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Development of High-Pressure Ratio and Wide-Operating Range 700bar Compressor

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ABSTRACT

Pre-Salt development projects continue to expand offshore Brazil into ever deeper waters year after year. Consequently, the required Discharge Pressures for Re-Injection Compressors aboard FPSO's are likely to exceed current 550 bar levels, and requirements for design of higher pressure compressors becomes more severe.

This paper presents development of super high pressure compressor and the results of verification tests. Super high pressure compressor can be used as re-injection compressor or injection compressor such as in the CCS plant and FPSO.

Rotor stability is the most considerable requirement for super high pressure operation because of the high density flow destabilization force in the compressor. From the point of view of rotor dynamics, the short bearing span and high shaft diameter are desirable for the stability of the rotor. On the other hand, from the point of view of aerodynamics performance, that rotor arrangement may decrease the performance. Some parametric study was conducted to optimize the boss ratio of compressor stage to get both high rotor dynamic stability and high aerodynamic performance.

Rotating stall can be the cause of sub-synchronous vibrations in super high pressure operation condition and it induces narrow operation range. To prevent such situation, vaned diffuser was applied for developed compressor stage and it can also help to reduce the diffuser radius space.

Seal leakage flow have large effects on the rotor stability and aerodynamic performance. For this item, the hybrid construction of hole pattern seal and step labyrinth seal was proposed. Hole pattern seal is well known as damping seal for high pressure compressor. CFD dynamic analysis and dynamic characteristics test for this hybrid hole pattern seal are introduced under the condition of high density by using hydraulic shaker device.

The extensive verification test of developed super high pressure compressor were conducted with impeller of optimized boss ratio, vaned diffuser and hybrid honeycomb seal, and has demonstrated the robustness for all field operation

condition including Molecular weight variation with high aerodynamic performance and small vibration.

INTRODUCTION

Pre-Salt development projects continue to expand offshore Brazil into ever deeper waters year after year. Consequently, the required Discharge Pressures for Re-Injection Compressors aboard FPSO's are likely to exceed current 550 bar levels.

The prerequisite criteria in the design of these higher pressure compressors for hostile marine conditions are invariably:

1. Reliability
2. Operability
3. Compact Design

In order to realize above requirement, OEM developed a High Pressure Ratio 700 bar Compressor with a wide range operational capability and incorporating the following features:

1. High stability and High reliability for volatile well conditions (Log decrement : more than 2.5).
2. Wide Operating Range for volatile well conditions (2 times wider than conventional types)
3. High Efficiency resulting in real energy savings (Polytropic Efficiency : app. 70%)
4. Compact design & reduced weight optimized for fixed and floating offshore facilities (50% less than conventional type)

This paper introduces more specifics about the applicable technologies, with details about the testing and operation of the developed super high pressure compressor.

LOAD MAP OF SUPER HIGH PRESSURE COMPRESSOR

Fig.1 illustrates and tracks the development of the OEM's high pressure in-line centrifugal compressor. The 1st generation design commenced with the manufacturing and delivery of a synthesis gas compressor for an ammonia plant in 1986. The 2nd generation was marked by the production and delivery of a CO_2 compressor for a Urea plant in 1994. This CO_2 compressor was equipped with a number of advanced design feature including 3-D high efficiency impellers, overhung dampers, swirl cancellers, etc.

The 3rd generation kicked off in 2000, with a phased program to develop a compressor ultimately targeted to achieve a discharge pressure rating of 700 bar. Shortly thereafter, in 2003, phase 1 verification shop testing was carried out on a casing at 450 bar. At that point in time, the highest needs in the global injection market were expected to be in the range of 450 bar. For this 3rd generation compressor, relatively new technology, such as high accuracy CFD, anti-rotating stall, firm nozzle design, and impeller brazing were applied.

In 2010, the 4th generation program was launched to develop the "super high pressure compressor" with a target pressure rating of 1000 bar. This 4th generation compressor is enhanced with proven high pressure compressor design features, such as, high boss ratio impellers, high damping seals and wide flow stages.

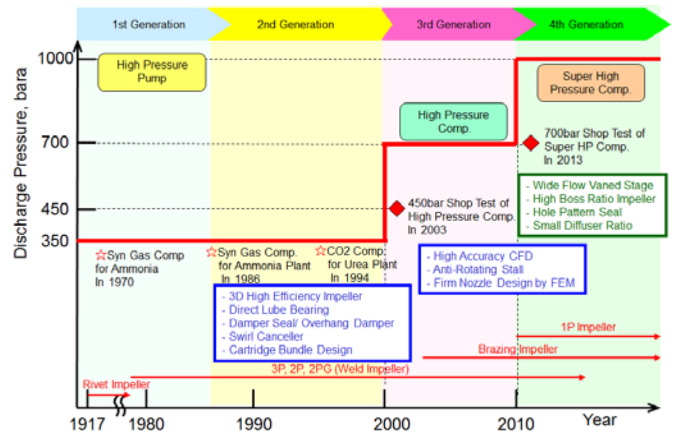

Figure 1. High Pressure Compressor Load Map

Fig.2 depicts the application range of the high pressure discharge compressor. The conventional high pressure compressor covers the blue region, which is a typical range for high pressure injection service. The super high pressure compressor covers the higher pressure rating in the red zone, as well as the high pressure rating in the blue zone.

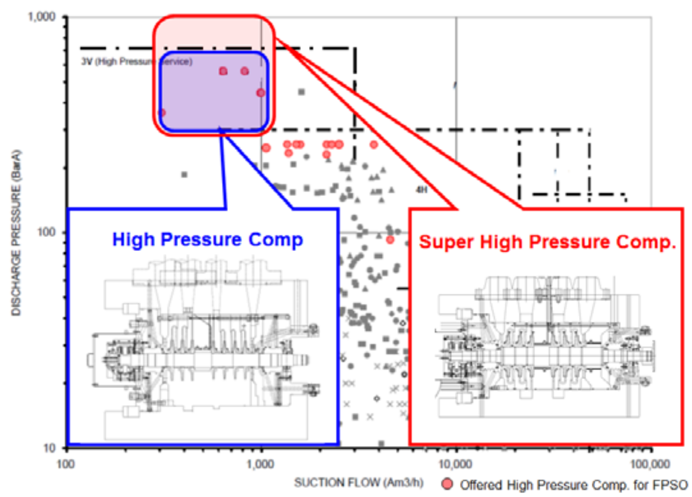

Figure 2. Application Range of High Pressure Compressor

DEVELOPMENT APPROACH OF HIGH PRESSURE COMPRESSOR

For the high pressure compressor, there are several papers about the evaluation of any potential risk which shall be considered in the design. Table 1 shows the brief summary of principal potential risks.

As the initial study of development of super high pressure compressor, OEM considered the prevention and mitigation program with value-added concept against each potential risks as shown in Table 2 to achieve the development objective. Circle shows the design features which were planned to apply as enhanced technology from conventional design. The detail explanation of enhanced technology and test results are shown in the subsequent paragraph.

Table 1. Potential Risks of High Pressure Compressor

Key factor	Potential Risk
High Pressure	Insufficient strength of static parts
	Deformation of static parts
	Leakage from sealing area
High Density Gas	Rotor instability
	Inadequate aerodynamic performance
	Appearance of rotating stall
	Insufficient strength of rotating parts
Thrust Load	Overloaded thrust load
Sealing System	Dry gas seal failure
	Dry gas seal system failure
Variation of Gas Well	Rotor instability
	Thrust load change
	Insufficient stability range

Table 2. Prevention & Mitigation Program against Potential Risk

Potential Risk	Prevention/Mitigation Program
-Insufficient strength of static parts	●Applying short length diffuser(compact design)
	Full 3D FEA analysis
-Deformation of static	Casing seal arrangement and material study
-Leakage from sealing	Hydrostatic test (1.5 x casing design pressure)
	Gas leakage Test
-Rotor instability	●Applying high damping & low leakage seal
	Component verification test of developed seal
	●Applying high boss ratio impeller design
	Conduct stability and lateral analysis
	Conduct sensitivity analysis
	Conduct full size, full pressure, full load test
	Conduct excitation test during full load test
-Inadequate aerodynamic performance	Aero. performance and stall assessment
	CFD analysis
-Appearance of rotating stall	Single stage performance test
	Conduct type.1 performance test
-Insufficient strength of rotating parts	Conduct FEA analysis (resonance analysis)
	Component verification test (resonance point)
-Overloaded Thrust Load	●Applying back-to-back arrangement
	●Applying double balance piston
	Conduct full load test
-Dry Gas Seal Failure	Standardization of dimension of dry gas seal
	Conduct full load test with seal system

DESIGN CONCEPT AND SPECIFICATION OF SUPER HIGH PRESSURE COMPRESSOR

The design concept of developed super high pressure compressor can be applied up to a pressure rating of 1000 bar. The super high pressure compressor was designed for 550 bar reinjection service which is currently the range with the highest demand globally, particularly offshore Brazil. Load testing was performed at app. 700 bar discharge which is the maximum pressure of the 550 bar casing design.

The specification details of the 700 bar casing are shown in Table 3.

Table 3. Specification of 700 bar test machine

Compressor model	3V-6B
Gas handled	CO_2+CH_4
Mol Weight	18~44
Discharge Pressure	550 - 700 bar
Pressure Ratio	4.3
Design Speed (MCR)	13,000 rpm
Required Power	10,000 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, and variations in shaft thrust force due to variations in operating conditions could also be reduced. A 3-dimensional view is rendered in Fig.3.

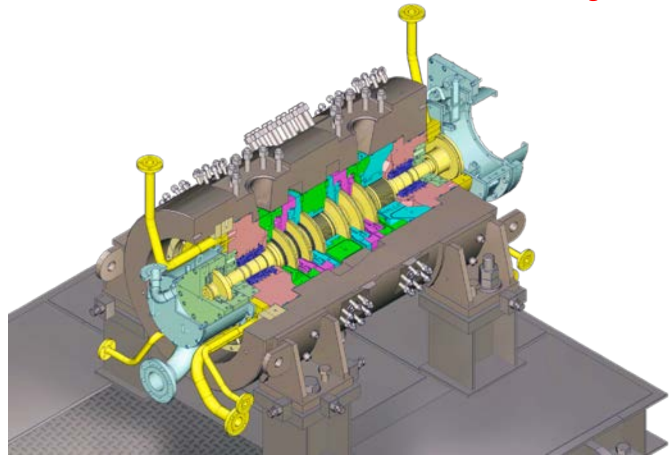

Figure 3. 3-Dimensional View of 700bar Compressor

The design of the super high pressure compressor features enhancements from the 3rd generation reinjection compressor as illustrated in Fig.4, and takes into consideration the followings to the design:

A) High Gas Density => High Destabilizing Force
B) Low Actual Volume Flow => Narrow Flow Path
C) High Pressure => High Stress
D) Variation of Gas well => Wide Range Operation

For item (A), high boss ratio impeller and hole pattern seals were applied. The dynamic characteristics of hole pattern seals were evaluated by CFD analysis and component testing.

For the item B), wide flow path impellers, high boss ratio impellers and vaned diffusers were applied. In order to verify impeller stage performance, single stage performance testing was conducted in keeping with manufacturer's own standard manufacturing practice.

For the item C), the compact design concept was applied and the strength of each part was evaluated by FEM analysis.

For item D), the double balance piston was applied to reduce a variation in the total thrust force acting on the shaft due to variation in gas well conditions. In addition to that, vaned diffuser can also help to avoid rotating stall.

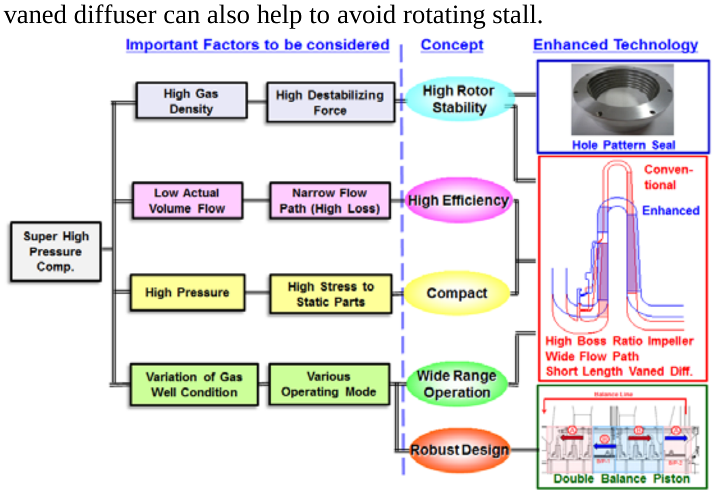

Figure 4. Design Concept of Super High Pressure Comp.

ENHANCED TECHNOLOGY

A. High Stability Design

As stable and robust rotor behavior throughout the operating range under high destabilizing force conditions is essential for reinjection compressors, the following two key advanced technologies were applied:

A-1.High Boss Impeller

Fig.5 charts comparison of rotor rigidity between original and enhanced rotor design for high pressure application (700bar discharge pressure, Mol. Weight=23 Condition). The horizontal axis shows the maximum continuous speed. And the vertical axis shows the shaft diameter divided by the square root of the bearing span. When the plot is above this line, the rotor becomes stable.

The light blue triangle shows the rotor rigidity of original rotor design, and the red circle shows the rotor rigidity of the enhanced rotor design. As shown in this graph, the rotor rigidity of the enhanced design to be higher than the original design, and it can also allow to increase the operating speed higher by applying high boss ratio impeller.

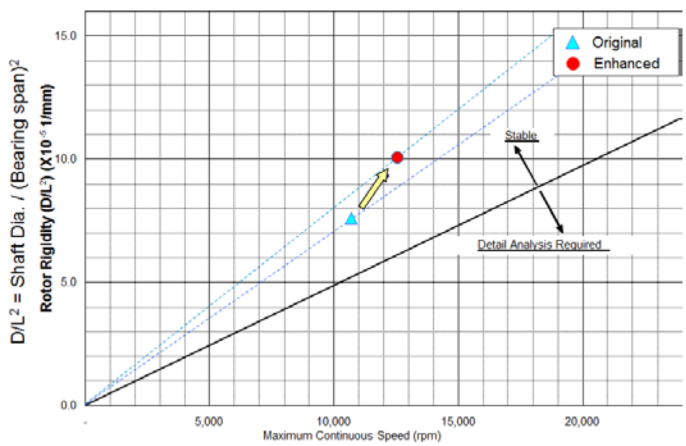

Figure 5. The Rotor Rigidity Map(Pd=700bar, MW=23)

Fig.6 shows the concept of high boss ratio impeller, and Fig.7 plots the relation between impeller boss ratio and log decrement & efficiency. The boss ratio was formulated to keep the log decrement more than 0.2 without hole pattern seals, and to keep the polytropic efficiency as high as possible.

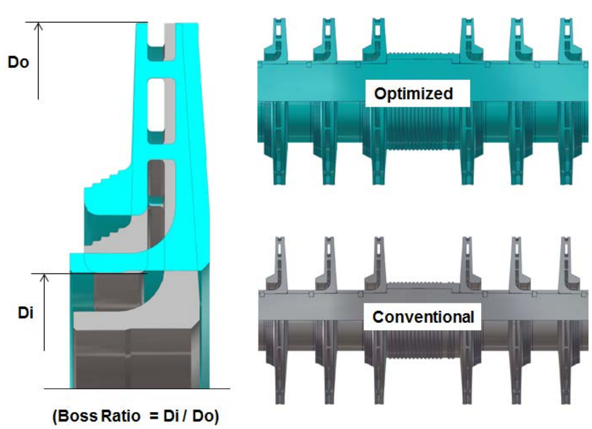

Figure 6. Concept of High Boss Ratio Impeller

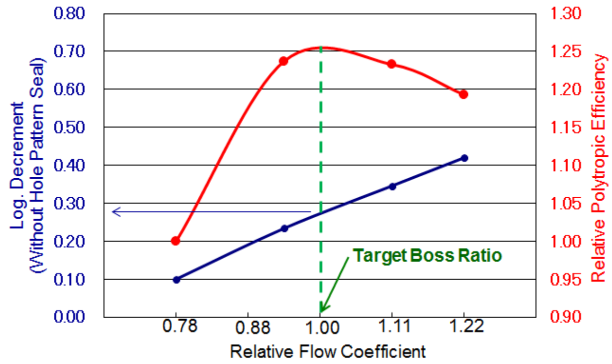

Figure 7. Relation between Impeller Boss Ratio and Log Decrement & Efficiency

A-2. Hole Pattern Seals

For the super high pressure application, the rotor dynamic stability is more important than other low pressure application in terms of high density gas and unstable excitation fluid dynamic force. The high rigidity of rotor is one of design requirement to have sufficient log. decrement as A-1. As the additional stability enhancement, the hole pattern seal on two balance pistons which has hybrid construction of hole pattern seal and step labyrinth seal were applied. The development approach of this type of seals is shown in Fig.8.

Verify the CFD analysis accuracy by internal component test data

⇩

Parameter study to find suitable configuration by CFD analysis

⇩

Decide the seal configuration

⇩

Evaluate static and dynamic characteristics and leakage rate of developed seal by component test in high density fluid

Figure 8. Hole Pattern Seal Development Approach

As first step, the accuracy of transient CFD analysis was verified by internal component test data of hole pattern seal in air condition. In the next step, the dynamic and static characteristics of several seals in the air condition were studied by CFD analysis to find suitable configuration from damping and internal leakage point of view. Through the parameter study of several seal, the most suitable configuration which has hybrid construction of hole pattern seal and step labyrinth seal was picked out as shown in Fig.9. Finally, before applying the hybrid hole pattern seal for actual compressor, a component test of hybrid hole pattern seal was conducted in high density fluid, and evaluated the static and dynamic characteristics of the seal, and verified CFD analysis accuracy.

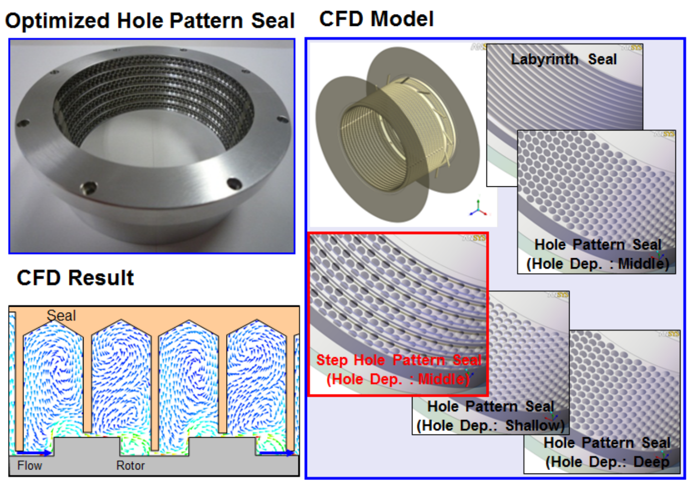

Figure 9. Hybrid Construction of hole pattern seal and step labyrinth seal

The seal component test parameters are shown in Table 4 and the test facility is illustrated in Fig.10. The dynamic characteristics of seals were measured and evaluated in accordance with the procedure described in detail in the references - [6] and [7]. The test apparatus utilizes water as the test fluid, because the fluid property of supercritical carbon dioxide in the high pressure compressor is between compressive and incompressive, and density is close to water.

The test stator, in which the developed hole pattern seal is set, is supported between the rotor bearing pedestals by two hydraulic shakers. The shakers are used to excite the stator with a random waveform in order to obtain the dynamic characteristics of seals. And effect of seal eccentricity, inlet swirl, rotating speed on the hole pattern seal were evaluated.

Fig.11 shows the component test result that was measured the seal dynamic characteristics and leakage rate through the seal. As shown in this graph, the effective damping measured during test was higher than the CFD analysis result which was estimated as incompressive fluid, and the leakage rate was lower than the CFD analysis result. That means the stability analysis result and impact on compressor performance based on CFD analysis represents a conservative approach and safe-side evaluation.

Table 4.Test parameter for hole pattern seal

Test Parameter		
(1)	Test Fluid	Water (close density to super critical CO_2)
(2)	Differential Pressure	10~ 20 bar
(3)	Rotor Speed	0 ~ 12,000rpm
(4)	Eccentricity Ratio	0% (optimum), 25%, 50%
(5)	Inlet swirl	without or with swirl (forward, backward)

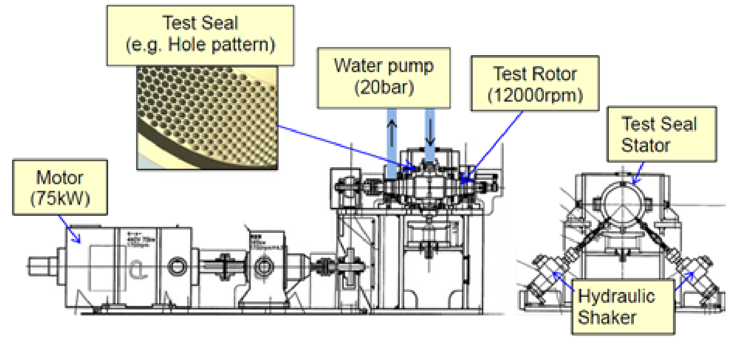

Figure 10. Hole Pattern Seal Component Test Facility

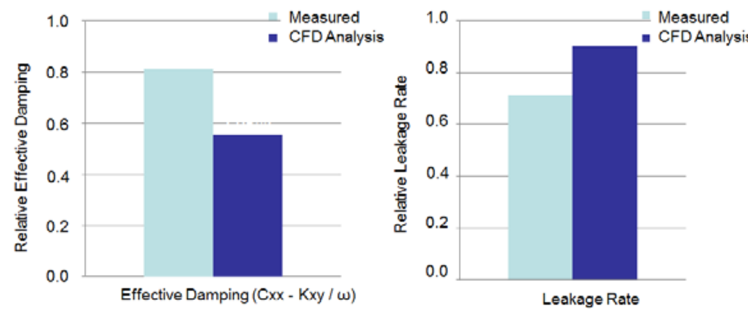

Figure 11. Comparison between Component Test Result and CFD Analysis

The analyzed characteristics by CFD which was estimated as compressive gas were implemented to the rotor dynamics design calculation. The analytical methods are based on those described by API 684 and 617. The tools used for this analysis include transient CFD, TAM XLTRC2 and OEM's in-house programs.

Fig.12 shows the calculation model and rotor assembly of 700bar compressor. For sensitivity analysis, a number of case studies were carried out in order to verify the design against operational robustness for stability, such as sensitivity to division wall/hole pattern seal deformation (divergence or convergence), sensitivity to wear and fouling on the hole pattern seal, and impact of impeller eye cross coupling.

Based on the analysis, a log decrement of 1.0 to 3.0 can be achieved. Also, even without the hole pattern seal, the log decrement can be kept greater than 0.2 by applying high boss ratio impellers.

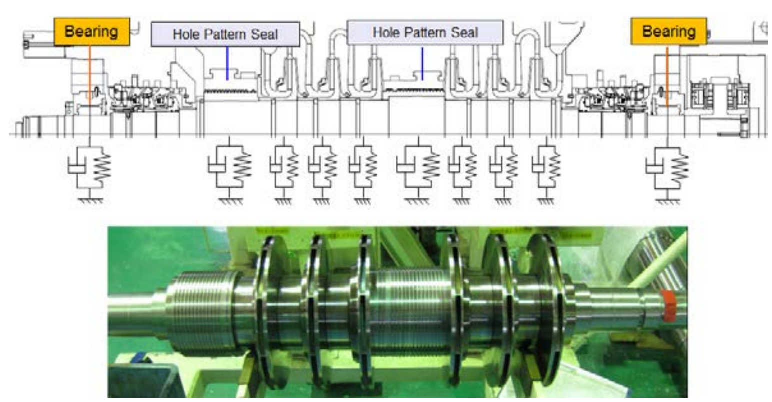

Figure 12. Calculation Model and Rotor of 700bar Compressor

B. High Efficiency Design

Generally speaking, from the performance point of view, when pressure increases, the impeller flow coefficient decreases, in turn, efficiency decreases, as shown in Fig.13. And impeller flow coefficient is given by the equation- (1).

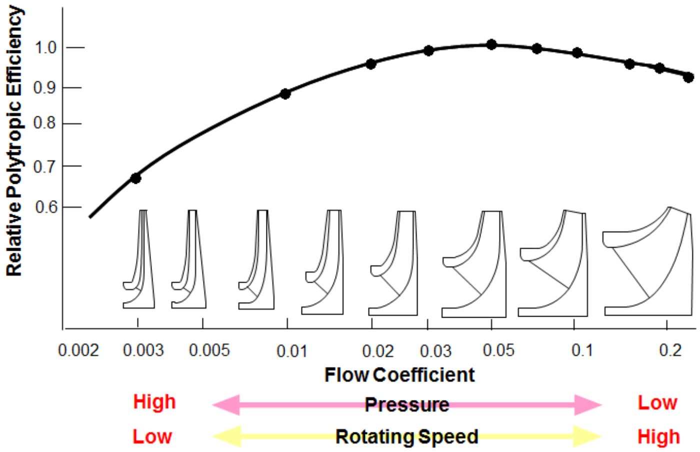

Figure 13. Relation of the Flow coefficient vs Relative Efficiency

$$\phi \propto \frac{Q}{D^3 N} \qquad \text{- (1)}$$

As shown in formula (1), decreasing of impeller diameter with high rotating speed is one of measure to increase the flow coefficient. However, it might move the rotor into an unstable region. Therefore, as a countermeasure, high boss ratio impellers and hole pattern seal are applied for super high pressure compressor as described in clause A "High Stability Design". These technologies can allow the operation under higher rotating speed since rotor stiffness and damping is dramatically increased. Consequently, flow coefficient of super high pressure compressor can be sifted to higher region, and efficiency is improved.

To further enhance performance, wide flow path impellers and vaned diffusers are applied. For the super high pressure compressor, impeller exit width of approximately 2 times wider than conventional design is applied. Thereby friction loss can be reduced and increasing efficiency more. However, if the flow path is widened, rotating stall needs to be taken into account in the aerodynamic design.

In general, in case of low pressure application, since surge phenomena is observed before occurring the rotating stall, vaned diffuser is not usually required from operation range point of view. In contrast, in high pressure application, operating range is limited due to the rotating stall which usually occurs before reaching the actual surge. Therefore, in order to avoid the rotating stall, vaned diffuser is applied as shown in Fig.14. Vaned diffuser can suppress the rotating stall, and can achieve a wider operating range than before.

In addition to that, labyrinth design was upgraded from a conversional to a step type to reduce recycle flow leakage.

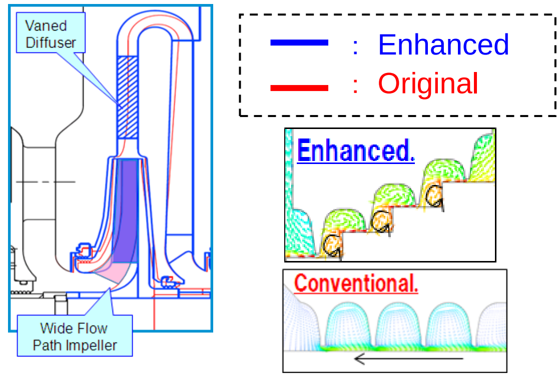

Figure 14, Model Comparison between Original and Enhanced Impeller Stage Design

The full flow path analysis using CFD of enhanced stage was carried out and verified the performance as shown in Fig.15. The upper column shows the CFD results of the conventional stage design, and the lower column shows the enhanced one. As can be seen, even though the relative inlet Mach number is increased due to the increasing of boss ratio, pressure was recovered adequately at the diffuser outlet with the short length vaned diffuser.

Fig.16 shows a photograph of the 700 bar Compressor Impeller (EDMed, 1-Piece Impeller) and single stage performance test results which include high boss ratio, wide flow path impellers and vaned diffuser with a smaller diffuser ratio. With the high boss ratio impeller design, the flow coefficient could be increased thereby achieving a higher efficiency than the original design. Furthermore, the enhanced wide flow path and labyrinth design contributed to the increase in efficiency.

As a result of single stage performance testing, a polytropic efficiency greater than 70% @ atmospheric air condition could be achieved. And the stability margin was more than 35% from the design point since rotating stall was suppressed with the vaned diffuser.

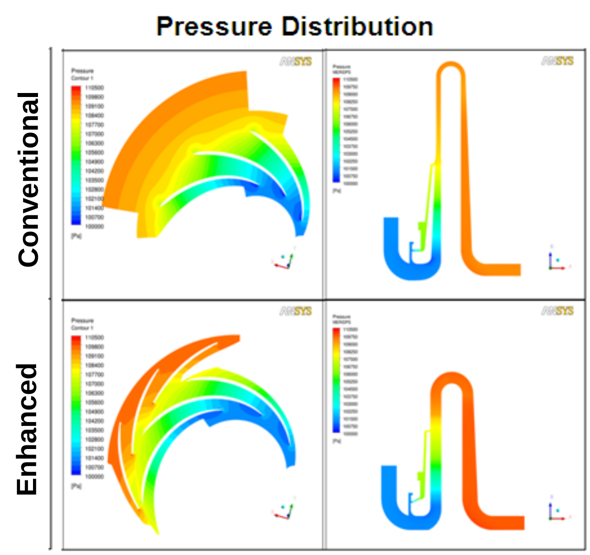

Figure 15. CFD Analysis Result of Conventional and Enhanced Impeller Stage

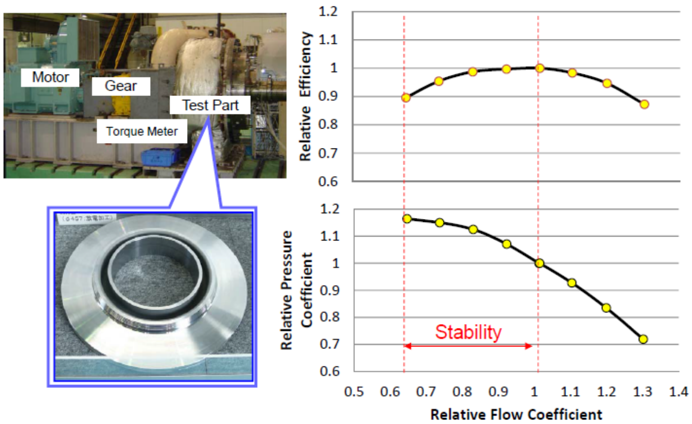

Figure 16. Impeller for 700bar Compressor and Single Stage Performance Test Result

C. Compact Design

C-1. Compact Design Concept

Fig.17 introduces that how to small the newly developed compressor in comparison with conventional compressor. Cross sectional drawing shows both conventional and enhanced compressor. In case of conventional design, casing inner diameter is relatively large. In contrast, that is reduced about 35% for enhanced design. This is resulted from application of short length diffuser for super high pressure compressor. Fig.18 shows the required casing wall thickness versus design pressure. In case the casing inner diameter is the same for all the pressure rating, the required casing thickness becomes bigger. However, by reducing the casing inner diameter and upgrading the material, the wall thickness is not necessary to increase. As a result, total weight of the compressor can be reduced to approximately half weight of conventional design even with the increase in pressure. This is the advantage for FPSO service.

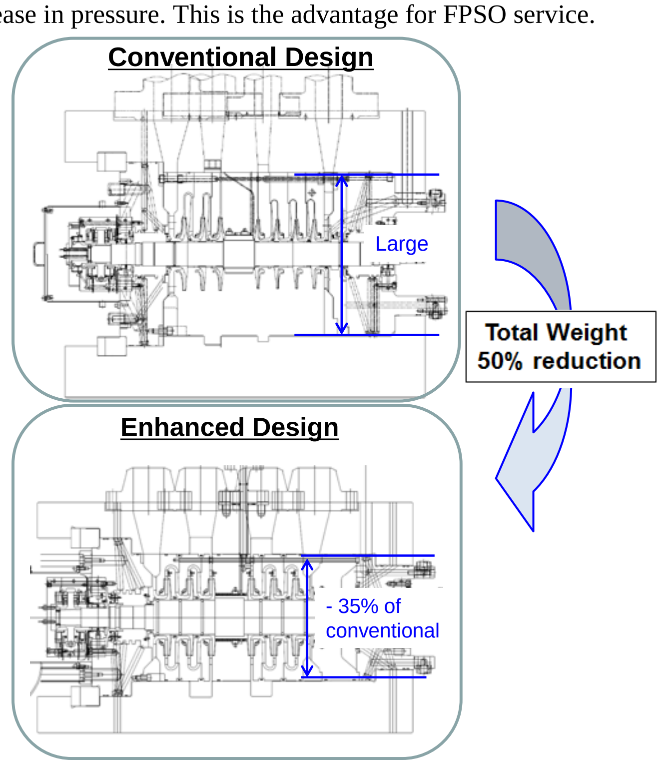

Figure 17. Compact Design Summary

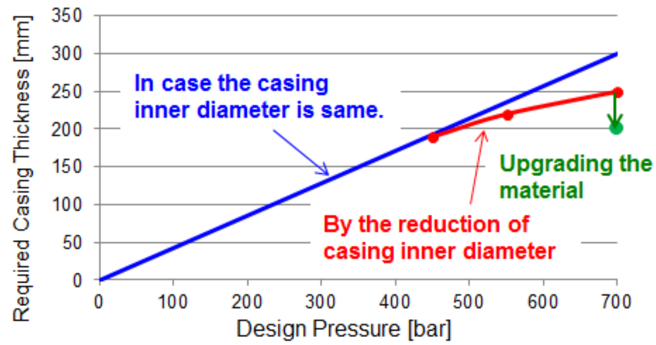

Figure 18. Required Casing Thickness vs design pressure

C-2. Strength of Static Parts

Casing design suitability has been confirmed by FEA as shown in Fig.19 and Fig.20. Shear ring, casing deformations, sealing and casing stress were checked in detail. And all the casing design parameters were confirmed to be suitable.

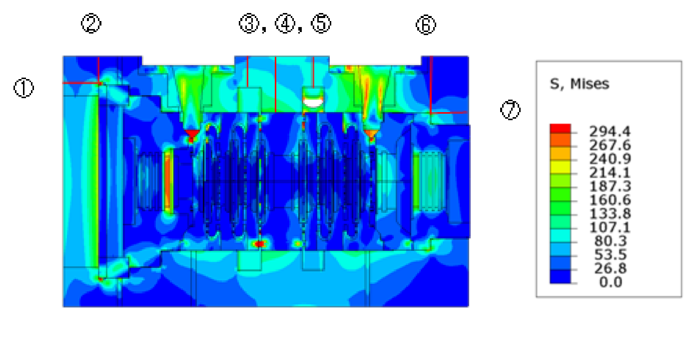

Figure 19. FEA Results of Static Parts

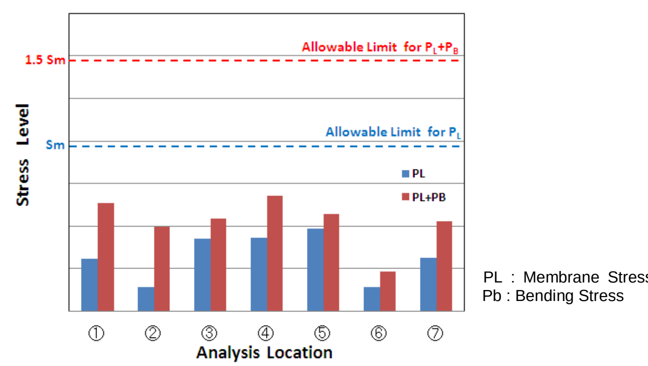

Figure 20. FEM Analysis Result

D. Wide Range Operation

Fig.21 shows the balance piston arrangement in a back-to-back type compressor. In a single balance piston configuration the balance piston is located between low pressure and high pressure sections. In this configuration, the total impeller thrust force of A and B is cancelled by the balance piston.

On the other hand, in a double balance piston configuration, the compressor has one more balance piston at the inlet of the high pressure section. In this configuration, the impeller thrust force of the low pressure section is cancelled by BP-2, and the impeller thrust force of the higher pressure section is cancelled by BP-1. This is manufacturer's standard configuration for CO_2 compressors in Urea plants. The merits of this double balance piston configuration are evident in Fig.22.

In the event of a significant change in the operating condition, the thrust balance between the low pressure and high pressure sections is changed. But even in that case, the impeller thrust force can be balanced. And also, the seal dynamic pressure can be the suction pressure for both sides by this design.

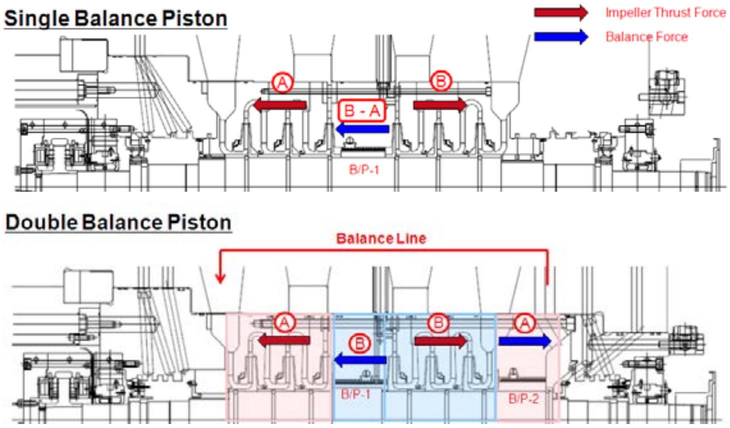

Figure 21. Balance Piston installation Model

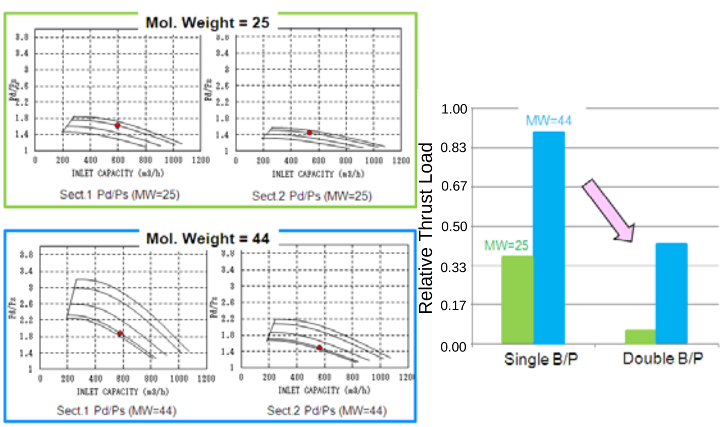

Figure 22. Thrust Force Comparison

LOAD TEST PREPARATION

Fig.23 shows a typical injection compressor arrangement at site. The mixture of the CO_2 and Natural Gas is compressed by the gathering compressor, after that the CO_2 and Natural Gas are separated to export or lift off the natural gas. After that the CO_2 and Natural Gas are mixed and compressed by the injection compressor.

Therefore, the injection compressor's molecular weight could be very wide range from 20 to 40. And density and pressure is very high.

Table 5 shows verification test program which has been successfully completed in October 2013. In order to demonstrate the robustness for all field operation condition including Molecular weight variation, load test with both low and high molecular weight cases including Type-1 performance testing, full load/ full speed/ full pressure/high density testing was carried out.

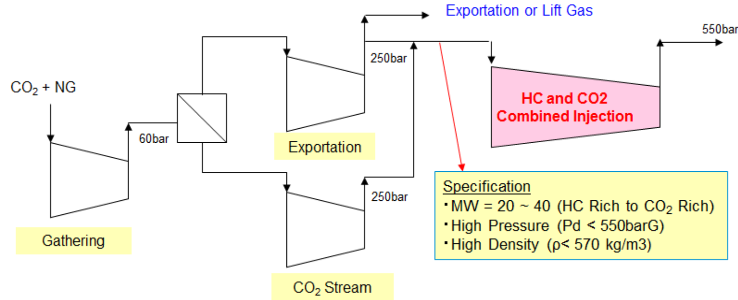

Figure 23. Typical Site Arrangement

Table 5.Verification Test Program

	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	N2	City Gas + CO2	City Gas + CO2	City Gas + CO2	City Gas + CO2	CO2	-

For the super high pressure compressor testing, the shop test facility used for the high pressure compressor in 2003 was upgraded. Fig.24 is a sketch highlighting the upgrades required to test the 700 bar compressor and measurement items for evaluation of compressor integrity.

The black color shows the original test apparatus. The red color highlights the new facilities to be added show PSV, FCV, HP Piping, HP CO_2 Liquid Pump with heating evaporation system. During verification testing, several items were measured such as compressor performance, pressure fluctuation to verify the rotating stall, thrust load by load cell, and rotor stability test by hydraulic exciter.

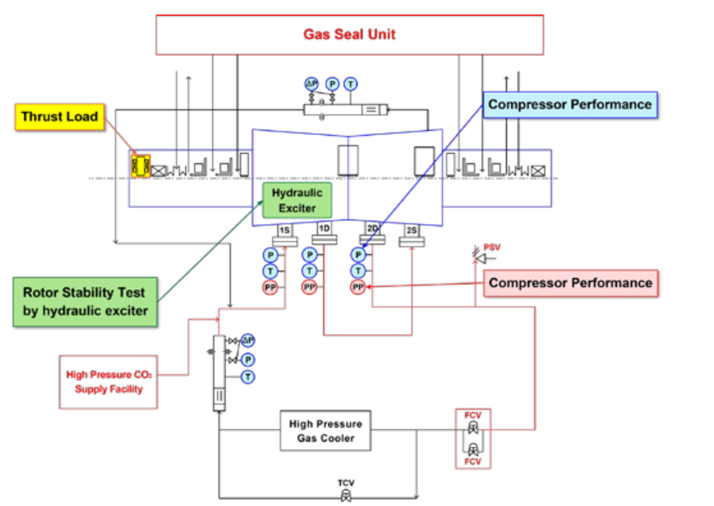

Figure 24.Test loop Arrangement and measuring items for 700bar verification Test

Fig.25 shows test stand overview and Fig.26 shows new facilities for super high pressure verification test. Test loop and each facility were checked to confirm no leakage and instrument system with sufficient accuracy.

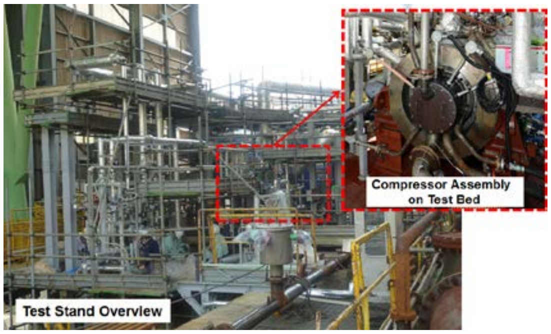

Figure 25.Test Stand Overview

Figure 26.Upgraded Facilities for Super High Pressure Test

TEST RESULT

A. Type.1 Performance Test

Type.1 Performance Tests were conducted and results are shown in Fig.27. The gas density was continuously measured to check the gas property during test, and gas chromatography analysis was also conducted occasionally. The expected efficiency and pressure coefficient coincided with the test result, and the overall efficiency flange to flange basis were over 70% under Pure CO_2 gas condition (MW=44). And the stability range from design point were over 30% except MW=44 condition. Fig.28 shows pressure fluctuation which was measured at 2nd discharge piping. Pressure fluctuation was measured at the expected surge point to confirm origination of rotating stall, and no pressure fluctuation around low frequency range was observed. That means rotating stall would be suppressed by vaned diffuser as expected.

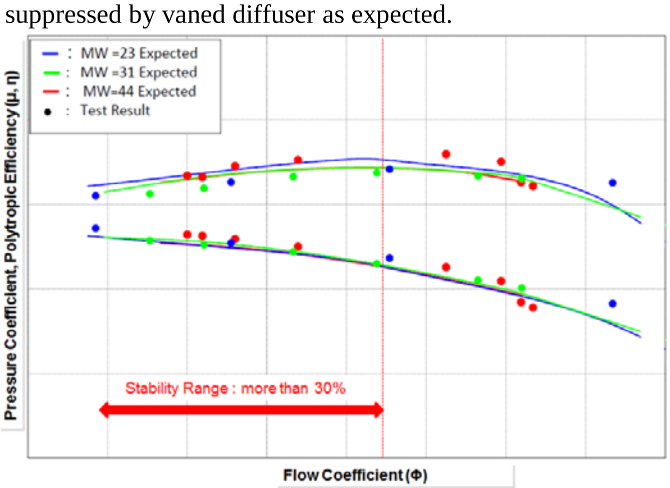

Figure 27.Type.1 Performance Test Result (MW=23, 31,44)

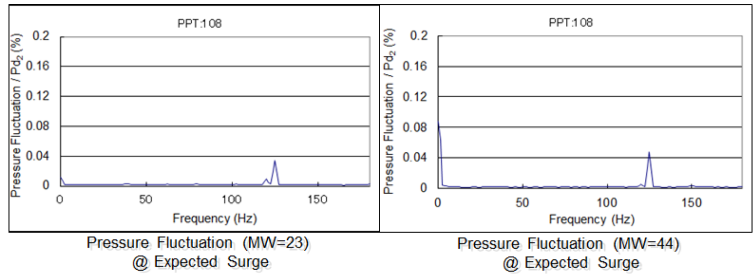

Figure 28. Pressure Fluctuation @ 2nd discharge piping

B. Full Load Full Speed Full Pressure Test

Full load, full speed and full pressure test and high density test used pure CO_2 gas was conducted and the discharge pressure was increased up to 680bar. Fig.29, Fig.30 and Fig.31, Fig.32 shows the operating trend data and waterfall plot under Molecular weight 23 and pure CO_2 gas condition. It was confirmed that the rotor vibration was very small 5 micron meter $^{p\text{-}p}$ without harmful sub-synchronous vibration and all mechanical operation data were within normal level and very stable.

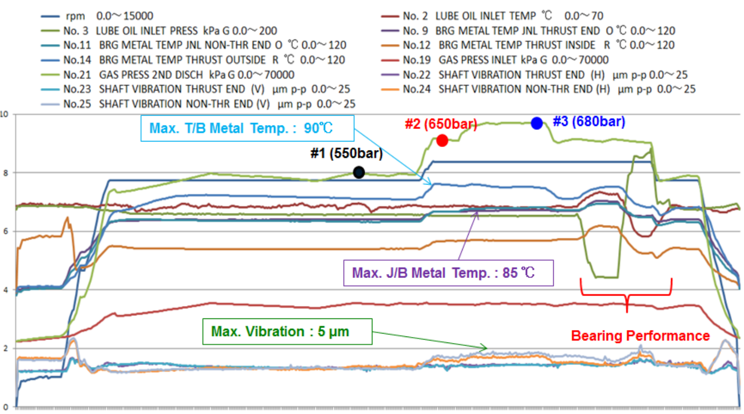

Figure 29. Trend Data at FL/FP/FS Test Condition (MW=23)

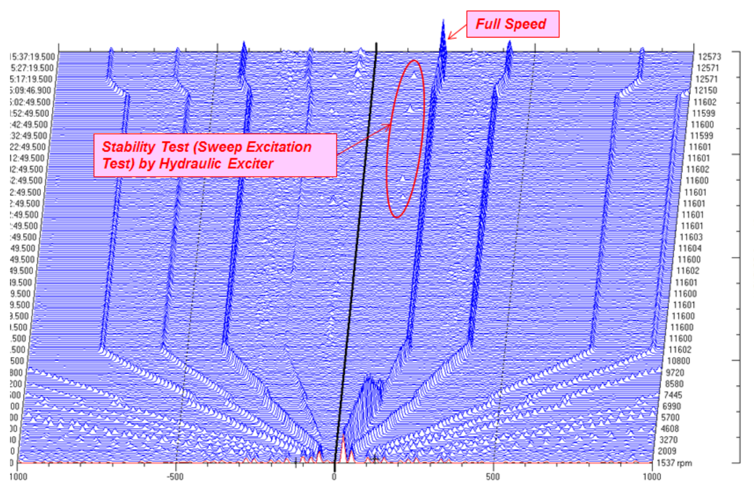

Figure 30.Waterfall Plot at FL/FP/FS Test Condition(MW=23)

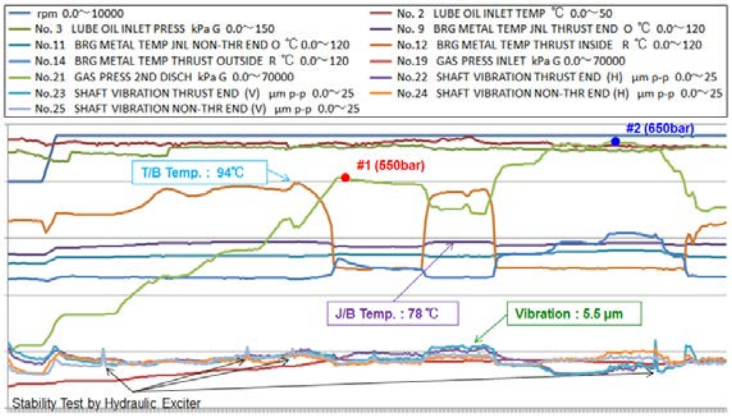

Figure 31.Trend Data at FL/FP/FS Condition (Pure CO_2)

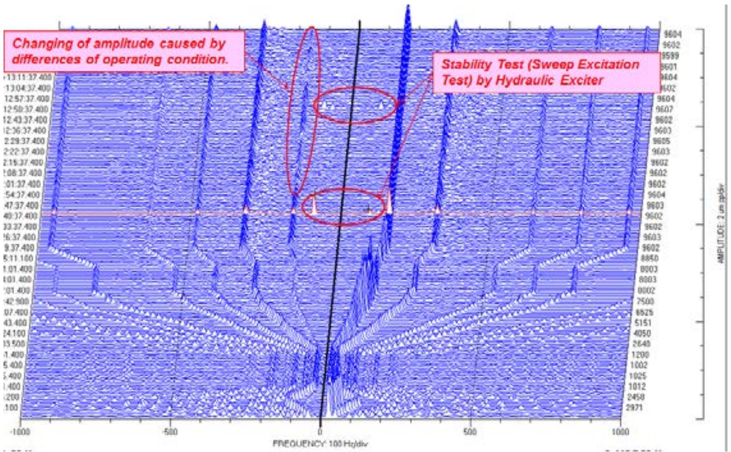

Figure 32.Waterfall Plot at FL/FP/FS Condition (Pure CO_2)

Fig.33 shows the method of rotor stability test by sweep excitation of casing. The rotor vibration was excited by two hydraulic exciters mounted on the casing (Horizontal direction and vertical direction). In order to measure the frequency and log decrement of the 1st critical mode at full speed, the first natural frequency was excited asynchronously by hydraulic exciters mount on the casing. The peak amplitude around 100Hz appeared in the waterfall plots were caused by this testing.

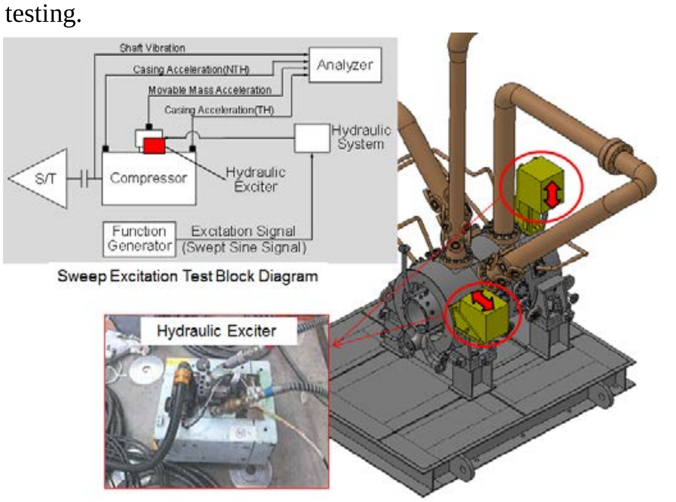

Figure 33.Method of Rotor Stability Test by Sweep Excitation of Casing

Fig.34 shows the rotor stability test result. The sweep excitation tests under no-load and full load condition were carried out. Under no load condition, 1st natural frequency was around 6600rpm and log. decrement was 0.4 which was evaluated by curve fitting of vibration response. On the other hand, in case of load testing, 1st critical mode was critically damped, and measured log decrement was increased to 2.5 with effect of developed hole pattern seal. Since vibration level under any operating condition was enough low level, developed compressor can provide stable and reliable operation for all field operation condition including Molecular weight variation. Moreover, since vibration response is relatively larger than background noise and input-output signal has good coherence which was more than 0.8, measured log. decrement would be considered reasonable and proper.

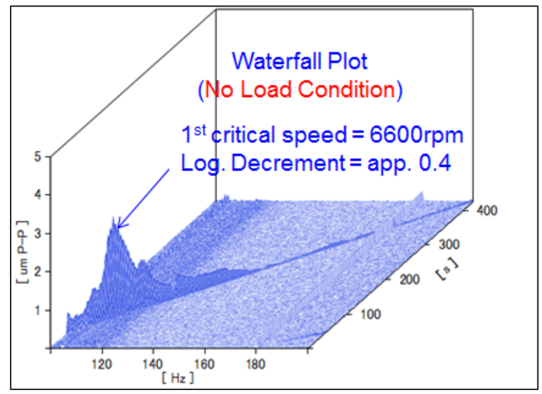

Figure 34-1.Rotor Stability Test Result by Casing Sweep Excitation (No Load Condition)

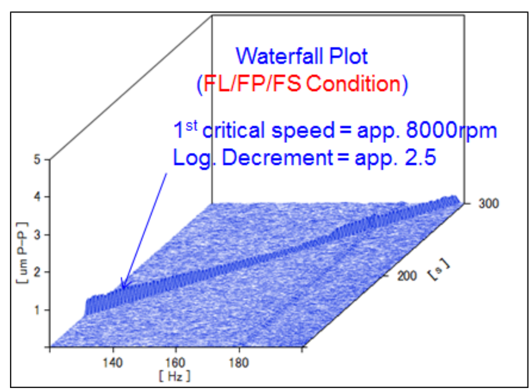

Figure 34-2.Rotor Stability Test Result by Casing Sweep Excitation (FL/FP/FS Condition)

Thrust load was also measured throughout load testing by loadcell to confirm the advantage of double balance piston and robustness to the thrust forces. As shown in Fig.35, the actual thrust force change was small and very lower than allowable level under any operating condition.

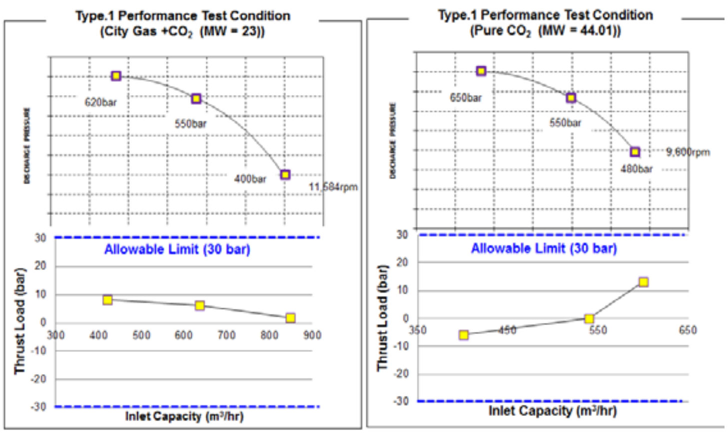

Figure 35.Thrust Load during Operating Condition Change

CONCLUSIONS

The super high pressure compressor was developed with following concept through the evaluation program against any potential risk.
1) "High Rotor Stability" with high boss ratio impeller and hole pattern seal.
2) "High Efficiency" with wide flow path and vaned diffuser.
3) "Compact" with high boss ratio impeller and short length diffuser.
4) "Wide Operation Range" with vaned diffuser
5) "Robust Design" with double balance piston.

Extensive verification test (No load M/T, Type.2 and Type.1 performance test, FL/FP/FS Test) was successfully completed as expected with;
1) High Rotor Stability (low vibration)
2) High Compressor Performance (Efficiency = app. 70%)
3) Compact and Light Weight (Half weight of conventional)
4) Wide Range Operation (Stability Range = more than 30%)
5) Low Thrust Load (Lower than allowable limit)

Super high pressure compressor has demonstrated the robustness for all field operation condition including Molecular weight variation from 23 to 44.

NOMENCLATURE

Φ	= Flow Coefficient
Q	= Volume Flow
N	= Rotating Speed
D	= Impeller Diameter

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