Shell U.K. Exploration and Productions Experience With Large Gas Injection Compressors on the Northern North Sea Platforms

This report highlights many difficulties, covering a period of two years, of gas compressor commissioning in the Brent Field. It is intended to give an insight of the problems encountered, with the objective to, wherever possible, avoid repetition. It is in most cases the DESIGN STAGE, where the lessons learned should be implemented, because it is often too difficult and sometimes impossible to make improvements in compressor modules of offshore platforms.

INTRODUCTION

With the discovery of the giant Brent oil field in the British sector of the North Sea, located half way between the Shetland islands and the Norwegian coast it became clear to the operators, Shell UK Exploration and Production (a 50/50 Shell/Esso venture) that the quantities of associated natural gas becoming available with the production of crude oil were of such an order that either transportation to shore for sales or preservation by recompression into the formation had to be realized.

For the time being, the findings of Shell UK Exploration and Production (hereafter abbreviated to Shell Expro) by exploration drilling in their allocated blocks of the North Sea did not yield natural gas in sufficient quantities economically to justify gas-handling equipment. Gas/oil ratios of production platforms of Auk and Dunlin oil fields are in the order of 200, while gas-quantities of Cormorant 'A', Fulmar and Cormorant-North oil fields can be predicted to be of such value that economic utilisation/preservation does not justify the installation of gas-handling facilities.

In relation to the other above mentioned discoveries the Brent field oil and gas reserves are of paramount importance, hence this paper is focussing primarily on the gas handling equipment installed on the production platforms of this oil field.

DESIGN CONCEPT OF GAS COMPRESSION MODULES

The development of the Brent oil field called for the installation of four production platforms, designated Brent Alpha, Bravo, Charlie and Delta. In view of the remote location it was decided in the early 1970's by Shell Expro engineering management that, as a basic principle, all major power requirements were to be provided by a central gas turbine driven power-plant, consisting of a number of gas-fired aircraft derivative gasturbo-alternators in the 12-16 MW range with a back-up of liquid-fired gas turbo-alternators (1-4 MW) for the time no gas is available in the initial stages and during production shutdowns.

As a consequence of the above philosophy it followed that all major pumps and compressors should not be gasturbo-driven, hence large electric motors were specified to power all compressors of the Brent field platforms.

1. Brent 'A'

Gas injection:

| Flow (total from 4 stages of gas/oil separation) | 62 Nm³/s |
| Suction pressure | 138 bar. abs. |
| Suction temperature | 24°C |
| Discharge pressure | 418 bar. abs. |
| Mol. weight | 21.501 |

Gas boosting:

| 1st Stage (from 4th Stage gas/oil separator) |
| Flow | 1.82 Nm³/s |
| Suction pressure | 2.07 bar. abs. |
| Suction temperature | 60°C |
| Discharge pressure | 6.55 bar. abs. |
| Mol. weight | 39.90 |

| 2nd Stage (from 3rd Stage gas/oil separator) |
| Flow (1st Stage + 2nd Stage together) | 7.10 Nm³/s |
| Suction pressure | 6.21 bar. abs. |
| Suction temperature | 23.9°C |
| Discharge pressure | 33.79 bar. abs. |
| Mol. weight | 30.90 |
Above process requirements have been realised with the following hardware:

<table>
<thead>
<tr>
<th>Stage</th>
<th>Flow (Nm³/s)</th>
<th>Suction Pressure (bar. abs.)</th>
<th>Suction Temperature (°C)</th>
<th>Discharge Pressure (bar. abs.)</th>
<th>Mol Weight</th>
</tr>
</thead>
<tbody>
<tr>
<td>3rd Stage (from 2nd stage gas/oil separator)</td>
<td>21.38</td>
<td>33.44</td>
<td>23.9</td>
<td>138.94</td>
<td>23.24</td>
</tr>
<tr>
<td>Flow (1st, 2nd and 3rd Stage together)</td>
<td>62</td>
<td>138</td>
<td>24</td>
<td>418</td>
<td>21.501</td>
</tr>
<tr>
<td>21.38 Nm³/s</td>
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<td>23.24</td>
<td></td>
</tr>
</tbody>
</table>

Flow (total from 4 stages of separation) 62 Nm³/s
Suction pressure 138 bar. abs.
Suction temperature 24°C
Discharge pressure 418 bar. abs.
Mol weight 21.501

Gas boosting:
1st Stage (from 4th Stage gas/oil separator)
Flow 2.65 Nm³/s
Suction pressure 1.72 bar. abs.
Suction temperature 37.8°C
Discharge pressure 6.53 bar. abs.
2nd Stage (from 3rd Stage gas/oil separator) 11.28 Nm³/s
Flow (1st Stage + 2nd Stage together) 6.21 bar. abs.
Suction pressure 23.9°C
Discharge pressure 33.82 bar. abs.
3rd Stage (from 2nd Stage gas/oil separator) 34.10 Nm³/s
Flow (1st, 2nd and 3rd Stage together) 33.44 bar. abs.
Suction pressure 23.9°C
Discharge pressure 138.95 bar. abs.

Numbers in italic are values from work sheets.

Fig. 1 A simplified flow diagram

2. Brent 'B', 'C' and 'D'

Gas injection:
Flow (total from 4 stages of separation) 62 Nm³/s
Suction pressure 138 bar. abs.
Suction temperature 24°C
Discharge pressure 418 bar. abs.
Mol weight 21.501

Gas boosting:
1st Stage (from 4th Stage gas/oil separator)
Flow 2.65 Nm³/s
Suction pressure 1.72 bar. abs.
Suction temperature 37.8°C
Discharge pressure 6.53 bar. abs.
2nd Stage (from 3rd Stage gas/oil separator) 11.28 Nm³/s
Flow (1st Stage + 2nd Stage together) 6.21 bar. abs.
Suction pressure 23.9°C
Discharge pressure 33.82 bar. abs.
3rd Stage (from 2nd Stage gas/oil separator) 34.10 Nm³/s
Flow (1st, 2nd and 3rd Stage together) 33.44 bar. abs.
Suction pressure 23.9°C
Discharge pressure 138.95 bar. abs.

Discharge pressure of 414 bar. abs. and a discharge temperature of 71°C.

The gas is handled in 2 stages from 138 bar. abs. suction pressure, 279 bar. abs. intermediate pressure to the 414 bar. abs. discharge conditions. The first stage is compressed at the head-end of each of the six cylinders, while the second stage compression takes place at the crank end side.

To boost the gas from lower gas-oil separators pressures to injection compressor suction one train of centrifugal barrel compressors has been installed, consisting of a tandem train compressor, directly coupled running at a speed of 8819 RPM, driven by a 7554 KW 1800 RPM electric motor through a Maag speed increaser.

Fig. 2 View of injection compressor

With ordering of the complete compression modules on 12th September 1974, three years of manufacturing, installation and hook-up were required before commissioning started in April 1978.

With the compressor module of Brent 'D' being the first to be ready after hook-up, it was decided to start with commissioning the reciprocating compressor and prepare it for gas injection.

During the initial runs the following defects were noticed:

A. Compressor Valve Cover Nuts
During routine assembly/dismantling of valves, valve cover nuts were found cracked. Initial cause was...
thought to be over-torqueing, but a more thorough investigation revealed that several nuts showed poor workmanship, inclusions of dirt etc. Materials of which these nuts were made was ASTM A-307 carbon steel, to be applied for structural purposes and low pressure equipment only.

As a result, all valve cover nuts had to be changed for nuts of medium-carbon steel ASTM A194-Gr.2H, equal to SAE 1045.

B. Valve Chair Failures

This problem was also noticed early, 1978. When the valve cover nuts were torqued to 1400 ft-lb and the valves later changed due to normal maintenance/commissioning, the chairs were found to have plastically deformed. The chairs yielded in the perforated area immediately above the valve. This region has the smallest cross section and was overloaded. The first response from manufacturer was that the damage was due to improper installation. The next suggestion was to turn down the diameter of the chair by 0.010" to avoid binding. The manufacturer was notified that the problem originated by mechanical weakness and that further weakening of the chair could hardly help. A new design review was initiated which resulted in the production of valve chairs of SAE 4140 materials with heat treatment. These new design chairs have performed satisfactorily.

The main impact of this design deficiency, beyond delays, was the massive amount of manhours required to extract the deformed and jammed chairs.

C. Pulsation Bottle Failures

The reciprocating compressors are fitted with 8 pulsation bottles; 2 first stage suction-bottles and 2 second stage suction-bottles located on top of the cylinders (see Fig.4) while 2 first stage discharge-bottles and 2 second stage discharge bottles are installed under the 6 cylinders.

During October 1978, after 392 running hours - 7 x 10^6 cycles - a 4 inch choke tube of the first stage discharge bottle of the 2-4-6 cylinder side. Fiberscope inspection indicated that one of the compartment baffles had come loose and moved about 12 inches within the bottle. Furthermore a choke tube supporting strap of the first stage discharge bottle had torn loose and the lost magnet was found in the second stage suction bottle inlet strainer. This time the second stage discharge bottle had to be removed and was replaced by a bottle from Brent 'C' injection compressor.

Further operation resulted in another baffle failure of the recently replaced second stage pulsation bottle on the 2-4-6 cylinder side of the injection compressor on Brent 'D' in January 1979. This failure took place after only 8 hours of operation and subsequent inspection showed a similar mode of failure as experienced with the first baffle failure of the original 2nd stage discharge bottle.

With this alarming situation the pulsation bottle design was reviewed, primarily by the design consultant, in addition to an independent engineering consultant.

Additionally, pulsation and vibration measurements were carried out on Brent 'A's injection compressor in March 1979.

An analysis of the pulsation bottle construction was also carried out by the independent engineering consultant and the analysis demonstrated quite conclusively that the failure could be attributed to fatigue, due to excessive stress in the failed location: a finite element model, loaded by pulsation pressures showed that the peak stress occurs at the
point of failure, with a predicted failure time in the order of hours.

In order to make an efficient repair possible a redesign was sought which could be carried out without having to cut open a bottle more than once. This led to a design with short choke tubes.

The baffles themselves were made differently: instead of the original 'inverted dish on skirt' baffle a weld cap was now taken. Due to time constraints different types of weld cap were used, and machined to fit.

The 'Short choke tube situation' was analysed on the analog simulation, results of which showed a marked decrease in the low frequency pulsation pressures and some increase in higher frequencies.

A sketch of the basic principles of the modification is shown in Fig.6.

![Fig.6 Pulsation Bottle Modification of Internals](image)

The repairs effected on 28 of the 32 pulsation bottles of four compressors turned out to be a major exercise. The inaccessibility, in particular, of the discharge bottles, combined with inadequate lifting and hoisting facilities, made dismantling, transportation and from the platforms, as well as re-installation an almost impossible task.

Repairs have been carried out by three pressure-vessel manufacturers, one situated in Rotterdam, The Netherlands; one situated in Scotland and the third situated in England. The biggest problem was to retain exact nozzle-positions after cutting, replacement of internals, welding and heat-treatment of the bottles.

Although no suction bottles failed during initial compressor operation, it was agreed to have their internals modified likewise.

On Brent 'D' compressor however, the suction bottles have so far not been refurbished and with running hours well over 4000 on that compressor and so far without any sign of suction bottle failure the repair of those bottles has been deferred until a more opportune period in the future.

D. Piston Rider Band Failures

Original Design. The original design was for two lead bronze rider bands 5/16" thick and 4.5" wide welded onto a piston of 4140 steel. The process specification was changed from sweet to sour gas which resulted in a change of piston material to 17-4 PH stainless steel. The rider band was changed to glass-filled teflon of 0.175" thickness with eight ventilation notches at each end of the band. Due to installation considerations the single band of 4.5" width at each end of the piston was replaced by two bands each 2.25" in width.

Running History. After 195 hours running the compressor valves were fouled by bits of deteriorating glass filled teflon rider bands. Severe deterioration had taken place due to excessive heat. All rider bands both crank end and head end had failed and all had several holes in them. The possible causes of this were:

1. Inadequate lubrication.
2. Too low a temperature limitation on the glass filled teflon.
3. Rider bands acting as compression rings due to inadequate ventilation.

Lack of lubrication was identified as the most probable cause in spite of lubrication feed rates of over 10 times the manufacturer's recommendation. It was found that the balancing valves in the lubrication system were improperly connected and essentially bypassed resulting in an unbalanced pressure system. The system was modified so that the original balancing valves were installed correctly, additional balancing valves were installed and the system re-piped for greater flexibility. The compressor manufacturers were of the opinion that lubrication was the sole reason for the rider band failures and confirmed the acceptability of the rider band design and the material for this service.

As the manufacturer could not deliver replacement rider bands in sufficient time, spares were obtained from a separate supplier. The material was changed to graphite filled teflon due to availability and later, when the new bands were installed, the manufacturer recommended this change from a design point of view. After a further 197 hours of running a pulsation bottle failure caused a shutdown and the rider bands were examined.

It was apparent that they were failing in a similar manner to the glass filled teflon bands. One crank end rider band had a single hole in it and from the general condition of the bands, it was estimated that the time between failures was increased by six times. The improved lubrication system was therefore helping but it was obvious that the original ventilation design was entirely inadequate. The bands were being forced out radially against the cylinder liner by gas under the band. This was causing excessive heat due to friction.

It was felt by Shell Expro that the best solution would be a two piece bronze rider band. This required piston machining and as a result the compressor was restarted on 17th November 1978 with three types of rider bands to test their suitability. The compressor ran for 143 hours before it was again stopped due to pulsation bottle failure. From the information gained from this test it was decided to modify all pistons to accept the two piece bronze rider band, 0.5 inch thick.

A short test has been carried out with 2 pistons fitted with a single-cut 0.5 inch thick graphite filled rider bands, but the satisfactory runs with the two piece bronze rider bands made Shell Expro decide not to apply teflon rider bands and standardise on bronze rider bands for the time being. A sketch of present piston design is given in Fig.7.
Fig. 7 Reciprocating Compressor Piston

E. Cylinder lubrication

The CLBA-6 reciprocating compressors are provided with a centralised lubricating system to service:

a) cylinders and pistons
b) pressure-packing boxes of piston-rod

A schematic layout is depicted in Fig.8 and its principle is the following:

A single HP pump provides a pulsating flow of cylinder lubrication oil which is distributed via divider blocks and balancing valves to the required lubricating points at the required injection pressures.

The system is protected from interruptions in flow by a proximity switch, while rupture discs are installed to prevent failures due to overpressurisation.

In view of the requirement to provide lubricant to the cylinders as well as the pressure packings, two pumps have been provided, one for each application, while balancing valves have been provided for the differences in pressure required at the injection points concerned.

From the beginning the system has been extremely troublesome, with following failures being the cause of shutdowns as well as undue wear resulting from lub. oil starvation.

a) Proximity switch failure, while no interruption of flow existed, causing unnecessary shutdowns.

b) Balancing valves set wrongly, causing inadequate amounts of oil to injection points concerned.

c) Leaking connections.

d) Burst rupture discs, mainly due to blocking distribution pistons.

E) NPSH problems at the pumps, mainly due to low ambient temperature and high viscosity of the oil.

f) Water and dirt in the system.

The situation was, and is, aggravated by the phenomenon that with the prevailing gas pressures the lub. oil tends to evaporate in the gas. For this reason the injection quantities (FIST-RULE: 1 pint/day for 2 x 10^6 sq. ft. of swept surface) have been increased ten-fold in an effort to maintain an oil film before the oil evaporates.

In order to eliminate above mentioned problems the following steps have been taken:

1. Lub. oil storage is heated, suction line to pump is heat traced.

2. Suction line has a stand-pipe in storage tank to prevent water/dirt from entering the pump.

3. The two-pump configuration will be replaced by three pumps; two pumps will service the HP and LP points of the pressure-packings SEPARATELY, so that balancing valves can be eliminated. For the cylinder lubrication also, no balancing valves are necessary.

4. An improved type of proximity switch will be introduced, while distribution blocks of an improved design have been ordered, facilitating venting and servicing.

Since items 3 and 4 are not yet implemented, the cylinder lubrication system is still extremely vulnerable.

Fig. 8 Typical Trabon System Application

F. Failures of Heat Exchanger Tubing

Particularly in the compressor module of Brent Bravo a number of heat exchanger tube leaks have hampered commissioning and operation.

Basic problems have been identified as:

a) millscale corrosion

b) improperly rolled tubes

The first problem has been caused by careless preservation of the heat-exchangers after their installation in the module. With millscale present at the bottom of the tubes, which should have been removed prior to shipment, and subsequent exposure to moist air during the long period between installation and commissioning, corrosion pitting progressed along the bottom of the tubing. This has been confirmed by removal and inspection of a leaking tube from one of the heat-exchangers.

It was initially assumed that millscale corrosion was the only cause of the tube leaks, further inspection revealed leakage between tube sheet and tubes. By executing re-rolling to an increased expansion of 7%, covering the full depth of the tubes in the tubesheet, further failures of this nature have been considerably reduced.

As a long term solution it is considered that replacement of heat-exchangers or their internals is the only effective way to prevent further failures.

G. Big-End Bearing Failures, Warped Crankshafts

With reinstatement of all pulsation bottles, efforts to start-up the reciprocating compressors on Brent 'A' and 'B' during 1979 were marked by big-end bearing failures. Although initially not recognised, these failures were caused by dirt in the lubricating oil circulating system. A major factor was that initial circulation was carried out, using the motor-driven standby lub. oil pump, having only one third of the flow capacity of the shaft-driven main pump. With dirt not being picked up in the down-stream piping, initial start-up and sudden increase of lubricant oil flow
caused dirt to enter the bearings.

Due to the fact that the big-end bearings were made of aluminium (with 6 pct. tin) any dirt is prohibited. The main bearings, being of white metal, have much better embedding properties, although it was established that with each big-end bearing wipe the adjacent main bearings were found slightly wiped as well.

Run-out checks on crankshaft journals also indicated that, due to heat developed during a big-end bearing wipe/seizure, the crankwebs distorted and the shaft warped to the extent that runouts over 0.020 inch were recorded. With assistance of the crankshaft supplier the shafts have been straightened in situ by applying heat with an acetylene torch. This exercise has been completed successfully four times.

By hooking up a motor driven portable pump of bigger capacity than the main lub. oil pump, it was possible to obtain clinically clean circulating systems, enabling normal commissioning of Brent 'A' and 'B' reciprocating compressors in November/December 1979. Brent 'D' reciprocating compressor suffered a big-end bearing failure and warped crankshaft in July 1980 after some 3000 running hours. Although not clearly proven, failure was considered to be caused by water in the lub. oil. The stacked paper disc type filter elements were completely water-logged, but did not create a significant pressure differential as might be expected with water blocking the filter.

It was however explained by the filter manufacturer that sudden blockage with water causes pressure differentials, while gradual blockage, such as condensate from the air mixing with the lubricant does not. Since the filter cartridges had logged 1500 hours, it was decided to change them at intervals of 750 running hours or three months time, whatever comes first.

CENTRIFUGAL BOOSTER COMPRESSORS COMMISSIONING

Following the injection compressors, the commissioning and start-up of the five centrifugal booster trains (1 train on Brent 'A', 2 trains on Brent 'B' and 'D' respectively) was a relatively easy job. The main problems concerned with them are described below.

A. Low Speed Coupling Lock

As detected by vibration at 30 Hz on the gearbox input side, it was soon evident that coupling lock was the cause. The reason for this was subsequently traced to interference between the motor-side coupling hub O/D and the I/D of the shroud of the corresponding coupling sleeve.

The shrink-fit between the hub and the E-motor shaft was too great, causing the hub O/D to grow beyond allowable tolerance when fitted. On four of the five motors the hub was extremely difficult to pull off after which the bore and the O/D had to be machined to obtain proper shrink-fit and proper clearance between hub and sleeve-shroud.

Subsequent runs proved to be very successful, no vibrations have been further encountered.

B. Inadequate Degassing of Contaminated Seal Oil Drain

All booster trains have one common lub/seal oil system with a 6000 gallons main tank, installed in the base of the skid. The contaminated drain from the inner seals is piped to a degassing tank, which is an integral part of the lub. oil tank.

Before the first runs were completed, Shell Expro were already aware of the possibility that, with the heavy ends of the gas dissolving in the contaminated oil drain, the degassing system could be grossly under-
A. Piston Failures

On 16th February 1980 the injection compressor on Brent 'A' suffered a broken piston in the No.2 cylinder after 1050 running hours since new.

The material of the pistons/integral piston rods was 17-4 PH stainless steel; originally the manufacturer offered delivery in SAE 4140 and this material was confirmed with the order, but in 1975 Shell Expro changed to 17-4 PH because of expected sour-gas duty with continued water injection.

The failure of the above piston caused only slight consequential damage, a bent connecting rod and a dent in the cylinder head. The decision was taken to crack-test each piston on all three platforms, resulting in detection of 3 pistons having a crack developing at the fillet of the land between the first two piston-ring grooves. Running hours were 1500 since new.

This alarming situation required drastic measures. A total of 24 pcs identical pistons/rods were ordered from the manufacturer of SAE 4140 material, and with clear instruction as to the required radius in the piston-ring land fillets. It was already evident that the broken piston showed very rough machinery marks, while failure mode indicated cracking initiation to originate from sharp corner of land neck.

A detailed investigation has been carried out on the broken parts of the piston by Koninklijke Shell Laboratorium Amsterdam (KSLA), who indicated that crack propagation, due to directional properties of the 17-4 PH pistons is expectable with ANY degree of surface finish, while for SAE 4140 the indications were similar, though not to such a marked extent as 17-4 PH. In the meantime there was no other option but to install the SAE 4140 pistons in the compressors of all three platforms and resume operation.

On every opportunity the pistons have been crack-tested, so far no cracks have been found with running hours now approaching 2000.

B. Undue Pressure Packing Ring Wear

The reciprocating compressors are fitted with standard leaded bronze packing rings. The bottom, being the pressure breaker, is followed by 9 sets of radial packing rings backed by axial rings.

Since commissioning the 3 compressors on Brent 'A', 'B' and 'D' the maximum life so far obtained on these rings is not higher than 1000 hours, which represents some 25 percent of what can be considered as normal for this duty.

At present a number of alternatives are being considered, to be implemented whenever opportune.

1. Application of teflon radial rings with metal backup rings.
2. An improved design of bronze rings comprising - 2 pressure breaker rings, 2 pressure relieved rings, 5 chamfered radial rings plus backup rings, 1 atmospheric radial/axial ring.
3. Metallic packing rings.

In view of the fact that none of these alternatives have yet been implemented, no further comments can be given. One factor which could have influenced packing ring life is the lubrication-rate, still many times the recommended quantity.

CONCLUSION

This report highlights many difficulties, covering a period of 2 years, of gas compressor commissioning in the Brent Field. It is intended to give an insight of the problems encountered, with the objective to, wherever possible, avoid repetition. It is in most cases the DESIGN STAGE, where the lessons learned should be implemented, because it is very often too difficult and sometimes impossible to make improvements in compressor modules of offshore platforms.