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Originally appeared in HYDROCARBON PROCESSING® JULY 2017 issue, pgs 73–79. Posted with permission.

Upgrade process compressors to enhance production

Centrifugal compressors are production-critical equipment in refineries, petrochemical plants, and offshore oil and gas production facilities. Increases in their capacity are easy to justify, even in a tight commercial market. Payback periods are very low—typically less than 5 yr—compared to greenfield projects.

In this specific case, which is from a field in the Caspian Sea, re-engineering and upgrading of the various components of the low-pressure compression trains enhanced the actual volumetric flow capacity of each compressor by approximately 60% (e.g., from 3,500 m^3/hr to 5,650 m^3/hr). **TABLE 1** provides a comparison of the old vs. new operating conditions.

The higher capacity was achieved by re-wheeling the compressors with a new set of impellers for the new operating conditions, upgrading the gearbox, installing a higher-rating motor and other related modifications to the facility.

Centrifugal compressor. The original compressor casing was designed to accommodate the axial length of seven impellers in series. Increasing the operating speed by 1,000 rpm could potentially develop an additional 22% of differential head across the compressor. This increase in operating speed to meet the new differential head duty allowed the supplier to reduce the number of impellers from seven to six. A lower number of impellers meant that the six impellers and the corresponding stators of greater width, suitable for higher volumetric flow, could be installed across the available axial length of the existing compressor casing (**FIG. 1**). The 12-in. compressor gas inlet and 8-in. outlet flanges were found to be suitable for the new operating conditions. The revised, higher gas inlet and exit velocities were below the maximum allowable gas velocity (norm) of 30 m/sec (**TABLE 2**).

The higher impeller tip speeds of 230 m/sec at the new operating speeds were within the supplier's allowable tip speed of 245 m/sec for sour gas applications. Material stresses were within the supplier's allowable design stress norms for A182 F6NM (NACE)-compliant impellers.[1] The impellers were overspeed tested in the supplier's shop at 115% of the new maximum continuous operating speed of 11,638 rpm (**TABLE 1**).

TABLE 1. Comparison of new vs. old operating conditions of the flash gas compressor

Description	Unit	Old condition	New condition	Change, %
Actual volumetric flow	m^3/hr	3,530	5,644	59.89
Efficiency	%	79	79.1	0.13
Operating speed	rpm	10,533	11,638	10.49
Increased head available from speed change	$h \propto speed^2$	10,533	11,638	22.08
Head	N–m/kg	138,210	133,631	–3.31
Number of impellers in casing		7	6	–14.29
Power required	kW	3,350	4,742	41.55
Old and new uprated motor rating	kW	4,500	5,360	19.11
Pressure ratio	Ratio	2.48	2.38	–4.06
Suction pressure	kPa	CONFIDENTIAL		–4.47
Discharge pressure	kPa			–8.35
Suction temperature	°C			6.25
Discharge temperature	°C			–1.5
Mole weight				–0.53
Mass flow	kg/hr			49.04
Standard volumetric flow	MMcfd			50.69

A compressor lateral vibration analysis of the new rotor design was conducted. The calculated values showed that further Level 2 stability analysis was not required.

- The actual logarithmic decrement value of 0.3 was higher than the minimum pass criteria of log decrement value of 0.1 required to limit the lateral analysis to Level 1 of API 617.[2]
- The ratio of minimum cross-coupling forces (to achieve zero damping; i.e., logarithmic decrement = 0) to actual cross-coupling forces was found to be greater than 2.0 ($Q_0/Q_A = 3.63$).
- The critical speed ratio [(CSR) maximum operating speed divided by first un-damped critical speed on rigid supports] was calculated to be 1.9. The average gas density inside the compressor was 25.6 kg/m^3. Plotting these values on the CSR diagram showed that the new rotor was within the specified region "A" indicated in API 617.

TABLE 2. Gas velocity at flange

Flange size	Flange size, in.	New gas velocity, m/sec
Gas inlet flange	12	21.5
Gas outlet flange	8	25.54

FIG. 1. A compressor package.

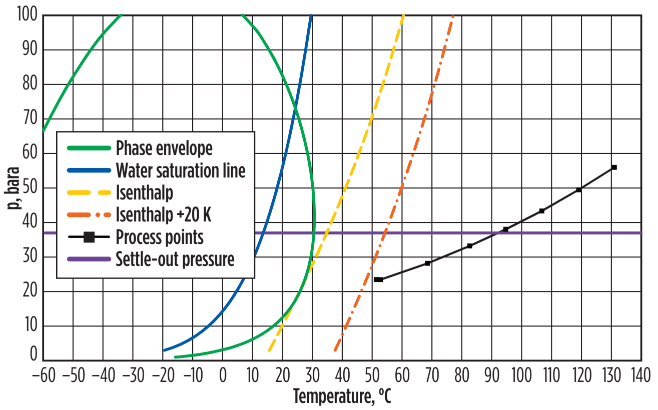

FIG. 2. Primary seal gas phase envelope diagram and separation margin.

Lateral analysis values of the new compressor rotor are shown in **TABLE 3.**

Couplings. Analysis revealed that both the existing low- and high-speed couplings were inadequate for the new power requirement and compressor configuration. Both couplings were redesigned to meet supplier and API 671 requirements.[7] A new quill shaft coupling was designed using a service factor of 1.5.

A low-speed, flexible coupling of higher stiffness and improved hub design, with a longer hub length, was installed between the new, higher-power motor and gearbox. To accommodate the new motor design, the axial length of the new coupling was increased by 8 mm, and the new spacer was built 23.5 mm longer.

Dry gas seals. The design of the existing dry gas seals was found to be suitable for the new operating conditions (**FIG. 2**). However, to reduce offshore production downtime and prevent a potential loss of production, new sets of dry gas seals were purchased for both compressors. Seal gas leakage rates are shown in **FIG. 3.**

TABLE 3. Lateral analysis design values of the new compressor rotor

Critical speeds	NC_1 = 5,300 rpm	NC_2 = 17,460 rpm	NC_3 = 18,260 rpm
Bending modes	First bending mode at compressor	First conical mode at compressor	First bending mode at coupling
Amplification factor	3.3	5.3	5.6
Separation margin	55%	50%	57%

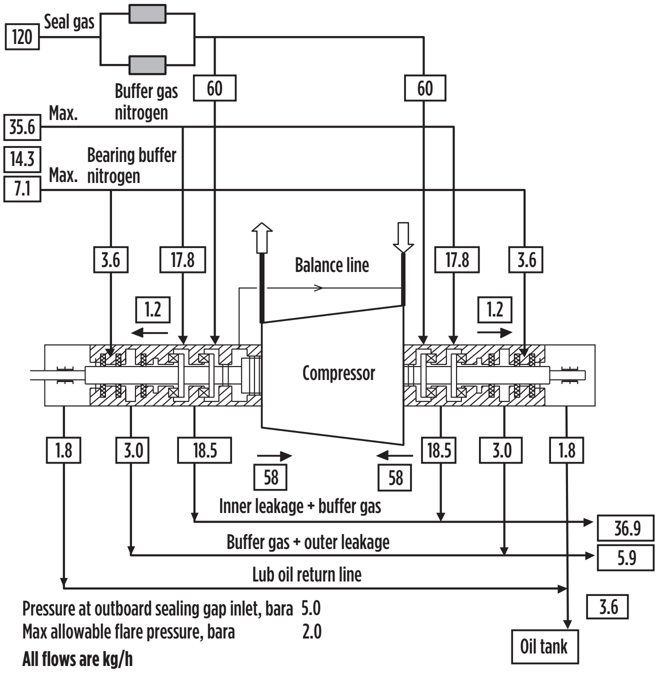

FIG. 3. Seal, buffer and separation gas leakage for new operating conditions.

Gearbox. A new gearbox (**FIG. 4**) that was suitable for a higher speed ratio and a higher power rating (40% increase) was required to meet the compressor capacity upgrade requirements. The comparison of the new design to the existing one showed that the higher power and speed needed to obtain a 59% increase in gas compression capacity can be accommodated in the same model gearbox casing. The transmission ratio changed from 7.1 to 7.82. One gearbox casing was reused with a new set of installed gear wheels. For the second compressor, a new gearbox casing with gear wheels was purchased.

> As per the schedule, the entire turnaround of the two compression trains was completed in four weeks. Lessons learned on the first train helped reduce the turnaround time for the second train.

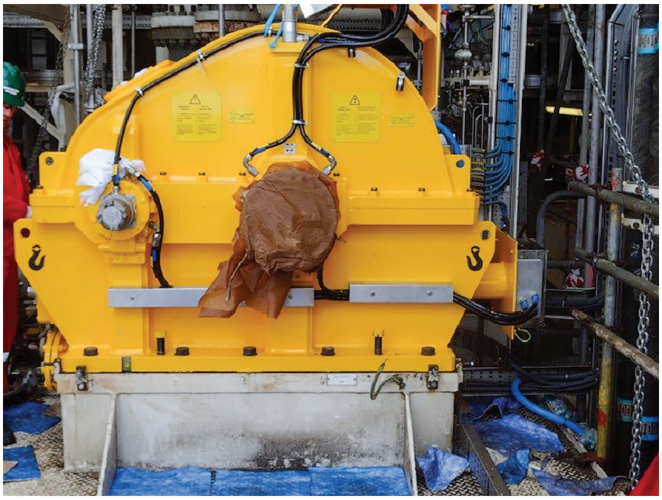

FIG. 4. To meet the compressor capacity upgrade requirements, a new gearbox that was suitable for a higher speed ratio and a higher power rating was required.

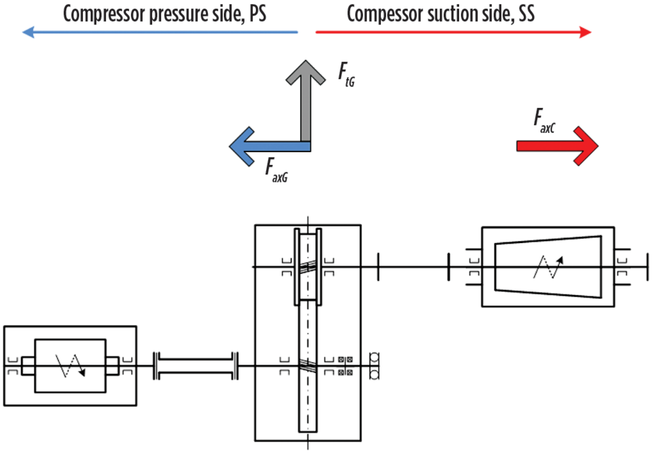

FIG. 5. Schematic view of thrust forces of high-speed train.

The design of the compression train is such that the axial thrust of the compressor is absorbed by the thrust collar of the high-speed gear (**FIG. 5**).

As can be seen in **TABLE 4,** the calculated thrust on the suction side (SS) and on the pressure side (PS), for the worst operating case (when one or more thrust balance components fail), were found to be within the design/allowable limits of the thrust collar design.

The maximum thrust on the discharge PS and the SS of the thrust bearing (mounted on the low-speed gear wheel) were found to be –9,670 N and –29,810 N, respectively. The thrust bearing installed on the low-speed train line is designed for 72,500 N, in both directions. It is sized for more than twice the actual thrust, therefore meeting API 617 criteria of actual thrust loads during normal operation, and not exceeding 50% of the design loading.

Base frame. The compressor casing with new internals, a new gearbox and a new motor were remounted on the existing common base frame with anti-vibration shock mounts (AVM). The weight of the compressor and high-speed coupling were unchanged.

The total weight of the equipment on the base plate was increased by approximately 4 metric t.

Induction motor. New motors with higher ratings (5,360 kW) were installed to meet the higher power demand for the upgraded compressor. Based on the mean torque difference of 40%, during startup the following times were determined:

- Starting time required to reach full speed (unloaded)—8 sec
- Starting time required to reach full speed (loaded)—11 sec
- Starting time required to reach full speed at 80% voltage—12 sec

TABLE 4. Thrust forces on high-speed line (compressor and high-speed gear pinion)

	Compressor thrust		Thrust collar limit		Margin	
	PS	SS	PS	SS	PS	SS
	[N]	[N]	[N]	[N]	[%]	[%]
Thrust collar	1,020	19,120	–10,673	38,227	110	50

PS = Thrust load toward compressor discharge pressure/flange side;
SS = Thrust load towards compressor suction/flange side; N = Newton

TABLE 5. Coupling safety factors—analysis confirmed that the complete train will operate safely for the design life

	Safety factor	Acceptance criteria
High-speed coupling shrink fit safety factor	9.81	≥ 1.15
Low-speed coupling hub shrink fit safety factor	5.3	≥ 1.15
Low-speed coupling peak torque (10,000-cycle) rating	3.52	≥ 1.15
Low-speed coupling momentary torque rating	4.03	≥ 1.0

- Motor safe stall time, when starting hot with 105% voltage (worst case)—13.6 sec
- Motor safe stall time, when starting cold with 100% voltage—20 sec.

The calculations estimated that the start times were lower than the worst case: stall time when the motor is starting hot. Field tests confirmed that the starting duration estimates were correct.

To fit the new motors on the existing baseplate, the new motors were designed to meet the following construction features:

- The centerline height of the new motor remained the same as the existing motor
- To use the existing baseplate, the new motor foot hole location was modified to meet the existing foot hole location.

To meet the limitations imposed by the existing baseplate and the compressor casing, the middle section of the new motor was lowered to meet the existing centerline height of 630 mm. The center of gravity of the new motor was 8 mm closer to the centerline, thereby improving the skid stability.

The new motor's first critical speed of 2,321 rpm was more than 125% of the motor rated speed of 1,488 rpm. The worst-case response due to the center body unbalance was 65 microns at 2,400 rpm, and 15 microns at the running speed. These were considered an industry-acceptable design by the motor supplier and met the requirements of API 541.[3]

The main motor terminal box was designed to accept the existing main terminal cable. No design changes were made to the connector housing or to the power supply cables. To improve the motor condition monitoring, an extra winding temperature measurement was added.

Torsional vibration analysis. A torsional vibration analysis of the compression train, including the couplings, the gearbox and the motor, was conducted. The resulting Campbell diagram is shown in **FIG. 6.**

A torsional harmonic response analysis was carried out, and the torsional stresses along the low- and high-speed shaft lines were calculated. The highest alternating stress in both shaft lines (low- and high-speed) did not exceed 0.1 N/mm^2, which was much lower than the stress limits of the materials used in the various shafts. Both shaft lines (low- and high-speed) were found to be suitable for the infinite number of starts.

TABLE 5 shows the calculated safety factors of the low- and high-speed couplings.

Lube oil system. The frictional losses of the new compressor, gearbox and motor bearing were similar to the old design, and the total lube oil requirement was marginally higher (**TABLE 6**). Therefore, no changes were made to the orifice(s) supplying the lube oil to the compressor, the motor or the gearbox. The existing lube oil pump capacity of 550 l/min was found to be sufficient for the new duty.

Compressor control system. The design and the operating philosophy of the compressor was unchanged. Suction throttling remained the primary mode of compressor performance control. To accommodate lower operating suction pressure and temperature without the formation of hydrates in the recycle line or the suction control valve, new alarms and trips were

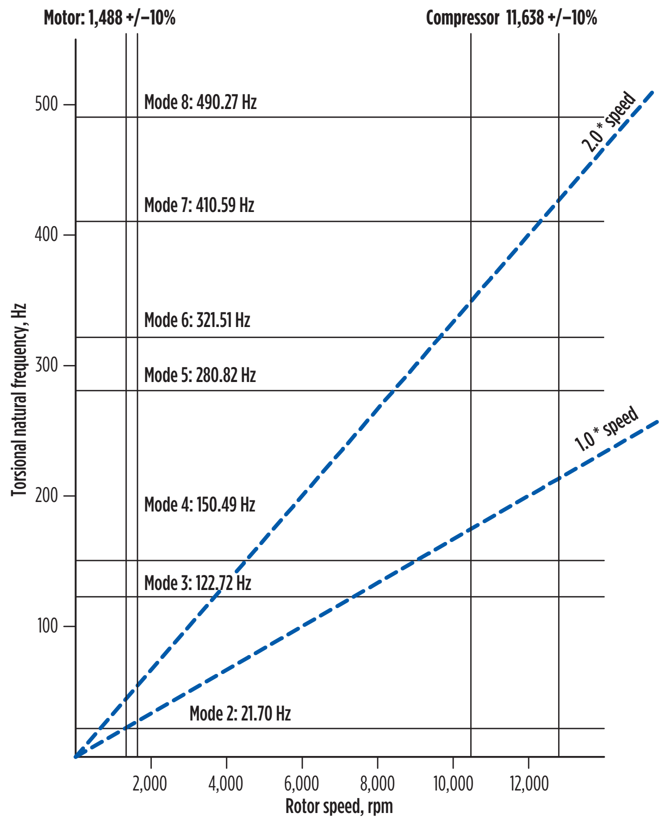

FIG. 6. Campbell diagram of the new compressor rotor, gearbox and motor.

TABLE 6. Lube oil system, friction heat generated and oil requirement

Description	Existing bearings design: Frictional loss, kW	Existing bearings design: Lube oil required, l/min	New bearings design: Frictional loss, kW	New bearings design: Lube oil required, l/min
Motor radial bearing	4	10	4	10
LS gear radial bearing	12	28	12	25
HS gear radial bearing	90	81	90	86
Gear axial bearing	3	9	3	9
Gear mesh	40	63	51	72
Compressor radial bearings	30	76	25	77
Total frictional heat	179		185	
Total lube oil required		267		279

included in the control logic. The new values were based on field experience and suitable process calculations.

Compressor surge control and protection system. This system required modification for the capacity upgrade. The supplier was asked to provide a comparison chart of the old vs. new performance curves. This chart assisted in visualizing and verifying the sizing of all associated equipment in the gas compression circuit (FIG. 7).

A dynamic simulation was carried out by a third party to simulate the three operating cases. The simulation took into account all piping (including recycle lines) and vessels between the compressor suction inlet isolation valve and the compressor discharge isolation valve. To accommodate both the process and the surge control requirements, the new antisurge control valve size was increased from 6 in. to 8 in. to accommodate the higher recycle flow. The performance requirements of the surge control valve were (FIG. 8):

- Opening time, via positioner: ≤ 3 sec
- Opening time, via solenoid: ≤ 1 sec
- Opening rate: ≥ 33%/sec
- Valve opening characteristic: linear.

To minimize noise, low-noise valve trims were purchased. The new trims reduced the pressure over the 14 stages. The flow velocity of 39 m/sec in the recycle line was above the normal design value of 25 m/sec, but well below the typical material erosion limiting flow velocity guideline limit of 75 m/sec.

The suction flow measurement transmitters of each compressor were recalibrated for higher flow, though no changes were made to the flow measurement element. The 8-in. recycle line size remained unchanged, although the size of the recycle valve was increased from 6 in. to 8 in. The existing pipe and valve supports were found to be adequate to serve the larger recycle valve.

The new surge control line was tested during commissioning, and the new surge control parameters were incorporated into the surge control system after testing.

Process equipment changes. The process heat load for the aftercooler increased from 2,430 kW to 4,513 kW. The surface area of the installed process gas cooler(s) was found to be fit for the purpose of the new duty. The internals of the low-pressure production separator and the compressor suction separator were upgraded. The control valve on the low-pressure production separator was replaced to accommodate higher flow.

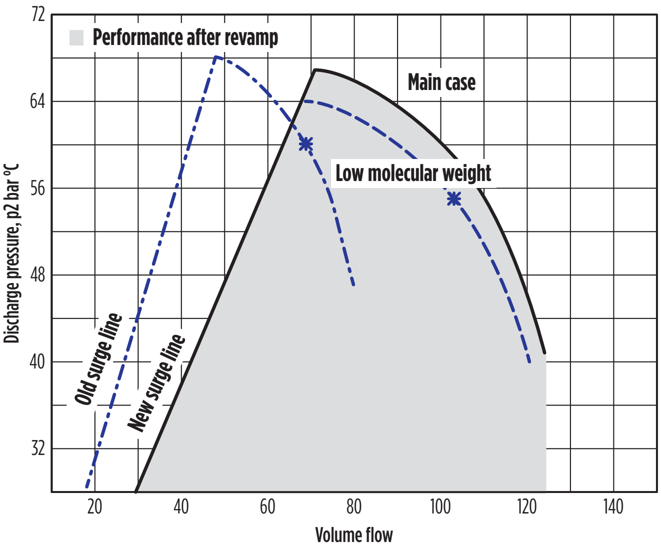

FIG. 7. Flow and pressure chart comparing old and new conditions.

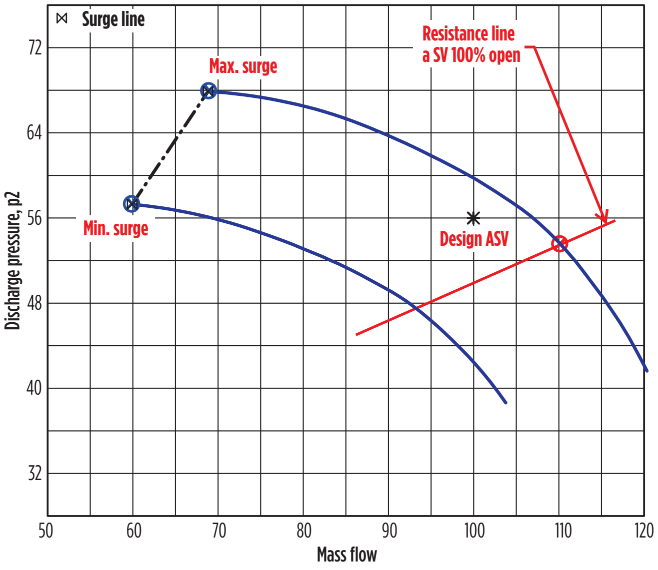

FIG. 8. Antisurge control valve (also called a surge control valve) design point.

TABLE 7. Field performance test results

Measurement	Tolerance per API 617	Guaranteed	Measured	Deviation
Flow, m³/hr	0/+5%	5644	5751	1.9%
Head, kJ/kg	0/+5%	133.63	137.58	2.9%
Power, kW	0/+7%	4742	4964	4.7%
Mole weight	±2%	Confidential		2.1%

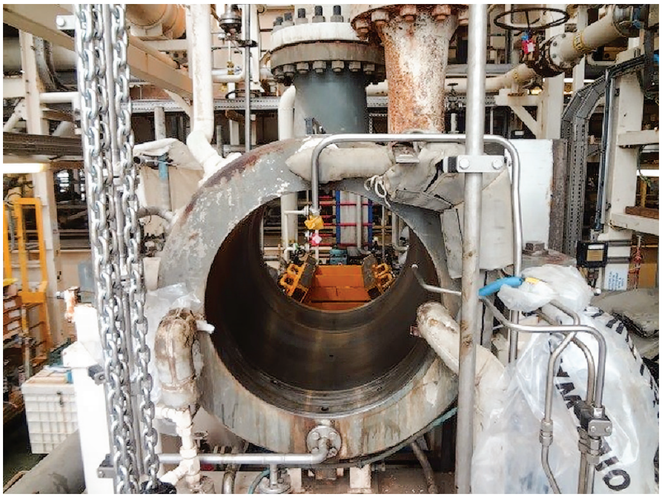

FIG. 9. A compressor casing ready for a new cartridge.

Installation, commissioning and testing. As per the schedule, the entire turnaround of the two compression trains was completed in four weeks. Lessons learned on the first train helped reduce the turnaround time for the second train. The time required to carry out each activity was discussed at length between the turnaround team and the compressor supplier. A survey of tool readiness and commissioning spares availability was accomplished onsite, eight weeks before the shutdown. A new rotor bundle removal device was purchased as part of the upgrade. The lube oil flushing and the uncoupled test run of the motor were concluded prior to the commissioning of the compression train.

The benefits of this extensive engineering verification and resources/materials planning became evident during field execution. The installation work was carried out within the allotted time, and the commissioning of the compressors was achieved without major technical difficulties (**FIG. 9**).

After the commissioning, a compressor performance test was carried out in the field, as detailed by PTC 10-2 and within the limitations of mounted instrumentation in the field.[5] Flow measurement was carried out as per ISO standard 5167-4 (measurement of fluid flow by means of differential devices).[6] A gas sample verified the gas analysis provided in the data sheet, which matched the predicted values within the allowable tolerance of the PTC 10-2 test. The gas properties were calculated using the Lee-Kesler-Plocker gas method.

Bearing losses were assumed to be in accordance with design calculations:

- Compressor bearings, motor bearings and dry gas seal losses (40 kW)
- Gearbox losses, including the bearings (156 kW).

Machinery parameters, such as motor current, bearing vibration and lube oil temperature, were monitored at 30-min. intervals. Process parameter, such as suction and discharge pressure, flow and temperatures, were also recorded and analyzed. The test results are shown in **TABLE 7.** Flow and head measured during the field performance test are shown in **FIG. 10.**

The analysis showed that the measured curves matched well with the predicted curves. The compressor developed approximately 3% more polytropic head than predicted. The upgraded compressor can deliver the predicted flow of 5,644 m^3/hr at the specified conditions.

A surge test was conducted; the calculated surge point is shown in **FIG. 10.** In the top left corner, the measured surge point is to the left (better) of the predicted value. The converted power of measuring Point 3 to the guarantee point shows a 4.7% higher power consumption at the motor coupling. This deviation is within API 617 limits. The higher power consumption is due to the slightly higher polytropic head and flow. For comparison, the electrical power consumption was measured at 2% lower than the heat balance. This difference is within the measuring tolerances.

Well-engineered and well-managed compression upgrade projects can significantly improve the profits of any operating site. HP

ACKNOWLEDGMENT

The authors would like to thank and acknowledge the contribution of their BP colleagues Tarlan Atayev, Zahid Nabiyev, Ramin Mammadov and Robin MacCuish; and Sunil Yaduka, formerly of BP. They would also like to acknowledge MAN personnel Mario Muss, Erick Schenk, Nigel Morgan and Martin Friedl for the engineering modification.

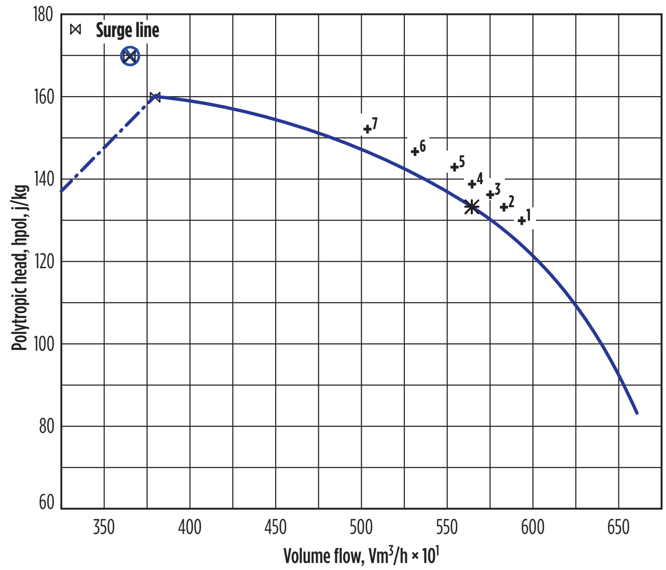

FIG. 10. Performance measurement in the field.

The authors also recognize the support of Muhammad Waqas (BP), Amal Ibadzada (BP), Parviz Tarverdiyev (BP), Nurlan Yusifzada (BP), Ilgar Abasov (BP), Werner Roth (MAN), Evgeny Vorobyev (MAN), and Reto Vorburger (MAN) for the installation and execution of the upgrade in the field.

Thanks are also due to BP C&EA/Legal and upstream engineering center and management teams for their support in obtaining the approval of the JV partners. The authors express their thanks to the JV partners and to SOCAR for agreeing to allow the publication of this paper.

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