ABSTRACT

The efficient operation of offshore gas turbine driven compressor trains is becoming more and more important. With the introduction of the CO₂ tax on fuelgas in Norway, the operating costs have dramatically changed. It is now necessary to focus on energy-saving operation and maintenance in order to include the influence of CO₂ taxation. In the Statfjord and Gullfaks fields, the LM2500 high efficiency aeroderivative gasturbines have been in operation for several years, and the operational experience is presented and discussed here with regard to efficiency and reliability. An optimised gas turbine maintenance program has been introduced in order to obtain energy-saving and cost reduction. The problems relating to the gas compression trains are discussed with regard to energy conservation and reliability. Off design operating conditions on compressors have caused problems, both with regard to power losses and machinery safeguard control systems. Energy-saving improvements have been implemented on the compressors and the system they are working in. The reliable operation of the compressors can be put at risk by the shaft-sealing system. It is of vital importance to ensure correct operation of seal system in order to obtain safe and economical operation. Improvements to shaft-sealing systems are discussed.

INTRODUCTION

Since 1986/1987, Den norske stats oljeselskap A/S (Statoil) has been the operator of the Statfjord field, the Gullfaks field and the Statpipe gas transmission system in the norwegian sector of the North Sea. The new fields of Veslefrikk and Sleipner were started up in 1989 and 1993 respectively, see also map in figure 1.

The purpose of this paper is to present operational experience with gas turbine driven compressor trains on offshore platforms, mainly in the Statfjord and Gullfaks fields. In addition, it is the purpose to discuss the energy-saving improvements and the influence of CO₂ tax on operation and maintenance.

During the last few years, it has become increasingly important to ensure an efficient operation of offshore gas turbines and compressors. There are two main reasons for this, the first one is the need for cost savings and the second is environmental demands which impose limitations on the exhaust emissions to the atmosphere. It is clear that even minor improvements in the gas turbine operation will give a valuable contribution to reduction of the total exhaust emission level. On offshore platforms, the fuel gas has traditionally been considered as an almost free energy source. Some years ago, the maintenance costs was therefore the main area for cost saving. With introduction of the CO₂ tax on fuelgas in Norway in 1991, the situation has dramatically changed.
DESCRIPTION OF GAS TURBINE DRIVEN COMPRESSOR TRAINS ON OFFSHORE PLATFORMS

Aeroderivative gas turbines

Gas turbines are the prime movers in the offshore installations, and they are of vital importance for reliable and efficient production. The gas turbines in operation in Statoil's installations are listed in Table 1. The list includes both gas turbines for offshore and landbased installations, and covers gas turbines in continuous service and stand-by service.

The offshore use of aeroderivative gas turbines has proved to be successful. However, several investigations were made at an early stage on different driver alternatives, but the advantages with regard to high efficiency, high power/weight ratio and reliable operation were decisive, see ref. 1 and ref. 2.

GAS TURBINES IN OPERATION IN STATOIL INSTALLATIONS - JAN.1994

<table>
<thead>
<tr>
<th>GAS TURBINE TYPE/ LOCATION</th>
<th>NO. OF GAS TURBINES</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. LM 2500 (General Electric)</td>
<td>42</td>
</tr>
<tr>
<td>Statfjord, Gullfaks, Veslefrikk, Sleipner fields</td>
<td></td>
</tr>
<tr>
<td>2. MARS (Solar)</td>
<td>2</td>
</tr>
<tr>
<td>Veslefrikk field</td>
<td></td>
</tr>
<tr>
<td>3. KG2 (Dresser Rand Kongsberg)</td>
<td>10</td>
</tr>
<tr>
<td>Statfjord field, Statpipe 16/11 platform, a.o.</td>
<td></td>
</tr>
<tr>
<td>4. AVON (Cobra2556)</td>
<td>3</td>
</tr>
<tr>
<td>Statpipe Kaarsto</td>
<td></td>
</tr>
<tr>
<td>5. Frame 6 (General Electric)</td>
<td>1</td>
</tr>
<tr>
<td>Statpipe Kaarsto</td>
<td></td>
</tr>
<tr>
<td>6. TORNADO (EGT Ruston)</td>
<td>2</td>
</tr>
<tr>
<td>Etzel / Emden</td>
<td></td>
</tr>
<tr>
<td>TOTAL</td>
<td>60</td>
</tr>
</tbody>
</table>

The aeroderivative gas turbines are used as drivers both for gas compression trains (22 units) and power generation (20 units).

This paper concentrates on the operational experience of the LM2500 gas turbine installations. The LM2500 gas turbines in the Statfjord field are of the old PC-version (19200 kW ISO rating), while the others are of the new PE-version (22600 kW ISO rating). The utilisation of LM2500 as prime movers on all the platforms has resulted in obvious benefits due to standardisation in operation and maintenance. Recently, 4 new LM2500 units have been ordered for the Sleipner Vest platform, which is in the construction phase.

Gas compression trains

The Statfjord and Gullfaks fields are oil fields with associated gas. The gas production is partly exported through the Statpipe gas gathering system and partly reinjected. The crude oil production is handled by an offshore ship loading system.

There are a total of 11 gas turbine driven recompression trains on the platforms in the Statfjord and Gullfaks fields. A typical offshore recompressor train and process plant is shown in figure 2. Each gas recompression train has 4 process stages. It has 3 compressor casings, where one casing covers the 2nd & 3rd process stages. The 4th stage is the pipeline compressor, and its power consumption is above 50% of the train power requirement. The gas turbine speed is governed by the suction pressure to the pipeline compressor. In addition, there are gas turbine driven or c.
A typical offshore compressor module is shown in figure 3. The centrifugal compressors are all equipped with oil film type shaft sealing systems. All the installations have separate lube and seal oil systems (with exception of Statfjord B & C). The LM2500 gas turbine has a separate lube oil system for synthetic oil.

**Waste heat recovery units.**

The LM 2500 has a high thermal efficiency, and its use has resulted in relative favourable exhaust gas CO$_2$ emissions. The gas turbine's thermal efficiency at site conditions has been measured at above 35% in the best operating point. The thermal efficiency can be increased further by use of waste heat recovery units. Hence, waste heat recovery units have been installed on 11 of the 42 gas turbines, see also figure 3. They are installed on the gas turbine driven generator sets on the Statfjord and Gullfaks platforms, and on the gas turbine driven pipeline compressors on Sleipner platform. The units are used for heating medium applications. The overall thermal efficiency has been raised to about 70% for one LM2500 with waste heat recovery unit. But the heat demand in an offshore process plant is limited, and it is therefore not possible to equip all the gas turbines with waste heat recovery units. Studies have been made on offshore combined cycle steam power plants. But it has not been possible to implement such plants so far, mainly due to the high weight and the high complexity. However, combined heat and power cycle have been installed for the Avon and Frame 6 gas turbines in Statpipe Kvaarstoe (landbased installation), and the overall thermal efficiency is approximately 78%, see ref. 4.
LM2500 GAS TURBINE EXPERIENCE DATA

Operating hours
Of the above mentioned offshore installations, Statfjord A was the first platform to be started up in 1979. The other platforms have been started up successively with one or two years interval. Sleipner A was the last platform to be started up in 1993. The following table 2 gives a status of the operational hours up to January 1994.

<table>
<thead>
<tr>
<th>UNIT / LOCATION</th>
<th>TOTAL HOURS, ACCUMULATED</th>
</tr>
</thead>
<tbody>
<tr>
<td>42 UNITS, LM2500 (Statfjord, Gullfaks, Veslefrikk, Sleipner offshore fields)</td>
<td>~1.5 mill</td>
</tr>
<tr>
<td>THE LM2500 OLDEST UNIT AT STATFJORD A (FLEET LEADER)</td>
<td>80000</td>
</tr>
<tr>
<td>Gasgenerator A</td>
<td>85000</td>
</tr>
<tr>
<td>Power turbine A</td>
<td></td>
</tr>
</tbody>
</table>

Performance deterioration
The LM2500 performance drop was relative high at an early stage of operation (5-8 years ago), but it has improved over the last 5 year period. Compressor cleaning is done every 750 hours. The cleaning is usually done with an off-line crank washing procedure. On-line washing system is recently installed in 9 gas turbine units.

The performance deterioration shown in table 3 is based on measurements at a constant firing temperature. Most of the gas turbine driven compressor trains are running at high loading, and the gas turbines are normally controlled by the firing temperature.

| POWER DROP | 3 -10% |
| EFFICIENCY DROP | 3 - 5% |

Maintenance Data
The maintenance concept of the LM2500 is based on a quick turbine removal and spare unit installation on the offshore platform. The gas turbine is brought onshore for repair in a turbine maintenance workshop. The data in table 4 is representative for the last 5 year period.

<table>
<thead>
<tr>
<th>TABLE 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>LM 2500 MAINTENANCE DATA</td>
</tr>
<tr>
<td>Periode between overhaul / Hot Section Repair Interval (HSRI)</td>
</tr>
<tr>
<td>Time required offshore for removal of gas turbine and install spare unit</td>
</tr>
</tbody>
</table>

AVAILABILITY FOR ONE LM2500 GAS TURBINE DRIVEN COMPRESSOR TRAIN

In the following definition for availability, the total outage hours include unplanned outage hours and planned outage hours (scheduled well in advance). The definition for running reliability is in accordance with a traditional unplanned outage formula (see for instance definition by Gas Turbine User's Association). The definition for reliability may differ slightly from the gas turbine manufacturer's formula, see ref. 5.

The experienced availability and running reliability of one gas turbine driven compressor train is given in table 5. The data are representative for the recompression train A on the Statfjord C platform, where two recompression trains are in parallel continuous operation (2x50%). The load factor is very high, which is typical for most of the gas compression trains. The availability and reliability have improved over the last years, but variations occur from year to year. The unplanned outages are mainly caused by instrumentation problems, which normally results in short stops. Further, the unplanned outages are caused by bearing and seal problems.

<table>
<thead>
<tr>
<th>TABLE 5</th>
</tr>
</thead>
<tbody>
<tr>
<td>LM2500 GAS TURBINE DRIVEN GAS RECOMPRESSION COMPRESSOR TRAIN A AT STATFJORD C</td>
</tr>
<tr>
<td>Availability = Installed hours / Total outage hours x 100%</td>
</tr>
<tr>
<td>Reliability = 100% x Installed hours / Unscheduled outage hours</td>
</tr>
<tr>
<td>Load factor = Energy produced / Rated power x Operating hours x 100%</td>
</tr>
<tr>
<td>Use factor = Operating hours / Installed hours x 100%</td>
</tr>
</tbody>
</table>
AN OPTIMISED PROGRAM FOR GAS TURBINE MAINTENANCE AND OPERATION

Traditional Operation and Maintenance.
Most aeroderivative gas turbines are designed for a hot section lifetime of approximately 25000 running hours. During this lifecycle there will be a non-recoverable degradation of the engine. The efficiency is reduced due to burnings and mechanical wear leading to increased internal clearances between stationary and rotating parts in the engine. To compensate for this degradation (efficiency drop, see table 4) and to be able to supply the required power, the fuel consumption and firing temperature have to be increased.

In a situation where the fuel costs were small and negligible, the efficiency of the engine was not so important as long as the engine was able to supply the required power. Under such operating conditions, the aim has been to obtain a maximum of running hours before removal and overhaul of the gas turbine. However, it normally resulted in a poor mechanical condition and performance at the end of the lifecycle. The expensive hot section parts have substantial damage, and a high change out rate of high cost parts is the result. Even though this traditional operation philosophy results in longer running hours between overhauls, the overhaul cost per running hour is higher due to the need for new parts for rebuilding the engines.

Maintenance costs reduction.
Since 1985, the aim has been to achieve a reduction in the gas turbine maintenance costs. In this period, there has been developed repair procedures and methods for almost all LM 2500 gas generator and power turbine parts. In parallel there has been a gradual build up of experience enabling the decision of the optimum overhaul point of the engines. An important element has been the ability to monitor the engine mechanical condition. Hence, the engines can be removed for overhauls at a time where high cost parts are within repairable limits and can be repaired at a far lower cost than the price of new parts. This philosophy has proved to be a very effective way to reduce the overhaul costs.

In fig. 4, overhaul costs for Statfjord gas generators are given for the period from 1985 to 1993. This method of operation will keep the engines at a high mechanical standard, and reduce the risk for damages and fatal breakdowns due to engine wear. Further, the engine mechanical condition and engine performance are normally closely linked together. Most of the installations are operated close to the maximum available power, and a high engine efficiency is therefore important in order to ensure a good operational flexibility and avoid production limitations.

Energy-saving and a new maintenance program.
The CO₂ tax was introduced in the Norwegian sector of the North Sea in 1991, and the efficient operation of the gas turbines became even more important. At present the CO₂ tax is 0.80 NOK per SM³ consumed fuel gas. In figure 5, an illustration of the level of the overhaul costs and the CO₂ tax as function of gas generator running time is given. The dramatic change in operational cost due to the introduction of the CO₂ tax can be seen. It must be underlined that the fuel gas value is not included. In most cases the sales gas value will be in the range from 0.5-1.0 NOK/SM³. The fuel costs will thereby be approximately the double the level of the CO₂ tax curve.

In figure 6, an illustration of the overhaul costs and increase in CO₂ tax due to engine performance drop versus engine running time is given. It underlines the importance of keeping the gas turbine performance drop within a certain limit (overhaul point), and if this limit is exceeded the result is high energy losses and thereby a high extra CO₂ tax.

Consequently, it is today very important to be able to monitor the engine performance in a reliable way. The operational philosophy has changed and to a much greater extent is based on total operational costs, including overhaul costs, CO₂ tax costs and fuel gas value. This situation represents a challenge to the gas turbine operator to aim for more energy conserving operation,
which will result in a reduction of the emissions to the atmosphere.

**CONDITION MONITORING SYSTEMS INCLUDING MONITORING OF ENERGY UTILISATION**

A thorough follow-up and condition monitoring system for the gas turbines represents a good investment, and can give a good payback in the form of a reduction of the total gas turbine costs. Condition monitoring systems are installed on most of the offshore gas turbines and compressors. Both performance monitoring and vibration monitoring systems are used. The condition monitoring of gas turbines is made on a selected number of key parameters. Difficulties have been experienced monitoring the centrifugal compressors due to varying off-design conditions. The performance curves are not always valid for all operating conditions. Flow measurements are also inaccurate due to the off design gas densities. This requires careful follow-up and supervision. The monitoring systems have also been extended to incorporate energy utilisation and recirculation losses.

**GAS COMPRESSOR TRAIN OPERATED AT OFF DESIGN CONDITIONS**

**Reliable operation at reduced efficiency**

One of the major operational problems which occurs on the recompression/pipeline compressor installations is the off design or unintended operating conditions. It is a fact that several recompression compressors are working today under substantially different operating conditions (gas composition, inlet pressure, inlet temperature, volume flow) compared to what they originally were specified and designed for. There are several reasons for these off design conditions, for example:

- Different wellstreams with other composition are phased in.
- Optimisation of the main oil/gas separation process at other temperatures and pressures.

The consequences for the compressors are as follows:

- Partly invalid performance curves, changed discharge pressures and changed compressor speed.
- Increased recirculation, poor efficiency and power losses occur.
- Anti surge control system may be inaccurate and must be rechecked.
- Condition monitoring system may be inaccurate.
It has been possible to ensure reliable operation of the compressor trains at the off design conditions. But this requires careful preparation and operational adjustments in order to match the multicasing compressor trains and the platform process plant, see ref. 3 and ref. 6. The reliable operation has often been made possible by compromising on compressor train overall efficiency, i.e. substantial recirculation or flaring losses have been experienced.

**Low molecular weights, new performance curves, changed discharge pressures and changed speeds.**

The manufacturer's tested performance curves for discharge pressure, power and massflow are only valid for the original specified design conditions, and cannot be used for the new off design conditions. In order to evaluate the safe operating limitations of the compressor, it is necessary to establish new performance curves (discharge pressure, power) for the off design conditions. The new performance curves can be made either by the use of inhouse computer programmes, or it can be obtained from the compressor manufacturers.

The off design operating conditions are caused mainly by low molecular weights and changed inlet pressures. Hence, the compressor train must run faster in order to obtain the specified discharge pressures. However, this may not always be possible due to limitation on speed and power from the gas turbine, and limitation on the other compressors in the train. This can be seen from fig. 7, where performance curves for both design conditions and off design conditions are plotted in the same diagram for all four stages. The off design curves have substantially lower discharge pressures. This is illustrative for most of the operating conditions on both Gullfaks A & C platforms. In some cases, the 4th stage/pipeline compressor might not reach the required discharge pressure even if the speed is increased as far as possible. If the required discharge pressure is 185 bara in order to export gas through the pipeline, then the compressor have to operate at speed above 100% (approx. 102%). Further, if the compressor wear over a 3 year period results in a substantial performance degradation, then it may not be possible to reach the required discharge pressure at compressor maximum speed of 105%.

![Fig. 7 Gas recompression train (Gullfaks C)](image-url)
Energy-saving, reduced recirculation losses and rewheeling of compressors.

The suction volume flows have changed substantially due to the off design operating conditions. For the 1st, 2nd and 3rd compressor stage, the suction volume flows are often substantially below the design values, and heavy recirculation around each compressor is required in order to maintain reliable operation. This leads to losses of power, which for one compressor train can be above 2 MW. Both the gas turbine fuel cost and CO₂ exhaust emission cost are therefore increased, at a corresponding cost of above 3 mill NOK/year.

Hence, it appeared logical to consider rewheeling of the compressors in order to match the new operating conditions. The pay-back time for such modifications has been between 1-2 years, which is very favourable. The highest recirculation losses have been experienced on the 1st stage compressors, and substantial energy-savings can be obtained by rewheeling of this stage. The 1st stage compressor power \( P \) and potential power-saving can be illustrated by the following equation:

\[
P = \frac{g \cdot \rho \cdot h_p (Q_F + Q_R)}{\eta_p}
\]

\( Q_F \) is the forward flow at suction conditions, \( Q_R \) is the recirculation flow at suction conditions.) By rewheeling, the recirculation flow \( Q_R \) is eliminated. However, further power-saving can be made by improvement of the polytropic efficiency \( \eta_p \). By rewheeling, it is possible to move the operating point away from the recycle line and closer to the best operating point.

In most of the gas compression trains, the 1st stage compressor is a double flow compressor with 7 impellers, see figure 8. The rewheeling studies have concluded that removal of impellers 1A and 1B can meet the Gullfaks new conditions, since the operation requires both reduced flow and reduced pressure ratio. On Statfjord, the new conditions can be met by removal of impellers 1B, 2B, 3B, changing impeller 4 to a single flow impeller and modification of the thrust balance piston. The latter rewheeling operation has already been implemented in the field.

In addition, other methods for energy saving are possible. The process parameters for the total production plant can be optimised, i.e. pressures, temperatures, molecular weights and flow are adjusted. Based on dynamic simulation, field test data and analysis of the process behaviour, it is possible to implement modifications to the process which result in a more efficient operation, see ref. 6.

Anti surge control system at off design conditions.

The safe and efficient operation of compressors is dependent on an accurate anti-surge control system. The anti-surge control system has to be based on a performance curve which is valid for all operating conditions, i.e. it must be able to compensate for the off design conditions. Most of the compressors are equipped with the "delta p" system (performance curve for \( \Delta p_{\text{oR}} \) versus \( \Delta P_{\text{oR}} \)). The "delta p" performance curve/surge line can compensate for changes in suction pressure. It can also to some extent compensate for changes in molecular weights and suction temperature. The surge line moves slightly to the left for lighter molecular weights, see ref. 7. A change in inlet temperature will also move the surge line slightly. It can therefore result in a too small surge margin for lighter gas. In addition, the "delta p" system is based on the assumption that the surge line is a parabola, but this may not be true for high pressure compressors. The system can therefore become inaccurate. Further, if the margin to the control line becomes too large, the recycle valve may open too early and give performance problems with high recycle losses.

In order to avoid problems in the field, the surge margin has to be checked or tested for any new operating conditions. In the field, it has sometimes been necessary to adjust the surge margin after all operating conditions are known.

Alternative anti-surge control systems have also been tried. A system which is based on performance curves with non-dimensional/normalized parameters can cover the
off design conditions, as long as the k-value change is moderate. It is used in some of the compressors, and may be retrofitted on other installation. Further, a anti-surge control system based on polytropic head versus volume flow \((h_p \text{ versus } Q^2)\) performance curve have also been tried. The system requires a larger amount of instrumentation and is therefore more expensive. It has been justified for complex installations with parallel compressor trains (Statfjord A).

The best type of performance curves for anti-surge control system has to be evaluated individually for each installation and operating parameters. In addition, the type of instrumentation and controllers have to be optimised to achieve best economical operation. In many cases, the "delta p" system has shown to be a relative robust and simple system, but it requires follow up and adjustments to new operating conditions.

**IMPROVEMENTS TO COMPRESSOR SHAFT-SEALING SYSTEMS**

**Oil film shaft-sealing systems**

Shaft-sealing systems represents another major problem-area for the gas recompression trains. The shaft seal systems are of critical importance for the safe and reliable operation of the centrifugal compressors. But seal oil systems are cumbersome and costly in operation. The seal oil systems require a high level of attention due to their complexity. All the existing compressor installations (Statfjord, Gullfaks) are fitted with oil film seal systems. Some of the major offshore related problems which have required improvements are:

a) **Degassing and reduced oil flash point.**

In oilfields with associated gas (Statfjord, Gullfaks), the compressors have to handle rich gas with a substantial amount of heavy hydrocarbon components. The heavy components are difficult to degas from the seal oil. It is difficult to maintain the oil flash point above a minimum value, and improvements are required. In some cases, a vacuum degassing unit has been installed next to the atmospheric degassing tank, and it has improved the results. Further, the introduction of nitrogen in the atmospheric degassing tank has also improved the flash point.

b) **Overhead tank on the weather deck.**

The tanks are normally located outdoor on the weatherdeck, and problems are connected to the low ambient temperature (normally between 0 - 15 deg. C). If the oil temperature during start-up is below 25-30 deg.C, then the oil viscosity becomes unacceptable. Further, the reference gas in the tank may condensate partly at these low temperatures, and mix with the oil or partly substitute the oil in the tank. Insulation and cladding of the tanks is normally not enough. Improved heating, heat tracing and weather protection walls are often required.

c) **Oil ingress into compressor.**

Oil ingress into the compressor can become an expensive maintenance problem. In some recompression trains, it has been necessary to refill 100-300 liters of oil per day. The oil disappeared either to the flare systems, or into the compressors and thereby caused fouling of the downstream process equipment and pipeline. The cleaning up costs can be substantial. It is important to ensure that compressor shaft labyrinths are in good condition and that the start-up and shut down procedures are satisfactory.

d) **Bacteria problems.**

Bacteria problems have been experienced in the seal oil systems, and they have caused damage to seal components. Bacteria problems may be caused by water or H₂S in the oil system, and can be cured by centrifuges or the addition of chemicals.

**Dry gas seal systems**

The dry gas seal system has the advantage of lower gas leakage rates, lower space requirements and lower complexity than the above mentioned oil film seal system. Statoil has today several compressor installations with dry gas seal systems, but the dry gas seals are for the time being only used on landbased installations. The dry gas seal installations are listed in table 6. The triple dry gas installations in Statpipe Kaarstoe have in general

<table>
<thead>
<tr>
<th>TABLE 6</th>
<th>DRY GAS SEAL INSTALLATIONS IN STATOIL PLANTS</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Location / Seal type</strong></td>
<td><strong>Start up</strong></td>
</tr>
<tr>
<td><strong>EXISTING INSTALLATIONS:</strong></td>
<td></td>
</tr>
<tr>
<td>Kaarstoe K-Lab</td>
<td>1987</td>
</tr>
<tr>
<td>Triple seal, Kaarstoe Salesgas</td>
<td>1993</td>
</tr>
<tr>
<td>Triple seal, Ettel gas storage</td>
<td>1993</td>
</tr>
<tr>
<td><strong>Tandem seal,</strong></td>
<td></td>
</tr>
<tr>
<td><strong>UNDER CONSTRUCTION:</strong></td>
<td></td>
</tr>
<tr>
<td>Sleipner Vest</td>
<td>1997</td>
</tr>
<tr>
<td>Tandem seal, Heldrun</td>
<td>1995</td>
</tr>
<tr>
<td>Tandem seal, Troll / Kollsnes</td>
<td>1996</td>
</tr>
</tbody>
</table>

* The numbers in brackets are static or settle out pressure.
performed well, see also ref. 4. The new Sleipner Vest offshore platform (under construction) involves 5 compressor trains with dry gas seal.

CONCLUSION

The availability of the offshore gas turbines and compressors has been improved during the last few years. The gas turbine driven compressor trains have been a major factor in the high production regularity of the Statfjord and the Gullfaks platforms.

In the future, it will be necessary to put more emphasis on efficient operation due to CO₂ tax, fuel costs, exhaust gas emissions and maintenance costs. It is necessary to ensure the overhaul of gas turbine before performance drop and energy costs get above a certain limit.

It has been possible to obtain reliable operation of the gas compressor trains at off design operating conditions, but it requires careful supervision and follow-up. The off design operating conditions have resulted in reduced efficiency and recirculation losses. Hence, performance improvements have been made to the compressors in order to obtain energy savings and to reduce emissions. Due to the impact of CO₂ tax, there is a very short pay-back time on modification to the compressors and process plants.

In order to obtain reliable operation, improvements are also required to the compressor shaft seal systems. Dry gas seal systems are being installed on new platforms that are under construction.

REFERENCES


NOTATION

\[ p_1 = \text{suction pressure} \quad \text{(bara or barg)} \]
\[ p_2 = \text{discharge pressure} \quad \text{(bara or barg)} \]
\[ P = \text{power} \quad \text{(W,kW)} \]
\[ t_1 = \text{suction temperature} \quad \text{(deg. C)} \]
\[ T_1 = \text{suction temperature} \quad \text{(deg. K)} \]
\[ Q_1 = \text{suction flow} \quad \text{(m³/h)} \]
\[ M = \text{molecular mass} \quad \text{(kg/kmol)} \]
\[ k = \frac{c_p}{c_v} = \text{Adiabatic exponent} \]
\[ g = \text{gravitational acceleration} \quad \text{(m/s²)} \]
\[ \Delta p_c = \text{pressure differential over compressor} \quad \text{(bar)} \]
\[ \Delta p_{oa} = \text{pressure differential over orifice} \quad \text{(mbar)} \]
\[ h_p = \text{polytropic head} \quad \text{(m)} \]
\[ n = \text{polytropic exponent} \]
\[ N = \text{speed} \quad \text{(RPM)} \]
\[ \rho = \text{density} \quad \text{(kg/m³)} \]
\[ n_p = \text{polytropic efficiency} \]