Optimum gas compressor selection and design to maximize Brent production

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ABSTRACT

Timely upgrading of the gas export compressors to suit the late life requirements of the reservoir is increasing hydrocarbon recovery from the Brent field in the North Sea. The strategy driving initial selection, re-wheeling and subsequent replacement of the compressors is explained. Advances in technology and modelling methods allowed each phase of the development to extend the operational envelope of the machine beyond that previously possible in a predictable manner. The design, test and offshore installation methodologies and challenges are discussed in the context of brownfield upgrade constraints. The vendor's rotor-dynamic optimisation of a critical compressor is outlined.

1 THE BRENT FIELD – FROM OIL TO GAS PRODUCER

The Brent field has been a cornerstone of the United Kingdom's (UK) offshore oil and gas industry since its discovery in 1971, some 500 km North East of Aberdeen. The co-venturers Shell UK Ltd and Esso Exploration and Production UK Ltd installed four platforms - Alpha, Bravo, Charlie and Delta - to develop the field with 600.10^6 m^3oe of oil and 290.10^6 m^3oe of wet gas initially in place (m^3oe = cubic meters oil equivalent). The platforms were originally configured for oil production with water injection. In the early 1990s it was decided to change the reservoir management strategy: Reducing the reservoir pressure from 380 bar-a to ultimately 70 bar-a would liberate incremental gas reserves of 58.10^6 m^3oe. The Long Term Field Development (LTFD) project installed an integrated two-stage separation/ gas compression/ dehydration process module on each platform Bravo, Charlie and Delta to enable reservoir depressurisation (1).

2 LTFD COMPRESSOR SELECTION

The need for a large gas compression capacity within limited weight and space constraints drove the LTFD equipment selection. Each platform was fitted with a single gas export compression train, which emphasized the need for an aero-derivative gas turbine driver to allow rapid turbine replacements during maintenance. Cooper-Rolls' RB-211/6562 gas turbine was selected as the prime mover and GE Oil & Gas Nuovo Pignone (GE-NP) supplied the gas compressor. The two-stage back-to-back compressor was coupled to the driver through a single-helical gearbox and fitted with tandem dry gas seals. The lube oil system of the compressor was shared with the gas turbine and gearbox.

Gas export availability was a key value driver for the LTFD project. A proven, reliable and readily maintainable high flow compressor was required. GE-NP designed a compressor rotor composed of 600 mm diameter, two-dimensional impellers in a three wheels per stage configuration (Table 1). The vertically split barrel type casing allows the bundle to be replaced without removing process pipework.

Several factors made the LTFD compressor a challenging design considering the experience and methods available in the early 1990s. The compressor was specified in accordance with NACE (2), as there was a risk of reservoir souring in later field life. The need to deliver high flow and polytropic head in a single casing resulted in the impeller tip velocity being relatively high. As a consequence, the impeller stresses were also significant and it was necessary to select materials with sufficient strength to sustain the stresses but also with low enough hardness to satisfy the NACE requirement. Material selection was further complicated by the presence of carbon dioxide and water.

Rotor-dynamic analysis was performed that took account of the destabilising flow effects in the labyrinth seals. This was particularly important in the area of the centre balance drum that separates the discharge volutes of the two compression stages. All three units underwent full load testing in the factory using the contract drivers. No stability problems were encountered in testing or in service. The compression trains were maintained for optimal performance by Rolls-Royce and GE-NP staff under a healthcare contract which supported an enhanced maintenance programme. None of the three compressors required a major unplanned intervention in over 20 years of cumulative run life.

3 DEVELOPMENT OPTIONS POST LTFD

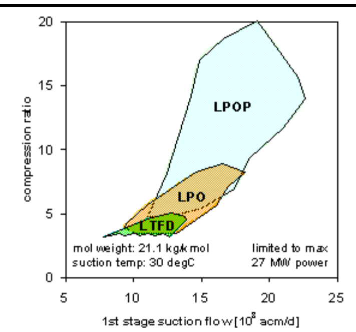
As the gas reservoirs depleted, well productivity declined below the design flow rate of the LTFD compressor. Lowering well backpressure enhances well performance and reduces the ultimate reservoir pressure, which in turn increases gas recovery. Options evaluated to reduce well backpressure whilst maintaining export pipeline pressure included multiphase pumps and various suction and discharge booster compressor configurations. All these concepts were found to be economically unattractive due to space, weight, power and flow assurance issues. The most cost effective approach was to modify the existing gas export compression train on each platform for a lower suction pressure by upgrading the compressor, retaining the gas turbine driver and reusing the base skid and the auxiliaries.

The compressors were upgraded for reduced suction pressure/ reduced mass flow in two phases to match the production decline of each platform Brent Bravo, Charlie and Delta. The shutdowns were staggered and aligned with other in-field activities to minimize avoidable production deferment:

Phase 1 : Upgrade the LTFD compressor as far as possible within the existing casing design. The Low Pressure Operation (LPO) project replaced the compressor bundle and gear internals. This well proven upgrade method only required a short platform shutdown.
Phase 2 : The Low Pressure Operation Plus (LPOP) project replaced the existing compressor with a new longer casing to accommodate additional impellers. This option only became feasible in recent years due to advances in rotor and labyrinth seal design techniques that enabled detailed evaluation of the behaviour of such a compressor. The upgrade required significant brownfield work and an extended shutdown to remove the original machine, install and align the new one. Piping spools, valves and gear internals were also replaced.

Table 1 / Figure 1 : Brent gas export compressors – comparison of key parameters

Project	Brent LTFD	Brent LPO	Brent LPOP
GE-Nuovo Pignone model	2BCL506/A	2BCL506/A	2BCL506-8/A
Maximum continuous speed (rpm)	8086	9083	9508
Number of impellers (per stage)	3 / 3	3 / 3	4 / 4
Impeller diameter (mm)	600	630	630
Bearing span (mm)	1953	1953	2166
Bearing diameter (mm)	130	130	150
Suction / discharge pressure (bar-a)	35 / 140	20 / 140	8 / 140
Polytropic head 1st / 2nd stage (m)	8500 / 8100	13000 / 11800	21000 / 18500
Flow (MM scm/d) @ max 27 MW	11.76	8.02	3.86
Log dec @ MCS	//	0.169	0.226
Wachel log dec @ MCS	//	0.021	0.056
Balance drum seal	stepped labyrinth	Stepped Labyrinth	tapered honeycomb
Shunt holes	no	Yes	Yes
Swirl breaks on impeller eye labyrinths	no	No	Yes
Impeller material	A182 Gr F22	A705 Gr 630	Virgo 38



The complete topsides process was reviewed for flow, pressure and temperature changes resulting from the new operating conditions. Increased compressor discharge temperatures affected pipe class and valve specifications as well as piping/equipment stress levels. Whilst the heat load on the gas coolers remained essentially unchanged (given no change to compressor driver power), these had to be re-rated for the increased temperatures. Fuel gas, gas dehydration and off-gas compression systems were impacted by lower pressure and were modified and/or operated differently.

4 LPO RE-WHEELED COMPRESSOR SELECTION

The polytropic head of the LPO compressor was significantly increased over the LTFD design by optimising impeller geometry and increasing rotational speed (Figure 1). The rotor line-up was again composed of two-dimensional impellers in three plus three configuration (Table 1). It was not possible to accommodate additional impellers within the length constraint of the existing casing. The selected impellers had the highest head geometry available within the GE-NP range of impeller families and an increased 630 mm diameter, the largest for the casing size. The impeller tip speed together with the NACE requirement resulted in the LPO bundle being the limit state design of the time.

In the absence of a gas turbine skid, compressor or gearbox casing it was not possible to factory test the new compressor or gearbox internals. Start-up risk was mitigated through design verification, use of proven impeller designs, GE-NP quality control (high speed verification and balancing of the rotor) and commissioning procedures. Offshore testing showed a good match with the predicted performance. The modification activities were completed within the planned shutdown durations and all three re-wheeled compressors proved to be as reliable in service as the LTFD units had been.

5 LPOP REPLACEMENT COMPRESSOR SELECTION

The following design objectives and constraints were specified for the LPOP compressor:
Objectives
O1 - Maximize polytropic head (minimize suction pressure)
O2 - Wide operating range (high start-up pressure required for fuel gas)
O3 - Materials suitable for sour service
Constraints
C1 - Maintain casing diameter and re-use the base plate of the existing compressor
C2 - Achieve a casing weight of less than 18 t (vs 21.5 t standard weight)
C3 - Sustain nozzle loads up to 5x NEMA
C4 - Maintain the size and location of the discharge nozzles

GE-NP studied all aspects of the compressor design to find the best compromise between the conflicting requirements while ensuring a safe and reliable compressor with a high level of performance. Initial efforts focused on meeting constraints C1 to C4. The complete casing design was reviewed and the main change to reduce weight (C2) was a reduction in the wall thickness of the second stage suction. The casing was validated for various load conditions (C3) by finite element analysis (FEA) and verified in a hydraulic test.

The polytropic head (O1) could only be increased by adding impeller stages (four plus four vs three plus three), and by increasing shaft speed (Table 1). Additional polytropic head was achieved by increasing the LPOP rated speed to the maximum continuous speed (MCS) of the LPO design (Figure 1). The increased gear ratio pushed the pitchline velocity and loading of the gear teeth to the maximum allowed by Shell. A special material known as Virgo was selected for the impellers to maximize resistance to sulphide stress corrosion cracking (O3) whilst achieving the required strength. The impeller interference fit geometry was optimized utilizing Design of Experiment and FEA techniques.

As a consequence of the compressor's wide operating range (O2), the residual axial thrust due to the pressure distribution over the rotor showed much greater variation and higher peak values than in standard GE-NP applications. Numerous evaluations were performed to optimize the balance drum design. Constraint C4 limits the interstage wall thickness while objectives O1 and O2 increase the differential pressure across the interstage diaphragm. Deformation of the diaphragm in the area of the balance drum seal had to be minimized since excessive movement would have had a detrimental effect on rotor stability. The axial length of the new compressor was minimized to prevent rotor stability issues. The aerodynamic design of the two inlet volutes was modelled using computational fluid dynamics and optimised within the space constraints.

6 LPOP COMPRESSOR ROTOR – DYNAMIC VERIFICATION

Rotor-dynamic stability issues due to the bearing span, rotational speed and polytropic head capacity of the LPOP machine had to be resolved. The criticality of the design is evident when comparing the compressor against GE-NP's historical database (Figure 2). All Brent machines are within GE-NP's "warning zone" in the Kirk-Donald plot (3), however, the LPOP compressor is close to the stability limit. Sections 6.1 to 6.3 discuss the design optimisation and verification done to avoid an unstable design. The analysis was conducted in parallel with a third party, the South West Research Institute (SwRI).

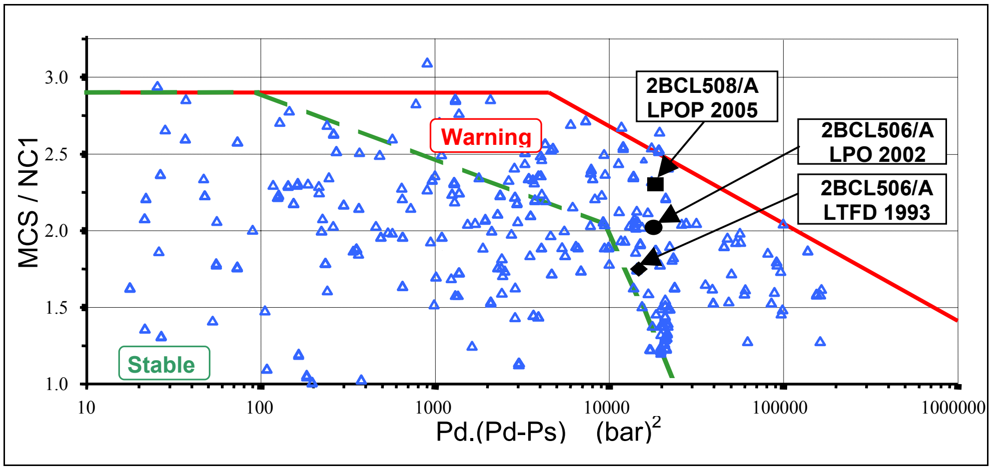


Figure 2 : NP-GE design references and Brent compressors in Kirk-Donald plot.

6.1 Undamped critical speed analysis

The starting point for the rotor-dynamic verification was an Undamped Critical Speed Analysis (UCSA) per API 617 (4) during the compressor feasibility design study. The first equivalent shaft was the result of a simple evolution of the previous (LPO) machine. The scheme was modified as the design proceeded to arrive at the final optimised shaft. The UCSA considers bearing characteristics purely in terms of their stiffness coefficients which were calculated using two different tools: an internal GE-NP tool that has correlated well over many years with GE-NP standard bearings, and XLTRC[2] from Texas A&M University (5). A good match was found between the two sets of calculation results.

On the basis of the UCSA, three decisions were taken:

- Increase journal bearing diameter to increase both the shaft stiffness and the damping effect from the bearings,
- Increase bearing journal clearances to increase the damping of the first mode,
- Reduce coupling weight to increase the second critical speed. This also resolved the issue of low damping ratios and low separation margins on the gearbox pinions at low load.

6.2 Rotor response to unbalance

The Rotor Response to Unbalance analysis adds the damping effect of the bearings to the system. Bearing characteristics were calculated using the same tools as for the UCSA and again a good match was found between the two calculations. Tilting pad journal bearings with 5 pads, load on pad, 0.6 offset pivot, high preload were selected. For this kind of journal bearing with a non centred pivot, stiffness and damping coefficients are not frequency dependent. This is in accordance with GE-NP's experience and technical literature (6), allowing a simplified synchronous stability analysis to be followed.

The results for the LPOP compressor confirmed the preliminary information obtained from the UCSA: the first critical speed (3550 rpm) was well below the operational range and lightly damped; the second critical speed (8700 rpm) was inside the operational range but highly damped. The log decrement of the rotor + bearings system at MCS was 0.226 and hence close to Shell DEP requirements (minimum acceptable value is 0.2) (7). Further stability analysis was required to include the effect of the seals and clearances.

Tests with different oil temperatures and bearing clearances showed that the second critical speed could sometimes be lower and more highly damped and sometimes higher and with less damping than in the nominal condition. Synchronous vibrations seemed in the worst scenario to be manageable with slight changes to bearing clearances in case of problems during the machine run test. Selection of the bearing clearance that offered the best response over the wide operating range of the machine completed the design work in this area with respect to synchronous rotor response aspects.

After selecting the internal seal for optimum rotor stability, a seal clearance study was performed in order to check the potential for rubbing at the interstage balance drum. The analysis considered gravity sag and rotor bearing eccentricity. Rubbing was not predicted even for the most onerous condition, which occurs when operating at the alarm limit (MPR).

6.3 Stability analysis and internal seals selection

A numerical model of the LPOP compressor was developed in XLTRC^2 (5) to include all sources of direct and cross coupling, stiffness and damping. The primary analysis was conducted at MCS/MPR with a check analysis at the MCS/choke condition. The following seal configuration was agreed between manufacturer, client and third party:

- Impeller eye seals = toothed labyrinths with nominal clearance
- Final balance drum seal = abradable seal with teeth on rotor
- Interstage balance drum seal = tapered honeycomb

The interstage balance drum seal near the centre of the rotor is critical for rotor stability and generally the most important source of damping in this type of back-to-back compressor (8). A tapered honeycomb design was selected for LPOP to guarantee rotor stability. The LTFD and LPO machines had used a stepped toothed labyrinth (Table 1). Moreover, in order to reduce the pre-swirl of gas entering the honeycomb, shunt holes were incorporated into the interstage diaphragm.

The tapered honeycomb design needs to be correctly modelled to avoid instability: In particular conditions a divergent honeycomb seal can exhibit strong negative stiffness, especially at low frequency. Effective damping can also turn negative at low frequency. The ISOTSEAL code used for the analysis closely predicts the appropriate behaviour and has been experimentally validated in the Turbomachinery Laboratory test rig (9).

The tapered honeycomb seal is sensitive to the difference in clearance between the inlet and the outlet of the seal. It was decided to design for a seal taper of 0.1 mm since this value is well centred in the stable region of the log decrement versus seal taper plot (Figure 4). The deformation of the interstage diaphragm, including the honeycomb seal and the balance drum, was calculated by means of a FEA that considered thermal, centrifugal and pressure effects (Figure 5). The definitive honeycomb seal machining (cold) was so determined to ensure that the honeycomb seal assumes the correct taper under working (hot) conditions.

At this point in the analysis, the compressor was theoretically proven to be stable across the full operating range. However, a further stability margin was incorporated in the form of swirl brakes on the inlet of each impeller eye labyrinth (Figure 3). A third party (SwRI) calculated that the swirl brakes increase the log decrement by 0.13 at MCS/MPR. PTC 10 Type 2 testing (10) of the final compressor with and without swirl breaks showed that the swirl brakes also had a positive effect on thermodynamic performance.

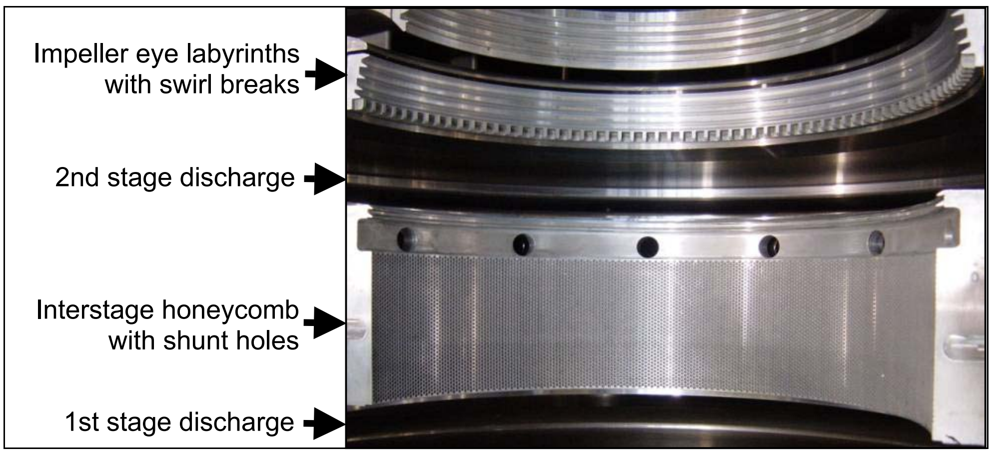


Figure 3 : Brent LPOP interstage diaphragm seal detail

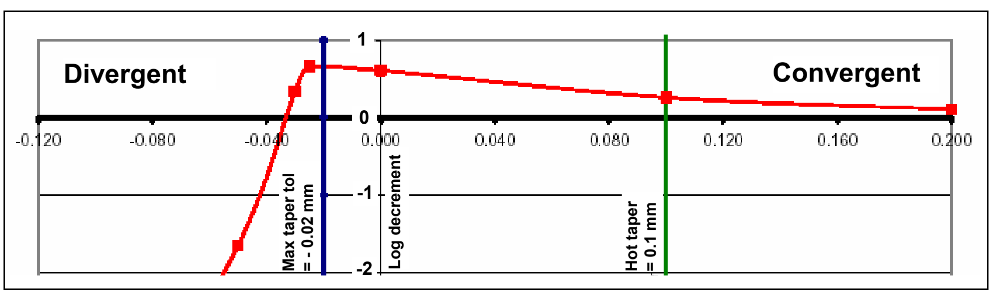


Figure 4 : Brent LPOP log decrement vs honeycomb seal taper at MCS/MPR

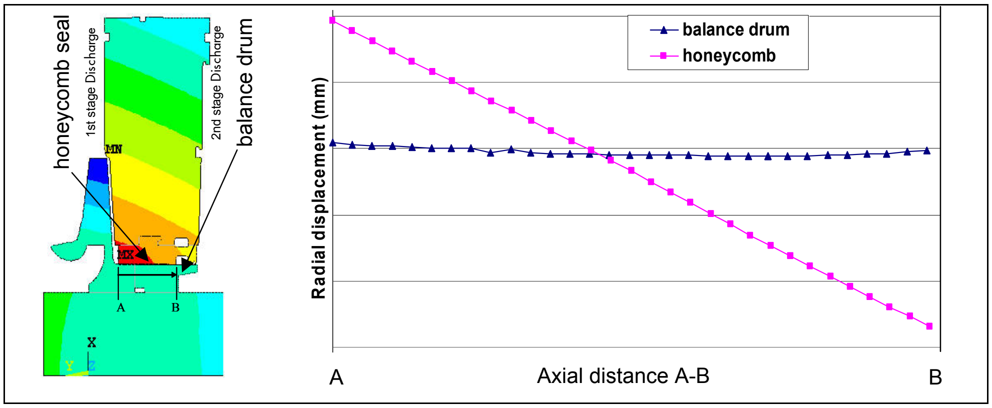


Figure 5 : Brent LPOP interstage diaphragm at MCS/MPR (hot condition)

7 LPOP BROWNFIELD CONSTRUCTION

The design constraints imposed on GE-NP limited changes to and maximized the reuse of existing equipment. Detailed design checks of the installed equipment for the new process conditions confirmed that no major equipment modifications would be required external to the compressor. However, the brownfield implementation posed its own challenges.

All lifts had to be performed using the platform cranes to avoid reliance on a lift vessel. Considering crane radii and dynamic factors, this limited the weight of a ship-to-platform lift to 18 t and of a platform inboard lift to 35 t. The compressors were therefore dismantled after shop testing and packaged into custom containers for transport. A frame with an integral gantry crane was built to provide a self-contained environment for offshore compressor re-assembly prior to the shutdown. The assembly frame replicated the compressor baseplate supports thereby allowing the contract special tooling to be used.

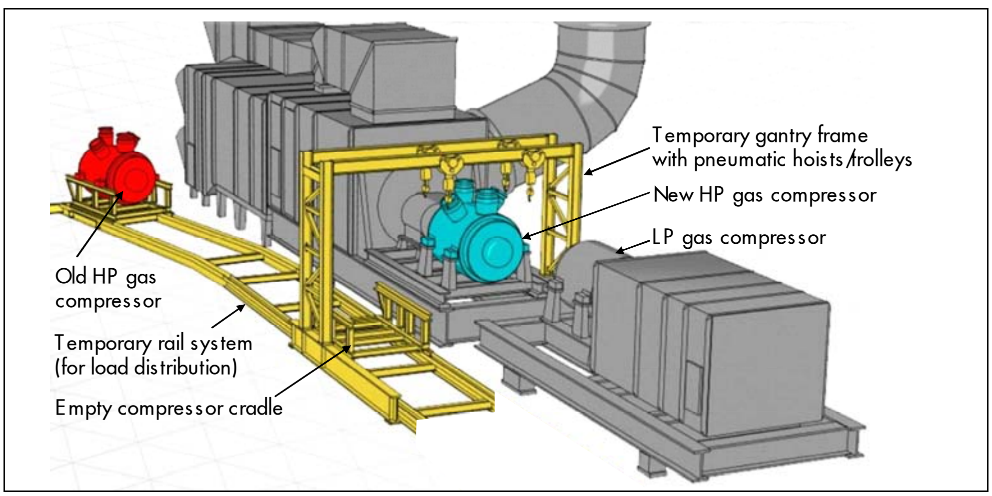


Figure 6 : Brent LPOP compressor mechanical handling

No offshore hot-work (welding) was allowed. This required all piping spools to be pre-fabricated to exact dimensions. A three dimensional (3-D) model of the new compressor within the module was developed based on an offshore laser scan survey of the existing configuration as well as a survey of the connection points on the new compressor. Particular attention was paid to the relative location of the spool termination flanges, flange angles and bolthole orientations.

Mechanical handling aids were purposely devised to facilitate a safe installation (Figure 6). The point cloud data from the offshore laser scan survey was used for clash checking and optimisation of the compressor installation path and temporary fixtures. The compressor had to be lifted above the existing skid supports before being translated off the skid and lowered onto the adjoining walkway. On removal of the compressor piping spools, a temporary gantry was constructed for this purpose within the module. Fail-safe pneumatic hoists and trolleys operated from a remote console ensured a controlled lifting operation. The gantry was load tested onshore, match marked and inspected on re-assembly to dispense with an offshore load test.

The compressors and the large tie-in spools were placed in cradles and transported through the module over a temporary rail system which was installed prior to the platform shutdown to minimise delay. The rail system distributed the compressor weight over the module structural members. The cradles had built-in jacks to help re-alignment on the rails for direction changes and were pulled on skates using a chain hoist attached to the rail cross-beams. All equipment and handling procedures were proved in onshore trials, thereby minimizing the risk of problems offshore.

It was not possible to perform an onshore string test of the LPOP compressors. Design verification, API 617 mechanical run tests of each machine (4) as well as a successful PTC 10 Type 2 (10) test mitigated the technical risk. An offshore stability trial procedure was developed based on dynamic process simulations to explore the operating envelope of the compressor in a controlled manner and safely identify any potential limitations. (At the time of writing, the first LPOP compressor installation was proceeding offshore but the machines had not been tested under full load conditions.)

8 CONCLUSIONS

The LTFD redevelopment laid the foundations for depressurisation of the Brent field. The two step LPO/LPOP compressor upgrade strategy minimized risk and lifecycle cost whilst allowing progressive modification of the machines for lower suction pressures in line with late life reservoir and production requirements. Each new compressor design leveraged the proven technology of its time so as not to jeopardise machine reliability and gas availability. The technical innovations incorporated in the LPOP compressor extended the operational envelope beyond that thought possible at LTFD. Advances in modelling techniques and design verification provided the confidence to install the LPO and LPOP machines without an onshore string test. Attention to planning, practice and procedures enabled the team to overcome the brownfield installation challenges and deliver high value projects that unlock additional gas reserves whilst enhancing operational efficiency.

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