indicates the integrity of the seal in the full load tests.

6. Conclusion

MHI has developed a new natural gas injection compressor, putting emphasis on reliability under the specific operating conditions of the unit.

Using natural gas as a test gas, full load shop tests were conducted at a discharge pressure of 450 bar which reproduces the same operating conditions as those on site. Based on the results of the tests, it was verified that the design and manufacture of the compressor were appropriate with respect to the mechanical properties including shaft vibration and aerodynamic performance, resulting in the compressor having high reliability.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