45TH TURBOMACHINERY & 32ND PUMP SYMPOSIA
HOUSTON, TEXAS | SEPTEMBER 12 – 15, 2016
GEORGE R. BROWN CONVENTION CENTER

Subsea Compression – Current Technology and its Use to Maximize Late Life Production

Markus Dettwyler
New Technologies
MAN Diesel & Turbo Schweiz AG
Zürich, Switzerland

Dirk Büche
New Technologies
MAN Diesel & Turbo Schweiz AG
Zürich, Switzerland

Urs Baumann
Head of Calculation and Development
MAN Diesel & Turbo Schweiz AG
Zürich, Switzerland

Markus Dettwyler graduated from the Swiss Federal Institute of Technology (ETH) in 1999 in Mechanical Engineering focusing on Turbomachinery and Lightweight Construction. Following his Dr. degree in Mechanical Engineering in the fields of Biomedical Engineering, he gained his first professional experience in the field of piezoelectric sensors as a team leader for pressure sensor development for commercial in-cylinder high-pressure monitoring sensors for large Diesel engines.
In 2010, Markus Dettwyler joined MAN Diesel & Turbo Schweiz AG as a team leader in Basic Product Development. Starting from 2011, he took over the position of Technical Lead Engineer for the detail design and execution of the world's first subsea compressor which took up commercial production in Sep. 2015 on the Norwegian Continental Shelf. He currently leads the New Technologies group at MAN Diesel & Turbo.

Dirk Büche received the Diploma degree in Aerospace Engineering from the University of Stuttgart, Germany, in 1999 and the Dr. degree in Mechanical Engineering from the Swiss Federal Institute of Technology (ETH) in 2003.
He taught various courses at the University of Applied Sciences Northwestern Switzerland (FHNW) in the field of Thermal Fluid Engineering and Information Sciences. He led various research cooperation projects between industry and academia.
In 2007, he joined MAN Diesel & Turbo Schweiz AG for the development of radial compressor stages. In 2010, he took over the lead for the mechanical and thermodynamic design of the world's first subsea compression project and now works on development projects in New Technologies. His fields of expertise are axial and radial compressors, system engineering, as well as thermo-mechanical simulations and numerical methods in general.

Urs Baumann is the Manager of the Calculation and Development department of MAN Diesel & Turbo Schweiz AG in Zurich, Switzerland. His responsibilities include the aerodynamic as well as the mechanical development and improvement of turbo compressors and associated components, as well as the implementation and maintenance of test stands and analytical tools needed to fulfill this task. His department also comprises a Product Development Group mainly focusing on high-speed motor-driven, magnetically suspended compressors. He is the owner of several patents related to high-pressure compression and high-speed oil-free motor compressors.
Before joining MAN Diesel & Turbo in 1996, Mr. Baumann worked for Sulzer Innotec, the Corporate Research and Development Center. For several years he was in charge of the machinery dynamics group which is responsible for the development, design improvement and troubleshooting on a wide range of Sulzer products.
Urs Baumann has a diploma (Mechanical Engineering, 1987) from the Swiss Federal Institute of Technology (ETH).

ABSTRACT

Oil and gas production from offshore fields has been established since the second half of the last century. The continuous need and search for new fossil resources has driven the technology to adapt towards harsher and more hostile exploration environments. Subsea processing and production facilities are the consequent continuation of this trend. Furthermore, adding compression for depleting fields boosts the production and may extend it for several years. As an efficient alternative to a classical top-side compression installation, the first subsea compressors are in operation.

This paper starts by illustrating the current state of technology in subsea compression. Background information as well as historical developments for the world's first subsea compressor are outlined. The concept relies on a hermetically-sealed compressor with an integrated electric motor and an active magnetic bearing system. This has been a standard solution for topside applications for almost 30 years and forms an ideal design basis for the subsea motor-compressor.

Copyright© 2016 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

In a second step, different approaches for multiphase subsea compression are compared. These are (1) separate compression and pumping of gas and liquid phases, (2) well-stream compression, and (3) multiphase pumping. Limitations of the concepts are outlined. While for compressors, well-streams are typically characterized by the liquid mass fraction (LMF), pumping literature uses gas volume fraction (GVF). The relation between the two parameters is discussed as well as their influence on compression/pumping.

Finally, an example of late life gas field production is given. The evolution of the required compression and pumping is estimated and the importance of the compressor selection in maximizing total field recovery is discussed.

INTRODUCTION

In the search for new sources of energy offshore oil and gas production became state-of-the-art in the middle of the 20th century. Starting from relatively shallow waters at the beginning, maturing technologies made the move towards increased water depths possible, allowing the effective tapping of energy resources that otherwise would have remained inaccessible. These days, subsea exploration activities can be found all around the world. Typical offshore sites are located in the North Sea and the Norwegian Sea, the Gulf of Mexico, east of Brazil, west of Australia and west of Africa. An area of big future development potential is the Barents Sea, where the availability of subsea processing technology is a clear and inevitable prerequisite for exploration. One can say today that subsea processing technologies have reached a sound level of technological readiness and are expected to become even more widely used in the oil and gas industry in the decades to come.

One of the main reasons why subsea processing technology is a key element in developing new exploitation opportunities is the fact that new discoveries of fossil resources are increasingly being made in more and more hostile environments with no easy access. The need for remote, fully automated operation, unmanned whenever and wherever possible, is clear and well recognized. Established offshore platform concepts become less attractive with increasing distances from shore.

Compression in Natural Gas Production

Many reservoirs contain natural gas as a predominant component. As briefly explained, there are strong arguments for the laborious placing of process equipment – including subsea gas compression – on the sea floor.

Natural gas, as it comes out of the well on the sea bed, normally contains a certain amount of liquid oil, also referred to as condensate, some amount of water and sometimes added hydrate preventers (e.g. MEG). These liquid components are typically in the range of up to 30 percent in mass for a natural gas field. In addition to the natural gas itself these liquids also need to be conveyed and processed. All equipment and systems need be designed accordingly to cope with this fact. This holds true for flowlines as well as for process installations on platforms, FPSOs and at land-based receiving facilities. Often the liquid and gas phases are separated and processed separately, which generally requires a tremendous technological effort.

In the case that natural gas and liquids are being transported in a common pipeline, which is an attractive solution to keep investments low, the flow regime within this pipe is of multiphase nature. Below a certain minimum flow velocity, the flow becomes increasingly unstable and undesired slugs can form within the pipeline (Ramberg and Hedne, 2016). The exact criteria determining when this happens vary from field to field and depend on specific variables such as pressure, temperature, pipeline diameter, and composition of the gas and liquid. The problem of slug flow is especially pronounced in the riser part of the flowline. Other undesirable phenomena such as natural gas hydrate formation are also adversely intensified by declining gas velocities and associated low temperatures in the pipeline. The avoidance of premature flow-velocity-decline is normally referred to as flow assurance and is accomplished by compressing the gas at a certain point in the pipeline, most beneficially close to the well. By thus increasing flow velocities and pressures in the pipeline the limit imposed by the incidence of unstable flow regimes can effectively be delayed. The production and hence the lifetime of the field is thus prolonged in an effective manner. Not only can the total amount of gas and condensate that can be extracted from the field (overall field recovery factor) be increased, but also the rate of production is enhanced. This benefit in addition to the physical need to maintain stable flow conditions underscores the economic advantages of gas compression and broadens the set of economically justifiable explorations.

For most subsea gas fields, compression becomes important at the end of the field life when the natural reservoir pressure decreases. At the beginning of production, the natural pressure in the gas reservoir is typically still sufficient to push gas and liquids from the wells to an onshore processing location.

Compressors may be installed either topside or subsea as depicted in Figure 1. Topside compression, as was the traditional technical approach up until now, requires platforms or FPSOs, which are large and costly constructions that require frequent maintenance activities and safety provisions. Even though there is a lot of equipment that needs to be submerged in the case of subsea compression, it remains an economically viable solution, both in terms of investment (CAPEX) and operational cost (OPEX). This can be explained by the fact that a substantial portion of the costly topside equipment becomes

Copyright© 2016 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

unnecessary, the subsea equipment is positioned directly on the seabed without the need for floating structures and remote operation requires no personnel with special offshore certificates on site. The reduction of on-site personnel is in any case a current trend that can be clearly observed at all operators. A further benefit of locating the compressors on the seabed, close to the well, is that the density of the gas in the pipeline is higher, the necessary flow velocities are lower, and thus smaller pipe diameters can be selected for new field developments reducing CAPEX.

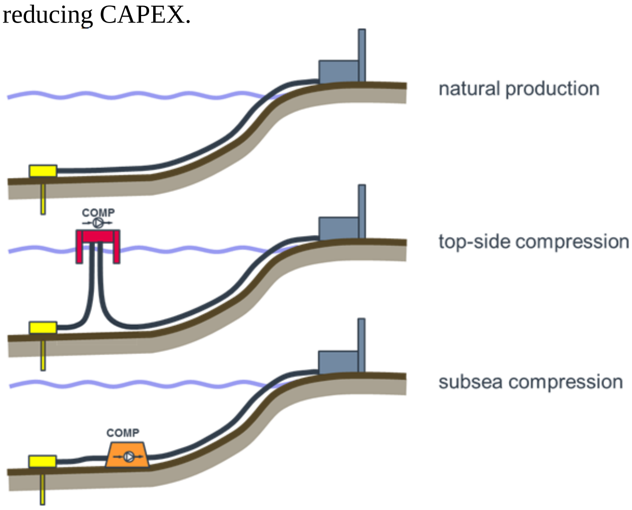

Figure 1: Top: Natural Production from a Subsea Natural Gas Well (yellow) to an Onshore Receiving Facility (gray). Middle: Topside Compression with Riser from the Subsea Well to the Topside Compressors on a Platform (red). Bottom: Subsea Compression with Compressor Station (orange) Located Close to the Subsea Well.

Needless to say, subsea compression can be applied both to already existing subsea fields (brown fields) and to fields yet to be developed (green fields). This gives this technology a great potential for the future.

ÅSGARD SUBSEA COMPRESSION, COMPRESSOR MODULE, CORE UNIT DESIGN

The successful uptake of subsea compression required intensive development and qualification activities that took into account the fierce conditions of submerged and remote operation. The author's company was involved into such a program in the Åsgard subsea project. The development and qualification activities are summarized in the following paragraphs.

Background and Historical Developments of the Core Unit Design

Early experience in wet gas compressor operation has been gained since the 1950s. At that time, solutions using centrifugal compressors for lean and rich CO_2 compression for soda ash production were established. As is typically the case for such applications, the high level of impurities in the gas necessitates permanent washing by water injection with a liquid mass fraction of up to 25 percent and drainage to prevent fouling in the compressor. As a direct result, and at this early stage, material selection was tailored towards these specific needs, qualified with respect to high erosion and corrosion resistance and finally successfully integrated in this harsh compression environment.

At the end of the 1980s, the first specific studies analyzed unmanned and remote offshore operations. Furthermore internal test series with wet gas compression were conducted in order to investigating the capabilities and restrictions of such applications (Casey, 1989).

In the 1990s, the concepts of remote operation were put into practice with the first oil-free compression installations. The first oil-free, hermetically sealed motor-compressor was installed in a gas transportation station as illustrated in Figure 2. The motor in this arrangement is directly coupled to the compressor shaft and is cooled with process gas. A separate gear box became obsolete and the whole shaft is supported by active magnetic bearings (AMB). This concept allows for the elimination of the lubrication oil system, requiring only electrical and process gas connection to the motor-compressor. Numerous commercial realizations followed in the years to come. Both the modular gas transport and gas storage product lines became a success. A detailed treatise on this machine concept can be found in (Kleynhans et al., 2005). Currently two motor frame sizes in the megawatt range are available from the authors' company.

Figure 2: First Integrated Pipeline Motor-Compressor at Station 100 (USA), 1990

Copyright© 2016 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

Driving Factors and Subsea Design Concept

When considering submerging and operating a piece of turbomachinery equipment in sea water several boundary conditions for the product are almost inherent. The use of a lubrication oil system is almost prohibitive, as is the use of gear boxes and shaft seals separating the process from the environment, which in this case is not air, but sea water. The selection of the mechanical drive is furthermore obviously restricted to electric motors. With the standard gas storage and gas transport product lines already available from topside applications, there was a seemingly perfect match with the new requirements. The compressor internals as well as the electric motor are hermetically encapsulated in a pressurized hull having only flanges for the process fluid and electrical penetrators for the main power supply lines, and for the active magnetic bearings (AMB). The presence of an AMB system provided the final essential pieces for the design of the subsea motor-compressor.

In the first decade of the 21st century, the author's company was approached by an oil company in order to explore the potential for realization of subsea compression. Front-end conceptual studies were launched and several machine arrangements were screened that could fulfil the requirements set by the process. In principle, the same process duty can be fulfilled with a small machine at higher speeds or with a larger machine at slower speeds. It was very clear from the beginning that the main focus needed to be put on maximum reasonably possible robustness of the machine as well as on reliable and safe operation. Even though there were benefits in terms of capital expenditures (CAPEX) and machine footprint (mainly size and weight) with the small, fast running concepts, the preference for a more robust, heavier, and slower-turning machine concept was clear, even at this early conceptual stage. OPEX are mainly driven by the price of fuel/electricity, the intensity of required service/maintenance activities and risk of loss of production, which is fundamentally related to machine availability. This is especially true in the case of subsea equipment where retrieving, repairing and re-installing is a very expensive scenario. Since there is little possible leverage on fuel/electricity costs, the main possibility to reduce OPEX is with a robust and reliable design. This went hand in hand with the low risk, slow-speed concept.

In order to reach the technological readiness level (TRL) required for realizing commercial subsea compression, a full-scale demonstrator was manufactured and thoroughly tested in close collaboration with the end-user (Kleynhans et al., 2016). The main focus areas of this testing and qualification campaign were:

- Material selection of metallic and organic materials, including the electrical insulation system of the AMB actuators and the electric motor
- Erosion testing of the impeller material
- Electric long cable step-out simulations
- Demonstration of compressor wet gas tolerance and performance testing
- Motor frame size verification

The involvement of the end-user into this product testing and improvement phase turned out to be very fruitful and opened new possibilities. Gradual design improvements were implemented where the original demonstrator showed shortcomings, followed by subsequent re-testing of the modification wherever possible. Alternative solutions were always kept ready in the background to be used if needed. Actual process conditions and the operation scenario were reproduced as closely as reasonably possible. The early testing and tailoring of the machine to the actual operation scenario proved to be very valuable at the end. A profoundly detailed understanding of the application specific requirements could be established early. Extensive testing campaigns were undertaken in the period between 2008 and 2011. More details of this conceptual product definition phase and the gradual design improvement steps can be found in (Kleynhans et al., 2016). These upfront testing campaigns took place on a slightly smaller machine size due to economic considerations. A final frame size verification campaign during the commercial project execution of Åsgard completed the technology qualification program (Figure 3).

Synergies from simultaneously ongoing qualification activities with other subsea technology qualification projects could be utilized since the end-user also was involved in parallel qualification activities with respect to marinization of the AMB electronics and HV-penetrators as part of the electric main power supply.

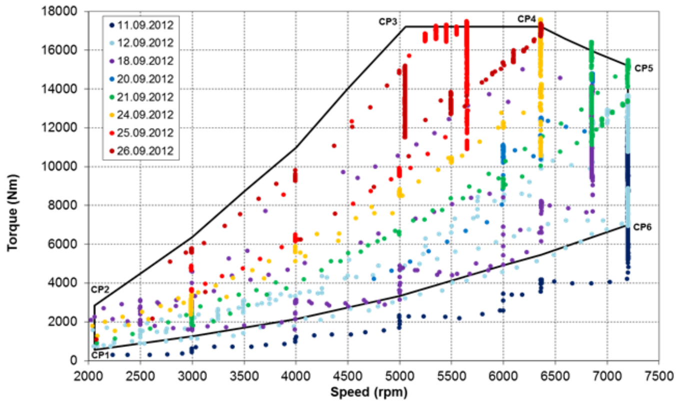

Figure 3: Subsea Motor Frame Size Verification 2012: Tested Torque vs. Speed Points. The Solid Line Represents the Åsgard Subsea Motor Envelope.

Copyright© 2016 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

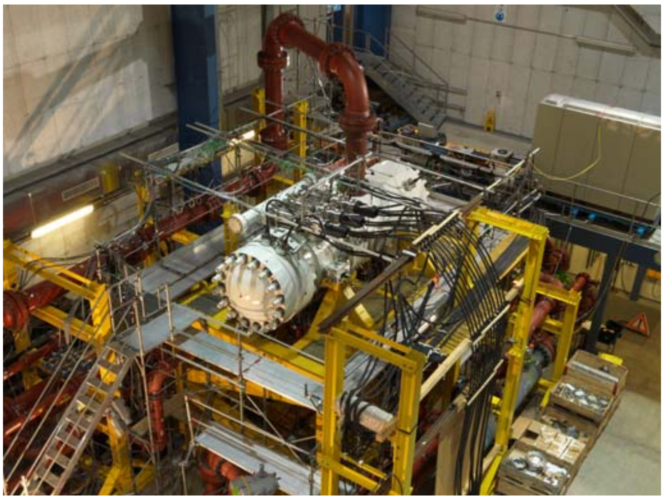

Figure 4: FAT-Testing at the Test Stand of the Authors' Company. The Motor-Compressor Unit is Mounted on a Tailored Test Frame in Order to Mimic the Later Subsea Module Stiffness of the Supports.

The qualification activities were finally rounded off with extensive pilot compression module submerged testing at the end-user testing facilities in Norway (Chellini, 2014). Once again, the submerged testing of the pilot module included real process gas conditions. Other focus areas were the operational envelope, wet gas tolerance, endurance testing, module vibration testing, motor cooling, motor compressor thrust and surge behavior, surge line identification, the trip sequence, and washing.

All delivered motor-compressor units had undergone thorough FAT testing before being delivered to the customer (Figure 4). One motor-compressor unit was even subjected to multiple drops at full speed and full load during FAT testing. This unit was disassembled and thoroughly inspected before being retested and delivered. A drop is a sudden power-off of the motor and the AMB system which causes the motor-rotor to "drop" into the backup roller bearings. These backup roller bearings allow safe short-term rundown in case of a system failure or of loss of control of the AMB system.

Åsgard System Description

The Åsgard production complex is situated approximately 200 km off the shore of Mid-Norway. Gas production through the semi-submergible platform Åsgard B, where incoming gas and condensate from the Mikel and Midgard field is processed began in 2000 (Beckman, 2015). With the addition of subsea compression, the field life can be prolonged until approximately 2030 and a surplus of approximately 300 MMboe is expected (Beckmann, 2015). The subsea compression station is installed at a water depth of approximately 300 m (1'000 ft). It consists of two identical compression trains operating in parallel. Each of the two trains consists of a variety of modules, such as an inlet cooler module, separator module, pump module, compressor-module, discharge cooler module, transformer module and other smaller modules (Kleynhans et al., 2016). Electric power is generated on the Åsgard A FPSO. The variable frequency drive (VFD) is situated on this vessel, the output of which is transformed to 33kV before being routed towards the subsea compression station 40 km (25 mi) away on the seabed. In the subsea compression station the voltage is then stepped-down before being fed into the compressor electric terminals. This, together with the relatively slow frequency of up 120 Hz allowed the cable losses to be kept at acceptable levels (Normann and Rongve, 2014).

The compressor module as depicted in Figure 5 weighs approximately 300 tons (660'000 lb). It is roughly 13m (43 ft) in height, 8m (26 ft) in width and 10 m (33 ft) in length. As well as the horizontal shaft motor-compressor unit itself, it also houses the AMB control pods, the motor cooling gas loop including a dedicated separator, an anti-surge valve, and various interfaces for process piping and electric power supply. The motor-compressor is gravity drained, which was achieved by arranging solely the suction process flange on the upper side of the motor-compressor. All other connections to the motor-compressor unit are located on the bottom side of the motor-compressor.

The key data of the motor-compressor are listed in Table 1. It consists of a 3-phase high-speed induction motor and a 7-stage inline barrel compressor. The compressor shaft is directly coupled to the electric motor shaft by means of a hydraulically mounted tapered bore rigid coupling. There are two radial magnetic bearings integrated in the electric motor and an additional one in the compressor. The axial magnetic bearing is located between the motor and the compressor. A tailor-made support structure on top of the motor casing houses the 3 dry-mateable power-penetrators. The electric motor has an internal star point, a Litz-wound stator with a specially customized insulation system and a solid rotor equipped with a copper squirrel cage.

Maximum shaft power	11.5 MW (15'400 hp)
Maximum shaft speed	7'200 rpm
Max coupling torque	17'200 Nm (12'700 ft·lbf)
Maximum supply voltage	7 kV
Maximum pressure ratio	2.8 [-]
Maximum volume flow	10'000 m^3/h (353'000 ft^3/h)
Weight	57 tons (126'000 lb)

Table 1: Key Data of the Subsea Motor-Compressor

Copyright© 2016 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

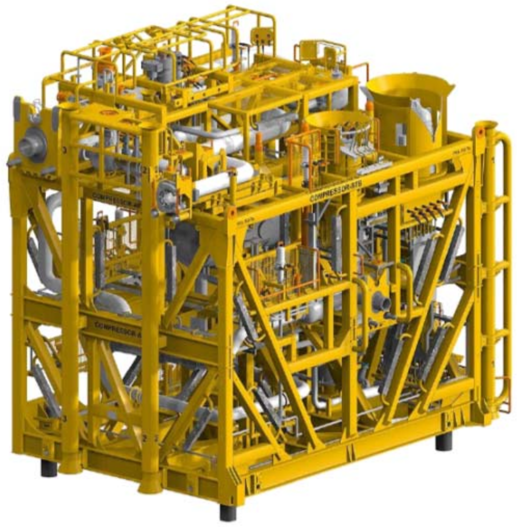

Figure 5: Åsgard Subsea Compression Module with Integrated Motor-Compressor (Vesterkjær, 2015, Courtesy of Aker Solutions)

Figure 6: Subsea Motor-Compressor Mounted on its Transportation Frame

Motor Cooling Gas System

The basic process flow diagram for the compressor module, including the cooling gas loop, is sketched in Figure 7. The electric motor is directly cooled with process gas. The cooling gas system is realized as an open loop. The motor is symmetrically cooled from both ends which results in a more effective utilization of the available cooling gas and, at the same time, leads to a symmetric temperature distribution in the motor.

The system is designed for maximum robustness and minimum complexity. The suction pipe (S) connects to the compressor from the top, the discharge pipe (D) from the bottom. A sophisticated inter-stage extraction after the second stage (1) delivers the cooling gas into the cooling gas loop. The compressor internal extraction was optimized such that only minimal amounts of liquid can enter into the cooling gas loop, especially in the case of online compressor washing. Nevertheless, a separator was included as a safe guard in the installation. After the cooling gas exits the separator (2), it passes a cooling gas valve (3) and is fed into the motor from both ends (4). A small fraction of the cooling gas is directly routed (7) towards the compressor radial AMB. The cooling gas return (5) from the motor, the leakage from the balance piston, and the drain from the separator, are joined and routed back into the compressor suction line without any further cooling or processing (6). This completes the open cooling gas loop.

The cooling gas extraction after the second stage turned out to be a good compromise, ensuring that enough driving pressure was available at the extraction, while not having to accept excessively high extraction temperatures or losses associated with excessive compression work for the cooling gas. Good cooling efficiency could thus be realized and the need for an additional cooler in the cooling gas loop could be avoided. Careful considerations covering the entire performance map and for the whole expected life of the unit are very important factors in ensuring a successful design. Special focus had to be put on the effect of the cooling gas extraction on thrust and stage matching when operating at different points on the performance line, and on the effect towards the end of the unit lifetime, when gas densities decline and the motor has to be supplied with a larger volumetric flow of cooling gas.

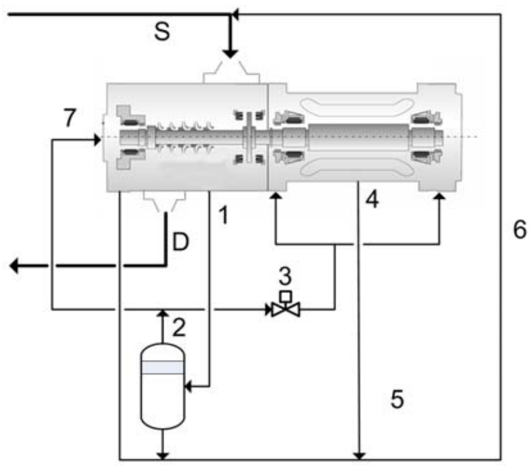

Figure 7: Motor-Compressor with Cooling Gas System

Copyright© 2016 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

Field Installation and Start of Production

A total of three operational motor-compressors plus a pilot unit and a spare electric motor were delivered to the customer in the years 2013 and 2014. After integration of the motor-compressors into the compressor module, the end-user performed further thorough testing in a dedicated test loop, including a full power 72h endurance test in submerged condition. Finally, after successfully passing this testing, compressor trains 1 and 2 were released for deployment and installed into the subsea compression station during the summer period of 2015 (Figure 8). Train 1 took up operation on September 16th, 2015, train 2 followed on January 28th, 2016 (Vinterstø et al., 2016; Ramberg and Davies, 2016).

Figure 8: Load-Out of the Subsea Compressor Module into the Norwegian Sea during 2015 (Courtesy of Statoil ASA)

TWO ALTERNATIVE SUBSEA COMPRESSION CONCEPTS: SEPARATION AND SINGLE PHASE BOOSTING VS. MULTIPHASE COMPRESSION (WELL-STREAM COMPRESSION)

When adding subsea or topside compression to an offshore natural gas field, there are two possible concepts for handling liquids. The traditional concept used in many topside installations is to separate the liquid phase from the gaseous phase and then apply a "gas-only" compressor to the gas, and a pump to the liquids. This concept was chosen in the Åsgard project and is well proven and considered the *standard solution*. On the other hand it is also possible to push simultaneously all gaseous and liquid phases through one single machine, which can be, depending on the liquid mass fraction (LMF) of the field, either a well-stream compressor (LMF typically below 50 percent) or a multiphase pump (LMF typically above 80 percent). This is denoted as *well-stream compression* and it has been successfully proven for wet gas compression with a centrifugal compressor on a prototype level. Results for a LMF of up to 30 percent are given in Kleynhans et al., 2016. These two concepts are illustratively discussed in the following paragraphs.

Standard Solution: Separation with Single Phase Compression and Pumping

A layout for a compression station for the *standard solution* is depicted in Figure 9. The actual layout is exemplary and in general dependent on the actual process conditions. The untreated well fluid is of multiphase nature and is typically cooled over an inlet cooler in order to decrease the required compression power. For some applications with sufficiently long upstream pipelines, the inlet cooler can be omitted as the hot fluid from the wells has already been cooled down by the water environment. The multiphase well fluid is then split up in the separator into a gaseous and a liquid phase. The gaseous phase is compressed by means of a compressor, whereas the liquid phase is handled by means of a pump. The power requirement for the pump depends on the LMF but is typically significantly lower than for the compressor due to the higher liquid density. This will be discussed below. It can be necessary to cool down the compressed gas before delivering it together with the pumped liquid into the discharge pipeline. This is especially the case if high pressure ratios are required and the discharge pipeline has protective coatings.

In the case that this standard solution is chosen, layout and operation, both for the compressor and pump, can follow well established design rules. Due to the single phase nature of the flow over the compressor and the pump, the flow rates can be metered precisely with appropriate respective devices. This allows for exact monitoring of performance or early detection of degradation, both for the compressor and for the pump. Separate recirculation loops for compressor and pump guarantee continuous operation, even if the production flow is below the minimum flow limit of either compressor or pump. This may occur due to changing liquid mass fractions (LMF) or during special operations.

There are further advantages of recycle loops, especially during commissioning, but also if parallel compressor trains need to be brought online. In such a set-up, adding an additional compressor to an already running process means that it can be started in recycle mode first. After reaching a certain pressure ratio, dictated by the already running process, the additional

Copyright© 2016 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

compressor can then be hooked into the production.

Figure 9 illustrates the mixing of both compressed gaseous and boosted liquid phases into a common discharge pipeline, which represents a cost effective solution. A single discharge pipeline does not, however, always have to be realized. Alternatively, separate discharge pipelines for gas and liquid could be considered. The risk of flow instabilities such as slug flow or liquid flow reversion can be reduced.

A further advantage of having a separate pump is the start-up scenario: If liquid has accumulated in the upstream piping between the well and the compression station for some reason, this can lead to fast changes in LMF or excessively high LMF at the entrance to the compressor station. This can be handled by operating the pump at different relative speeds compared to the compressor.

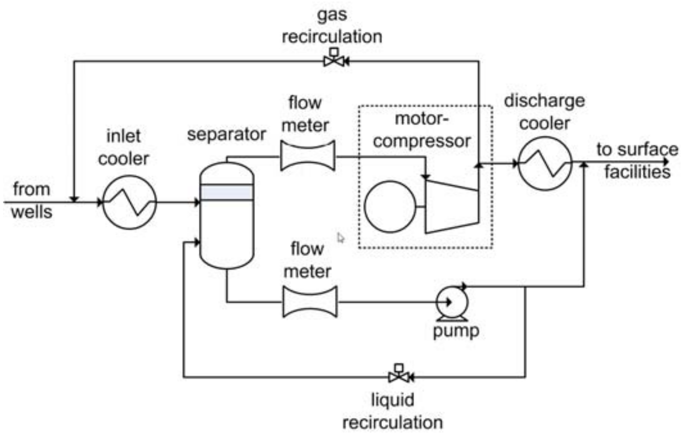

Figure 9: Layouts Denoted as Standard Solution

The disadvantage of this standard solution (Figure 9) is the associated high system complexity and cost. There is the need of a pump in addition to a compressor, along with additional power supply and pipework, a separator and increased structural framework. This all adds considerably to the CAPEX, which is especially relevant for smaller size installations, where a single machine provides sufficient power. For larger installations, the provision of pumps along with several compressors can still be cost-effective solution. This is because from the compression efficiency point of view, there are advantages when compared to well-stream compression, as briefly outlined later in this paper. The pump might contribute to limit the number of required compressors. The total number of machines can thus, depending on the circumstances of the actual field, potentially be kept lower, which impacts both CAPEX and OPEX.

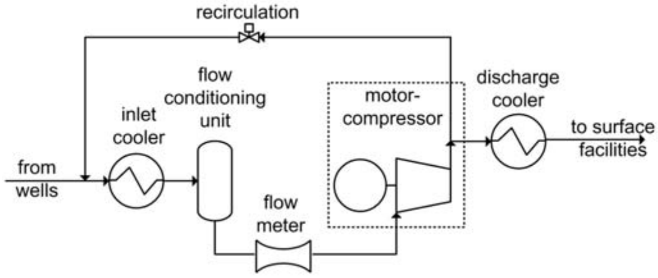

Figure 10: Layouts Denoted as Well-Stream Compression

Well-Stream Compression

Wet gas compression has been studied and used for decades in some specialized areas of applications such as soda ash production, as was briefly outlined in the introduction of this paper. Compressor online washing, inlet fogging or overspraying are other examples. The amount of liquid injected is continuous and is normally strictly controlled. In contrast to these experiences, *well-stream compression* is a relatively new process concept for the oil and gas industry. Extensive testing with a multi-stage radial compressor under real upstream conditions is reported in Kleynhans et al. (2016).

A possible process flow diagram of a well-stream compression train is given in Figure 10. A well-stream compression facility may optionally be equipped with an inlet cooler in order to reduce the required compression power. As the LMF in the well-stream may be subject to abrupt changes, a flow conditioning unit (FCU) in close proximity to the compressor inlet is required to smooth out these fluctuations. The size of the FCU is mainly dictated by the required buffer volume for the liquid phase and has to be sized according to the expected variations of LMF in the incoming well-stream. Fast changes in LMF at the compressor inlet have a direct impact upon the torque and power of the motor as speed cannot be adjusted beyond a certain rate. The FCU's purpose is to keep the changing rates in LMF at the inlet to the well-stream compressor below a certain level and to account for incoming slugs. The flow at the compressor inlet can be measured with the aid of a multiphase flow meter. Monitoring the actual compressor performance, or effects such as degradations, is, however, generally more difficult. This is due to higher uncertainties in the multiphase flow measurement when compared to single phase measurements, and due to the more unsteady nature of operation of the compressor under temporally varying LMFs. The well-stream compression layout also contains a recirculation line and an optional discharge cooler for the same reasons as discussed for the standard solution.

Well-stream compression is a beneficial concept especially for small fields, where a single well-stream compressor is capable of compressing all fluids, reducing system complexity com-

Copyright© 2016 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

pared to the standard solution requiring an additional pump. Furthermore, it can be beneficial for fields with low liquid content, where the influence on the compressor performance map is typically minor (Kleynhans et al., 2016). However, clear disadvantages arise from the flow measurement point of view, from adverse operability under massive inlet slugging, and from the lack of operational experience in the industry.

Typical Limitations in Pumps, Multiphase Pumps and Well-Stream Compressor Operation

Pumps are operated with incompressible liquids of typically high density. The capability to move fluid from a low suction pressure level towards a discharge level is given as a pressure rise $\Delta p = p_2 - p_1$. The commonly used pressure ratio $\pi = p_2/p_1$ in compressors is not used for pumps because the fluid is incompressible and the density is known. Neglecting pump efficiency in a first rough approximation, the power P of the pump can be directly expressed as the product of pressure rise Δp and volume flow rate $\dot{V}$:

$$P = \dot{m} \cdot \Delta h \approx \frac{\dot{m} \cdot \Delta p}{\rho} = \dot{V} \cdot \Delta p = Q \cdot \Delta p \qquad \textbf{Equation 1}$$

Note that in the pump industry, the volume flow rate $\dot{V}$ is most often denoted with the letter Q. Neglecting viscous losses, the change of enthalpy of the fluid can be approximated to be dependent on the change of pressure since the density is inherently constant for incompressible fluids. Limiting the discussion to fluid-dynamic pumps, the speed of the fluid and thus the volume flow rate through a turbomachine is governed by shaft speed. By keeping volume flow rate and shaft speed constant, Equation 1 shows that the pressure rise and the power in a pump will decrease with lowering fluid density. This is typically encountered in multiphase pumping with increasing gas volume fractions (Turpin et al., 1986). The breakdown in pressure rise can either be countered with provisions for liquid recirculation or by an increase in speed.

Given a certain required pressure rise dictated by the process, shaft speeds in pumps can be kept lower compared to compressors because of the higher densities of the liquid. The pumping equipment is designed accordingly. As a consequence, pumps that are operated with fluids that are considerably lower in density than that assumed during design of the pump will not be able to deliver the required pressure rise because their speed cannot be readily increased due to mechanical or thermal restrictions. The situation with compressors is opposite to this. Compressors are designed to operate with lower density and compressible fluids. Increasing the fluid density will result in excessive torque levels and mechanical loading of the compressor internal components. It follows that pumps operated at constant speed with lower-than-design density fluids, as is encountered, for example, in multiphase flow, will not be able to deliver the required pressure rise. On the other hand, compressors operated with higher-than-design density fluids, as is, for example, the case in wet gas flow, will see an excessive torque load with a potential drop in speed if the drive cannot deliver this excess torque. Therefore, for both multiphase pumps and well-stream compressors, the multiphase nature of the flow has to be considered in the design phase in order to be able to operate the machine successfully under such conditions. This, however, may lead to compromise solutions in order to cover a certain multiphase flow range.

There have been developments both on the pump side as well as the compressor side in order to render the machine tolerant to multiphase flows, i.e. clearly off the standard design case of the respective machine type. Typical liquid pumps require LVF’s higher than 90 percent in order not to encounter unacceptable break down of pressure rise or phenomena such a gas lock, which is a complete blockage of impeller passages with gas pockets (Turpin et al, 1986). Specifically tailored multiphase pumps can operate down to LVFs of approximately 15 percent corresponding to GVFs of up to approximately 85 percent without liquid recycling (Vesterkjær, 2015).

For compressors, as mentioned above, LMFs of up to 30 percent were experimentally demonstrated with a large scale prototype. Of course, a compressor being able to run such high LMF levels comes at a price. Compressor internal components need to be designed accordingly in order to withstand this extra level of load and increased risk of erosion and corrosion. Also the drive system needs to have the required margin in order not encounter speed breakdown. Several authors state a direct relation between the mean suction density and the required torque (Brenne et al., 2005, Bertoneri et al., 2012). Adding 10 percent in mass of liquid to the gas increases shaft torque by approximately 10 percent for constant shaft speeds.

A direct numerical relationship between GVF and LMF can be established with $\gamma = \rho_{LIQ}/\rho_{GAS}$ denoting the density ratio between the liquid and the gaseous phase.

$$LMF = 1 - GMF = \frac{\gamma \cdot (1 - GVF)}{\gamma \cdot (1 - GVF) + GVF} \qquad \textbf{Equation 2}$$

or equivalently,

$$GVF = 1 - LVF = \frac{\gamma \cdot (1 - LMF)}{\gamma + (1 - \gamma) \cdot LMF} \qquad \textbf{Equation 3}$$

This relationship is graphically visualized in Figure 11 for typically encountered density ratios γ in well-stream compression. By assuming a liquid phase density of 800 kg/m^3, (50 lb/ft^3) and natural gas, low values of $\gamma \approx 5$ are present at high

Copyright© 2016 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

pressures of approximately 150 bar (2’180 psi) and high values of $\gamma \approx 40$ at low pressures of approximately 20 bar (290 psi).

Multiphase pumping is clearly located in the high-LMF region which is why typically the gas volume fraction (GVF) or equivalently liquid volume fraction (LVF) are specified (Moen, 2015) to distinguish a multiphase pump from a standard pump. In multiphase pumping, which can be associated with very high LMF above 80 percent, the presence of gas (GVF) has a negative impact on pump head. Depending on the pressure and thus density ratio between liquid and gas, a GVF of 20 percent may be equivalent to a gas mass fraction (GMF) of only 1 percent.

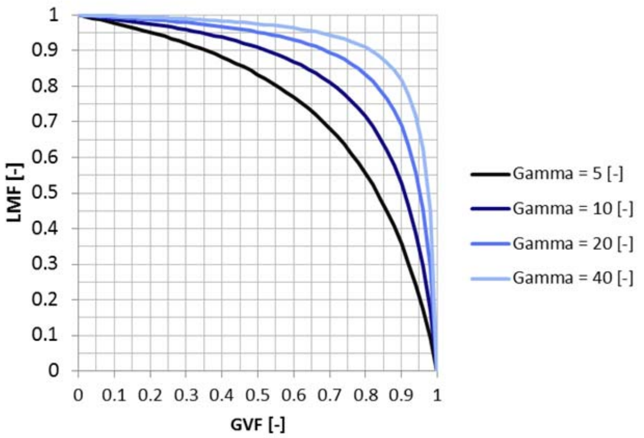

Figure 11: Relationship between GVF and LMF for typical Density Ratios

In contrast to multiphase pumping, wet gas compression is located in the high GVF region, which is why LMF is normally specified. Liquids here can have a positive effect on the pressure ratio of the compressor and their influence is typically given as a function of liquid mass fraction (Kleynhans et al., 2016).

LATE LIFE FIELD EXPLOITATION AND RELATED CHALLENGES

Figure 12 illustrates a production profile of a typical natural gas field. After startup, the initial production phase between (1) and (2) is referred to as the natural plateau. During this phase, the production rate is mainly limited by the topside or downstream processing facilities and thus the production rate is hence largely constant over time. The natural pressure of the reservoir is sufficient to drive the flow, which often even has to be throttled. With ongoing depletion, the natural pressure of the reservoir drops below a certain level such that constant production cannot be maintained any longer (2). Production thereafter continuously decreases over time. At a certain point in time (3) the production has to be halted. This can be because a critical economical limit is reached, or it may result from physical limitations such as maintaining stable flow conditions within the pipelines.

Adding compression power as shown in Figure 12 allows for prolonging the plateau production from (2) to (2’). This typically starts with suction pressures in the range $p_s = 50$ bar to 120 bar (725 psi to 1’740 psi) and pressure ratios over the compressor of $\pi = 1.2$ to 2. Discharge pressures are hence typically in the range of $p_d = 80$ bar to 170 bar (1’160 psi to 2’470 psi). In case of subsea compression higher water depths and larger step-out distances typically go along with higher absolute pressure levels.

Even with added compression power, at a certain later point in time (2’), the field pressure is insufficient to maintain plateau production and the production typically declines. Production has to finally be abandoned at (3’). Suction pressure may be as low as $p_s = 15$ bar (217 psi) at this point and pressure ratio has increased to typically $\pi = 3$ to 6. The light blue shaded area in Figure 12 reflects the gain in production compared to the case where the production is only dependent on the natural pressure of the reservoir (gray shading in figure). The gain in production is very valuable as it can in most cases be achieved by using mainly existing processing infrastructure.

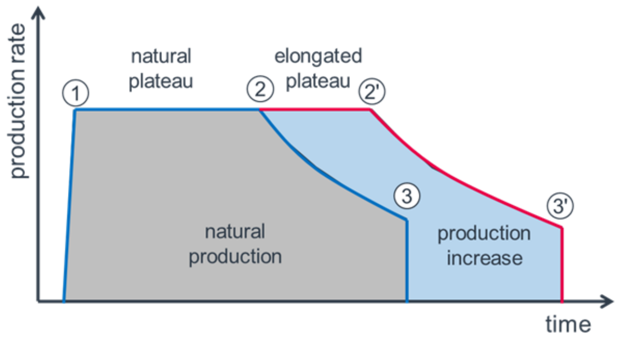

Figure 12: Production Profile for Natural Production and Enhanced Production with Added Compression

Adding Compression Power to a Natural Gas Field

In order to enhance production, compression power is ideally added at point (2) in Figure 12 to prologue the plateau. Figure 13 illustrates the required pressure ratio typically corresponding to such a scenario. At the beginning of compression at time (2), only a very limited pressure ratio is required in order to keep the production plateau, but a high volumetric flow rate is required at the same time. The compressor has to be able to operate at such high flows with low polytropic head, i.e. close to choked condition, as illustrated in Figure 14.

Copyright© 2016 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

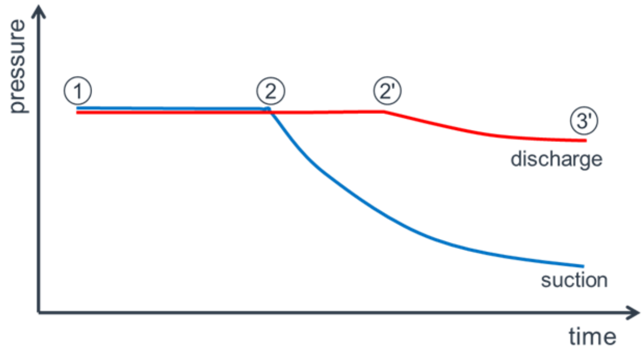

Figure 13: Exemplary Suction Pressure and Required Discharge Pressure of a Compression Station for the Production Profile in Figure 12

Over time, as the compressor operates with decreasing suction pressure but largely constant discharge pressure, the operating point in the compressor map shifts towards higher head and thus towards the center region of the performance map (Figure 14). The compressor speed is increased continuously over the years until the maximum allowable speed is reached. The compressor selection is typically optimized for maximum performance (efficiency and production) over a time period of typically 5 to 10 years, this period being a typical interval between maintenance activities. Therefore, the motor normally operates with a speed close to maximum shaft power. The wide operating range of radial compressors when compared to axial compressors is highly beneficial for enabling large variations in operation. This is especially the case when uncertainties concerning the production profile of the field exist. Furthermore, the broad performance map of centrifugal compressors generally allows for smaller efficiency variations over wide areas of operation.

An example of a compressor modification strategy is shown in the subsequent paragraphs. In this example, the compressor needs to be re-bundled or reconfigured around time (S) as illustrated in Figure 14. The changing production necessitates the compressor to be modified towards higher polytropic head, and together with the power limitations of the motor-compressor, lower absolute mass flow rates will result.

Ideally, the new bundle or compressor configuration has an overlapping performance map with the first bundle. This is in order to guarantee a compressor exchange window of several months or even years. As the suction pressure decreases from (2) to (S), the same electric motor can now drive a larger compressor or, equivalently, a larger number of compressor stages with both larger volumetric flow as well as higher pressure ratio as illustrated in Figure 14. This modification allows for continued exploitation of the field. Furthermore, it allows for a certain reuse of equipment in order to minimize CAPEX, in particular the reuse of the electric power supply system comprising variable frequency drive, transformer, umbilical and motor. While the rebundling typically requires an exchange of the compressor internals, the reconfiguration also allows for the addition of an additional compressor casing on the second shaft end of the electric motor.

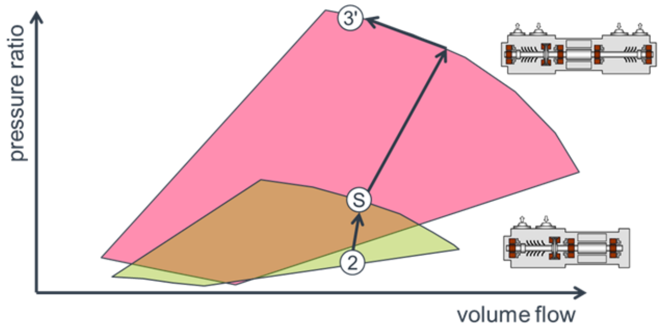

Figure 14: Typical Compressor Performance Maps for Two Different Layouts

Which is the Boosting/Compression Concept to Choose?

In the following, 3 different concepts are considered for production from oil and gas field in the range of 0 to 100 percent LMF which is equivalent to 100 to 0 percent GVF. The concepts are:

1. Separation, gas compression with a standard compressor and liquid boosting with a standard pump
2. Well-stream compression with a well-stream compressor (assumed max. LMF of 50 percent)
3. Boosting with a multiphase pump (assumed max. GVF of 85 percent)

The limitations of these three concepts are graphically visualized for a suction pressure of 30 bar (435 psi) which is typically encountered towards the end of a natural gas field life, when adding compression becomes necessary in order to prolong field operation, as explained earlier. The colored bar in Figure 15 represent possible operation of the respective concept. The standard solution, i.e. relying on separation together with a standard dry gas compressor and a standard pump is capable of handling the full range of GVF or LMF respectively. The operation of future well-stream compressors is assumed to be realistically reasonable somewhere in the region from 0 to 50 percent LMF, multiphase pumps are known today to reasonably tolerate a GVF of up to 85 percent, assuming no liquid phase recycling. However, arguing solely at the level of GVF distorts the picture considerably, because the value of the production relates to the produced masses and mass fractions. At a pressure of 30 bar (435 psi), a LMF of 50 percent turns the multiphase pump solution into a non-viable solution, whereas the well-

Copyright© 2016 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

stream compressor can still handle this situation. Put in another way: a multiphase pump relies on a liquid mass fraction of more than 80 percent.

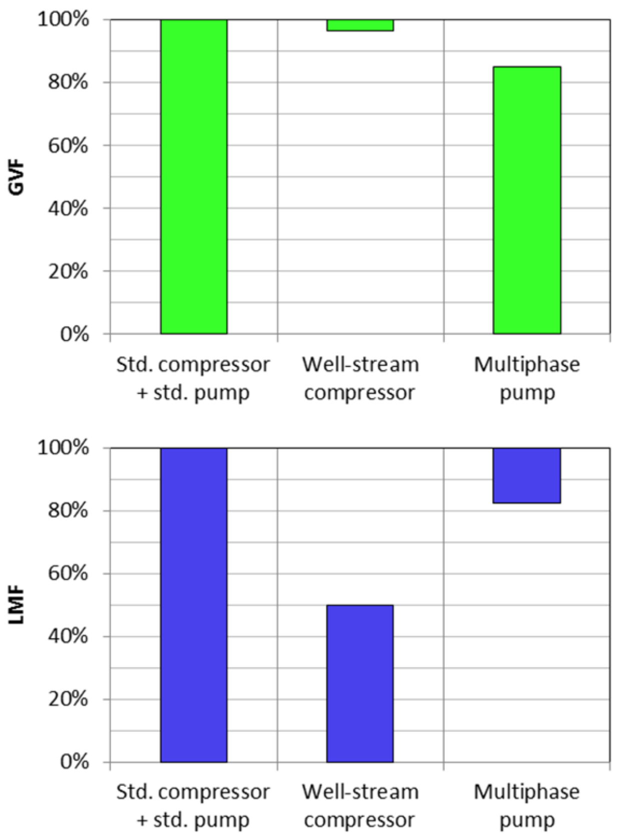

Figure 15: Possible Field Operation Concepts for an Assumed Gas Pressure of 30 bar

SIZING OF COMPRESSOR AND PUMP – SOME SIMPLE SAMPLE CALCULATIONS

When operating a field, the compressor and pump layout has to be generated for predicted operation over several years. This chapter introduces simple equations for compressors and pumps and conducts computations accordingly for an assumed but typical natural gas field behavior. The suction and discharge of pump and compressor are denoted with subscript 1 and 2, respectively. The *standard solution* with compressor and pump is described first, and then compared to *well-stream compression* and *multiphase pumping*.

Standard Solution

The *standard solution* comprises a gas compressor and a liquid pump. Gas compression is often approximated by the well-known polytropic process. It is assumed that pressure and density of the gas follow the equation

$$\frac{p_2}{p_1} = \pi = \left(\frac{\rho_2}{\rho_1}\right)^n \qquad \textbf{Equation 4}$$

where n is the polytropic exponent. The required polytropic head $h_{GAS,pol}$ to compress a gas from pressure p_1 to p_2 is then given by

$$\Delta h_{GAS,pol} = \frac{n}{n-1} \cdot \left(\frac{p_2}{\rho_{GAS,2}} - \frac{p_1}{\rho_{GAS,1}}\right) \qquad \textbf{Equation 5}$$

For a pump, the liquid density is largely constant such that the required pump head simplifies to:

$$\Delta h_{\mathrm{LIQ}} = \frac{(p_2 - p_1)}{\rho_{\mathrm{LIQ}}} \qquad \textbf{Equation 6}$$

The required shaft power P for compressor or pump is then the product of head and mass flow:

$$P = \frac{\dot{m} \cdot \Delta h}{\eta} \qquad \textbf{Equation 7}$$

where η is the efficiency for compression and pumping, respectively.

With these equations different compression scenarios are computed. First, the standard compression process (separation with compressor and pump) is addressed and then direct well-stream compression is considered. For simplicity, we set the following variables to typical constant values:

$$T_1 = 280\ \mathrm{K}$$

$$n \approx 1.55$$

$$\eta_{\mathrm{comp}} \approx \eta_{\mathrm{pump}}$$

$$MM_{\mathrm{gas}} \approx 20\,\frac{\mathrm{kg}}{\mathrm{kmol}} \quad (44.1\ \frac{\mathrm{lb}}{\mathrm{mol}})$$

$$z = 0.85$$

$$GMF = 70\%\ (\mathrm{i.e.}\ LMF = 30\%)$$

$$P_{\mathrm{comp}} \approx 10\ \mathrm{MW}\ (13'400\ \mathrm{hp})$$
(the pump power is then computed from the given LMF)

Copyright© 2016 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

The suction and discharge pressure are input values into the calculation and arbitrarily set as shown in Figure 16. Figure 17 shows the corresponding pressure ratio π for the compressor and differential pressure Δp for the pump. Let the process start at a high suction pressure of 100 bar (1'450 psi) with a pressure ratio in the compressor of $\pi = 1.4$ and a differential pressure of 40 bar (580 psi) in the pump. Let furthermore the suction pressure decrease significantly more than the discharge pressure over time. The pressure ratio over the compressor thereby increases to $\pi = 4.8$ while the differential pressure in the pump reaches a maximum of about 70 bar (1'020 psi).

For an assumed constant compressor power of 10 MW (13'400 hp) the mass flow of both the gaseous and the liquid phases must decrease over time (Figure 18) and the pressure ratio in the compressor increases (Figure 17).

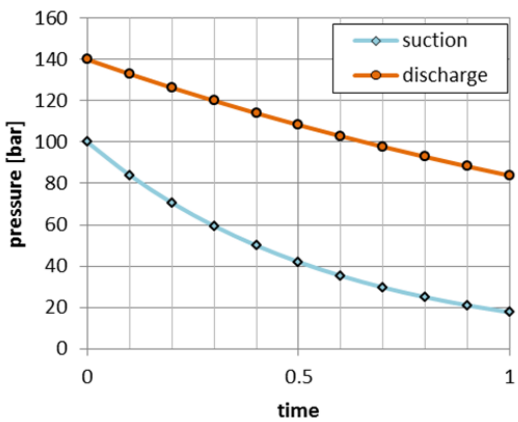

Figure 16: Suction and Discharge Pressure for the Chosen Application Case

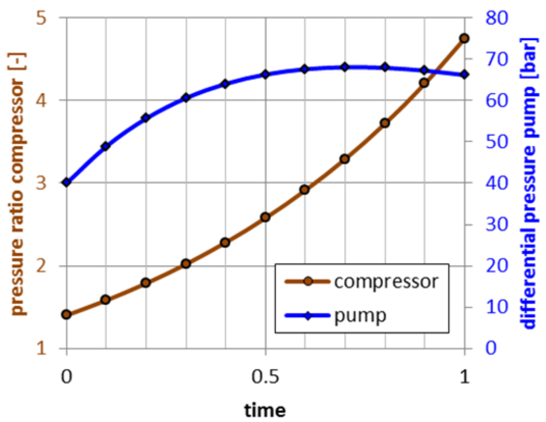

Figure 17: Pressure Ratio for the Compressor and Pump Differential Pressure for the Chosen Application Case

Even though the system LMF is set to 30 percent over the entire time, the system LVF is low and decreases from 5 to 1 percent over time as shown in Figure 19.

The compressor and the pump head are shown in Figure 20, the power in Figure 21. The specific head increases, especially for the gaseous phase (Figure 20). Due to the decreasing suction pressure, the volumetric suction flow in the compressor is nearly constant, while it is decreasing in the pump. This is shown in Figure 22 .

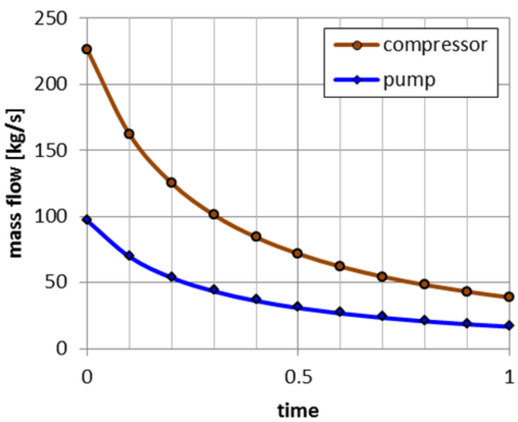

Figure 18: Mass Flow for Compressor and Pump for the Chosen Application Case

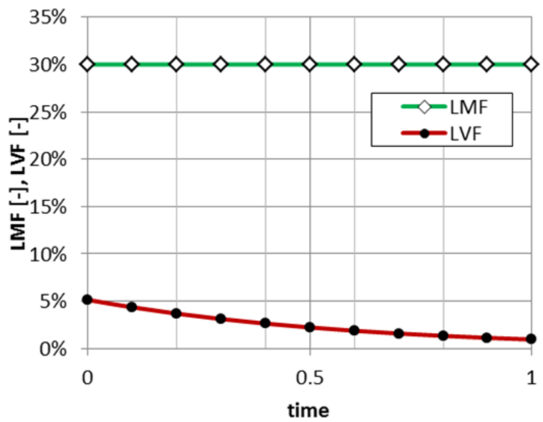

Figure 19: System LVF and LMF for the Chosen Application Case

Copyright© 2016 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

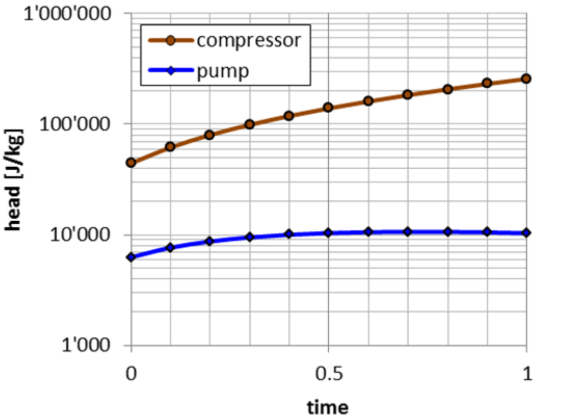

Figure 20: Compressor and Pump Head for the Chosen Application Case

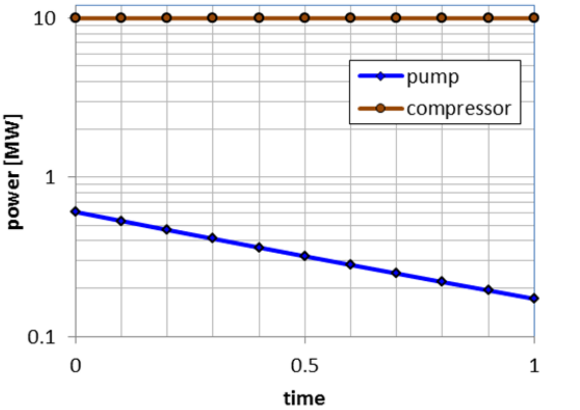

Figure 21: Compressor and Pump Power for the Chosen Application Case

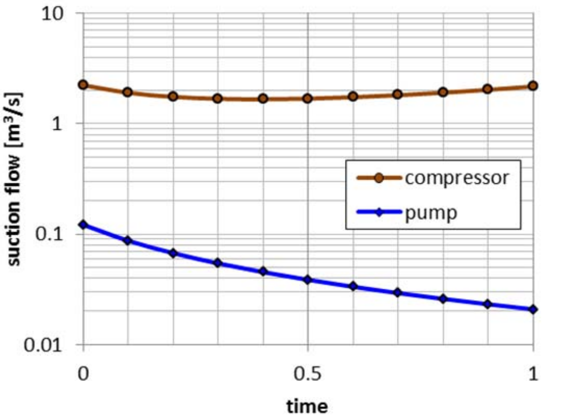

Figure 22: Suction Volumetric Flow for Pump and Compressor for the Chosen Application Case

Summarizing, due to the lower density of the gas compared to the liquid, the power requirement is dominated by compression of the gaseous phase. The power for pumping the liquid is in the range of a few percent of the compressor power. This allows for oversizing of the pump in order to be able to handle liquid transients or liquid build-up at start-up, which may have accumulated in the suction pipeline.

Well-stream Compression

Predicting compressor characteristics and power is well known for gas compressors. However, equivalent models for a wet gas or well-stream compressor have not been fully established yet. The first published experimental results for wet hydrocarbon gas compression can be found in the work of (Brenne et al., 2005). One clear result is that the shaft power consumption typically increases almost linearly with the added liquid mass if speed is kept constant. At first glance this seems not to be ideal, as this would require approximately an additional 30 percent torque and power for 30 percent LMF, compared to the dry gas compression case. Adding a pump for the same pressure duty would require only 2 to 6 percent of the compressor power. However, the increase in power in well-stream compression is compensated by an increase in pressure ratio as shown, for example, by Brenne et al. (2005) and Kleynhans et al. (2016). The latter report quantitative results: increasing LMF from 0 to 16 percent while maintaining the pressure ratio at a constant value of 2.2, required a torque increase of 8% and a power increase of 4 percent, which means that shaft speed could be lowered by 4 percent.

The compressor selection can be based on the pure gas and on standard layout procedures. No detrimental impact on stage matching and compressor range could be observed with multi-stage wet gas compression to date. The effect of adding mass to the gas, however, needs to be considered in order to avoid excessively high torque levels on the motor. A design trimmed towards maximum robust design is hence clearly favorable. Depending on the actual case, similar overall power levels for the well-stream compressor can be achieved, but at reduced shaft speeds.

The provision of a flow conditioning unit (FCU) placed closely upstream of the well-stream compressor is considered a necessity in order to ensure constant operating conditions of the well-stream compressor.

Well-stream compression with compressor derivative designs are not yet in commercial use, however, first prototype level large scale testing shows promising results (Kleynhans et al, 2016).

Copyright© 2016 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

Multiphase Pumps

As outlined above, multiphase pumps are not considered to be the optimum solution for the chosen application case due to the low LMF. The pressure increase of such devices is rather limited for high GVF levels. Therefore these pumps are not further discussed for this case.

CORE UNIT MODULARITY AND ADAPTABILITY TO CHANGING PRODUCTION

When generating a layout of a motor-compressor for specified operating points of a field, the flexibility is typically in the compressor design. For the motor the maximum speed and maximum power or torque can only be varied within small boundaries. In contrast, the compressor design has a high flexibility allowing for different stage selections, impeller diameters and numbers of stages. This allows for variations according to main compression parameters such as mass flow, suction volumetric flow, and polytropic head. The evolutions of mass flow rate and volumetric flow rate with increasing pressure ratio for the chosen application from above are illustrated in Figure 23 and Figure 24.

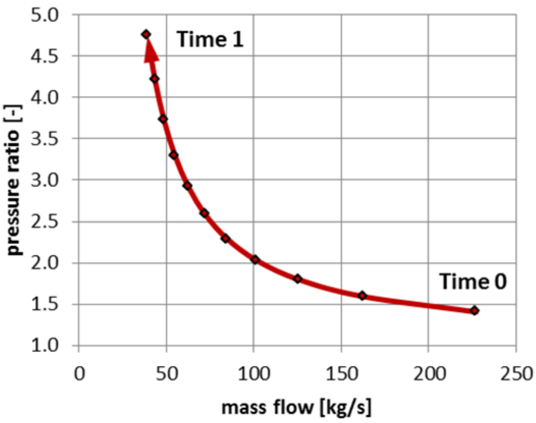

Figure 23: Dependence of Pressure Ratio on Mass Flow Rate for the Chosen Application Case from Above.

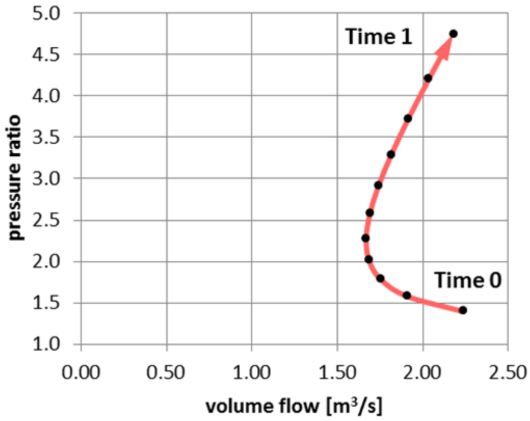

Figure 24: Dependence of Pressure Ratio on Volume Flow Rate for the Chosen Application Case from Above.

Figure 25 shows 3 different compressor configurations and a line of constant shaft power. These configurations cover the different time periods of the production of the chosen application case, beginning with high mass flow rate at a low pressure ratio and ending with low mass flow rate at a high pressure ratio. For compressors, typically pressure ratio or polytropic head is plotted as a function of volumetric flow in a non-dimensional form. However, in this figure the mass flow and pressure ratio is used as this relates to power as given by Equation 5 and Equation 7.

The first configuration is denoted as “single” because a single barrel compressor is connected to the motor. It is shown in blue color in Figure 25. A typical barrel may contain between 3 and 8 stages and allows, due to the comparably low required pressure ratio (i.e. polytropic head), for higher mass flows. When higher pressure ratios are required, e.g. for depleting fields, a second compressor casing may be added to the second shaft end (green in Figure 25). For intermediate pressure ratios, this may be an overhung stage, as given by the green configuration in Figure 25 or for highest pressure ratios, a second barrel compressor as illustrated by the red configuration. The latter is denoted as “tandem” as two multi-stage barrels are connected to the motor. As the suction pressure may drop faster than mass flow, the suction volumetric flow may be higher for the high pressure ratio configurations as described by:

$$\dot{V} \approx \dot{m}/\rho \qquad \textbf{Equation 8}$$

Not shown in Figure 25 is a configuration for even higher mass flows with lower head, as typically used for pipeline motor-compressors. In these configurations, overhung impellers are used on both shaft ends of the motor.

Copyright© 2016 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

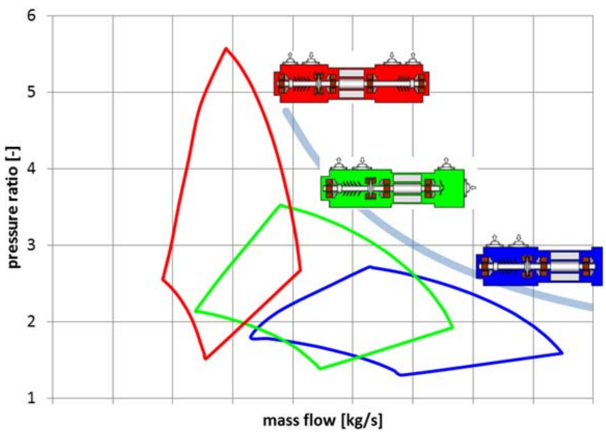

Figure 25: Pressure Ratio vs. Mass Flow for Different Compressor Arrangements using the same Electric Motor. A Line of Constant Power is Included with a Shaded Line.

The example above clearly indicates that compressors for various pressure ratios are required. For topside applications fixed and variable speed motors are used. For this application a variable speed motor has clear advantages.

FUTURE DEVELOPMENT TRENDS AND CHALLENGES IN SUBSEA COMPRESSION

Oil and gas producing companies have realized that standardized solutions bear an enormous potential for keeping project related risks and costs low. This trend has already impacted on subsea gas compression being developed on a conceptual level at present. Future subsea compression systems are being streamlined with the focus on robustness, standardized interfaces and packages, system simplicity, weight and size minimizations and the elimination of unnecessary system complexity. Having delivered a subsea compression system for the Åsgard project and thereby acquired valuable knowledge and expertise, the main involved suppliers are now cooperating to simplify and optimize the existing system (Vesterkjær, 2015). Significant potential for simplification and optimization which can be applied without major changes to the qualified core functionality has been identified. For example, the size of the compressor module can be significantly reduced resulting in a reduction of size and weight. A further yet still to be qualified system simplification can be realized by moving towards the above-mentioned well-stream compression concept and thus removing the need for the separator and the pump.

CONCLUSIONS

The potential of subsea compression has been a focus area of offshore natural gas production and intensive qualification programs were needed before the first commercial units took up production. The technology is expected to change this industry in the years to come. A modular machine concept allows for flexible in-service updates and for maximizing late life production.

This paper describes a subsea compression technology with its broad applicability for small and large fields. Depending on the LMF of the field as well as on the total compression power required, the compressor can either be used directly as a *well-stream compressor* or in combination with a pump in the *standard solution*. One characteristic feature is the high flexibility in the compressor layout. The possible combination of one or two barrel type or overhung compressor stages allows for adaptation to the highly varying requirements of operation over the late-life production years. The variation is typically characterized by changes in volumetric and mass flow as well as pressure ratio. The goal is to find suitable compressor layouts over the field life while keeping the modification effort at a minimum. Production rates as well as total recovery of the field can thus be substantially increased.

Multiphase production adds a further dimension to the compressor and pump design. The effect of multiphase flow is opposite for compressor and pump due to the density effect: Adding a small volume fraction of liquid to the gas drastically increases the mean density, resulting typically in increased torque, power and pressure ratio of the compressor for a given shaft speed. Maintaining a constant pressure ratio results in a speed reduction of the process gas compressor. In contrast, multiphase pumps show a significant decrease in pressure rise with increasing amounts of gas, even at only minor mass fractions of gas. To compensate for this, the pump speed has to be significantly increased and/or liquid recirculation has to be considered.

NOMENCLATURE

Variables

h	= Polytropic Head (Enthalpy)	(J/kg)
$\dot{m}$	= Mass Flow	(kg/s)
MM	= Molecular Mass	(kg/kmol)
n	= Coefficient of Polytropic Compression	(-)
$\dot{V}$	= Volumetric Flow Rate	(m^3/s)
Q	= Volumetric Flow Rate (Pumps)	(m^3/s)
z	= Real Gas Factor	(-)
η	= Efficiency	(-)
γ	= Density Ratio	(-)
π	= Pressure Ratio	(-)
ρ	= Density	(kg/m^3)

Copyright© 2016 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

Indices

1	= Inlet
2	= Outlet
COMP	= Compressor
GAS	= Gaseous Phase
LIQ	= Liquid Phase
pol	= Polytropic

Abbreviations

AMB	= Active Magnetic Bearing system
CAPEX	= CAPital EXpenditure
FAT	= Factory Acceptance Test
FCU	= Flow Conditioning Unit
FPSO	= Floating Production, Storage and Offloading
GMF	= Gas Mass Fraction
GVF	= Gas Volume Fraction
HV	= High-Voltage
LMF	= Liquid Mass Fraction
LVF	= Liquid Volume Fraction
MEG	= Mono Ethylene Glycol
MMboe	= Million Barrels Oil Equivalent
OPEX	= OPerating EXpenditure
TRL	= Technology Readiness Level
VFD	= Variable Frequency Drive

REFERENCES

Baggerud, E., Sten-Halvorsen, V., Fantoft, R. 2007, Technical Status and Development Needs for Subsea Gas Compression, Offshore Technology Conference, OTC-18952.

Beckman, J., 2015, Subsea Compression Prolongs Gas Production at Åsgard Offshore Norway, Offshore, December Issue 2015.

Bertoneri, M., Duni, S., Ransom, D., Podesta, L., Camatti, M., Bigi, M., and Wilcox, M., 2012, Measured Performance of Two-Stage Centrifugal Compressor under Wet Gas Conditions, ASME Turbo Expo, Copenhagen, Denmark.

Brenne, L., Bjørge, T., Gilarranz, J.L, Koch, J.M., Miller, H., 2005, Performance of a Centrifugal Compressor Operating Under Wet Gas Conditions, 34th Turbomachinery Symposium, Houston, USA.

Büche, D., Aho, T., and Dettwyler. M., 2014, From Design Aspects through to Testing of the MAN Diesel & Turbo Subsea Motor-Compressor for the Åsgard Subsea Project, ASME Paper Number GT2014-26380, Gas Turbine Conference.

Casey, M. V., 1989, Multiphase Compression of Oil and Gas Flows with a Gas Volume Fraction between 90% and 100% - A Preliminary Technical Study, TK-0266, internal report.

Chellini, R., 2010, MAN achieves Breakthrough in Upstream Compression, COMPRESSORtech^2, December Issue 2010.

Chellini, R., 2014, First Submerged Tests of Subsea Compressor, COMPRESSORtech^2, October Issue 2014.

Chellini, R., 2015, World's First Subsea Compressor in Operation, COMPRESSORtech^2, December Issue 2015.

Kleynhans, G., Pfrehm, G., Berger, H., 2005, Hermetically Sealed Oil-Free Turbocompressor Technology, 34th Turbomachinery Symposium, Houston, USA.

Kleynhans, G., Brenne, L., Kibsgaard, S. and Dentu, P, 2016, Development and Qualification of a Subsea Compressor, Offshore Technology Conference, OTC-27160-MS.

Moen, F., 2015, Development of a Novel Multiphase Pump Technology, Underwater Technology Conference, Bergen, Norway.

Normann, T., Rongve, K., 2014, Long Step Out Power Supply System, Underwater Technology Conference, Bergen, Norway.

Ramberg, R. M. and Davies, S., 2016, Statoil Operating Gullfaks, Asgard Subsea Compression Systems, Oil & Gas Journal, May Issue, 2016.

Ramberg, R. M., Hedne, P. E., 2016, Subsea Compression – Long Term Technology Development, Offshore Technology Conference, OTC-27201-MS.

Turpin, J., Lea, J. and Bearden, J., 1986, Gas-Liquid Flow Through Centrifugal Pumps – Correlation of Data, 3rd International Pump Symposium.

Vesterkjær, R., 2015, Subsea Well Stream Compression Development, Underwater Technology Conference, Bergen, Norway.

Vinterstø, T., Birkeland, B., Ramberg, R. M., Davies, S., Hedne, P. E., 2016, Subsea Compression – Project Overview, Offshore Technology Conference, OTC-27172-MS.

ACKNOWLEDGEMENTS

The authors would like to thank Statoil ASA and Aker Solutions for their continuous support.

Copyright© 2016 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station