Verification Test of 700 bar Super High Pressure Robust Compressor

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It is expected that the resource and energy field, which accounts for approximately half of the compressor market, will continue to grow in the future. In this field, operating experience is important, and high robustness and reliability are required of equipment. Among such equipment, super high pressure range gas injection compressors have higher technical difficulty and therefore have many technical issues that need to be solved. We have investigated the challenges that need to be overcome and have developed super high pressure gas injection compressor (hereinafter referred to as the super high pressure compressor) with high robustness. We have also implemented verification tests that simulated all on-site operating conditions including CO_2 concentration, pressure, rotating speed and changes in gas quantity in order to verify the robustness. This report describes these technological development and verification test results.

1. Introduction

In the resource and energy field, which accounts for approximately half of the compressor market, development needs for FPSO (Floating Production Storage & Offloading) offshore gas and oil fields including the Pre-Salt area in Brazil have been increasing significantly. At these plants, super high pressure compressors have been installed for EOR (Enhanced Oil Recovery) and CCS (Carbon Capture and Storage). As development of gas and oil fields has proceeded to deeper sea areas year after year, there are an increasing number of cases where the pressure requirements for the compressor exceed 550 bar.

This super high pressure compressor has higher technical difficulty and its robustness and reliability are important. We have therefore clarified its technical issues and special requirements, developed super high pressure compressors with high robustness by investigating and solving the issues and requirements, and implemented verification tests at 700 bar in order to verify the high robustness. This report describes the high pressure compressor development concept, the specific technological developments and the verification test results.

2. Development approach and development concept

2.1 Development approach

As the first step of the development of a super high pressure compressor, we studied literature related to high pressure compressors and clarified the technical issues and latent risks. As the second step, we researched and sought solutions to the technical issues that had additional value unique to Mitsubishi Heavy Industries Compressor Corporation (MCO). **Table 1** shows the clarification results of the technical issues and their respective solutions. The development

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approach of a super high pressure compressor with high robustness was to solve all these technical issues.

The black circles in the table represent additional value (features) unique to MCO. The development proceeded in consideration of not only the creation of a super high pressure compressor, but also producing competitive and attractive products for customers through the application of additional value unique to MCO.

Table 1 Solutions to technical issues

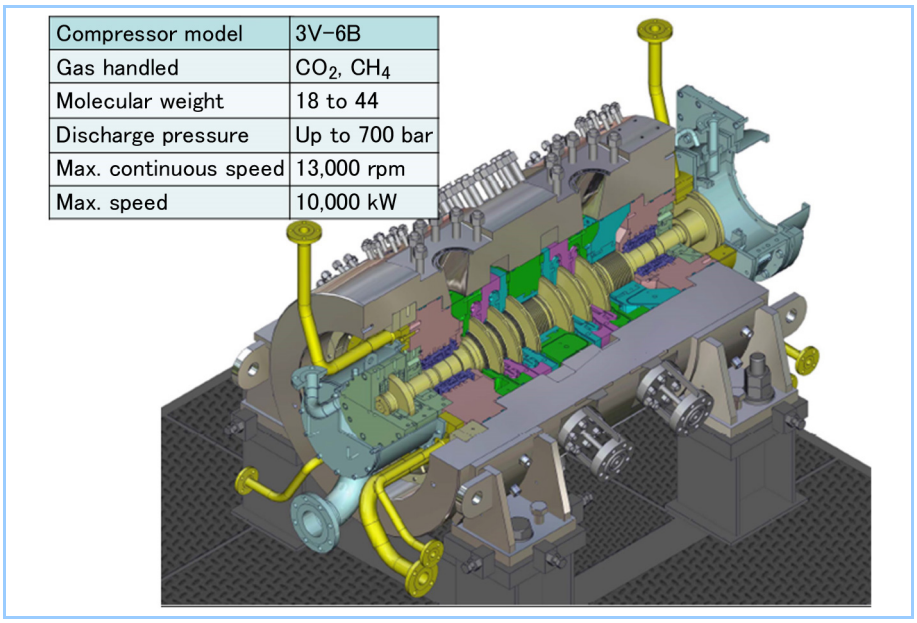
Key factor	Technical issue	Applied technology and its verification method
High pressure	− Insufficient strength of stationary parts − Deformation of stationary parts − Leakage of machine internal gas from seal	● Employment of low diffuser diameter ratio impeller stage
		○ Three-dimensional FEM analysis
		○ Review of method and material of casing seal
		○ Hydrostatic test (1.5 x casing design pressure)
		○ Gas leakage test
High density gas	− Destabilization of rotor (high destabilization force)	● Employment of grooved hole pattern seal
		○ Element test of grooved hole pattern seal
		● Employment of high boss ratio impeller stage
		○ Rotor dynamics calculation (stability/lateral)
		○ Sensitivity analysis for stability
		○ Compressor soundness check through full load test
		○ Rotor stability check through vibration test
Aerodynamic performance	− Inadequate aerodynamic performance − Decrease of operating range caused by rotating stall	○ Estimation of aerodynamic performance and rotating stall
		○ CFD analysis
		○ Single stage performance test of development matrix
		○ ASME PTC-10 Type 1 performance test
Impeller strength	− Insufficient strength of rotating object	○ FEM analysis (resonance estimation, response analysis)
		○ Element test (check of fluid compound natural frequency)
Thrust load	− High thrust force	● Employment of back-to-back impeller arrangement
		● Employment of double-balance piston
		○ Measurement of thrust force through full load test
Seal system	− Defect of gas seal body and failure of control system	○ Standardization of gas seal main dimensions (structure allowing seals from other vendors to be used)
		○ Verification test using actual seal and control system

2.2 Development concept

Figure 1 shows the specifications and a three-dimensional drawing of the developed machine. The developed machine has three impellers for each low and high pressure section in a back-to-back arrangement in order to minimize the thrust force and internal leakage. This compressor was developed so that it could deal with gases with a wide range of molecular weight, from methane-rich gas to CO_2 gas, based on the assumption of various kinds of actual operating environments.

Figure 2 shows the design concepts and enhanced technologies from the conventional design which were applied to the developed compressor. To provide a competitive and attractive compressor, the five design concepts were set as countermeasures to the technical issues, and advanced technologies were employed to attain the concepts. The concepts and the corresponding technologies are also shown below.

A) High stability design : Step hole pattern seal (high damping seal), high boss ratio impeller
B) High efficiency design : High boss ratio impeller, wide flow path impeller stage
C) Compact design : Short length diffuser
D) Wide range operation design : Vaned diffuser
E) Low thrust force design : Double-balance piston



Compressor model	3V-6B
Gas handled	CO_2, CH_4
Molecular weight	18 to 44
Discharge pressure	Up to 700 bar
Max. continuous speed	13,000 rpm
Max. speed	10,000 kW

Figure 1 Specifications and three-dimensional drawing of developed machine

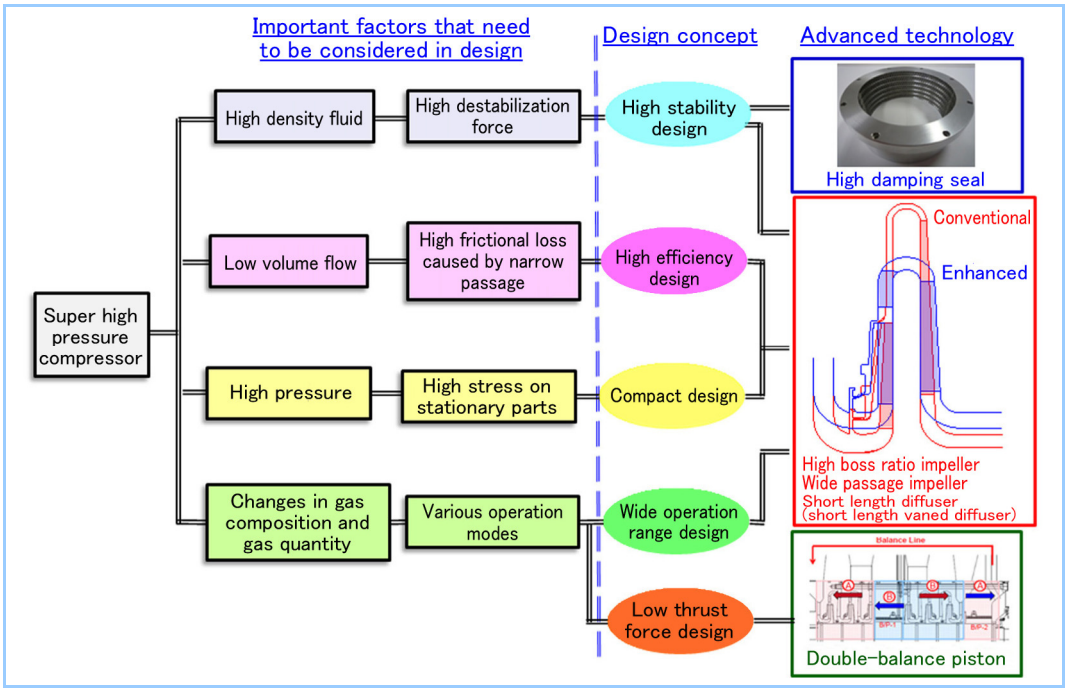


Figure 2 Design concepts and advanced technologies of developed compressor

3. Technological development

3.1 High stability design

The super high pressure compressor deals with high density gases and then generates a large destabilization force. Therefore a rotor design with high stability that enables stable and robust operation under all on-site operating conditions is essential. For the attainment of such a rotor design, the developed machine employed a high rigidity shaft and enhanced the damping of the shaft system.

3.1.1 High boss ratio impeller

Figure 3 shows the concept of high boss ratio impeller. The employment of a high boss ratio impeller intended to enlarge the shaft diameter and then improve the rotor rigidity. **Figure 4** shows the relationship between the impeller boss ratio and the rotor logarithmic decrement/the relative polytropic efficiency. The impeller boss ratio was optimized so that high efficiency can be maintained (refer to Section 3.2) while ensuring the compressor rotor logarithmic decrement of 0.2 or more (not taking into account the hole pattern seal effect). As a result, rotor rigidity approximately 30% higher than the conventional design has been attained.

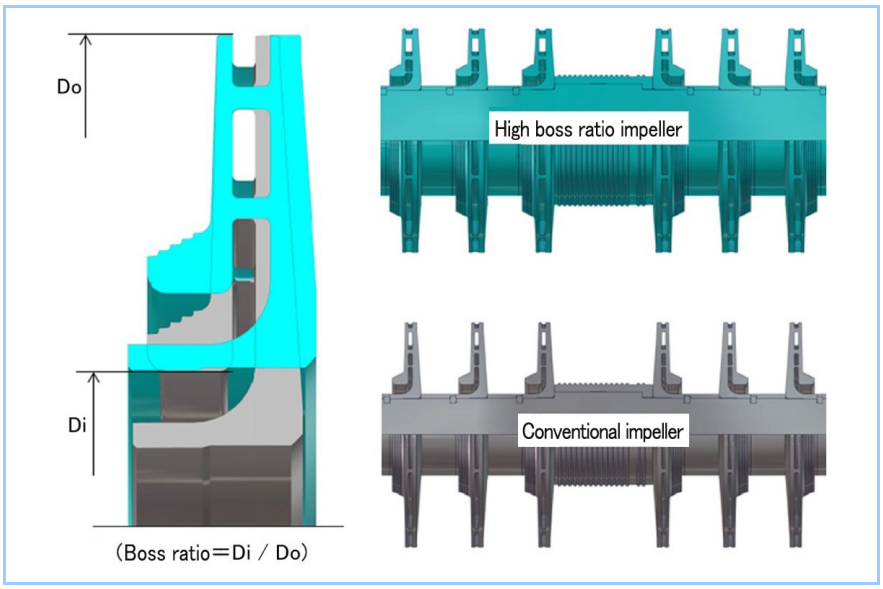


Figure 3 Concept of high boss ratio impeller

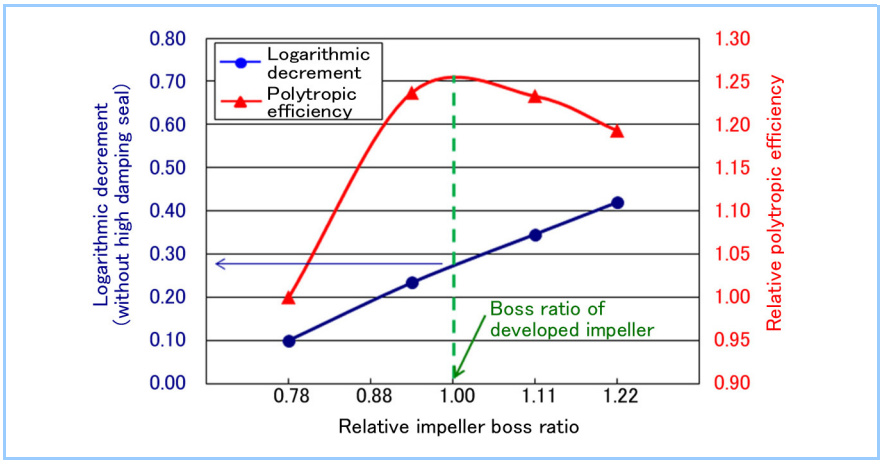


Figure 4 Relationship between impeller boss ratio and rotor logarithmic decrement/polytropic efficiency

3.1.2 Step hole pattern seal

Figure 5 shows a step hole pattern seal used for the developed super high pressure compressor. The developed machine employed a hole pattern seal in order to enhance the damping of the shaft system. In addition, the hole pattern seal used for the developed machine has a hybrid construction of a hole pattern seal and a step labyrinth seal in order to reduce the leakage rate. In the development of this seal, parameter studies for hole diameter, hole depth, etc., were performed using CFD in order to determine the optimum seal shape for the leakage and dynamic characteristics. In addition, a component test with the optimum shaped seal was performed in order to verify the accuracy of the CFD analysis.

Because the properties of the supercritical carbon dioxide in the high pressure compressor are in the region between compressive and incompressive and its density is close to water, this component test used water as the test fluid to evaluate the level of effect of seal clearance and the existence of swirl at the seal inlet on the leakage rate and the dynamic characteristics of the seal. **Figure 6** compares the component test results and the CFD analysis results, and indicates that the test results were favorably equivalent or better than the CFD analysis results for both the leakage and dynamic characteristics.

The stability under on-site operating conditions was calculated using the calculation model shown in **Figure 7**. As a result, it was verified that the rotor had high stability that could maintain sufficient logarithmic decrement under all on-site operating conditions of CO_2 concentration, pressure, rotating speed, changes in gas quantity, etc. (For the dynamic characteristics and the labyrinth exciting force of the hole pattern seal, the CFD analysis results were used.)

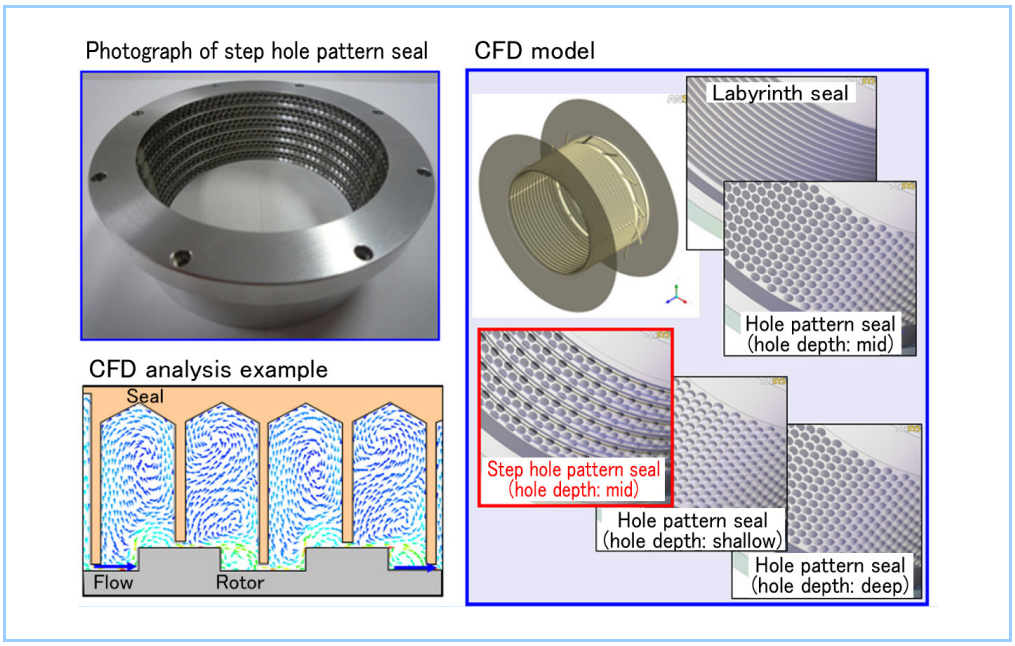


Figure 5 Photograph and CFD analysis of step hole pattern seal (high damping seal)

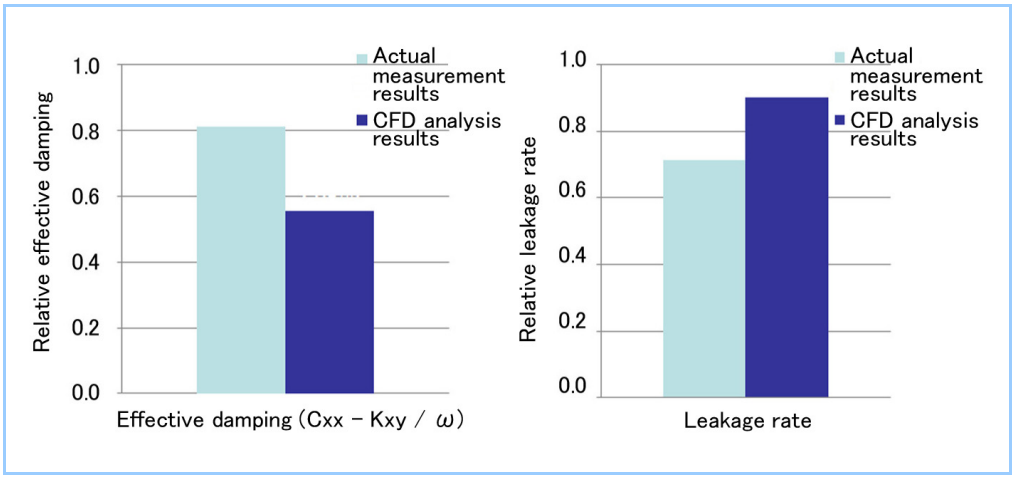


Figure 6 Comparison between results of CFD analysis and component test of step hole pattern seal (high damping seal)

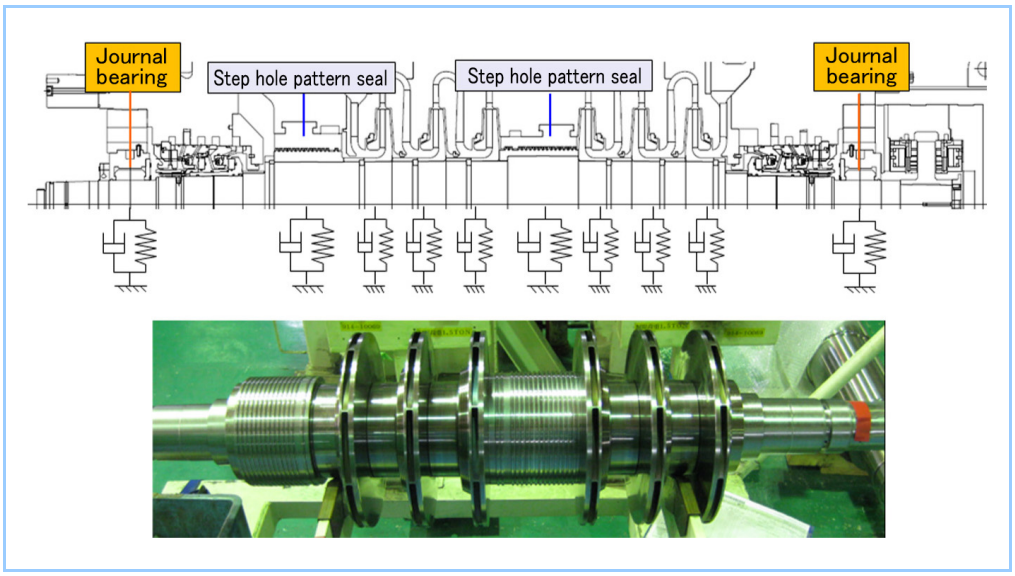


Figure 7 Stability analysis model and picture of super high pressure compressor rotor

3.2 High efficiency design

One of the factors that lower the efficiency of a high pressure compressor is the growth of frictional loss on the wall surface of the narrow passage impeller stage and the increase of labyrinth leakage caused by high pressure difference. As a countermeasure, the developed machine employed a wide flow passage impeller stage and a stepped labyrinth. **Figure 8** compares the flow coefficient and the impeller efficiency and indicates that the efficiency improves when the operating point is shifted toward the higher flow coefficient side. An effective way to increase the flow coefficient is to increase the rotational speed, but this also leads to the destabilization of the rotor. However, by increasing the impeller boss ratio by 30% and employing a hole pattern seal for the further improvement of stability, the developed machine ensures sufficient stability and allows for high rotation speed operation. Typically, the increase of the boss ratio causes the deterioration of efficiency as a result of an increase in Mach number at the impeller inlet. However, by shifting the operating point toward the higher flow coefficient side and through the employment of a wide flow passage impeller stage and a stepped labyrinth, we have developed an impeller with an increased efficiency of 10% or more in comparison with conventional machines (**Figure 9**). The developed impeller can change the commonly held belief that increasing the boss ratio causes the deterioration of efficiency.

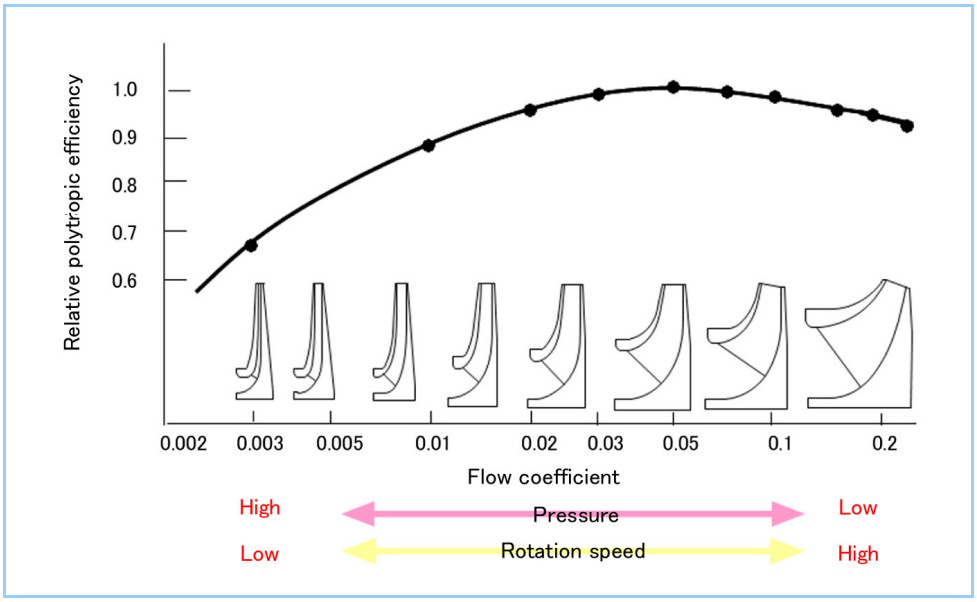


Figure 8 Relative comparison of flow coefficient and impeller efficiency

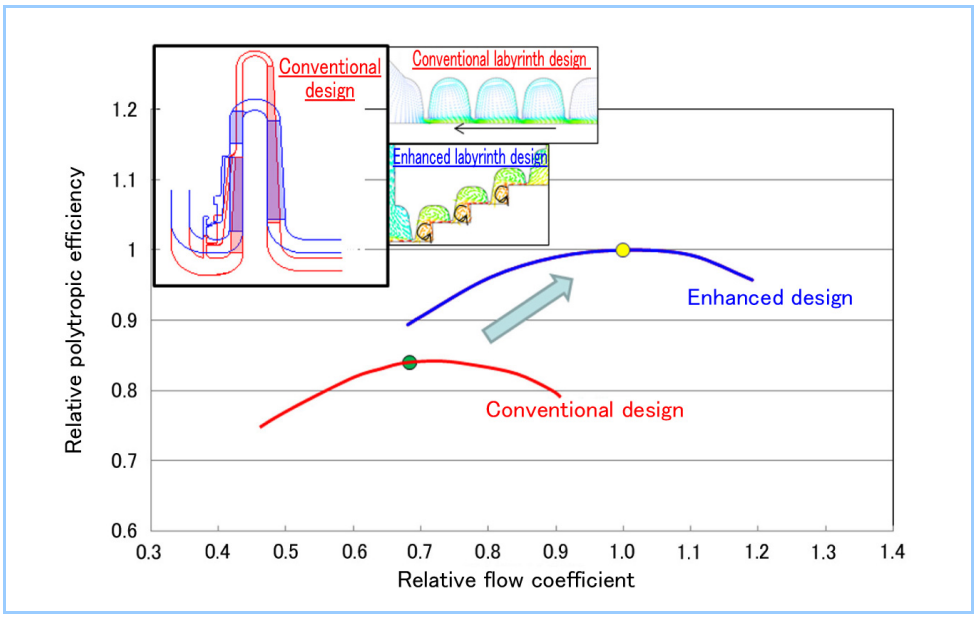


Figure 9 Comparison of polytropic efficiency between conventional and improved structures

3.3 Compact design

For offshore plants including FPSO plants, one important factor is the reduction of the weight and footprint of the compressor. To attain this, the employment of an impeller stage with a short length diffuser was considered. However, the decrease of the diffuser length led to the deterioration of the efficiency of the compressor and a decrease in the operating range in terms of static pressure recovery. As a countermeasure, the developed machine employed a vaned diffuser to allow the impeller stage to enable sufficient static pressure recovery even with a short length diffuser and to ensure high efficiency. **Figure 10** compares the cross-sectional views of the conventional design and the developed machine, and indicates that the employment of an impeller stage with a short length diffuser has successfully enabled a decrease in the diameter of the compressor by 30% or more and a reduction in the weight of the whole machine by half (50%) in comparison with the conventional design.

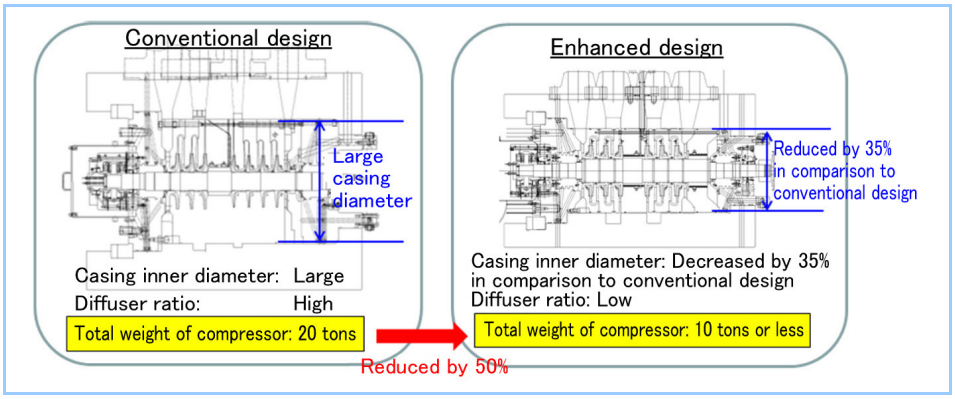


Figure 10 Comparison of weight between conventional and developed machines

3.4 Wide operating range design

One of the operating range restricting factors of a high pressure compressor is the rotating stall. As described in Section 3.2, the developed machine employed a wide flow passage impeller stage to improve efficiency, but it led to a decrease in the impeller outlet angle and the promotion of rotating stall. As a countermeasure, the developed machine employed a vaned diffuser that can shift the originating point of rotating stall to a lower flow rate and can achieve a wider operating range than the conventional design. **Figure 11** shows the results of a single stage performance test of the developed impeller stage (impeller stage with high boss ratio and short length vaned diffuser). These results indicate that an impeller stage that has a peak efficiency of 70% or more and a stability range of 35% or more has been successfully developed.

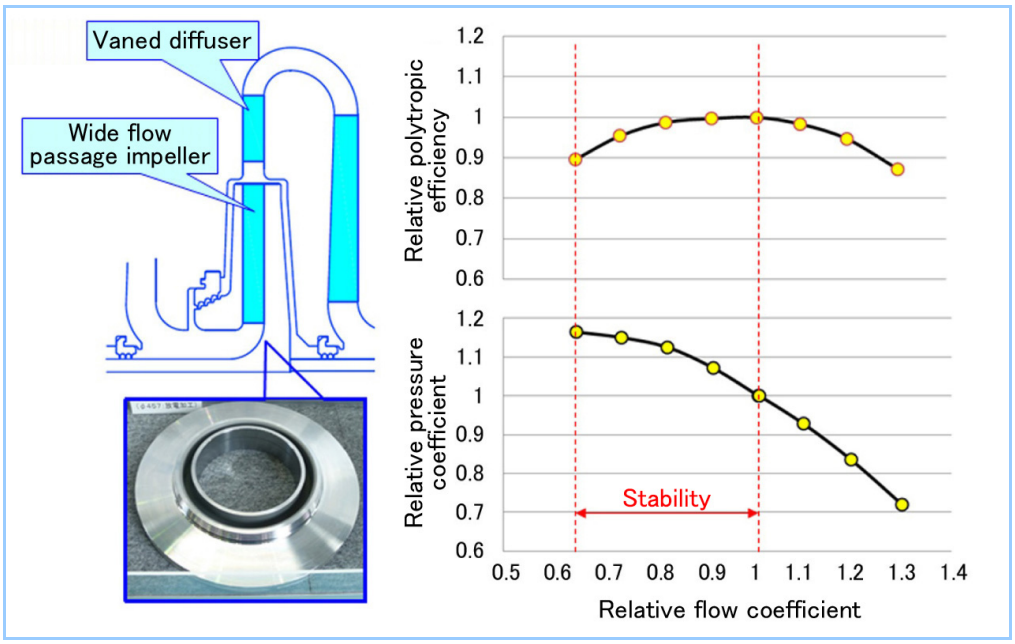


Figure 11 Single stage performance test results of impeller stage with high boss ratio and short length vaned diffuser

3.5 Low thrust force design

A super high pressure compressor is required to flexibly deal with changes in gas properties and gas generating quantity at a well site. Because of high pressure, even a slight change in operating conditions may cause unbalanced thrust force and make continuous operation difficult. However, the employment of a double-balance piston structure instead of the conventional single-balance piston structure (**Figure 12**) has allowed thrust balancing in each of the low and high pressure sections and minimized the unbalance of thrust force caused by changes in gas properties and gas generating quantity (**Figure 13**). The developed super high pressure compressor is equipped with this double-balance piston structure and also attains high robustness against changes in thrust force.

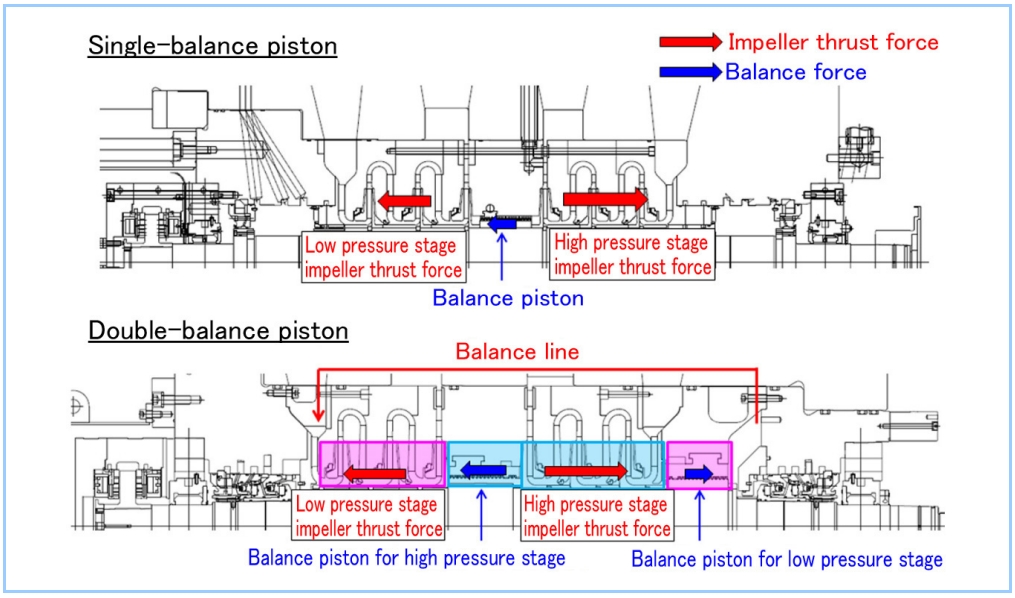


Figure 12 Comparison of single-balance piston structure and double-balance piston structure

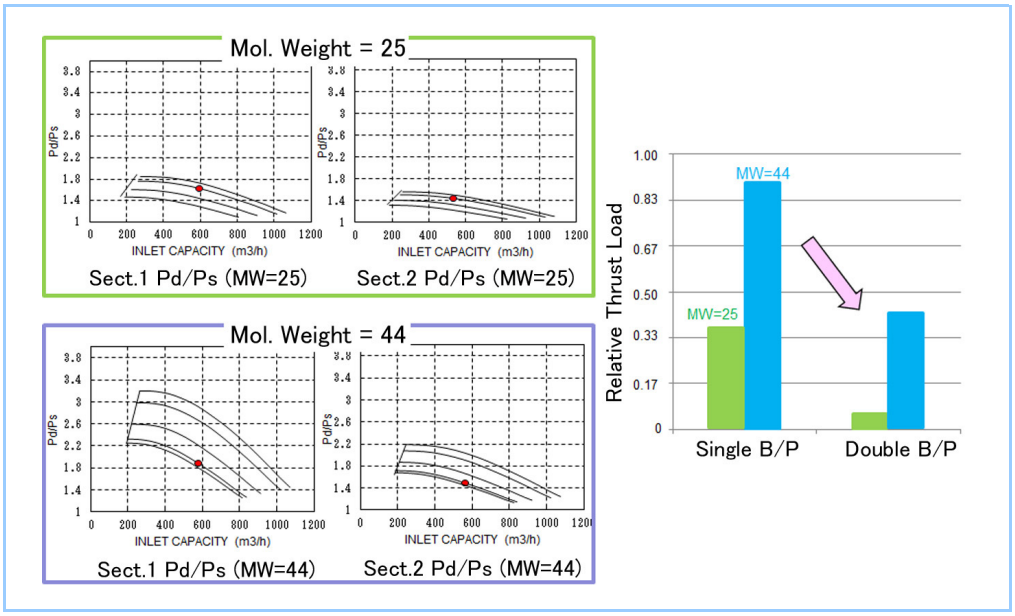


Figure 13 Comparison of thrust load between various molecular weights

4. Verification test at 700 bar

An actual scale super high pressure compressor equipped with the technologies described above was manufactured and a verification test that simulates the on-site operating conditions was implemented. In this verification test, all possible on-site operating conditions were simulated using high pressure town gas and high pressure CO_2 gas, as well as changing the CO_2 concentration, pressure, rotation speed and gas quantity. We then checked the soundness of the compressor and

verified its high robustness (**Table 2**). In addition to measuring the performance and shaft vibration, the measurement of pressure fluctuation to determine the occurrence of rotating stall, load cell measurement to check thrust force and the measurement of hydraulically excited casing to check rotor stability were performed.

Table 2 Verification test items

	Test-1	Test-2	Test-3	Test-4	Test-5	Test-6	Test-7	Test-8
Test Item	No Load M/T	Type.2 P/T (MW=28)	Type.1 P/T (MW=23)	FL/FS/FP (MW=23)	Type.1 P/T (MW=31)	FL/FS/FP (MW=31)	High Dens. (MW=44)	Overhaul Inspection
Exp. Pressure (barG)	Vacuum	340	550	680	550	680	650	—
Exp. Power (kW)	0	1,500	7,000	8,800	8,000	9,800	9,800	—
Exp. Speed (rpm)	12,500	7,400	11,900	12,500	10,910	11,800	9,600	—
Gas Property	Vacuum	N_2	City Gas + CO_2	City Gas + CO_2	City Gas + CO_2	City Gas + CO_2	CO_2	—

Figure 14 shows the results of ASME PTC-10 Type 1 performance testing that simulated on-site operating conditions. The total efficiency of the compressor including the internal leakage was approximately 70% and a stability range of 30% or more was ensured.

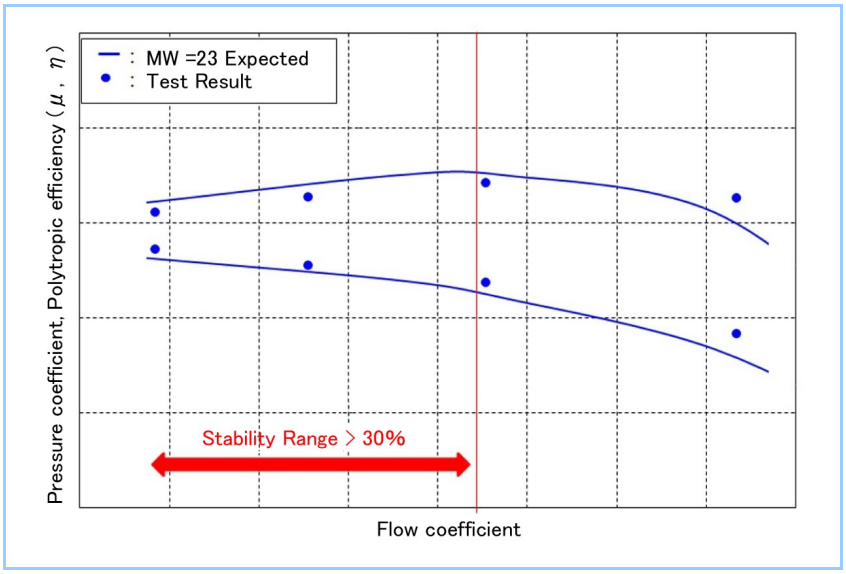


Figure 14 ASME PTC-10 Type 1 test results

Figure 15 shows the trend of shaft vibration and bearing metal temperature in the full load test under conditions with a molecular weight of 23. These test results prove that very stable operation was achieved with the shaft vibration level of 5 μm P-P or less and the metal temperature of 90 °C or less (the allowable limit was 120 °C) under all operating conditions.

During the verification test, a rotor stability test using the casing excitation method shown in **Figure 16** was conducted to measure rotor logarithmic decrement and verify the validity of the logarithmic damping calculated by the stability analysis. **Figure 17** shows the results of logarithmic decrement measurement in the no-load running test and the full load test. The logarithmic decrement in the no-load running test was 0.4, while the vibration response was very small and the logarithmic decrement increased to 2.5 in the full load test. The measured value was close to the calculation results and this indicates that the hole pattern seal was effective and the stability calculation method was reasonable.

Figure 18 shows the measurement results of thrust load in several load tests. The measured thrust load was 10 bar or less under all operating conditions from an overload condition to a surge range and well below the allowable limit.

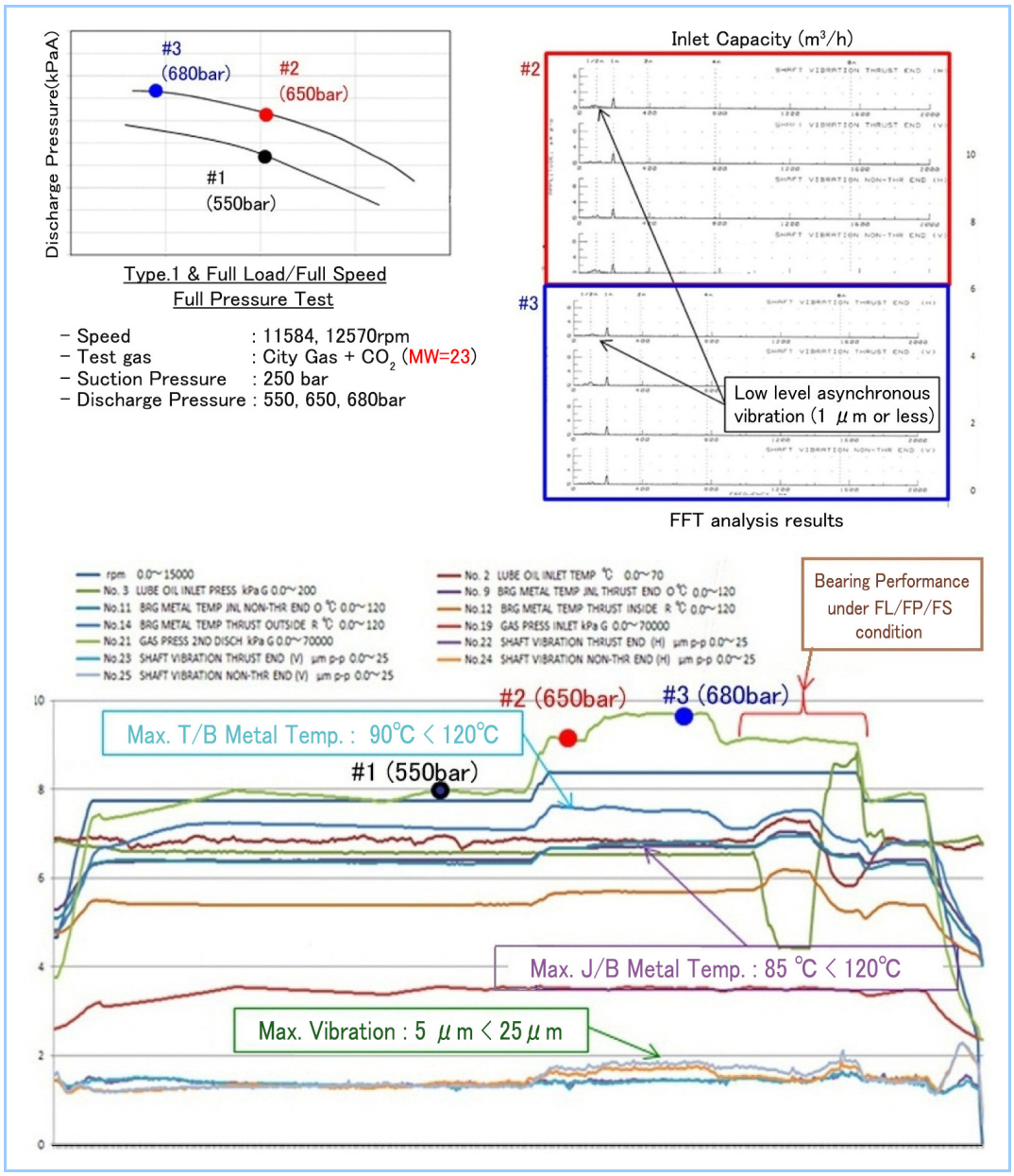


Figure 15 Mechanical data trend in actual load test

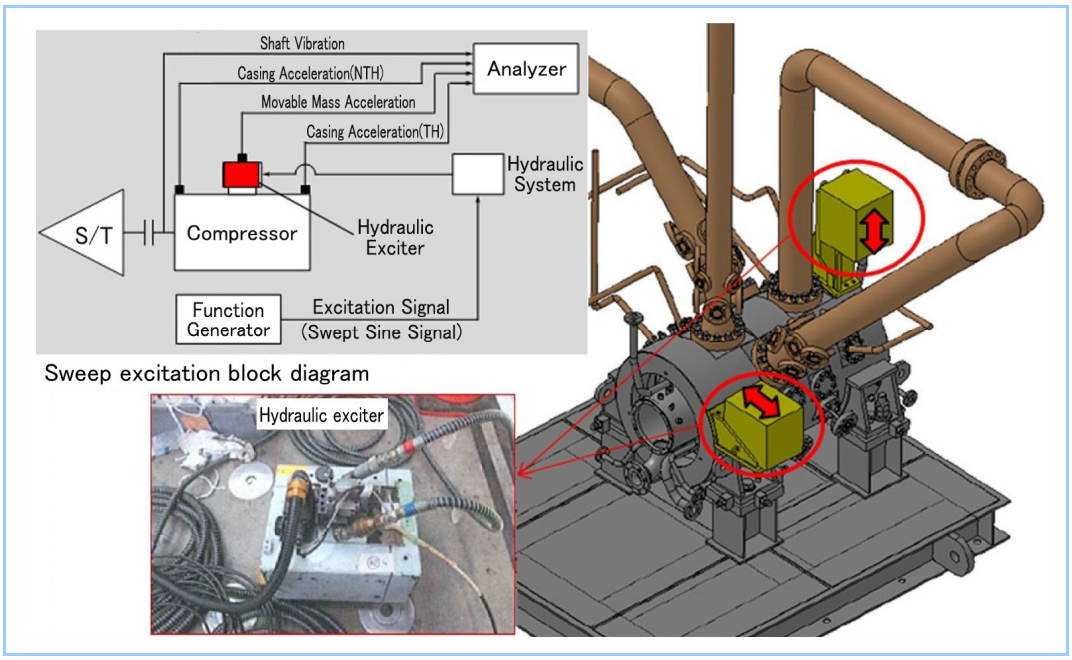


Figure 16 Casing excitation test method

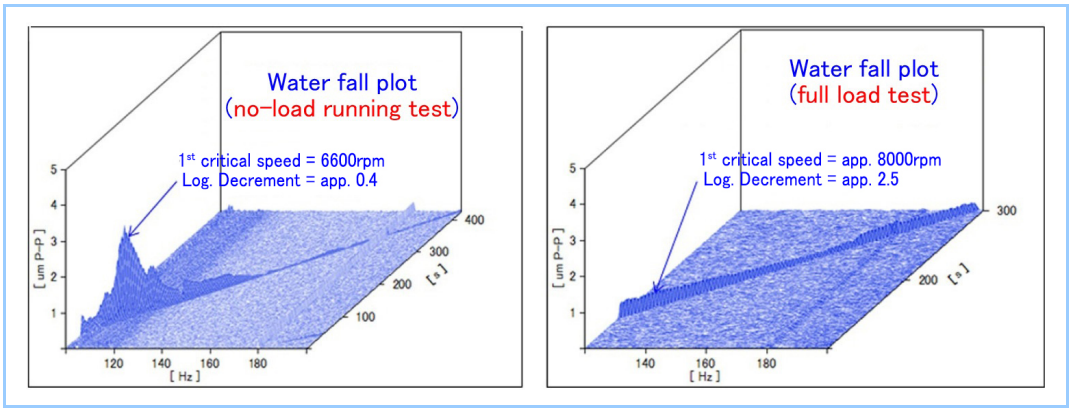


Figure 17 Measurement results of logarithmic decrement in casing excitation test

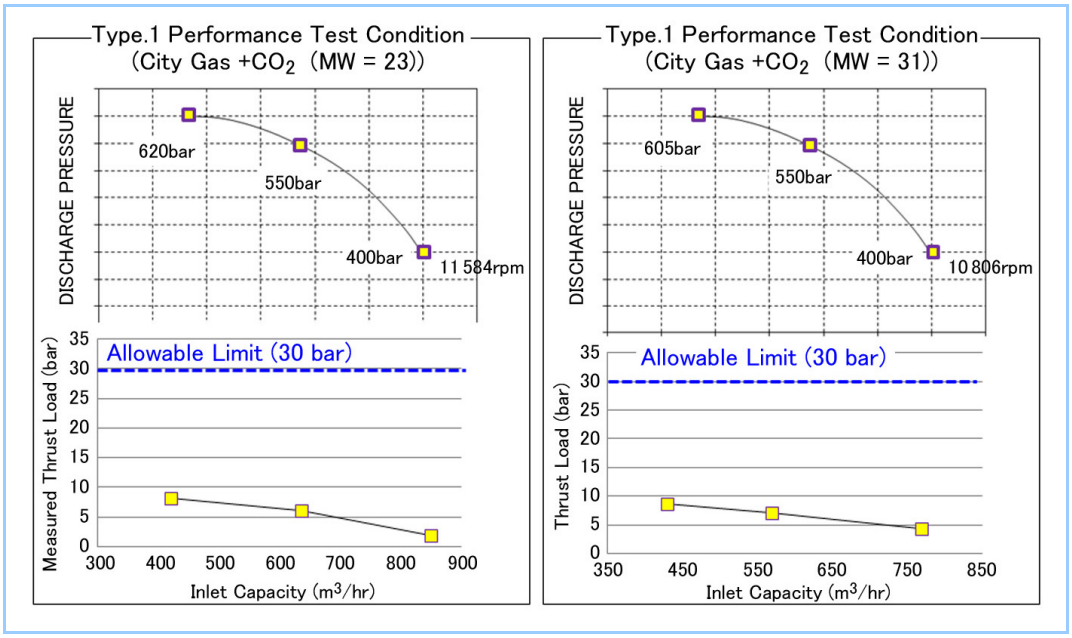


Figure 18 Measurement results of thrust load in full load test

5. Conclusion

MCO developed and manufactured a super high pressure compressor for use in EOR and CCS, and performed extensive verification testing at pressures up to 700 bar. As a result, the developed compressor can achieve highly reliable and highly robust operation successfully under all operating conditions from a methane rich gas condition to a 100% CO_2 gas condition, while having the following features:

(1) High stability design: Shaft vibration of 5 μm P-P or less, Logarithmic decrement of 2.5 or greater
(2) High efficiency design: Approximately 70%
(3) Compact design: Reduced by 50% in comparison with the conventional design
(4) Wide range operation design: Turndown range of 30% or greater
(5) Low thrust force design: 10 bar or less

We conducted the verification test in the presence of our customer to dispel all concerns about technical issues and successfully received an order for a gas injection compressor with a discharge pressure of 550 bar from Petrobras/ MODEC.

We are willing to continue to improve efficiency and reliability further in order to offer compressors with high robustness to customers.