SUBSEA CENTRIFUGAL COMPRESSOR DEVELOPMENT—BREAKTHROUGH TECHNOLOGY FOR OIL AND GAS OFFSHORE PRODUCTION

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ABSTRACT

As the offshore market faces deepwater production challenges, the oil and gas industry is investing in new technologies to bring down the costs in the subsea fields exploitation. A breakthrough technology solution is the development of a subsea centrifugal compressor module installed on the seabed, transporting the wellstream to a central platform or directly to shore. The typical application of the module is gas boosting, but it can also be used in the reinjection service. The subsea module concept is made of a package, a compression train (centrifugal compressor, gearbox, electric motor), and a separator as an option.

Due to the particular installation site, reliability is an essential requirement to guarantee the module operation. In order to validate this innovative design solution and minimize any risks in the field on future units, thermodynamic and mechanical performances of the machine must be performed over the entire operating range in the test facilities.

INTRODUCTION

Typically, the exploitation of subsea marginal fields and fields in deep water (<500 m [<1640.42 ft]) is not at present profitable because the platform costs are too high for the expected return, thus these fields remain unexploited. For this reason some governments, notably the Norwegian, are funding and leading programs to develop subsea systems and seabed processing for deep water and to accelerate the commercialization of research and development (R&D) in upstream oil and gas.

DEMO 2000, launched in 1999, is one of the programs aimed at qualifying deepwater technology by pilot demonstration, which has been selected and approved to finance a certain number of projects deemed to represent a technology breakthrough. Some of the most important companies participate in an operator committee that recommends projects and decides how many pilots could be hosted.

The Subsea Centrifugal Compression Module Development has been one of the projects approved; for this project GE Oil & Gas Nuovo Pignone and Kvaerner Eureka I have codeveloped a new advanced technology to exploit, in an economically way, certain types of subsea fields.

Specifically, the scope of Nuovo Pignone and Kvaerner is the development of a 2.5 MW (3352.56 hp) module composed of a package and a classical compression train architecture: electric motor, gearbox, and centrifugal compressor (BLUE-C compressor line). The project was started with the refurbishment and testing of a 850 kW (1139.87 hp) unit realized in 1992 by Nuovo Pignone/Kvaerner to validate the concept and to identify technology gaps before completing the detailed design of the new module. The 2.5 MW (3352.56 hp) prototype construction and testing will be realized by a pilot installation and final qualification.

This paper describes how the compression technologies, based on the experience gained on the 850 kW (1139.87 hp) unit (Figure 1), can be applied to subsea services, especially to the 2.5 MW and 5 MW (3352.56 hp and 6705.11 hp) units that are under development.

Figure 1. 850 KW (1139.87 HP) Subsea Compressor Module Refurbishment.

PRODUCT CONFIGURATION

The product architecture of the subsea module is made of a centrifugal compressor train and package, including a separator as an option. The compressor train, vertically arranged, is made up of a centrifugal compressor, a planetary gearbox, and a variable speed electric motor to adjust the speed during operations (Figure 2). All the components of the module are oil lubricated: power is transmitted from the electric motor to the compressor and the oil pump through a gearbox wheel.

One of the main features of the module is that all three casings (centrifugal compressor, gearbox, and electric motor) are directly joined together with screws, and the entire module is internally pressurized at compressor suction pressure. In order to facilitate the assembling of the gearbox and centrifugal compressor casings, a planetary gearbox has been chosen. This allows centrifugal compressor and gearbox shafts to be aligned so the casings can be easily joined to one another. The reference pressure of the module is not the ambient pressure but the suction pressure of the compressor.

Subsea Compressor

The subsea compressor was studied starting from an existing project of a standard compressor: it is a barrel-type compressor with two flanges (one suction and one discharge) and six stages

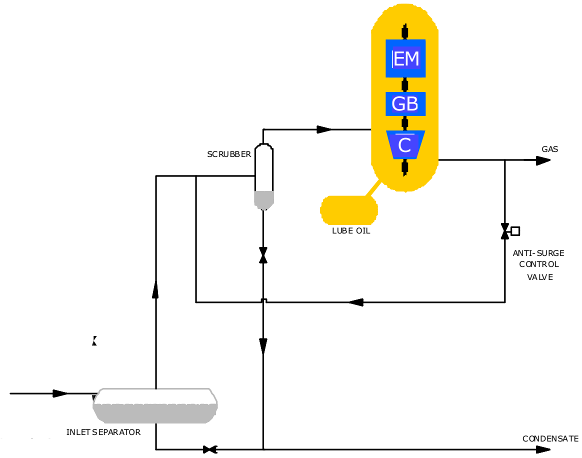

Figure 2. Subsea Module Concept.

(for the 2.5 MW [3352.56 hp] unit) in line. It consists of a static unit (casing, casing heads, covers, diaphragms, seals, and bearings) and a rotating one (rotor formed by shaft, impellers, and balance drum).

The centrifugal compressor architecture is standardized and well proven: impellers are fitted on the shaft shrinkage. In order to reduce the net thrust on the thrust bearing, the compressor, as standard, is equipped with a balancing drum on the discharge side. The back chamber of the balancing drum is connected to the suction through the balancing line in order to ensure the same pressure on the two ends of the machine.

One of the most important aspects to be carefully considered in the development phase of the compressor design is the reliability of the machine. From the machine point of view, it means to use proven and well-referenced components and safe and demonstrated mechanical behavior. From a mechanical point of view, it is crucial to realize a machine with a safe lateral behavior, and characterized by very low vibration levels in order to avoid any possibility of sealing damage and consequent maintenance needs. In order to obtain such a machine, great attention has been given to rotor stiffness maximization, i.e., few stages and large shaft diameters.

The centrifugal compressor (Figure 3) has been designed to match API requirements. Inlet design flow of the machine is 900 m^3/h (31,783.20 ft^3/h), suction pressure is 65 barA (942.74 psi), and delivery pressure is 130 barA (1885.49 psi). The gas is methane with molecular weight close to 19. Casing design pressure is equal to 200 barA (2900.75 psi).

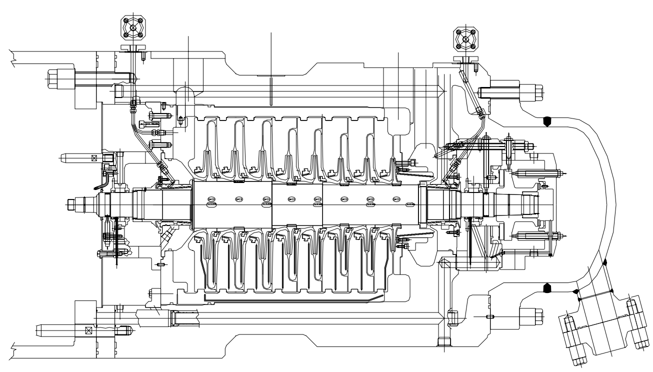

Figure 3. 850 KW (1139.87 HP) Subsea Compressor Cross Section.

The size of the impellers is close to 350 mm (1.15 ft) and rated speed is 12,000 rpm. All the impellers are 2-D type, widely used in reinjection applications. Flexibility ratio (the ratio between maximum continuous speed and first critical speed) has been minimized in order to maximize stiffness and reliability. Impeller bore diameter is close to 120 mm (0.39 ft), while journal bearing diameter is 80 mm (0.26 ft). Thrust bearing size is equal to 8 inches. Rotor length is close to 1700 mm (5.58 ft). The machine is equipped with standard oil tilting-pad journal and double-side leveling plate thrust bearings.

All the components of the subsea module are pressurized at the compressor suction pressure; in principle no differential pressure exists at the end of the shaft so there is no need for end seals. In order to stop any oil flow between journal bearing and process gas, the shaft of the compressor is equipped on the two sides with buffered end labyrinth seals. The end seals are standard buffered labyrinth seals.

Some gas taken from the compressor delivery is reduced in pressure and sent to the middle of the seal in order to create a barrier between the process gas and the oil side. Oil consumption is hence minimized. As a matter of fact, oil consumption is another crucial aspect of the project: the oil tank is filled during the commissioning phase of the module and no additional refilling is allowed during module operation and before maintenance (no connection is provided between the oil tank and the platform). Unexpected extra oil consumption could determine the need of unscheduled maintenance with an important impact on field productivity.

The differential pressure between buffer gas and suction pressure can be maintained to minimum values (down to 0.2 bar [2.9 psi]) in order to reduce as much as possible the buffer gas consumption (to increase compressor efficiency). Oil from journal and thrust bearings is then collected and sent to a pressurized tank directly connected to the bottom of the compressor. Oil tank pressure coincides to the suction pressure of the compressor: the oil vent pipe is connected to the process suction line (upstream of the separator in order to recover oil particles).

Due to the fact that the "ambient" pressure of the compressor is the suction pressure, there is no need for dry gas or oil seals. The only sealing system that is used on the end of the compressor, as said, is the labyrinth seal system to avoid oil migration. The absence of dry gas mechanical seals represents an important simplification in the machine architecture improving reliability and maintenance interval maximization with a positive impact on rotor-bearing span.

Except for the different number of stages and the power, the 2.5 MW (3352.56 hp) unit design has been derived from the tested 850 kW (1139.87 hp) unit. Some improvements in the end seal design and in the oil process and instrumentation have been introduced in order to further reduce oil consumption. In particular it was noted that great attention must be put in the control of oil flow and pressure to the gearbox and the compressor bearings. While compressor bearing oil inlet pressure has to be carefully controlled in order to avoid oil migration toward the machine, gearbox oil inlet pressure has to be maintained higher (the compressor's pressure) to ensure correct lubrication.

As far as material selection is concerned, presently the gas is "sweet" (there is no hydrogen sulfide (H_2S) in the gas), but the presence of carbon dioxide (CO_2) calls for the use of materials able to withstand general corrosion.

It has to be considered that, even if during normal operation no liquid water exists in the gas, during transient or steady conditions (pressurized shutdown), the presence of water around the compressor casing determines the great possibility of having free liquid water inside the gas. For this reason all static components of the compressor are made of stainless steel. Impellers as well have to be built with materials able to withstand CO_2 and water. In any case the compressor technology already available allows H_2S presence inside the gas without any reduction in the life/reliability of the machine.

As far as the thermo-fluid dynamic behavior of the machine is concerned, one of the main differences between this and standard ground applications is the important heat exchange between the process gas and the external cold water environment (down to -1°C [-30.2°F]). This effect has to be carefully investigated and taken into consideration during the design phase of the machine in order to avoid any mismatching between the stages (caused by the lower stages' suction temperature).

In practice, the gas coming from one stage and before going to the next is cooled as if compression would interrefrigerate. If correctly evaluated, this effect could determine an increase in the efficiency of the machine compared to ground applications.

Upstream of the compressor flange there is a gas liquid separation system followed by a cyclone scrubber. The purpose of this system is to protect the compression module against potential liquid intrusion in an efficient and reliable way. As already mentioned, the risk of having liquid particles inside the compression module has to be avoided. Actual requirements in terms of gas dryness are the same as for ground applications.

The next steps in the development of the subsea module will be to qualify impeller materials able to tolerate a high percentage of liquid and to eliminate any oil need. The low-speed electric motor will be replaced with a variable high-speed one in order to eliminate the gearbox. Oil bearings will be replaced with magnetic bearings.

COMPRESSOR RELIABILITY APPROACH

Reliability is a very important aspect for a product like the subsea centrifugal compressor module.

Consequently particular focus on centrifugal compressor reliability, with a specific plan for this item, is an essential requirement to guarantee machine performances. During the reliability study, different steps have been followed across the preliminary design and design review phase.

On the one hand, the reliability design consisted of a preliminary flow down (FD). This has been realized considering the list of the main items that are to be taken into account in order to avoid system working condition misunderstandings during the following reliability block diagram (RBD) realization.

On the other hand, a failure mode and effects analysis (FMEA) of the subsea compressor has been performed based on an existing project of barrel class compressor of the same casing. Then considering the particular underwater machine application and configuration, some gauge parameters were introduced to consider the environmental impact in failure situations. Moreover, additional stressing conditions have been considered due to the fact that no corrective maintenance actions can be performed on the system's items.

During the RBD realization, canonical configurations as well as series, parallel, cold standby, and k/n (k item should work in order to have the system at least working) were considered in order to evaluate the preliminary design reliability value, and to give feedback to the engineering department with the aim of increasing the whole system reliability figures by means of redundancy strategies where possible or by higher reliability component utilization. RBD simulation was performed with different software tools.

Availability figures are not the main aspect of this study because, due to the particular machine site location, no multiple corrective maintenance actions should be considered (zero failure is envisaged) in order to guarantee customer expectations.

Flow Down

The first step in this compressor's reliability analysis is the creation of its flow down. To design this tool, the starting point is the preliminary process and instruments diagram (P&ID) of the machine where all the components are present. The compressor is formed by two first level subsystems: the kernel and the auxiliaries. In Figure 4, the kernel's flow down is presented, with each component's quantity.

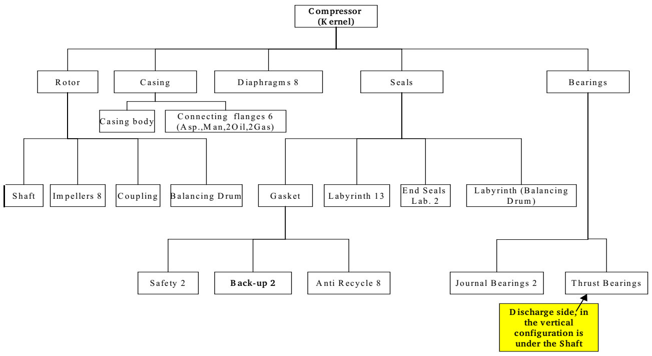

Figure 4. Subsea Compressor: Example of the Kernel's Flow Down.

The auxiliary system is made up of four second level subsystems: seals, sensors, lube/oil, and antisurge system. Once all the machine's components have been individuated, the challenge is to find their failure modes, to estimate the effects of unwanted failures, and to assign them a criticality for the overall system.

Failure Modes and Effects Analysis

To find the solution to this challenge, it is necessary to create an FMEA based on similar machines. Such an approach needs engineering brainstorming to understand where the differences between this application and a conventional one lie. The values of occurrence (O), severity (S), and probability of detection (D) are assigned to every component, finding the components with the highest and lowest risk priority number (RPN):

$$RPN = O \times S \times D \tag{1}$$

In this study, particular attention is given to the concepts of environment class (E) and environmental impact (EI). Working 350 m (1148.29 ft) underwater, the environmental consequences of a failure in a component need a careful study to prevent and avoid any risk of sea contamination. Hence great importance is given to the value of risk priority number and environment (RPNE):

$$RPNE = O \times E^2 \times EI \tag{2}$$

The values of *E* and *EI* are taken from tables properly built in the range of 1 to 5, in which the highest figures describe situations of environmental risk without any alarm, decreasing to the cases (1 to 3) of eventual failures under control, whose consequences are avoided by corrective maintenance actions. After the machine's flow down and FMECA creation, it is possible to modify the preliminary P&IDs version, coming to their final structure.

Figure 5 is an example of FMEA for a single component, in this case labyrinth seals installed between the compressor's impellers.

Reliability Block Diagram

RBD realization is very important in introducing data into reliability and availability software, which simulate the life of a system with a Monte-Carlo method of random numbers generation. The software generally needs to know failure distribution, repair distribution, and scheduled maintenance policy of every component, producing an output including uptime, downtime, PM downtime (planned maintenance downtime), reliability, availability, failure rate, mean time to first failure (MTTFF), and mean time to repair (MTTR) of every component.

With this analysis, it is possible to perform a reliability importance study as well, which, in the first design stages, allows modifying the component's quality or introducing redundancies to improve system reliability. In this case (subsea application), the mission reliability target is very important and difficult to reach

FMECA(E-EI)

System:	Labyrinth	Identification Number:	
Function/ Mission:		Date :	14/01/02
		Analyst:	A.Masi; M.Mugnaini

Modes of Failure				Failure Effects						
Serial Number	Failure Mode	Cause	Frequency of Occurrence	Local Effects	Overall Effects	Failure Detection Method	Degree of Severity	Probability of Detection	Environment Class	Environmental Impact
	Thermal expansion	Temperat.	1	Gas leakage	Loss of performances	Inspection	6	10	1	1
	Breaking	Vibration	3	Gas leakage	Loss of performances	Inspection	6	10	1	1

Criticaly		
Criticality value	RPN	RPNE
6	60	1
18	180	3
0	0	0
0	0	0
0	0	0
0	0	0

Corrective Actions

Operating Condition

A = Normal Condition
B = Start-Up after pressurized shut
C = Start-Up after no pressurized sh
D = Shut Down pressurized full equa
E = Shut Down pressurized partially
F = Shut Down no pressurized

Figure 5. FMECA: Example of Labyrinth Seals.

with a high level of confidence; but, with the sensitivity analysis, it is possible to have the opportunity of optimizing the design solution.

Starting from the flow down structure and P&IDs, the RBD was created considering only the components whose failure causes the machine to stop, without considering noncritical components. Critical components are described by failure distributions (exponential, Weibull) derived from a proprietary reliability database. This database was created analyzing field data, obtaining reliability information on every component, as well as with the support of another reliability database. The subsea compressor's kernel RBD is presented in Figure 6.

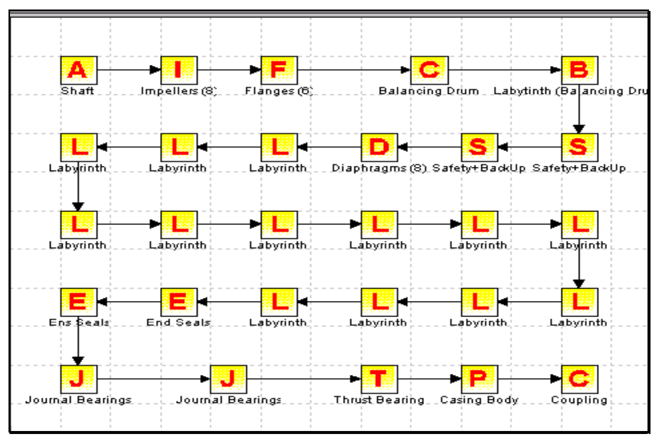

Figure 6. Subsea Compressor Kernel's RBD.

Simulations Results

Reliability and availability simulations are performed by three software programs. These programs are fundamental in studying such systems because it would be too expensive to perform accelerated life tests. Moreover, there is the possibility of computing and presenting the reliability simulation output with a confidence level (95 percent, 90 percent, . . .).

FULL LOAD MODULE TESTING

The innovative design solutions required for the development of these subsea centrifugal compressor modules and the desire for validating these design solutions to minimize any risk of activities in the field on the future units, drove the authors to exhaustively confirm the thermodynamic and mechanical performance of the machines over the entire operating range with a complete testing campaign.

The purpose of the tests is to verify before construction of the new units (2.5/5 MW [3352.56hp/6705.11 hp]) that the design solution will work properly in the subsea environment. Various types of tests are available to verify with different levels of detail the performance of the units. The most exhaustive test is the full load string test of the entire train. In this kind of test, all the machines are coupled as in the field, and, in general, the most critical auxiliary systems are also installed. For this special application, the authors have realized a complete test campaign based on a full load string test underwater during which an API 617 Mechanical Running Test, an ASME PTC10 Type 1 test, and a 500 hour endurance test were performed. In this type of test, the process conditions are reproduced very closely: process gas characteristics (flows, pressures, temperatures), machine speeds, power absorbed, etc. This test allows an almost complete analysis of the thermodynamic and mechanical behavior of the entire module, including compressor performance curves, surge limit, machinery vibrations in both steady-state and transient conditions, power absorbed, and efficiencies. In addition, the auxiliary systems included in the test can also be fully verified. As described later in more detail, these tests are technically challenging, requiring specific design, erection and execution skills, experience, and know-how.

Test Arrangement

The subsea compressor module was installed in a dedicated testing facility (Figure 7) consisting of a special tank (in which the module is placed underwater) and a process gas closed loop with all the auxiliaries required. The test environment can be divided into the following subsystems:

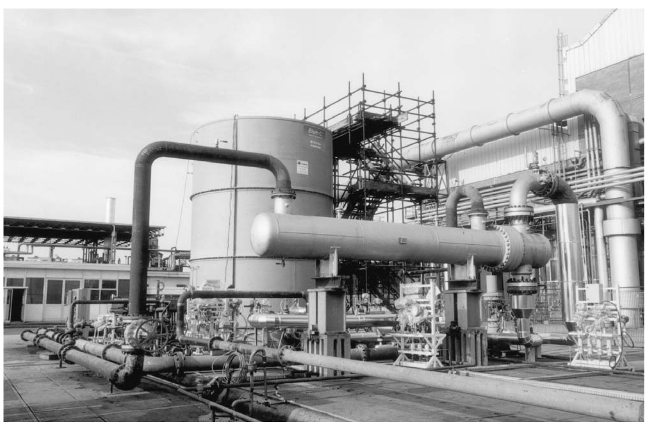

Figure 7. 850 KW (1139.87 HP) Subsea Compressor Module Facilities in Florence.

- Subsea compressor module including main auxiliaries installed underwater
- Process gas loop with shop related auxiliaries (throttling valves, water/gas heat exchangers, process flow measurement devices, safety valves, gas analyzer)
- Transfer and filling system for process gas mixture (natural gas and CO_2)
- Additional lube oil and seal gas auxiliaries
- Water cooling system (for process gas and water tank)
- Additional instrumentation

The main test facilities dedicated to the subsea module full load test are shown in Figure 8. The following main subsea module auxiliaries were installed underwater for the outdoor string test:

- Electric motor, gearbox, and compressors assembled with piping on the module structure

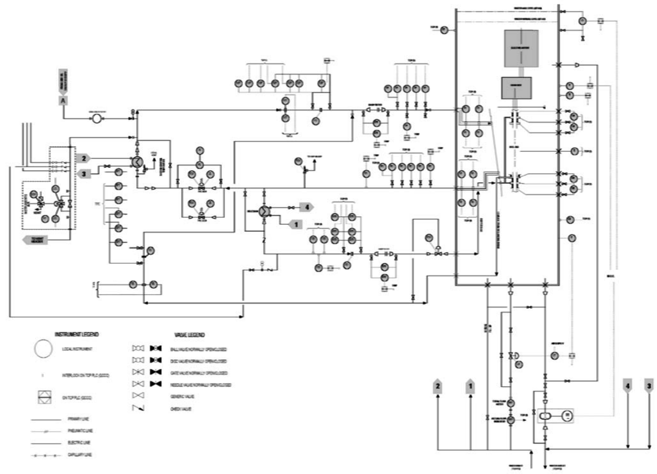

Figure 8. 850 KW (1139.87 HP) Process and Instrumentation Diagram of Testing Facility.

- Complete lube oil system (composed of a mechanical pump driven via gearbox, a water-oil cooler, and a pressurized tank)
- Complete seal gas system

Additional shop auxiliaries were installed out of the water to complete the test facilities. These additional systems were installed to drive the unit in safety condition during the experimental activities related to the validation of the new design solution, especially for lube oil and seal gas systems.

Test Preparation

The preparation of the full load endurance test started with an engineering phase carried out with a cross functional job with product engineering. The first step was the plant and equipment investment definition: the authors started by reviewing the test facilities to define requirements for special equipment and/or upgrading of existing facilities to cover the new issues related to the underwater test.

The second step was specifying the thermodynamic design of the test, which was carried out jointly with compressor engineering to define the pressures, temperature, and flows test conditions. The thermodynamic design is not so critical, especially for the 850 kW (1139.87 hp) unit, so the design was quite simple.

The third step was the design of the major components of the test facility. During this step, the authors selected dedicated throttling valves, coolers, safety valves, and piping using standard coded materials as much as possible. During test engineering, special attention was dedicated to the lube oil, seal gas, and water exchange systems. A detailed analysis was performed to define the additional auxiliaries needed to run the unit properly during the experimentation of the innovative design solution, and a major effort was required to define the running conditions over the entire range and during steady-state or transient conditions.

The lube oil system of the subsea module is based on a mechanical pump driven via gearbox with a water-oil cooler that exchange the heat directly with the external water, taking the oil from a tank connected to the module. Due to this connection, the oil tank has an internal pressure equal to the compressor suction pressure, and all the oil system is pressurized at the compressor inlet pressure plus the extra pressure of the main pump. This system is designed to run underwater with very few components to increase the reliability of the unit, but to verify the right functionality. During the tests, the authors added a prelube pump, an additional pressure control valve, a flow control valve with bypass to recirculate the extra flow, and some additional instrumentation inside the compressor to verify in every condition the real lube oil pressure on the bearing. With this complete facility, the authors were able to analyze and to characterize the subsea compressor lube oil system integrated on the module.

Another area of detailed engineering was the buffer gas filling system. Due to the design solution of the new seals, it was necessary to use additional instrumentation inside the compressor to verify the right flow direction under the labyrinth seals and to avoid the risk of an oil flow from a bearing to the internal part of the compressor. The buffer gas system is based on a special valve that maintains the right buffer pressure (0.2 bar [2.9 psi] up to inlet pressure). To validate this valve during the tests, two additional lines were also installed, the first with an orifice, the second with a throttling valve.

This complete system gave the capability of testing the final seal gas system, checking the functionality of the complete system in cooperation with the oil system to define the best case: very low buffer gas flow with no oil flow inside the machine. After conclusion of the engineering, the authors performed a complete design review with the entire team, product management, product engineering, test engineering, and test execution.

Test Execution

The installation quality process was based on a four-level checklist. The first was a mechanical installation check out, the second covered all the wiring loop check and functional check starting from the instrumentation on the machine up to the shop unit control panel verifying the correspondent signal. Two other checklists were used to verify all the parameters of the complete module during the running period and the status of all the activities during the full load test execution. The execution of the test was planned in detail during the engineering phase to manage the high technical needs of the running period. Every step—starting with filling of the gas loop, through the run up period, the thermal stabilization, the point readings and the shut down—were planned and reported in a detailed procedure. With this approach, it was possible to obtain a complete test of the module to give all the information to product engineers to validate the compressor module project.

The goals of the test also included checkouts of the mechanical behavior of the complete train, by means of measurement of vibrations, of bearing babbitt temperature, and lube oil system parameters. A camera, a real-time vibration analyzer, a tape recorder, a digital vector filter, and an oscilloscope were installed in the shop control room in order to monitor and analyze all the noncontact probe signals of the constitutive machines of the shaft line during the complete duration of the tests. Vibration diagrams were plotted (frequency analysis at 100 percent speed, rundown and runup diagrams) with a shop vibration data gathering system.

Test Results

The characteristic curves of the compressors measured during the test are reported and compared with those predicted during the engineering phase, and the results confirm that each compressor satisfied performance requirements (Figure 9).

During these tests, special care was dedicated to examining the mechanical behavior of the compressor after the endurance test of 500 hours. The mechanical behavior of the machines was fully compliant with the requirements, both in steady-state and transient conditions. In particular, the vibration level (Figure 10) of the rotors was very low and in accordance with API specs.

The auxiliaries' systems also performed properly under all the operating conditions, and all the testing data were collected to validate the final technical solution for the new generation of subsea compressor based on 2.5 and 5 MW (3352.56 hp and 6705.11 hp) drivers.

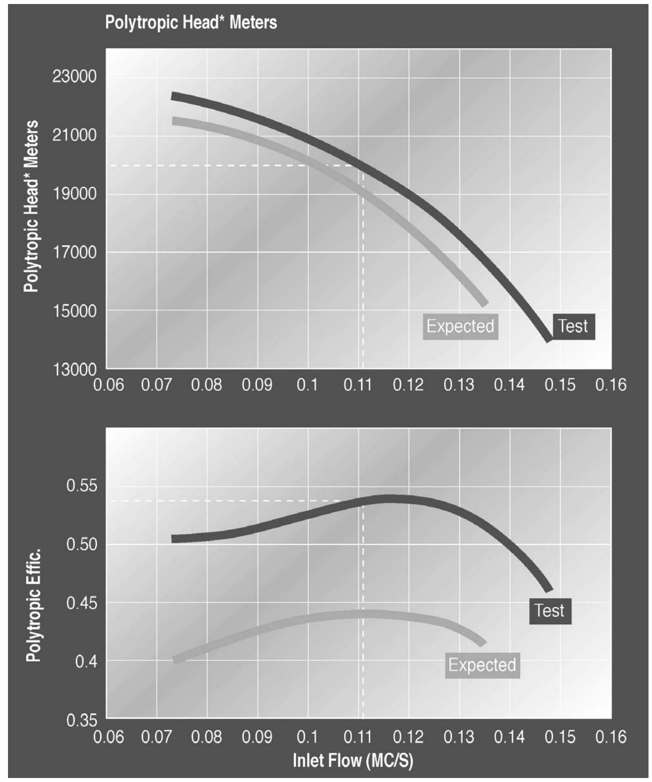

Figure 9. 850 KW (1139.87 HP) Subsea Compressor Performance Curves.

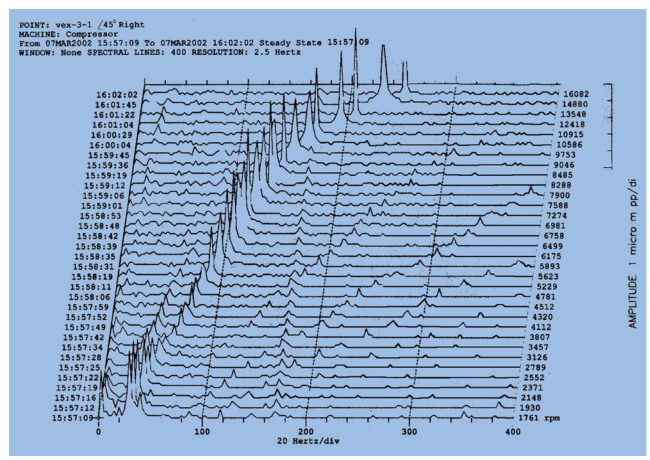

Figure 10. 850 KW (1139.87 HP) Vibration Reading During Steady-State Conditions.

CONCLUSION

The first subsea module will follow a technological qualification process under real operational conditions in a subsea gas field. Initially, the module will be installed on a platform to perform prequalification tests. After the prequalification phase, it will then be submerged to a depth of approximately 300 m (984.25 ft) (Figure 11), and the qualification protocol will then be continued.

In the meantime, feasibility studies with some oil companies have already been performed for several potential opportunities in the North Sea, for both gas boosting and gas reinjection.

Looking at the offshore future and the deepwater fields assets, current technological developments are leading the increase of activities on the seabed rather than ones on the platform. This is clearly a tough challenge, but it can represent a real breakthrough with a mutual benefit for the years of production to come.

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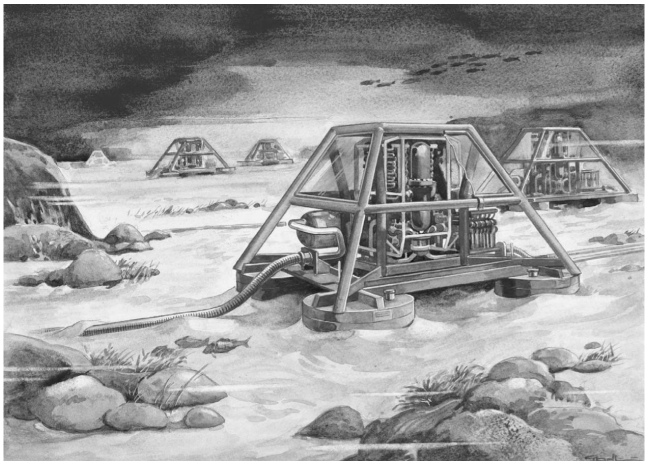

Figure 11. 2.5 MW (3352.56 HP) Subsea Module.

Development Manager Subsea Rotating Equipment) and Mehdi Navidi (Kvaerner Eureka Project Manager of the Subsea Centrifugal Compressor Module).