Improving Compressor Performance to Reduce Loss Production Opportunities in Oil and Gas Production

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Abstract   This paper focuses on improving compressor performance of two compressors in operation by reducing discharge pressure while one compressor is shut down. The aim is to minimize Loss Production Opportunity (LPO) while carrying out turnaround maintenance on one out of three compressors in parallel connection. Historic data was studied and analysed with particular attention to compressor performance improvement as a function of suction pressure, discharge pressure, power and efficiency. Discharge pressure was considered as the preferred decision variable for improving the compressor performance due to its ease of control and its minimal impact on the process facilities upstream of the gas compressor. A step change of 5 bar at a time, up to 15bar (drop from 360 to 345 bar) was simulated with prediction that 1bar change in pressure will gain about 0.634kg/s or 2282.4kg/hr of flow rate (2.0mmscfd) for the two compressors. ACompSIM simulation software was utilized for this study. The results were validated with field data from recent turnaround maintenance of one of the gas compressors with percentage deviation of 83% of the predicted throughput. A graph of pressure ratio plotted against efficiency indicated an improvement in efficiency, though minimal, as pressure ratio decreased. The actual LPO saved was about 11MBOPD which amounts to about 230MBOPD for the 21 days duration of the turnaround. The cost saving was estimated at $21MM. The reduction of discharge pressure as a control variable, considering the allowable limits due to reservoir constraint, improves the performance of centrifugal compressors.

Keywords   Compressor, Performance, Improvement, Oil and gas

1. Introduction

Loss of production is defined as the difference between the potential production and the actual production (Berdahl, 2011). The potential production rate at a time can simply be described as whatever is less of (i) delivery rate from reservoir, (ii) passing rate through subsea equipment, (iii) processing rate on the platform, and (iv) the export rate/storage capacity. Each of these stages in the upstream process of oil and gas can be a bottleneck for the actual production on an offshore platform. Accordingly, reliability for a continuous production plant should be the fundamental driver (Berdahl, 2011). Under Nigerian legislation driven by the Department of Petroleum Resources (DPR), operators are required to inspect all pressure vessels within a 5 year period / cycle based on prescriptive philosophy. Due to bed space limitation on offshore facilities, multiple turnaround projects are planned and carried out by identifying outage opportunities. This will reduce the work scope of the major turnaround before the expiration of the 5 years cycle. In this case, there are three gas compressors of four compression

stages each, with natural gas stream as the working fluid. Each compressor has a capacity of 150mmscfd. These gas compressors were installed for gas re-injection to maintain reservoir pressure. Natural gas re-injection has become widespread over the last ten years (Arnold and Maurice, 1999). One of these three gas compressors is scheduled for a shutdown to carry out turnaround maintenance. Usually, in such operations where gas is separated as a result of oil production with the constraints of zero or minimum environmental impact due to gas flaring, oil production is reduced to compensate for the gas compression requirement lost to the shutdown of one compressor due to turnaround maintenance. See system schematic in Figure 1 below.

In a previous work, compressors have been classified into two categories: positive displacement and dynamic compressors (Hansen, 2008). Positive displacement compressors in essence work by entrapping a volume of gas and subsequently reducing this volume which in turn increases the pressure. Dynamic compressors generally work by imparting movement to the gas (Hansen, 2008; Bendinelli et al, 2001). That is, kinetic energy is transferred from the machines internals to the gas. By subsequent reduction of this velocity the kinetic energy is converted into potential energy (pressure). The two main types of dynamic compressors are axial and centrifugal compressors. As the name implies, axial compressors impart movement to the gas

in the axial direction. This is done by a series of rotors. Each rotor is followed by a stator where the kinetic energy, imparted to the gas by the rotor, is converted into pressure. Centrifugal compressors, on the other hand, work by imparting movement to the gas in radial direction by an impeller. This outward velocity is then converted in a diffuser. For centrifugal compressors, changes in pressure ratio have a significant effect on the actual flow through the machine. Thus, for a centrifugal compressor operating at a constant speed, the head or pressure ratio is reduced with increasing flow (Kurz et al, 2010). The centrifugal compressor causes the air to travel in radial direction from the impellers and pass through diffusers between stages prior to discharge. The air can be effectively cooled between stages by cooling the housing, resulting in near ideal compression stages. Except in the largest sizes, the overall efficiency of centrifugal compressors is less than that of positive compression machine. Centrifugal compressors provide a stable discharge pressure with wide variation in flow rate. Inter-stage cooling of centrifugal compressors is normally accomplished by water circulation through the casing. Units that discharge at less than 400 kPa (gauge) do not normally require cooling. Centrifugal compressors operate at high speeds with most commercial units running at approximately 20000 revolutions per minutes (rpm). In the aircraft and space industries speeds of 100000 rpm can occur. The advantages of centrifugal compressors become significant at flow rates in excess of 1200 L/s. The major advantages are large capacity, low vibration, compact construction, oil-free air discharge and self-limiting of the units. The major disadvantages include the necessity of a speed increaser (unless turbine driven), close running clearance, and high cost maintenance.

Studies have also shown that reducing system pressure drop to increase suction pressure or decrease discharge pressure allows more gas to be compressed through the compressor without modifications (Goldenet al, 2002). On compressor capacity and driver power, it was also reported that Compressor driver power requirements can also limit the compressor maximum flow rate. The compressor suction pressure in the propane refrigeration cycle has been identified as the main decision variable for optimization of the compressor throughput (Hassan et al, 2009). The objective of the model was to minimize the total power cost for the refrigerant compressors. The low performance of centrifugal compressors operating in the oil and gas production facilities, compared with the original design has been examined (Akhtar, 2006). However, the focus of this work is to reduce loss production opportunity during planned turnaround of one out of three gas compressors, by improving the performance of the other two gas compressors in parallel connection with the compressor to be shutdown. This will be achieved by determining the actual capacity of the three gas compressors, evaluating the performance of the remaining two gas compressors (when one is shut down) for the possibility of accommodating the extra volume of gas within the shut down period, and then determine the operating conditions with respect to maximum throughput by altering the key variables of suction pressure, temperature, discharge pressure, and molecular weight. A schematic diagram of the gas stream is shown in figure 1.
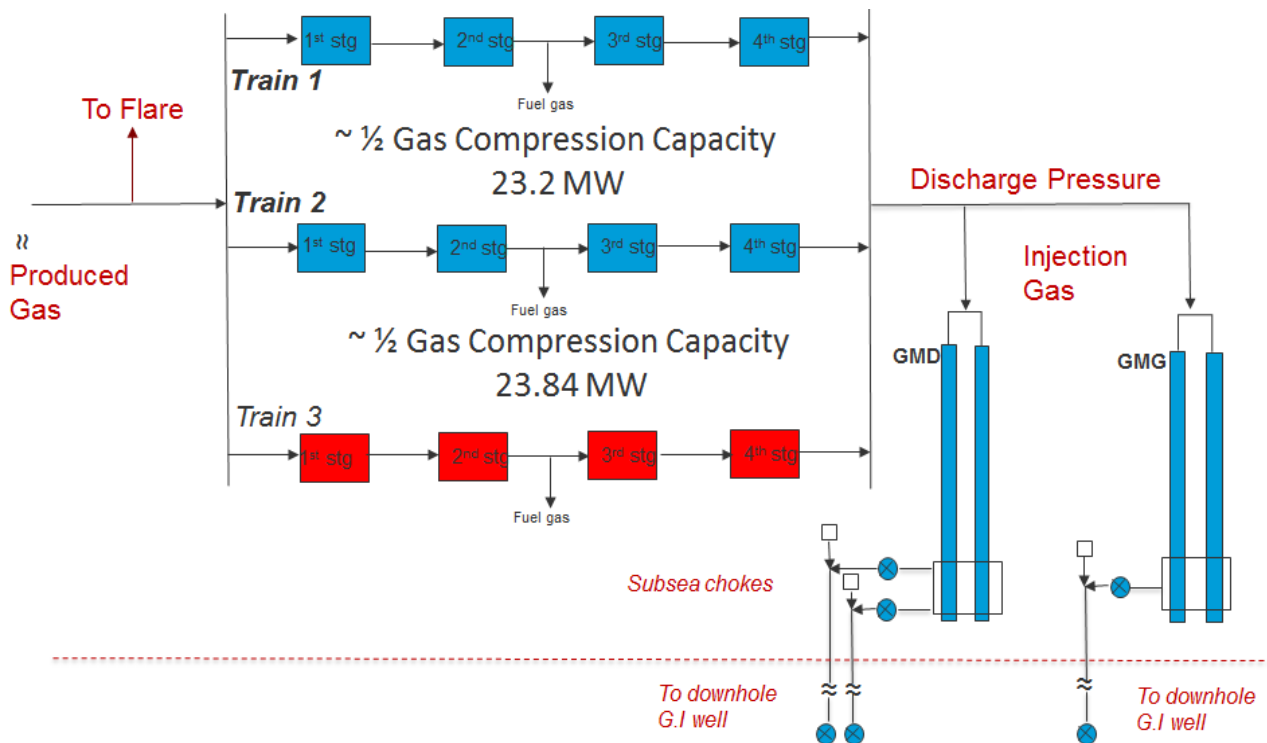

Figure 1. Simplified Schematic of Gas Stream

2. Materials and Methods

With all the fundamental concepts established, a working model that represents the gas compression system for gas re-injection (*CompSIM)* designed for simulating compressor performance and for analysis of data historian was used in running the cases generated in the course of this work. The methodology included the following:

1. Data gathering and identification of performance parameters: the required data profile gathered includes: power, speed, mass flow, gas turbine compressor (GTC) discharge pressure and temperature. The source of data historians were compressor operating maps from the proprietary software, machinery support centre and distribution control centre (DCS)

2. The obtained historic data during the previous shutdown was evaluated and analyzed to build cases for simulation, considering the objective function, variables and constraints.

3. The expected total amount of gas to save by improving compressor performance based on data historian was estimated.

4. Simulation cases were run on *CompSIM* to predict effect of discharge pressure reduction on gas throughput.

5. The estimated performance improvement result on compressor -01 and 02 while compressor-03 is down for turnaround maintenance was verified.

6. Calculation of the actual gain in compression throughput from field test carried out during the turnaround maintenance work was done and compared with the isentropic efficiency.

7. Performance improvement results and good conclusions were obtained.

Table 1. Cases at 23 MW of Available Power

Cases	Discharge Pressure (Bar)	Discharge Pressure reduction relative to Base case (Bar)
Base case	360	0
Case 1	355	5
Case 2	350	10
Case 3	345	15

In order to address the objective function of reducing "Loss production Opportunities", minimum discharge pressure, surge limit, choke limit were considered as constraints. Surge and choke limits were built into the CompSIM model while the base cases were built within the confine of the base case discharge pressure of 360 bar and the minimum discharge pressure of 345 bar required for gas injection (as determined by the reservoir pressure of which a discharge pressure below 345 bar would limit volume of gas injected and may cause a back pressure which can damage the compressors).

Out of all the performance parameters, only discharge pressure was considered as the appropriate decision variable for improving the compressor performance due to its ease of control and its minimal impact on the process facilities upstream of the gas compressor. From the head formula in Equation 1, head is a function of discharge pressure and as a result, lowering the discharge pressure will lower the head. The following Table 1 lists the cases.

Hand or spreadsheet calculations can be performed if gas properties are known, and will give operators a good idea of how well a machine is performing. Compressor head is presented as in equation 1 (Campbell, 1998).

The following formulas (in SI Units) were adapted in the calculation spreadsheet to determine and compare the isentropic efficiencies of the cases as follows:

Polytropic Head for a Centrifugal Compressor:

$$H_p = \frac{R}{MW}\, Z_{AVG} T_1 \left(\frac{n}{n-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right]$$

Eqn. (1) (Bendinelli et al, 2001)

where
 H_p = Polytropic Head
 T_1 = Suction Temperature
 Z_{avg} = Average Compressibility Factor $(z_1 + z_2)/2$
 R = Universal Gas Constant
 n = Polytropic Exponent
 MW = Gas Molecular Weight
 P_1 = Suction Pressure
 P_2 = Discharge pressure

Absorbed Power:

$$Pw = \left(\frac{(m)(H_p)}{\eta_p}\right)$$

Eqn. (2) (Campbell, 1998)

where:
 Pw = Gas Power
 $\dot{m}$ = Mass Flow Rate
 H_p = Polytropic Head
 η_p = Polytropic Efficiency

Polytropic exponent, n:

$$\left(\frac{n}{n-1}\right) = \left(\frac{k}{k-1}\right) \text{ x } \eta_p$$

Eqn. (3) (KLM, 2011)

where:
 n = Polytropic Exponent
 k = Isentropic Exponent
 η_p = Polytropic Efficiency
 Isentropic efficiency, η_s:

$$\eta_s = \frac{\left[\left(\dfrac{P_2}{P_1}\right)^{\left(\frac{\kappa-1}{\kappa}\right)} - 1\right]}{\left[\left(\dfrac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right]} \qquad \text{Eqn. (4) (KLM, 2011)}$$

A review of the compressor map and historic data establishes an approximate target gain 15.5% of the compressors throughput as an opportunity for performance improvement when two compressors are in operation. Discharge pressure was considered as the appropriate decision variable for improving the compressor performance due to its ease of control and its minimal impact on the process facilities upstream of the gas compressor. Therefore a step change of 5 bar at a time, up to 15bar (drop from 360 to 345 bar) of the suction pressure was simulated with prediction that 1bar change in pressure will gain about 0.633kg/s or 2278.8kg/hr of flow rate (2.0mmscfd) for the two compressors. *Comp*SIM simulation software was used to model the compressors based on current set points and operation (see table 2). In order to validate the simulation result, a test was carried out during the shutdown of one out of three gas compressors for turnaround maintenance.

3. Results and Discussions

Table 3 shows the impact of GTC discharge pressure drop on mass flow when two compressors were operating. The graphical presentation is shown in figure 2.

Table 2. Calculation for head, polytropic exponent, power and efficiency

Simulated Base Case

	Input
	calculated

Standard Conditions	
Temperature, T (oC)	15
Pressure, P (Pa)	101325

Extract from DCS

Parameters	Stage 1	Stage 2	Stage 3	Stage 4	Comments
Suction Pressure, P_1 (Bar)	21.01	58.02	113.15	214.34	
Suction Temperature, T_1 (°C)	53.43	50.78	51.41	48.85	
Discharge Pressure, P_2 (Bar)	60.92	116.05	217.24	360.09	
Mass Flowrate, $\dot{m}$ (mmscfd)	150.95	141.79	132.25	137.28	
Compressibility, Z_{avg}	0.939	0.849	0.804	1.003	
Isentropic Exponent, k_{avg}	1.26	1.408	1.559	1.558	
Molecular Weight, MW (kg/kg-mol)	23.45	23.45	23.45	23.45	
Polytropic Efficiency, η_p	0.86	0.77	0.76	0.68	

Calculations (in SI unit)

| Parameters	Stage 1	Stage 2	Stage 3	Stage 4	Comments
Suction Pressure, P_1 (kPa)	2100.80	5802.32	11314.64	21434.11	Converted from Bar to kPa
Suction Temperature, T_1 (°K)	326.43	323.78	324.41	321.85	Converted from °C to °K
Discharge Pressure, P_2 (kPa)	6092.32	11604.64	21724.11	36009.30	Converted from Bar to kPa
Discharge Temperature, T_2 (°K)	422.00	420.15	442.11	423.15	Converted from °C to °K
Mass Flowrate, $\dot{m}$ (kg/s)	48.97	46.00	42.90	44.54	Converted from mmscfd to kg/s
Mass Flowrate, $\dot{m}$ (kg/hr)	176300.99	165602.13	154457.49	160333.75	Converted from kg/s to kg/hr
Compressibility Z_{avg}	0.924	0.803	0.701	0.714	Z = Average compressibility factor $(z_1 + z_2)/2$
Gas Constant - R (kJ/kmol.K)	8.314	8.314	8.314	8.314	Constant
Discharge Temperature, T_2 (°K)	149.00	147.15	169.11	150.15	$T2 = T1 (P_2/P_1)^{(n-1)/n} - 1]$
Compression Ratio (P_2/P_1)	2.90	2.00	1.92	1.68	P_2/P_1
T_2/T_1	1.29	1.30	1.36	1.31	T_2/T_1
Polytropic Exponent, n	1.318	1.602	1.903	2.116	$n/(n-1)=(k/(k-1)) \eta_p$
Polytropic Head, H_{poly} (kJ/kg)	129.77	73.01	61.63	48.64	$H_{poly} = 8.314/MW \times T_1 \times Z_{avg} \times (n/(n-1)) \times [(P_2/P_1)^{(n-1)/n} - 1]$
$(k-1)/k$	0.21	0.29	0.36	0.36	$(k-1)/k$
$(n-1)/n$	0.24	0.38	0.47	0.53	$(n-1)/n$
Power, Pw (kW)	7427.68	4356.49	3499.34	3190.25	$Pw = \dot{m} \times H_{poly}/\eta_p$
Total Gas Power per Train (kW)	18473.75				For each Train
Total Mass Flow Rate, $\dot{m}$ (kg/hr)	352601.97				For Two Trains
Isentropic efficiency, η_s	0.84	0.75	0.73	0.65	$\eta_s = [(P_2/P_1)^{(k-1)/k} -1]/[(P_2/P_1)^{(n-1)/n} -1]$
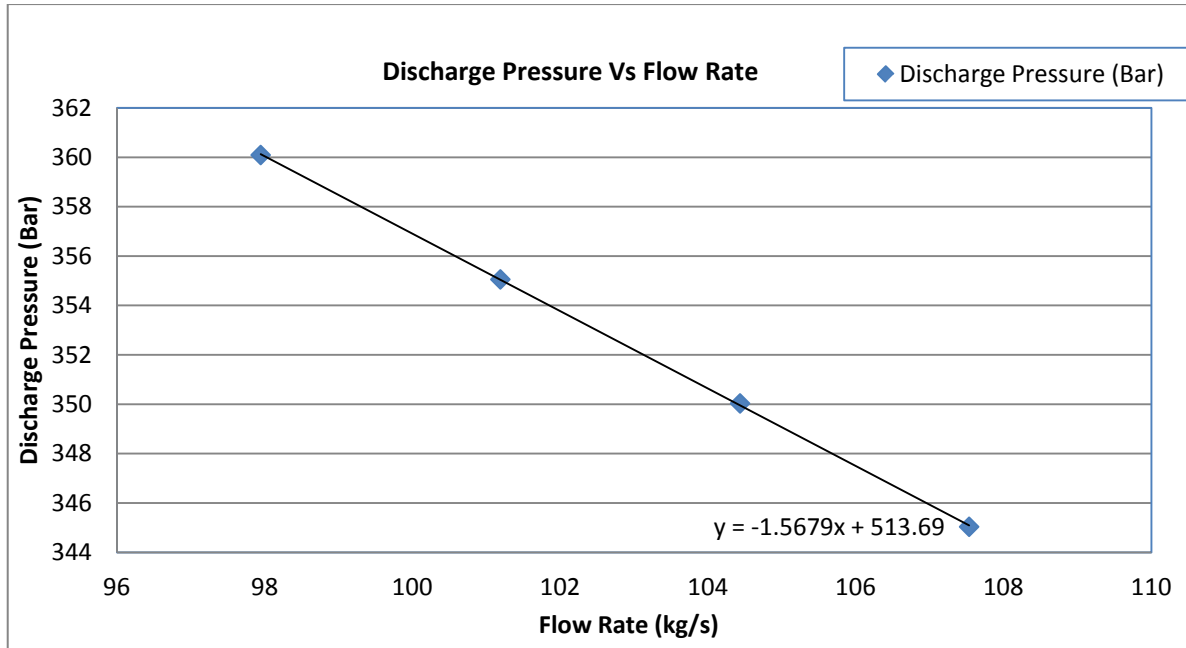
Figure 2.   Graph showing the effect of simulated discharge pressure reduction on flow rate

Table 3.   Effect of reducing discharge

Cases	*Injection Gas throughput (Kg/s) Due to Discharge pressure reduction*	**Discharge Pressure (Bar)**
Base case	98	360
Case 1	101	355
Case 2	104	350
Case 3	108	345

The governing equation for the graph is:

$$y = -1.5679x + 513.69 \qquad \text{Eqn. (5)}$$

$$\frac{\partial \gamma}{\partial x} = -1.5679$$

Based on Figure 2 and the resultant straight line equation above, it implies that for every one step change in y (pressure), there will be a 0.634 (1 divided by 1.5679) step change in x (mass flow). That means 1bar change in pressure will gain about 0.634kg/s or 2282.4kg/hr of flow rate (2.0mmscfd) for the two compressors at suction conditions. If then we can drop 10bar, it means 6.34kg/s or 22824kg/hr (20mmscfd). The implication is that there is an opportunity to reduce LPO by 6.34 – 9.51kg/s (10 – 30mmscf/d) equivalent of 1.723 – 5.169 MBOPD at a pressure drop of 5 -15bar with two compressors.

Note: 1 MMSCF of natural gas = 172.3 barrels of crude oil equivalent (Hart Group, 2006).

Results of Simulation

Table 4 is the result of the CompSIM steady state simulation that was carried out with a gain of about 10 kg/s at a discharge pressure of 345 bar.

Actual Performance of the two Gas Compressors

In order to validate the simulation result, a test was carried out during the shutdown of gas compressor-03 for turnaround maintenance while compressor-01 and 02 were still in operation. The results are as reported in Table 5. Actual Performance result of effect of discharge pressure on flow rate for the two compressors in operation compared with base case and Predicted case indicates an improvement in compressor throughput by 83% of the prediction. It was also observed that absorbed power increased by 2 MW from 18 MW of the base case to 20 MW of actual case.

As seen in Table 5, the final case of total drop of 15bar of discharge pressure from 360 bar to 345 bar yielded a throughput of 114 kg/s with an additional compression of 12 kg/s. The relationship of the discharge pressure and the flow rate is as presented in the Figure 3 below.

The base case, predicted case and actual case were put side by side on Table 6 to examine deviation of actual case from the base and predicted cases.

Effect of Compressor performance improvement effort on the Production Facility

The effect of lowering the discharge pressure is such that less oil production is curtailed to carry out turnaround maintenance since more produced gas can be compressed at an additional rate of ~12 kg/s indicating an obvious increase in gas flow rate.

Lowering the discharge pressure enables the Power Turbine speed to increase thus increasing the mass flow rate even at limiting environmental conditions

Increase in Power Turbine Generator speed also improved the power utilization of the compressors.

Table 4.   Simulation result from *CompSIM*

Cases	Total Absorbed Power from 1st to 4th stage (MW)	Discharge Pressure (Bar)	Mass Flow Rate Saved (kg/s)	Barrels of Oil Equivalent Saved /day (MBOPD)	Mass flow rate (kg/s)	Mass flow rate (MMSCFD)
Base Case	18	360	0	0	98	302
Case 1	19	355	3	~ 1.7	101	312
Case 2	20	350	6	~ 3.4	104	322
Case 3	20	345	10	~ 5	108	331

Note: 1 MMSCF of natural gas = 172.3 barrels of crude oil equivalent (Hart Group, 2006)

Table 5.   Actual Performance result of effect of discharge pressure on flow rate for the two compressors in operation

Cases	Total Absorbed Power from 1st to 4th stage (MW)	Discharge Pressure (Bar)	Mass Flow Rate Saved (kg/s)	Barrels of Oil Equivalent Saved /day (MBOPD)	Mass flow rate (kg/s)	Mass flow rate (MMSCFD)
Actual Base Case	19	360	0	0	102	302
Actual Case1	19	355	3	~ 1.6	105	312
Actual Case 2	19	350	7	~ 3.8	109	324
Actual Case 3	20	345	12	~ 6.1	114	337

Note: 1 MMSCF of natural gas = 172.3 barrels of crude oil equivalent (Hart Group, 2006)
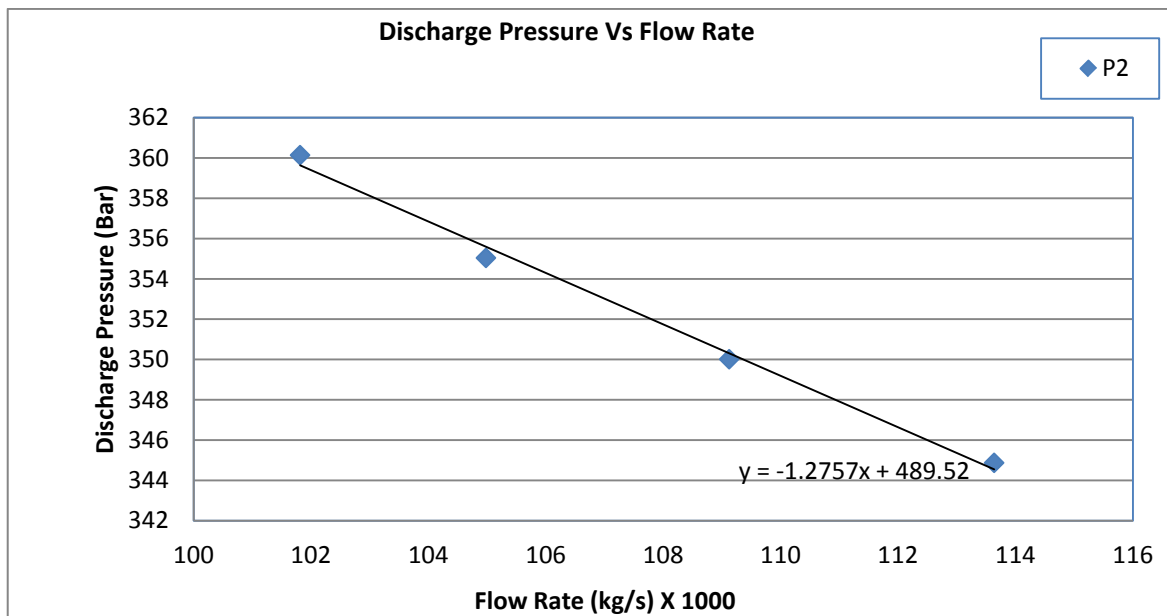
Figure 3.   Graph showing the effect of actual discharge pressure on flow rate

Table 6.   Result comparison of flow rates of the actual case, base case and predicted case at discharge pressure of 345 bars with two compressors in operation

| Cases	Absorbed Power (MW)	Mass Flow Rate Saved (kg/s)	Barrels of Oil Equivalent Saved /day (MBOPD)	Mass flow rate (kg/s)	Mass flow rate (MMSCFD)
Base Case	18	0	0	102	302
Predicted Case	20	10	~ 5	108	331
Actual Case	20	12	~ 6.1	114	337

Note: 1 MMSCF of natural gas = 172.3 barrels of crude oil equivalent (Hart Group, 2006)

Gas compression performance comparison

Comparison of flow rates with 3 compressors and 2 compressors at 360 bar and comparison of the Normal operation, Base Case, Predicted Case and Actual Caseis as tabulated below:

Table 7. Performance comparison

Cases	No. of Compressors up	Discharge Pressure (Bar)	Mass flow rate (MMSCFD)
Normal operation	3	360	398
Base Case	2	360	302
Predicted Case (reduced discharge pressure)	2	345	331
Actual Improved Case (reduced discharge pressure)	2	345	337
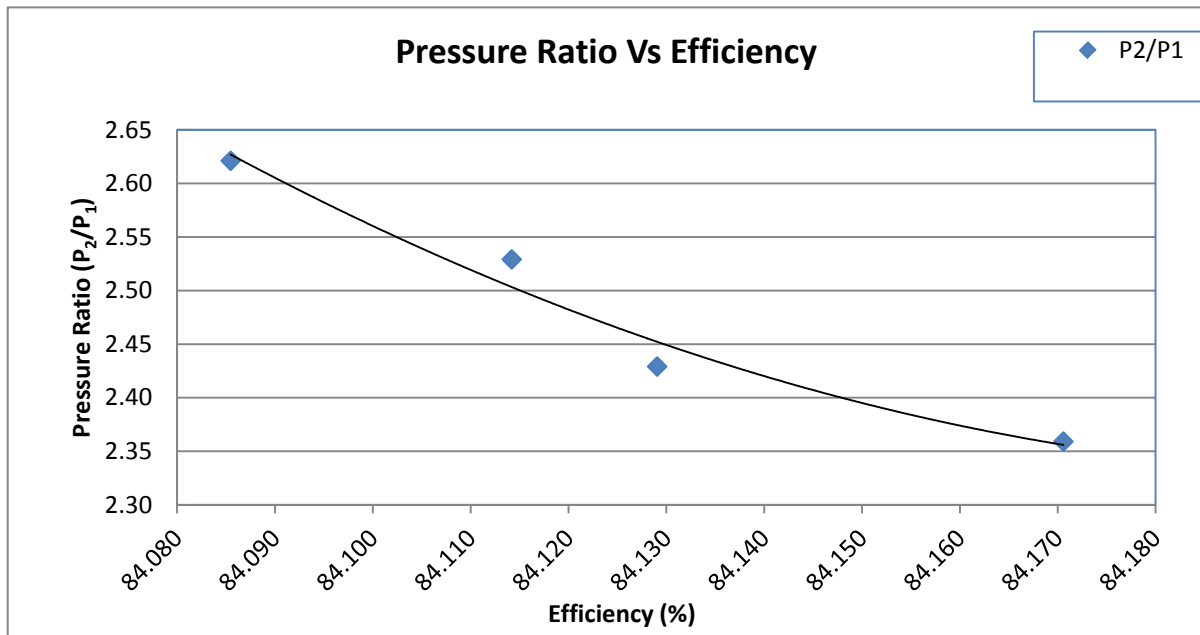
Figure 4.   Pressure Ratio Versus Isentropic Efficiency

The reason why this compressor performance improvement effort cannot be applied to all three compressors in operation at this time is because the field produces oil and gas at a certain Gas/Oil Ration (GOR) and the three compressors are designed to adequately compress the produced gas. Since the 2 out of the 3 compressors cannot compress 100% of the produced gas, the 3 will always be put to use for load sharing with approximately 88% capacity utilization of each compressor. Therefore, the need to carry out this improved control with lower discharge pressure at normal operation is remote, unless during failure or Turnaround as in the case study.

Isentropic Efficiency:

In order to calculate the compressor efficiency of the cases, polytropic exponent was calculated based on the design polytropic efficiency using equation 3. Isentropic efficiency was calculated using equation 4. Figure 4 below shows a graph of pressure ratio against efficiency indicating improvement in isentropic efficiency as pressure ratio reduces.

Deviation

Comparing results of actual flow rate with the predicted case yielded a percentage deviation of 81%.

Table 8.   Deviation between estimated versus actual flow rate gained

	Improved flow rate due to reduction in discharge pressure by 15bar (kg/s)	Barrel of oil equivalent (MBOE)
Estimated	10	5
Actual (MMSCFD)	12	6
Variance	2	1

Plausible reasons for deviations

Model was based on designed gas density of 23.45 kg/kg-mol while the actual gas density was found to be 24.35 kg/kg-mol.

Cost Benefit

Since the aim of this paper is to minimize LPO which is the highest cost element from the list above, reference was made to the definition of Loss of Production by Berdahl (2011) as the difference between the potential production and the actual production. The cost benefits analysis is as shown below (table8). The givens and actual data as gathered

from the field were put into a spreadsheet to determine the cost benefits.

Table 9. Cost Benefit Analysis

	Function	Value	Unit
	GOR	1.483	
	TD	21	Days
Values at normal production with 3 compressors	POPR	236	MBOPD
	PGPR	350	MMSCFD
	BOEP	60	MBOE
	PPR	296	MBOPD
	PP	6216	MBO
Values before performance improvement, while 1 of 3 compressors is shutdown	AOPR	204	MBOPD
	AGPR	302	MMSCFD
	BOEA	52	MBOE
	APR	256	MBOPD
	AP	5377	MBO
Values after performance improvement, while 1 of 3 compressors is shutdown	AOPR′	227	MBOPD
	AGPR′	337	MMSCFD
	BOEA′	58	MBOE
	APR′	285	MBOPD
	AP′	5986	MBO
Loss Production Analysis	LP	839	MBO
	LP′	230	MBO
Cost Value	Average price of oil per barrel	93	$
	Cost Savings	21,355,860	$

Note: 1 MMSCF of natural gas = 172.3 barrels of crude oil equivalent

4. Conclusions

Increase in the centrifugal compressor mass throughput by approximately 0.634 kg/s (2 mmscf/d) per 1 bar drop in discharge pressure for the two gas compressors was observed, resulting in an actual LPO savings of about 11MBOPD. A graph of pressure ratio plotted against isentropic efficiency indicated an improvement in efficiency, as pressure ratio decreased. Available Power was 23MW while maximum consumed power at improved volumetric performance was 20MW. Decrease in polytropic head was observed at fixed suction pressure and decreasing discharge pressure. About 11MBOPD of loss production opportunity (LPO) was prevented. This translates to about 230MBO savings for the turnaround period of twenty one (21) days duration. At the rate of $93 per barrel of oil, about $21MM dollars was saved from. This improvement is in support of increase in Return On Investment (ROI) as LPO is reduced (Abraha, 2011).

The study shows that there exist the potential of improving the performance of centrifugal compressors by reducing the discharge pressure with the objective of minimizing loss production opportunities in gas injection applications. This engineering principle of improving compressor throughput can also be engaged in planning for potential compression demand, as well as Prediction of performance improvement by correlating results of simulated discharge pressure reduction cases and the resultant flow rates in a graph, with the help of a straight line equation.

Nomenclature

AGPR - Actual gas production rate before performance improvement

AGPR′ - Actual gas production rate after performance improvement

AOPR – Actual oil production rate before performance improvement

AOPR′ - Actual oil production rate after performance improvement

AP′ - Actual production after performance improvement

AP - Actual production before performance improvement

APR′ - Actual production rate after performance improvement

APR - Actual production rate before performance improvement

BOEA – Barrels of oil equivalent of actual gas saved before performance improvement

BOEA′ - Barrels of oil equivalent of actual gas saved after performance improvement

BOEP – Barrels of oil equivalent of potential gas saved

DCS - Distribution control centre

DPR – Department of petroleum resources

GOR – Gas oil ratio

GTC – Gas turbine compressor

k - Isentropic exponent

LP – Loss production

LP′ - Loss production after performance improvement

LPO – Loss production opportunity

MBOE – Million barrels of oil equivalent

MBOPD – Million barrels of oil per day

MBO - Million barrels of oil

MMSCF – Million standard cubic feet

MMSCFD – Million standard cubic feet per day

n - Polytropic exponent

p – Pressure

PGPR – Potential gas production rate

POPR – Potential oil production rate

PP - Potential production in the turnaround window

PPR – Potential production

TD - Turnaround duration

η_s– Compressor isentropic efficiency

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