DESIGN AND VERIFICATION TESTING OF NEW BALANCE PISTON FOR HIGH BOOST MULTIPHASE PUMPS

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ABSTRACT

In 2011, the first Subsea Multiphase Pump (MPP) able to deliver a differential pressure of 2175 psi (150 bar) was introduced to the market. This major innovation for the subsea industry conquered the 17 year old 725 psi (50 bar) delta pressure barrier for existing multiphase pump technology. This achievement was made possible by the successful development of the first subsea MPP equipped with a balance piston, a technology which allows compensation of the axial thrust generated by the impellers, thus opening the way to pumps with high pressure differentials.

This paper presents the background for the design of a balance piston that is required to function with a mixture of gas, water, viscous crude oil and sand, and also the rotordynamic issues related to pumping this type of raw process fluid. The paper also addresses the test set-up as well as the results from the verification testing of the rotordynamic behavior with a full scale 2400 BHP (1.8 MW) high boost multiphase pump.

INTRODUCTION

Multiphase pumping on the sea bed is gradually becoming the most efficient way to produce deep offshore oil & gas fields. However, operators are now facing new challenges as the future subsea fields will be more difficult to produce due to remote locations, increased water depth, and higher viscosity of the process fluids. There is now an increasing demand from the industry to develop pumping systems with larger boosting capabilities, as high as 2900 psi (200 bar) and beyond. Multiphase pump (MPP) manufacturers are being challenged by the O&G Majors into bringing the required technology to the market.

The paper initially presents the background for the selection of the balance piston solution, given the design requirements. For MPPs these are generally spanning over a wider range than that commonly considered for conventional pumps or compressors. Further challenges are related to the varying Gas Volume Fraction (GVF), viscous multiphase fluid, expansion of the gas phase over the balance piston, leakage rates, heat generation and not least rotordynamics during the different operating conditions.

An important part of the design process was to develop a good simulation model which allowed considering both the multiphase fluid behavior as well as the complex balance piston geometries, with respect to mechanical, thermodynamic, sand tolerance and rotordynamic aspects. Several balance piston geometries were considered and various solutions tested.

The paper addresses the test set-up as well as the results from the verification testing of the full scale 2400 BHP (1.8 MW) high boost helicoaxial multiphase pump (Figure 1 and 2).
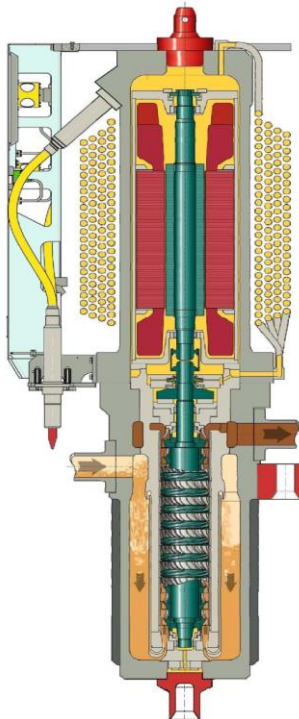

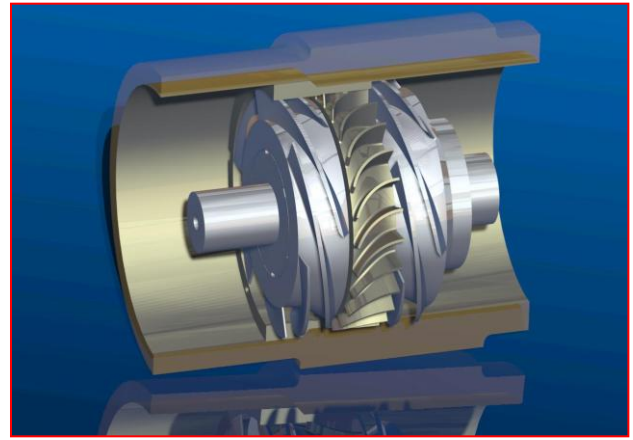Figure 1. Illustration of a Multiphase Pump

Figure 2. Helico Axial Pump Hydraulics
(Poseidon Technology – IFP/Total/Statoil)

THRUST BALANCING

The axial force due to the thrust load is a major challenge in the design of a multiphase pump providing a high differential pressure. If all the impellers of a multistage pump face in the same direction, the total theoretical hydraulic axial thrust acting towards the suction end of the pump will be the sum of the individual impellers. The unbalanced resultant axial force must be counteracted mechanically and/or hydraulically. The mechanical thrust load absorbing device is usually in the form of a thrust bearing. However for the differential pressure and forces in question, this would require a thrust bearing out of proportion both structurally and with respect to rotordynamic effects. Some form of axial thrust balancing must therefore be applied. To handle the thrust induced by impellers providing a differential pressure of 2175 psi (150 bar) or more, various possibilities have been screened:

- One option was to develop a pump with impellers in a back-to-back arrangement. This was abandoned due to the high level of complexity of this solution for a multiphase pump where the volumetric flow reduces with the pressure ratio. Each impeller thus operates with different relative flow, and doesn't produce identical axial force. These forces also vary with GVF.

- Another option was to design a MPP with low diameter impellers, running at very high speed. This was also abandoned because the flow would have been limited, while challenges related to rotordynamic performance and increasing wear rate would become more critical.

- The third option considered was to develop a pumping system with two conventional "Medium Boost" MPP's in series. This was not considered further due to the additional complexity related to this solution.

The principle of thrust balancing by utilising a balance piston was thus selected (Figure 3). However, the general rules for balance piston design were not sufficient to finalize the design. The operating conditions of the balance piston for a MPP were not comparable with the conventional design requirements for a single phase liquid pump. Therefore the detailed balance piston design was split into several critical design parameters, and each one carefully assessed.
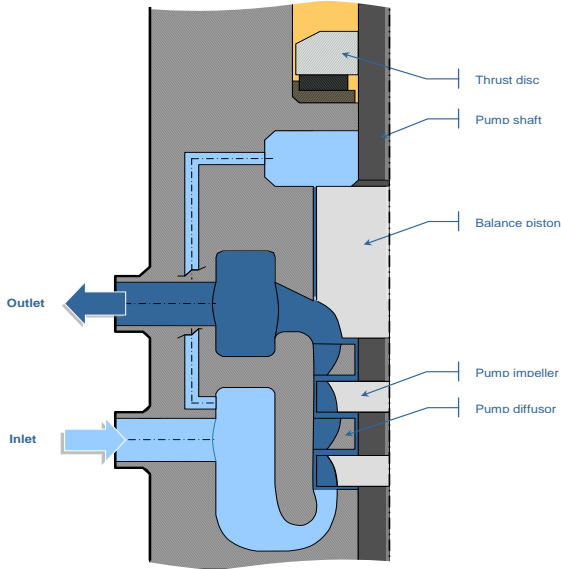

Figure 3. View of the Balance Piston

The first design parameter to take into consideration is the balance piston diameter, where two main requirements are present. The diameter is selected in order to limit the thrust forces at high differential pressure. From this requirement a minimum diameter is identified. The other requirement is to avoid negative thrust forces. These can potentially appear when operating at low differential pressures, hence a maximum diameter is identified.

Figure 4 shows a typical residual thrust profile. In order to secure reliable rotordynamics and avoid axial movement of the shaft, a positive residual thrust is secured over the full operating envelope.
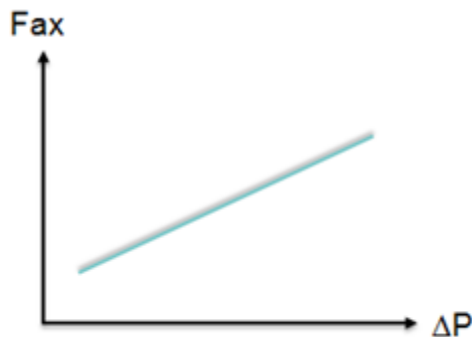

Figure 4. Typical Residual Thrust Profile

The diameter was selected in the upper part of the allowable diameter range in order to have a margin on thrust forces at high differential pressure, and also to have the possibility to increase the differential pressure beyond the base case limits.

MINIMIZED LEAKAGE LOSSES AND TEMPERATURE EFFECT

The target for fluid leakage rates past the balance piston was less than 10 percent of the main flow for operation at the required differential pressure and for the expected level of gas fraction in the fluid. At the same time the pump shall be able to run at high speed at a low differential pressure without risk of high temperatures or rotordynamic instabilities due to low flowrate through the balance piston. In a classical single phase application, the leakage rate through the balance piston is controlled by the main parameters length, diameter, clearance and wall surface roughness. For multiphase flow the volumetric leakage rates will change significantly with the level of GVF in the fluid due to the different densities and viscosities of the phases. Several effects were identified as a consequence of operating at different GVF's. An obvious benefit of minimizing the GVF in the balance piston is to reduce leakage rates. On the other hand the liquid rich part of a multiphase flow also includes the majority of particles in the fluid and this can lead to undesirable wear rates. By maximizing the GVF in the balance piston, the risk of particles is expected to be negligible but the leakage rates will be unacceptable during normal operation. The target is therefore to achieve the same GVF in the balance piston as for the main flow in the multiphase pump. From a thermodynamic point of view this is also beneficial as fluid flow past the balance piston will be maintained at all operating conditions. This provides cooling even in extreme operating conditions with pure gas/low differential pressure as well as with low GVF/high differential pressure.

Studies of different surface roughness were reviewed and none of the conventional hole pattern or honey comb designs were found to be robust enough to withstand solid particles or multiphase fluid. As both hole pattern or honey comb designs will lead to particle accumulation or liquid accumulation, only marginal benefits were found for these concepts. Therefore a smooth wall surface was selected to ensure a robust design.

To identify where best to place the balance piston inlet, CFD calculations were undertaken to simulate the balance piston inlet, and to determine the liquid holdup and particle path through the pump outlet section. The CFD model included the last impeller and the last diffuser to create the exact flow pattern in the area of the balance piston inlet (Figure 5).
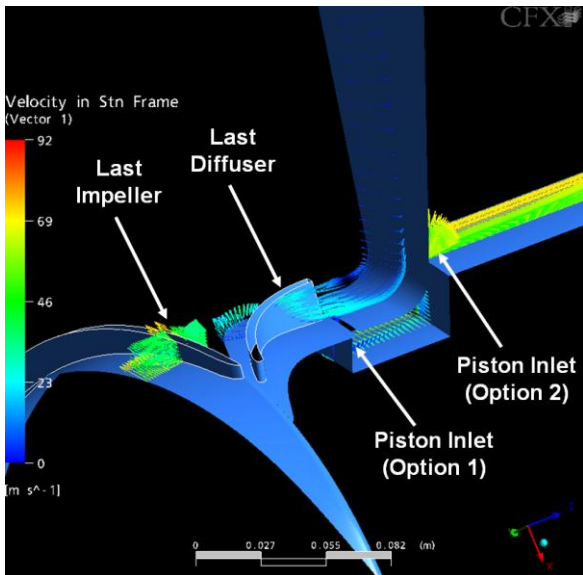
Figure 5. View of the CFD model

For these simulations, the Ansys CFX software tool was used. This software facilitates several multiphase modeling alternatives. Due to the high velocity gradients, the "homogenous" multiphase model was used. The simulated fluids were both live fluids and test fluids as used in the test loop. In the test loop water and nitrogen were mainly used, but for higher viscosities, mineral oil and air were used. The geometry of the model includes both large cavities, such as the pump outlet and small clearances as for the balance piston. The mesh had to be modeled carefully in order to be able to detect the large variations in pressure and velocity gradients. The mesh was generated by use of the Ansys ICEM blocking software, making it possible to use pure hexa mesh for the complete geometry. For turbulence modeling, the SST model was used thereby taking advantage of both the k-ω and the k-ε turbulence model.

The results from the multiphase simulations (Figure 6) clearly showed the distribution of the "liquid rich" and "gas rich" areas at the exit of the last diffuser, thus enabling correct design of the balance piston inlet with respect to the required GVF.
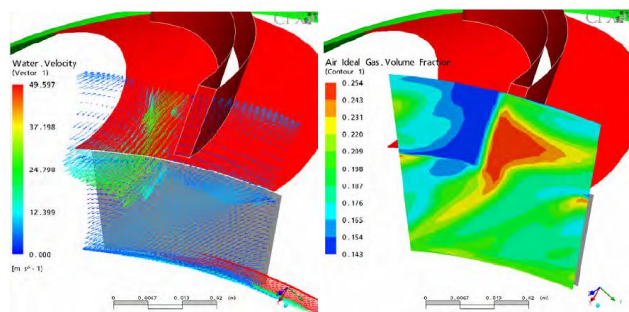

Figure 6. View of the blue "Liquid Rich" and red "Gas Rich" zones

WEAR MITIGATION

The need to create a design that will tolerate particles and deposits in the fluid stream was highlighted from the project start. It was therefore obvious that wear resistance had to be evaluated. Materials were selected to maximize wear resistance. Static parts were made of solid tungsten carbide, while rotating surfaces were coated with tungsten carbide. However, it was not sufficient to only look at the balance piston design with respect to achieving 100% wear mitigation. The number of particles entering the balance piston had to be minimized to reduce wear. Therefore a study of flow conditions upstream the balance piston was used to address this issue. CFD calculation was carried out, with particle tracking (Figure 7), for different particle sizes with a constant particle concentration of 200 ppm. A solution for minimizing wear was found by taking advantage of the centrifugal forces in the fluid swirl just downstream the last impeller. The fluid swirl combined with the selected diffuser design ensured a high particle concentration at the external diameter of the flow path. Hence the major part of the particles passed by the balance piston inlet and followed the main stream into the pump outlet cavity.
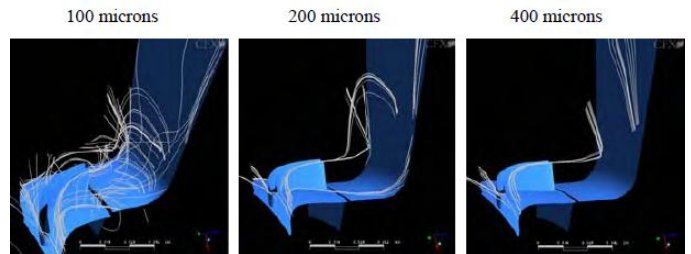

Figure 7. CFD with Particle Tracking (3 particle sizes)

The standard method for selecting a swirl brake design was found to be too vulnerable to erosion and abrasion for this application. A separate study was performed focusing on a new swirl brake design to avoid thin walled swirl brake segments but still achieving the required swirl control. After several concept suggestions, a specific design was found which met the requirements without compromising the swirl control (Figure 8). A swirl factor of zero was achieved for several operating conditions.
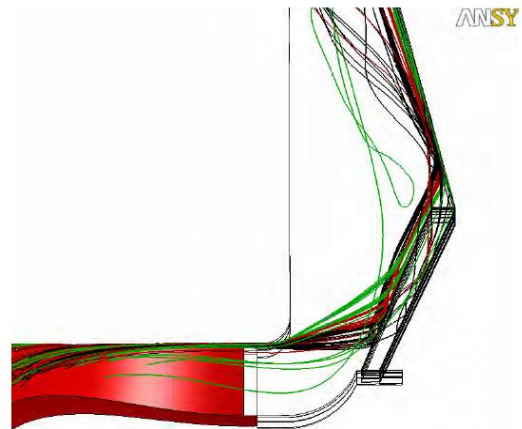

Figure 8. CFD with Particle Tracking on the Inlet Swirl Brake

ROTOR DYNAMIC STABILITY

The most important part of the balance piston design was to ensure stable rotordynamic performance of the balance piston and to identify how the balance piston parameters would affect the whole rotor assembly when operating at any of the specified conditions over the full range of fluid mixtures. Due to the relatively large dimensions of the balance piston, its rotordynamic parameters could have a significant impact on the shaft-bearing system.

As for all balance piston designs it is fluid induced forces that dominate the rotordynamic performance. With the large range of possible fluid compositions, gas fractions and differential pressures, this factor implied a large variation of rotordynamic performance for the balance piston for the different operating conditions. In addition, the requirements for thrust balancing and leakage rate control, resulted in a relatively large L/d ratio for the balance piston.

The main objective, in rotordynamic terms, for the balance piston was to define the critical parameters required to achieve a robust design, and to be able to document and validate the stable performance with both simulations and testing. As neither the fluid mixture nor the selected geometry could be called conventional for a balance piston, it rapidly became apparent that it was not possible to use conventional simulation tools to validate the selected design. The typical bulk flow simulation models are not suitable for simulating multiphase fluids or complex geometries. Therefore several CFD based simulation tools were used to validate and optimize both the various concepts and specific final designs. The benefit of CFD methods is that both specific fluid properties and specific geometries are taken into consideration. Some of the CFD calculations were quite time consuming, and a separate simulation tool to check leakage rates and thermodynamic results was therefore developed to be able to check a wider range of operating points.

With respect to rotordynamic stability, the target for the balance piston design was to reduce the large cross coupled stiffness that is typical for high L/d ratios, and to increase the direct stiffness by the means of clearance profiles and balance piston inlet design. As a starting point, the inlet area of the balance piston was simulated in a CFD model including the impeller and diffuser upstream the balance piston. This was to ensure that the correct GVF was entering the balance piston and also to ensure that the majority of the particles follow the main stream and do not pass through the balance piston.

After verifying the inlet conditions, the next step was to validate the swirl brake design by simulating the local flow pattern around a set of swirl brake teeth. A well designed inlet with a swirl factor close to zero maximized the Lomakin effect and hence contributed to optimised direct stiffness for the balance piston.

A remedy for the adverse effects of the high L/d ratio was to split the piston into three independent segments (Figure 9) and thereby achieve a lower L/d ratio for each of the segments. As the cross coupled forces increase with a factor of approximately three with increasing L/d, it was beneficial to have three balance pistons of reduced L/d rather than one balance piston with high L/d. The segments could be defined as rotordynamically independent due to a cavity, including swirl brakes, implemented between each segment. The cavities stabilize the pressure field in the circumferential direction and hence suppress the Bernoulli effect.
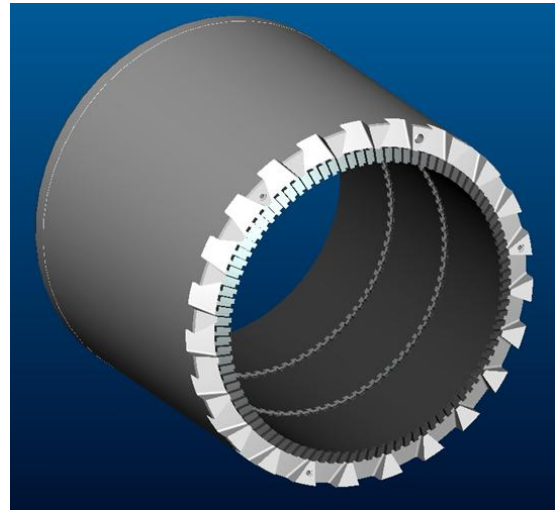

Figure 9. Balance Piston Liner in Three Segments

After validating that the swirl brake design had an acceptable velocity field and inlet pressure drop, the complete balance piston assembly, including all three segments, was simulated in a larger CFD model. This simulation was performed with a perturbation method designed to extract the rotordynamic parameters from the fluid pressure fields as the shaft was excited in an eccentric position. The most used simulation method is the IPM method where the shaft undergoes vibration excitation at several whirl frequencies. Several sensitivity analyses had to be performed to document the optimum geometry. One example of such a study can be seen in Figure 10. The cross coupled stiffness for the three segments was found for two different intermediate cavity designs. As can be seen, the improvement for case two is seen as both reduced cross coupled stiffness and equal values for all three segments. The equal values represent a fully de-coupled pressure field for the segments, and the target of independent segments was thus achieved.
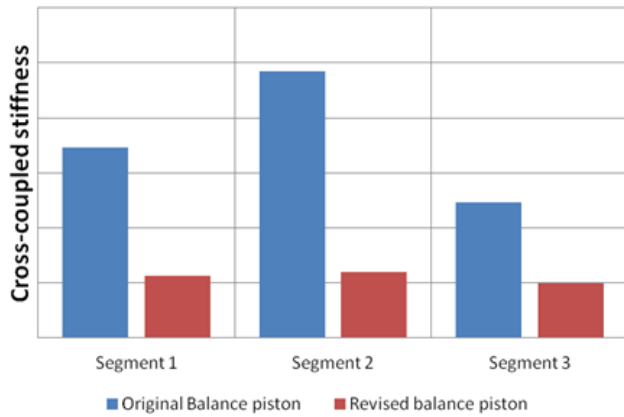
Figure 10. Cross Coupled Stiffness for Two Different Balance Piston Cavity Designs

As the balance piston now had three independent segments it also had three inlets with low swirl factor. This resulted in a direct stiffness that was almost three times higher than for a balance piston with only one segment.

The presence of multiphase flow in the balance piston also raised other concerns that had to be addressed. With a homogenous multiphase fluid entering the balance piston clearance the fluid is exposed to significant centrifugal forces which will result in phase separation after a given axial distance into the clearance. The liquid phase is forced out to a high diameter and is mainly covering the static surface. The light gas is thereby mainly covering the inner rotating surface. This effect increases with increasing liquid viscosity as the Reynolds number is reduced and laminar flow can be expected. There are several disadvantages of this phenomenon but the most critical will be non-linear stiffness effects. For a given level of shaft eccentricities the rotating surface will "hit" the liquid rich area and will suddenly be exposed to a fluid with totally different viscosity and density. This will give a step change, especially in cross coupled stiffness, and can result in uncontrolled rotordynamic behavior. The solution here was to control the multiphase fluid mix and ensure a homogenous mixture. For example, each segment is made short enough to avoid significant phase separation and the intermediate swirl brakes and the swirl brake cavities ensure good fluid mixing before the fluid is enters the next segment.

TESTING / VERIFICATION

During the design and analysis of the balance piston, several critical parameters regarding inlet conditions were found to have an impact on piston performance. By building a test-rig containing only the balance piston, these parameters would not be verified during testing. Also the high pressure required during testing, would result in a test set up with both a compressor and pump to deliver the high pressures for both gas and liquid supply lines. The best test set-up was found to be a full scale 2400 BHP (1.8 MW) multiphase pump, with 13 impellers of 12 inches (305mm) OD, running at 4600 rpm, and fitted with the balance piston. Hence achieving the correct inlet conditions and the required pressures at balance piston.

The test loop can be seen in Figure 11 with the single phase measurement sections, mixing point, test object, choke valve and separator. The special feature for this test was the introduction of a multiphase meter at the balance piston outlet/return piping. The flow meter facilitates accurate measurements of gas and liquid volume fractions passing through the balance piston. Hence the simulated performance could be verified through the test program. A multiphase flow meter was also used at the liquid line to measure any gas carry over present in the liquid line due to reduced separation capacity. These additional flow meters were critical to the verification of accurate and stable flow conditions.
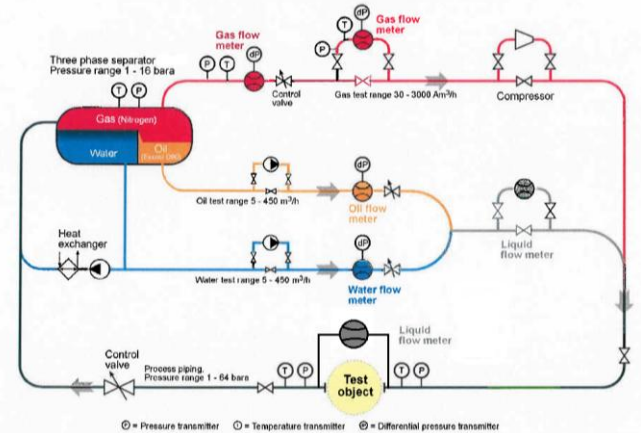

Figure 11. Test Loop and Test Set-up

The pump casing (Figure 12) was equipped with housing vibration probes at both ends of the pump. Pump shaft relative vibration was measured with proximity probes at both ends of pump shaft, close to the radial tilt pad bearings. A key phasor probe, at the shaft non drive end, was also used for accurate speed measurement and vibration data processing.
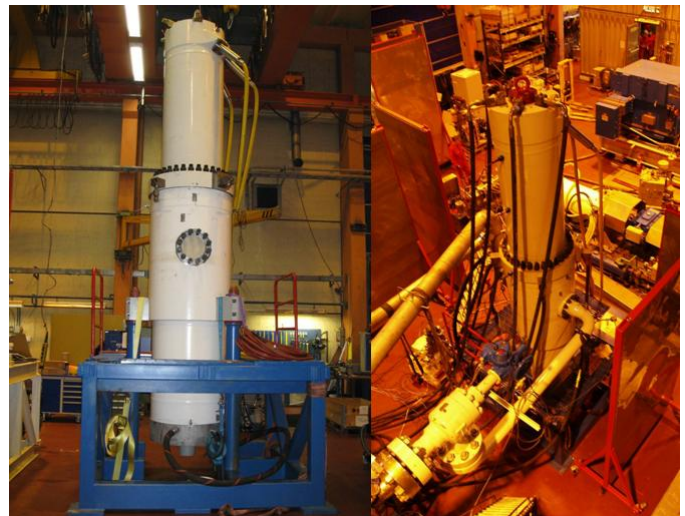

Figure 12. Test MPP installed in the Multiphase Loop

The base case target test conditions were stated early in the project to be operation at 2175 psi (150 bar) differential pressure when running with multiphase fluid with 30 percent

GVF and low viscosity liquid. Testing showed a good hydraulic performance even at high gas volume fractions (Figure 13).
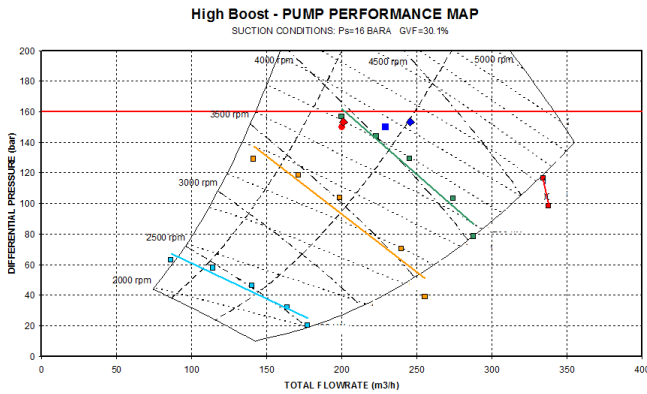

Figure 13. Performance Map for GVF 30%

The target flow rates and gas volume fractions in the balance piston were met and matched well with simulation results. Figure 14 shows gas volume fractions at pump inlet and in the balance piston. The test also shows close to identical GVF at pump inlet and for the flow passing through the balance piston. Only at test points with high GVF, can a slightly lower GVF in the balance piston be seen.
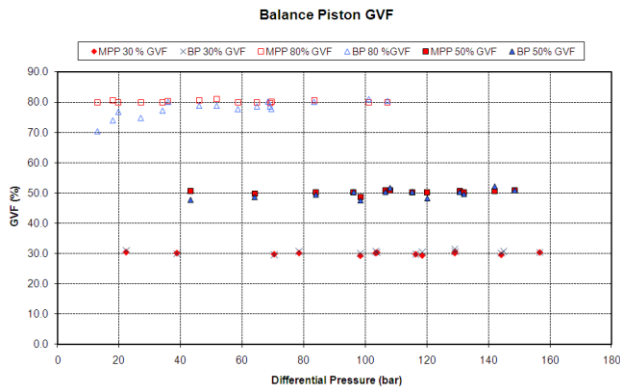

Figure 14. GVF passing through the Balance Piston vs GVF at Pump Inlet

The base case operational requirements were met with good results for both hydraulic, rotordynamic and mechanical performance. All parties in the development project were eager to take the qualification project an important step further by qualifying both the design and simulation tools to also be valid for multiphase flow with higher viscosities.

One of the challenges then became thermal effects due to higher viscous losses. The balance piston axial length could be reduced and/or the clearance could be increased to mitigate this effect. Such changes would be acceptable as leakage rates will in any case decrease with increasing viscosity. The more demanding consequence of high viscosity was the changes in rotordynamic characteristics due to significant changes in fluid induced forces. The stabilizing Lomakin effect is dependent on

having a significant inlet pressure drop at the inlet of each segment. The inlet pressure drop would now be marginalized at the expense of increased frictional losses in the clearance. The de-stabilizing Bernoulli Effect will be more prominent with increased viscosity/shear forces. As mentioned above, the risk of phase separation will increase.

Testing showed the expected decrease in balance piston leakage flow and also the increase in outlet temperature due to higher fluid shear forces. The pump performance was mapped for a large variety of speeds, differential pressures, GVFs and viscosities. The pump and balance piston were found to be rotordynamically stable for most areas of the operating envelope, but rotordynamic instabilities were found for certain operating conditions. Figure 15 shows the pump operating envelope for the full speed range 1500 to 4600 rpm. Some sub synchronous vibrations were found with multiphase flow and for relatively low differential pressures. In conditions with sub synchronous vibrations, the balance piston rotordynamic parameters show a whirl frequency ratio of more than 1 and the mode shape with negative damping ratio in Figure 16 shows highest amplitude at the drive end. This indicates a de-stabilization force at the balance piston location on the pump shaft. The test thus gave useful information regarding the effects of multiphase flow and high viscous fluids. Important information regarding "worst case" operating conditions was now documented. After the test period, the final design for high viscosity applications was completed.
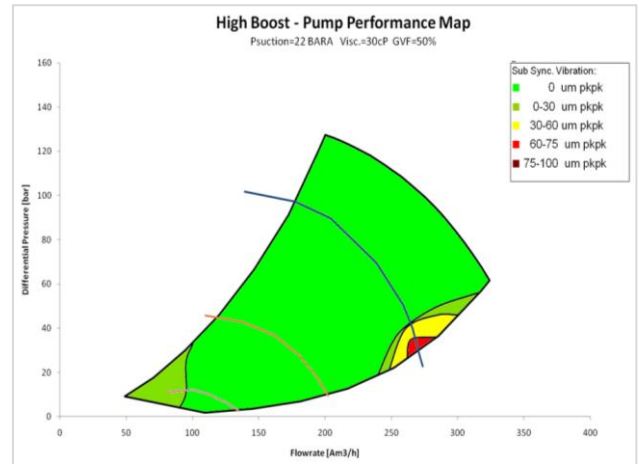

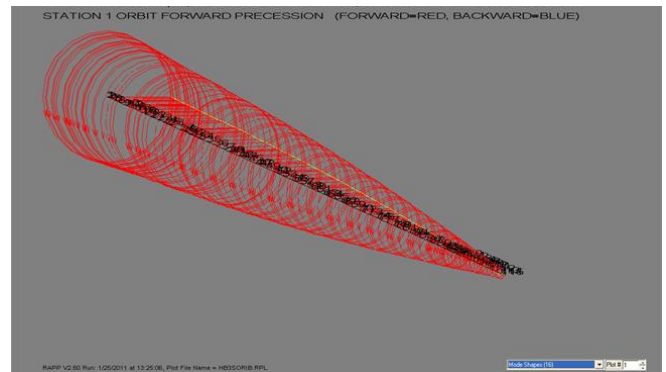Figure 15. Subsynchronous Instabilities – Transverse Mode

Figure 16. Mode Shape - Subsynchronous Instabilities

The balance piston design for higher viscosities was simulated with acceptable damping factors and tested with stable rotordynamic performance as shown in Figure 17.
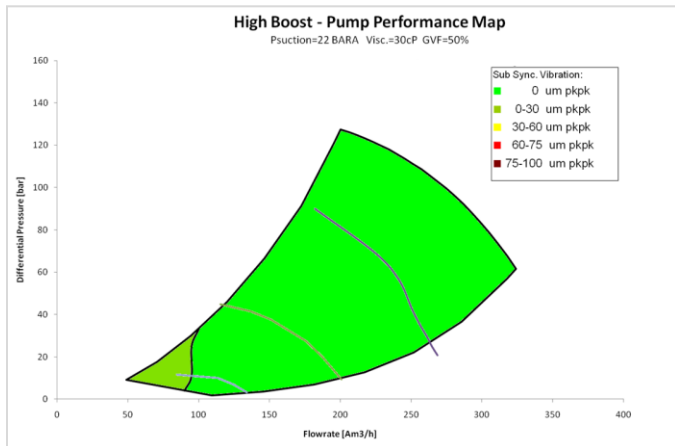

Figure 17. Operating Envelope with Stable Rotordynamic Performance.

To demonstrate stable performance at worn conditions, a balance piston with larger clearances and worn swirl brakes was tested and showed stable operation. Results verified that a critical change in rotordynamic behavior is not present for worn conditions.

SUMMARY / CONCLUSION

From a hydraulic performance standpoint, the high boost MPP has largely met the target by generating a differential pressure of 2175 psi (150 bar), with an inlet GVF up to 60%.

From a rotordynamic perspective, the design phase of delivering a full scale high boost MPP with a balance piston, has revealed a wide range of parameters to keep under control. A MPP operates in an untreated well stream and is exposed to a large variety of conditions. These have to be taken into consideration to achieve a robust balance piston design. It is therefore essential to have a system approach when performing such a design. A key element of this successful development was that it has been driven both by the Pump Designer and the End Users (as part of the JIP). For the Pump Designer, this ensured that all relevant design requirements coming from the process and the operations were identified. For the End Users, this was a guarantee that the qualified equipment will fulfill the required flexibility, robustness and reliability, knowing that a subsea pump shall have at least 5 years interval between regular maintenance.

Due to the required complex geometries and the challenging process conditions, it has been found that existing bulk flow models cannot be used to model the present design. A new design approach, with new simulation methods, has been verified during this development phase and will also be used in upcoming commercial pump applications.

NOMENCLATURE

CFD	:	Computational Fluid Dynamics
FAT	:	Factory Acceptance Test.
GVF	:	Gas Volume Fraction.
IPM	:	In-stationary Perturbation Method
L/d	:	Length-diameter ratio
MPP	:	Multiphase Pump
MW	:	Megawatt
O&G	:	Oil and Gas

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