Determining the real performance of centrifugal compressors operating in oil & gas production facilities

Dr M Sib Akhtar
MSE (Consultants) Limited
Carshalton, Surrey UK SM 5 2HW
Sib.Akhtar@mse.co.uk

ABSTRACT

This paper examines reasons for the low performance of centrifugal compressors operating in the oil and gas production facilities, compared with the original design. It compares the differences in factory test and field test results. The flow measurements in the field in particular are influenced by higher internal leakages within the machines, which distort the performance values. The paper argues the case for adopting volumetric efficiency concept from reciprocating compressors to rectify this problem. Case studies have been presented to explain the effect of performance loss on production and how it can affect machine reliability.

1. INTRODUCTION

With rising oil prices, operating companies are under pressure to ensure that their production systems operate at their maximum capacity. The performance and availability of centrifugal compressors is vital to the operation of oil and gas production facilities. Over the past 20 years, the availability of these machines has been the focus of attention and performance issues have taken a back stage. Even API 617 (1) does not require the vendor to guarantee the compressor efficiency, only the absorbed power with a +/- 4% tolerance.

While fuel prices were low, the performance of these machines in oil and gas sector at least, has never been a central issue. Operators were content to see that the machines kept running and did not fail. The general perception remained that the loss of compressor performance can be compensated for by an increase in engine power or higher fuel consumption. Since fuel was available in abundance and was cheap, the economic case for performance improvement of operating machines was weak until now.

With the rise in fuel prices, the fuel consumption could no longer be ignored and became an issue. The operators also discovered that the production capacity was no longer constant and that there was a link between compressor efficiency and the production capacity.

2. CONSEQUENCES OF COMPRESSOR PERFORMANCE LOSS

During the factory test, the compression power of a single stage compressor is determined as a product of the mass flow and polytropic head divided by the polytropic efficiency, where the compressor absorbed power has a 4 % tolerance.

Compressor Power = (Polytropic Head x Mass Flow) / Polytropic Efficiency …………(1)

Once installed in the field, operators discover that the power of the driver is fixed with an upper limit and therefore the actual mass flow delivered is largely a function of compressor efficiency and pressure ratio. The relationship expressed in equation 1, which was useful for the manufacturer, had to be rearranged for field testing as follows:

Mass Flow = (Engine Power x Polytropic Efficiency) / Polytrophic Head… …………….(2)

Since polytropic head for a given compression process is relatively constant, and the engine power has an upper limit, the mass flow delivered therefore becomes a function of the compressor efficiency. If the efficiency falls the production rate or the mass flow decreases; so does the operator’s revenue, as the following example will illustrate.

2.1 The effect of performance loss on a well-head & gas export compressor

Case study 1

A well-head & gas export compressor, driven by a gas turbine, with a site rating of 23.7 MW was delivering a lower volume of gas than anticipated and was approaching the surge line. The flow could not be increased (by increasing engine power) because the engine was operating at its power and exhaust gas temperature limit.

The compressor was designed to deliver 515 MMSCFD (14.58 MSm3/day) when pumping gas from the wells at 30 bar suction and 85 bar discharge. With a factory test efficiency of 80 %, the head required is 137.5 kJ / kg. The operator found that the actual flow delivered was significantly lower than the anticipated flow and was much closer to the surge line. Fouling exacerbated the problem, which meant that the compressor had to operate at a higher speed than required by the factory performance curves. The efficiency loss resulted in operation at a reduced capacity at a constant engine power. Table 1 shows the effect of reducing compressor efficiency from the original design. A reduction of efficiency from 80 % to 60% decreases the flow from 515 MMSCFD to 386 MMSCFD and approaches the surge line.

At reduced efficiencies, the head required will increase and power will be higher but this is a second order effect and has not been considered for clarity. Thus at 70 % efficiency, the flow delivered reduces from 515 to 450 MMSCFD or a drop of 12.7%. Most operators will not notice the effect of fouling and efficiency loss. However in this example, the combined effect of fouling and efficiency loss caused the operating point to approach the surge line sooner than expected; once it reaches the surge control line it is not possible to lower the capacity any further as the machine will go in surge.

This point is illustrated in Figure 1, which shows the operating point on a compressor map for a clean and a fouled machine. The clean machine could run for longer periods. The fouled machine, which suffered from 15% - 20 % fouling and loss in efficiency, could not operate because of proximity to surge and had to be re-wheeled.

In this respect fouling and efficiency loss need to be explained.

In this paper, fouling is understood to decrease the head generated by the compressor at a given speed and flow compared with the design. Thus a fouled machine needs to operate at a greater speed than required by design to generate the required head. For example, a compressor operating at 100 % speed and at 100 % volume flow develops 90 % polytropic head, instead of the design 100 % head. The machine will therefore need to run at 105 % speed to generate the full 100 % head. Such a machine is considered to have 10 % fouling.

Similarly efficiency loss means that the compressor will require greater power to compress a given volume of gas compared with its design. Thus a compressor operating at 100 % speed and 100 % capacity suffers from a 10 % loss in efficiency and will require approximately 10 % more power to achieve the design compression ratio.

Both fouling and efficiency loss assume that thermodynamic parameters i.e. gas compositions, temperature and compressibility at inlet are unchanged.

Fouling distorts the compressor performance curve shape and surge line. It is possible to generate the fouled curve if a proper performance test is performed.

The original compressor design (Figure 1) was intended to operate for 12 years without requiring a re-wheel and was therefore a cost effective option. However, the original selection did not anticipate the impact of fouling or performance loss and therefore had to be re-wheeled earlier.

2.2 Effect of performance loss on gas lift compressor

Gas Lift compressors normally consist of two or three stages of compression and are driven by a common driver, usually a gas turbine or electric motor if the compression powers are small. In these situations the effect of performance loss is very similar and all the operating point will be driven towards the surge line. The stage with the smallest turn down to surge will experience the surging first, as the next example will illustrate.

Case study 2

A platform with three gas lift compressor trains was experiencing an availability problem. Only two out of three machines could operate with the third machine in a constant state of repair and refurbishment. An examination of the failure history revealed that the LP compressor had an unfortunate history of thrust bearing failures; 6 thrust bearings were damaged in as many years.

All compressor trains were performance tested; LP compressor efficiency was good but the operating point was close to the surge line. The HP casing was substantially down in performance but the operating point was well removed from surge. The gas turbine, a 14 MW engine, operated close to the exhaust gas temperature limit. During the tests the LP compressor experienced severe surging. The event was recorded. Detailed analysis showed that the surging occurred on the train where the engine was suffering most from loss of performance. This was attributed to incorrect Inlet Guide Vane (IGV) schedule on the turbine. Loss of engine power exacerbated the surge problem. The engine did not have sufficient power to deliver enough power to the LP compressor stage to keep it away from surge. Surging is very damaging to the compressor internals; it causes flow and pressure reversals through the machine. These events often wreck the balance piston and thrust bearings.

3. FIELD TESTING OF COMPRESSORS

Field-testing of centrifugal compressors is not something to be taken lightly. Tests that use the field installed instrumentation can lead to highly unsatisfactory results. This is due to a combination of factors including poor maintenance, lack of proper calibration and the design range of the instruments.

Flow measurements are a recurrent and a difficult problem to resolve. The flow devices are normally optimised for measuring surge flows. Under normal operations compressor flows are some 30 – 40 % higher than surge flows. Since orifice differentials are proportional to the square of the volumetric flow, the output of the flow controllers becomes saturated and the signal output goes out of the scale, hence accurate flows cannot be measured. Temperatures measurements and especially discharge temperatures are also subject to errors and this can produce unrealistic performance results.

For this reason, vendors are vehemently opposed to field tests and insist that testing of compressors should only be carried out in the factory under controlled environment. They have a valid point and while their argument is genuine, it is not practicable to remove the machine from the site and send it to factory for a test. Therefore practical measures are needed to find alternatives.

MSE has developed procedures for conducting tests on compressors operating in the field. More than 100 machines have been tested over a long period of time. Many problems were encountered in conducting field tests. However, with careful planning and with the support and cooperation of the operating company staff, it is possible to overcome these problems.

In general **field test are performed in accordance with ASME PTC 10 Class I** (2) test and should follow the same procedures. The gas handled is normally the design gas; gas pressures and gas densities and the operating conditions are as per original design. Some times it is necessary to normalise for off-design operation.

Method of calculations follows the ASME PTC 10 Class 1 procedures. In the great majority of cases, machines are operating with the design gas composition and therefore it is not necessary to correct for differences in molecular weight and temperatures.

For light gases the Benedict, Webb Ruben and Starling (BWRS) equation of state is used. It is reliable and is commonly used by the more experienced compressor vendors. For higher molecular weight gas applications, it is desirable to check with SRK method, which is favoured by engineering contractors. For very high-pressure and sour gas it is necessary to use other equations of state such as Lee Kessler or their derivatives.

Gas composition samples must be obtained during tests and repeated where necessary.

Pressure and temperature measurements should be verified by installing additional test instrumentation, which should be calibrated before and after the tests. Oil and gas production facilities, which export gas, are normally equipped with fiscal metering systems, which are properly calibrated and well maintained. It is recommended that flow-measuring devices should be verified against the fiscal meters wherever this is possible. Figure 2 shows details of proposed flow and power measurements for field tests.

3.1 Resolving differences between field and factory tests

Factory tests are conducted on simulated gases at low operating pressures in accordance with ASME PTC 10 Type II procedure. In general terms, factory tests verify the aerodynamic performance of a compressor stage. The tests reproduce the velocity triangles the Mach Number and equivalent tip speeds in test gases, which have a higher molecular weight than the field gas. However, because factory tests are conducted at very low suction pressures and lower speeds, the inter-stage leakages as well as balance piston leakages and centre seal leakages are assumed to be negligible.

The field-tests however are conducted at the normal design operating pressures and flows, where the machine experiences significantly higher internal leakages; the leakages being a function of gas density, clearances and pressure ratios. The labyrinth clearances are very likely to have increased due to normal wear and tear. This increase in the internal flows is not visible and cannot be measured but it affects the performance of the machines adversely.

Experience has shown that higher internal leakages introduce large flow distortions, up to 15-20%, which have a detrimental effect on the performance of the machine. The increased internal flow circulation requires up to 25% extra power but also shifts the operating point to the right of the design and into deep into the stonewall area where the compressor efficiencies are very low.

The leakages can adversely affect the performance of multi-stage compressors, where it declines steeply due to "snowball" effect associated with multiple wheels.

3.2 Effect of higher internal leakages on the performance of a four-stage gas lift compressor

Case Study 3

A four-stage compressor suffered from reduced flow capacity. The 3^{rd} and the 4^{th} stages were housed in a single back-to-back casing separated by a centre seal. The flow in the 1^{st} and 2^{nd} stages was found to be in agreement with mass balances when account was taken of water and condensate knockout. The mass balances of 3^{rd} stage and 4^{th} stages however did not agree. The 3^{rd} Stage flow was 971 kSm3/day (34.29 MMSCFD) and the fourth stage flow was 1121 kSm3/d (39.58 MMSCFD). The flow meters were calibrated and verified against design specifications. There was no side stream and no additional separator flow introduced. The discrepancy was found to be purely due to increased leakage through the balance piston and centre seal as shown in Figure 3.

These leakages distort the real flows passing through the compressor and can lead to very erroneous results. It became necessary to estimate the internal leakages and compensate for these by building a compressor model with allowances made for stage flows as well as centre stage and balance piston leakages. By making allowances for these leakages, the apparent flow discrepancy could be resolved.

As shown in Figure 3, the flow measured at the third stage inlet is 971 kSm3/day and consists of process gas and does not include the balance piston flow; which is shown as a dotted line. However the flow meter installed at fourth stage inlet includes both the centre seal leakage and the balance piston flow in addition to the process flow through the compressor. Thus the fourth stage (which measured 1121 kSm3/day) appears to pass more flow than was measured at third stage inlet. Please refer to Table 2.

If we include all of this extra 15.4% flow to enter the fourth stage, the polytropic head calculated from the equation of state is considerably greater than head derived from compressor characteristics and operating speed line. Please see Figure 4a and 4b. A closer examination of the flow paths will show that not all of this 15.4 % extra flow enters the 4^{th} stage; a proportion of this leakage is recycled through the balance piston shown as dashed lines, and bypasses the fourth stage. Once a correction is made for the balance piston flow, the operating point falls on the compressor performance map.

Calculations were carried out for potential leakages across the centre seal and balance piston from the design data available. The clearances were increased to bring the level of leakages in line with the flow measurements. This resulted in mass balance and power balance to be achieved for the test conditions.

4. APPLYING VOLUMETRIC EFFICIENCY CONCEPT TO CENTRIFUGAL COMPRESSORS

The above case study demonstrates the need for redefining the performance test procedures for centrifugal compressors. Factory tests do not take into account the effect of increased leakages especially the balance piston and centre seal effect that will occur in the field. These leakages effectively reduce the net flow capacity of a compressor. The design flow capacity of the compressor remains unchanged but more and more of this design capacity is used to compress the leakage flows.

The volume flow and head characteristics of a compressor impeller is fixed by its design (i.e. its flow coefficient) and cannot vary without changing the speed or diameter, therefore if impeller has to accommodate leakage flows due to an increase in the balance piston clearances, this will result in a reduction in the net gas volume delivered by the impeller.

In this respect the centrifugal compressors are analogous to the piston compressors, where volumetric efficiency is calculated to differentiate between the total swept volume and net volume flow.

4.1 Piston compressors

In piston compressors, the net volume delivered is a product of swept volume and the volumetric efficiency. Where the volumetric efficiency is defined as follows (3)

Volumetric Flow Delivered = Swept volume x Volumetric Efficiency…………………..(3)

VE = 96 – R – C (Zs / Z d (R ^ 1/k) – 1) ……………………………………………….(4)

Where

VE = Volumetric efficiency for piston compressor

R = Ratio of compression

Zs = Gas compressibility at suction

Zd = Gas Compressibility at discharge

C = Clearance as a percent of cylinder volume

In reciprocating compressor industry, the concept of clearances and volumetric efficiency is well understood and is extensively used in capacity control.

4.2 Centrifugal compressors

In centrifugal compressor application, the concept of volumetric efficiency has not been considered. The vendors do differentiate between the internal flow and flange-to-flange flow, but this information is treated as "proprietary" and is not shared with the end user.

To apply this concept would require re-writing the test procedures. However, the changes proposed are simple and can benefit the end user and making more accurate estimates of the compressor capacities in the field. The Net volume flow can be defined as follows:

Net Volume Flow = Compressor Design Flow X Volumetric Efficiency ……………..…(5)

Compressor Design flow is a function of the impeller design flow coefficient, its speed and diameter and is inclusive of all leakages.

Net Volumetric Flow (or flange-to-flange flow) is Compressor design flow minus the internal leakages i.e. balance piston and centre seal leakages whichever is applicable.

Volumetric efficiency can then be defined as follows and is explained graphically in Figure 5.

VE = Compressor Net Volume Flow / Compressor Design Volume Flow ………………(6)

Compressor Design Volume Flow = Flow coefficient x speed x diameter ^3 ……………(7)

The flow coefficients are available from the manufacturers or estimated from computer programs (4). The volumetric efficiency will be close to 100 % in factory test and lower in the field depending upon application, operating pressures and sizes of the machine and pressure ratios etc. This concept can help to explain some of flow anomalies found during field tests and reconcile the differences in head, efficiency and power found in field tests.

Before leaving this point, it is important to remember that there is a fundamental difference between volumetric efficiency of piston compressor and centrifugal compressor.

In the piston compressors, the volume of gas trapped in the clearance spaces is compressed and expanded. Thus during compression it will absorb power and during expansion it will deliver net power and therefore if the frictional effects are ignored the volumetric efficiency in piston compressors does not increase power consumption pro rata. Although frictional effects may be present which can distort compressor efficiency.

In centrifugal compressors however, the situation is very different. The high-pressure gases, which leak past the balance piston and centre seals are throttled back to the inlet pressure and unlike piston compressors, do not contribute to the net power. Thus all the recycled gas has to be recompressed from suction pressure to discharge. In addition, the recycled gas arrives at a higher temperature than the process gas and therefore it leads to an increase in inlet temperature and required compressor head.

Therefore when estimating compressor head, efficiency and the driver power, it is necessary to account of the internal leakages or volumetric efficiency of the compressor.

It is suggested that the vendors should provide performance curves based on the full compressor design capacity, without deducting any balance piston leakage. During factory test, where the balance piston leakages are small, no corrections are needed and this should verify the base line for the maximum design capacity of the compressor. When correcting from factory test to field data, allowance should be made for the increased leakages, which would occur due to higher operating pressures and gas densities as well as due to wear and tear of the labyrinths. In fact API 617 requires thrust bearings design calculations to be based on double the balance piston clearances. A similar approach can be made for estimating compressor performance in the field.

Using the volumetric efficiency, field power can be predicted by modifying the equation number 1 discussed earlier.

Compressor Power = (Polytropic Head x Mass Flow) / (Polytropic Efficiency x VE)…...(8)

Similarly the net mass flow delivered can be estimated by modifying equation 2 as follows:

Mass Flow = (Engine Power x Polytropic Efficiency x VE) / Polytropic Head…………. (9)

When determining the polytropic head and polytropic efficiency a consistent set of volumetric flow should be used as shown in Figure 5.

It is possible to calculate the volumetric efficiencies for the new design when clearances are small as well as allowing for the effect of increased clearances.

The volumetric efficiency concept will not solve all field related performance issues. However, it may help to establish correct mass flows through the machine and that is one step in the right direction. For medium pressure and high-pressure back-to-back compressors in particular, the internal leakages can be considerable and are underestimated. This distorts the real performance of the machine. Volumetric efficiencies in such cases can provide a means for resolving performance short falls in the field.

5. MODEL BASED ANALYSIS HELPS OVERCOME MEASUREMENT DIFFICULTIES

MSE models of compressor trains integrate the driver and driven equipment. This normally includes a thermodynamic model of gas turbines and a separate model of the gas compressor with balance piston and centre seal leakages. The models are calibrated for the original design conditions when they use the factory test data for the compressor and the driver. However when applying these models to analyse the field data, without including internal leakages, discrepancies were found in mass and energy balances.

The introduction of internal leakages made it possible to vary mass flows through the machine and examine its effect on head and energy balances. It was found that by carefully analysing the compressor field performance and taking due care of the internal losses, it was possible to calibrate the computer model and achieve energy balance between the compressor and the driver using the vendor provided performance maps.

The case study 3 highlighted the flow and power discrepancies. However once leakage corrections were applied the model provided a consistent set of results.

6. THE JOINT INDUSTRY PROJECT

Recognising the need for a better understanding of centrifugal compressor performance, as related to operating machines in the field, MSE launched a Joint Industry Project (JIP) entitled, "Pointing to the optimum performance of centrifugal compressors for oil & gas ". The JIP was supported by five operating companies namely BG Group, BP, Centrica, ConocoPhillips & Lasmo.

Design and operating data was obtained and analysed for more than 50 compressors and drivers. The analysis included performance of compressors and the surrounding systems i.e. recycle system, process equipment including knockout drums and liquid carry over issues, gas coolers, gas turbines and electric motors.

This data was incorporated in a large database, which was made available to the JIP group. Database was designed to protect the anonymity of the operator and their machines. Performance and history of the machine was analysed over an 18-month period. This provided a very large volume of data. Analytical tools were developed for the analysis of compressor performance, processing equipment performance and gas turbine performance. The data was rationalised using the MSE model of a compressor and gas turbine and this helped to resolve many issues with flow, pressure and temperature measurements. Statistical techniques were applied to find the mean performance for a large population of data. The analysis of data and results were presented to the JIP group in three separate workshops in which guidelines were presented for performance retention of centrifugal compressors.

7. COMPARISON WITH DESIGN PERFORMANCE

One important outcome of the JIP has been to compare the compressor design performance with that actually achieved in the field.

Figure 6 is a plot of compressor efficiency versus average inlet flow coefficient for the compressor stages. It shows the guaranteed efficiencies quoted by the vendor as well as the actual efficiencies recorded during the JIP. The scatter in the recorded efficiencies indicates that some machines are operating close to the original design values whilst others are significantly under performing.

In general terms the actual efficiencies and guaranteed efficiencies tend to converge at higher flow coefficients >0.05. The highest actual efficiencies have been recorded for flow coefficients greater than 0.1. Conversely measured efficiencies diverge from the guaranteed values as flow coefficients decrease below 0.05. In the database the lowest efficiencies recorded were for flow coefficients <0.02. In all cases the actual efficiencies were lower than the guaranteed efficiencies.

Although the amount of data available is limited and more data is needed, it is interesting to note that variation of efficiency with flow coefficient for installed machines shows a similar trend as for the new machines. However the actual efficiencies are significantly lower than values quoted by the manufactureres.

The JIP database was used to determine if there is a pattern in performance degradation, which is independent of the manufacturers and which is common to all the machines operating in the field. The results of the JIP appear to show that this is indeed that case.

For example, the belief that presence of liquid carryover in the compressor adversely affects its performance was verified by analysing the performance of the knock out drums (KOD) which are installed upstream of the compressors. The larger diameter KOD’s were found to be more effective in removing liquid carry over which in turn gave rise to a better compressor performance. Conversely, smaller diameter KODs resulted in higher liquid carry over which exacerbated the performance loss and increased fouling of the machines.

For the operating companies, knowing the real performance of their machines is a great step forward. This information helps in making accurate production forecasts. It allows decisions to be made to remedy the production shortfall problem with greater confidence i.e. do we need to re-wheel the compressor to increase the capacity or change the driver since we need more power. Making correct decisions is critical to the successful operation of the oil and gas assets.

Manufacturers need to focus on machinery performance as it relates to operation in the field. Higher efficiency is of no value if it can only be achieved within the factory gates and cannot be reproduced in the field.

Concept of “volumetric efficiency” as described in the paper can prove useful in establishing realistic efficiencies in the field and avoid incorrect designs to be implemented in the field.

8. CONCLUSIONS

The main conclusions of this article are summarised as follows:

- Determining performance of compressors operating in the field is a difficult task and requires a different approach from factory testing.
- The flow measurements in particular are very difficult to verify without exhaustive analysis of the compressor internal designs.
- Volumetric efficiency concept, adapted from piston compressors, can prove useful for correcting field performance provided it is applied correctly.
- However, unlike piston compressors, a reduction in the volumetric efficiency increases compression power for centrifugal compressors as well as head requirements.

- Problems with field testing and measurement inaccuracies can be resolved by building accurate models of compressors and drivers and their calibration with design and test data.
- Although not apparent, loss of compressor efficiency is a major contributing factor to a reduction in production capacity of oil & gas facilities.
- Regular performance testing and monitoring of compressors should be carried out. This can help to resolve field operating problems for example, engine failure on overload, repeated failure of thrust bearings, surging of compressors and inability to meet the export pipeline pressures etc.
- Experience of JIP machines has shown that the performance losses experienced by centrifugal compressors are substantially higher than 4% allowed for in the API 617.

9. ACKNOWLEDGMENT

The author would like to acknowledge the following companies who supported the Joint Industry Project: - BG Group, British Gas Hydrocarbon Resources Ltd wholly owned subsidiary of Centrica, Conoco (U.K) Limited now known as ConocoPhillips, BP Exploration Operating Company Ltd & Lasmo Netherlands (now owned by CH4 Netherlands).

Effect of loss of compressor efficiency on flow capacity

Table 1.

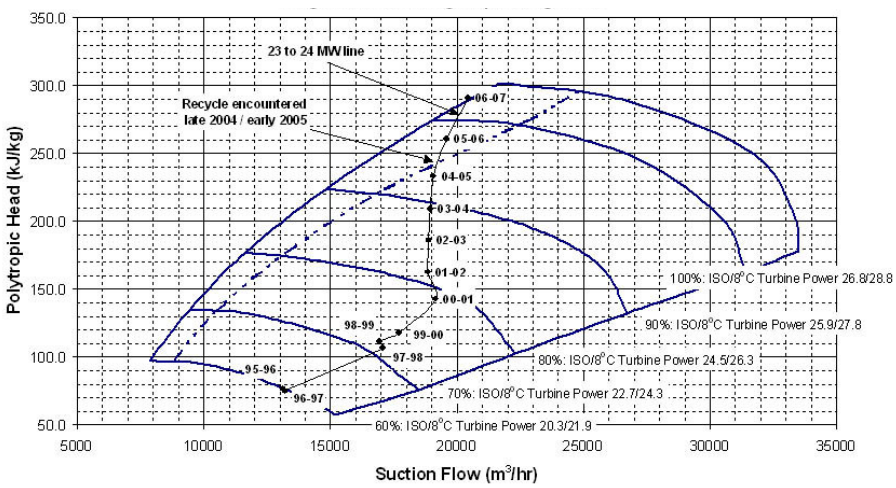
Polytrophic Efficiency %	**Standard Flow MMSCFD**	**Actual Volume Flow M3/hr**	**Polytrophic Head kj/kg**	**Driver Power**
At design 80%	515	18 932	137.5	23.7
During operation 75%	483	17 760	137.5	23.7
During operation 70%	450	16 578	137.5	23.7
During operation 65%	418	15 392	137.5	23.7
During operation 60%	386	14 208	137.5	23.7
Assumptions				
Suction pressure	30 bar			
Discharge pressure	85 bar			
Mole weight of gas	19.3			
Variation in head	ignored			
Engine max. power	23.7MW			

Table 2.

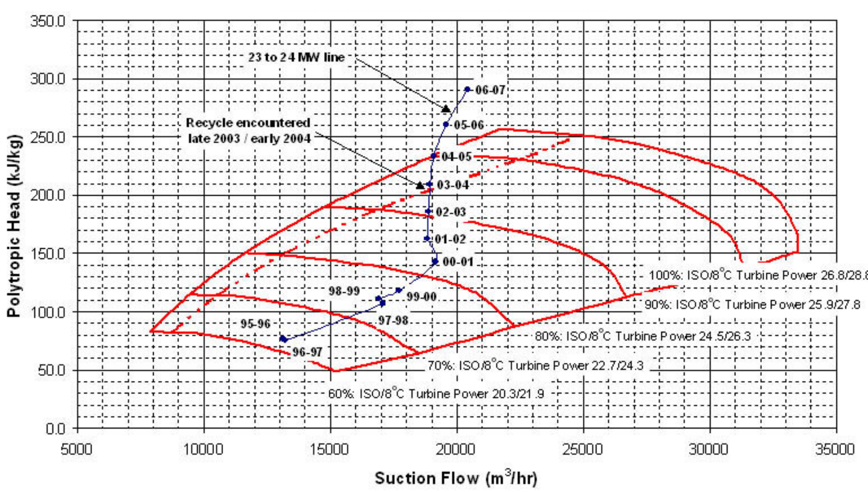
	Stage 1	Stage 2	Stage 3	Stage 4	Discharge
Pressure (bar)	2.00	5.5	15.7	36.5	
Discharge pressure (bar)	6.00	16.2	37.00	72.50	
Inlet flow (kSm^3/day)	1165	1048	971	1121	920
Molecular Weight	25.6	25.8	26.9	24.1	

<u>Wellhead / Export Compressor</u>

As Designed
Progression of Design Operating Point

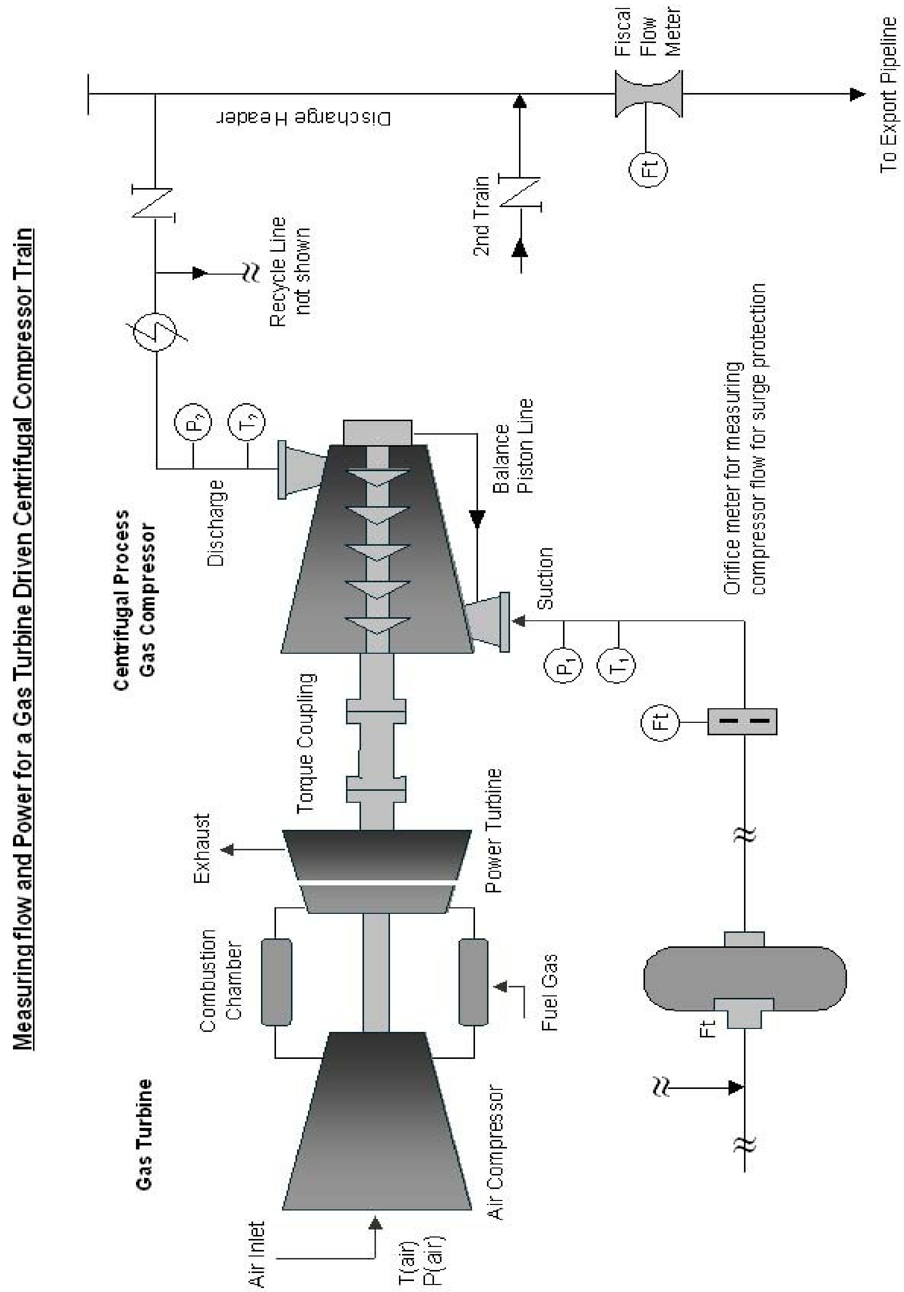


Fouled (15%)
Progression of Design Operating Point



For efficiency loss, see Table 1

Figure 1



Measuring flow and Power for a Gas Turbine Driven Centrifugal Compressor Train
Gas Turbine
Air Inlet
T(air)
P(air)
Air Compressor
Combustion Chamber
Fuel Gas
Exhaust
Power Turbine
Torque Coupling
Centrifugal Process Gas Compressor
Discharge
Suction
Balance Piston Line
P_2
T_2
P_1
T_1
Ft
Ft
Ft
Orifice meter for measuring compressor flow for surge protection
Recycle Line not shown
Discharge Header
2nd Train
Fiscal Flow Meter
To Export Pipeline

Figure 2

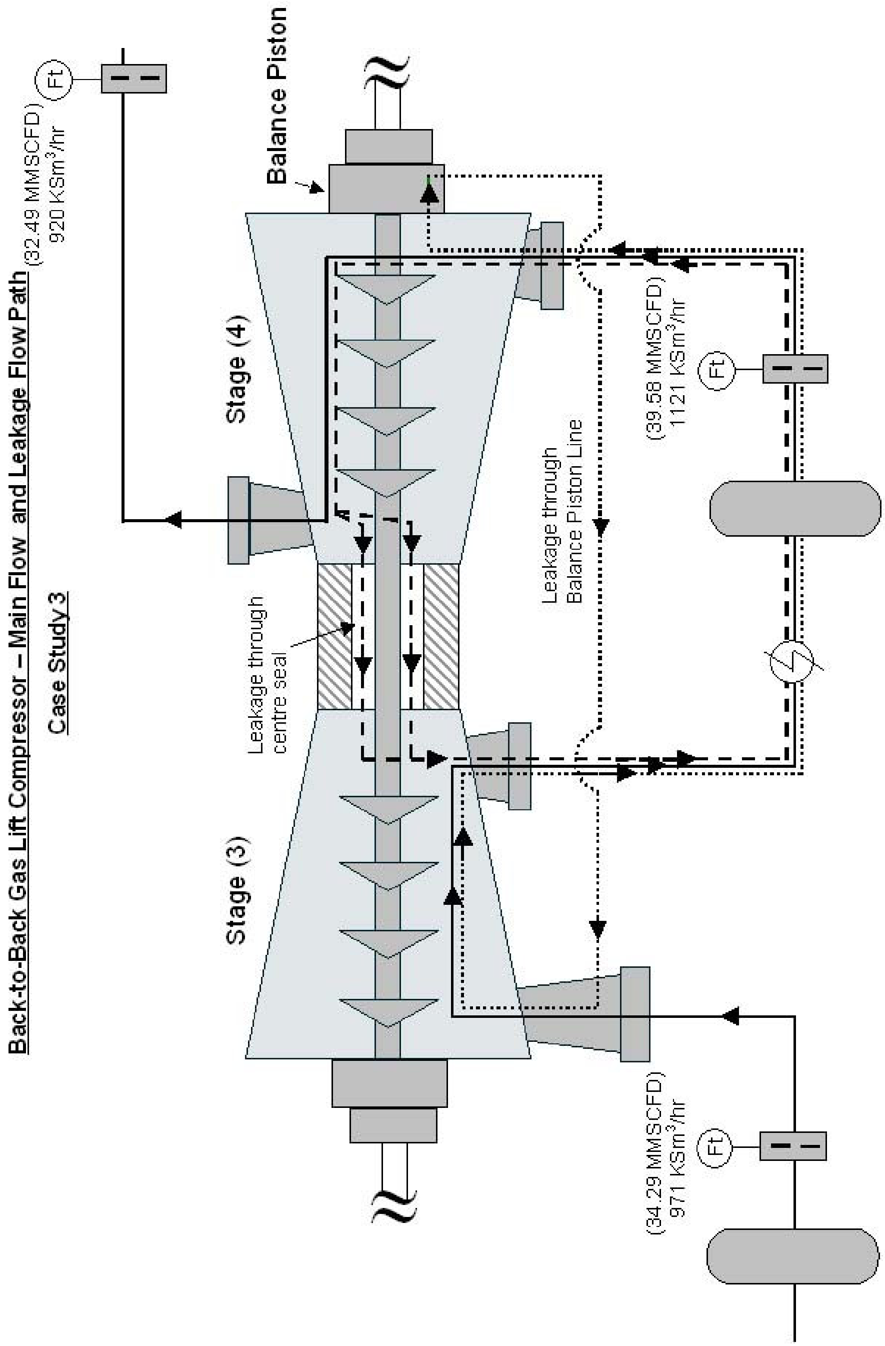

Back-to-Back Gas Lift Compressor – Main Flow and Leakage Flow Path
Case Study 3
(32.49 MMSCFD)
920 KSm3/hr
Ft
Balance Piston
Stage (4)
Stage (3)
Leakage through centre seal
Leakage through Balance Piston Line
(39.58 MMSCFD)
1121 KSm3/hr
Ft
(34.29 MMSCFD)
971 KSm3/hr
Ft

Figure 3

Performance Discrepancies in a Gas Lift Compressor

Case Study 3

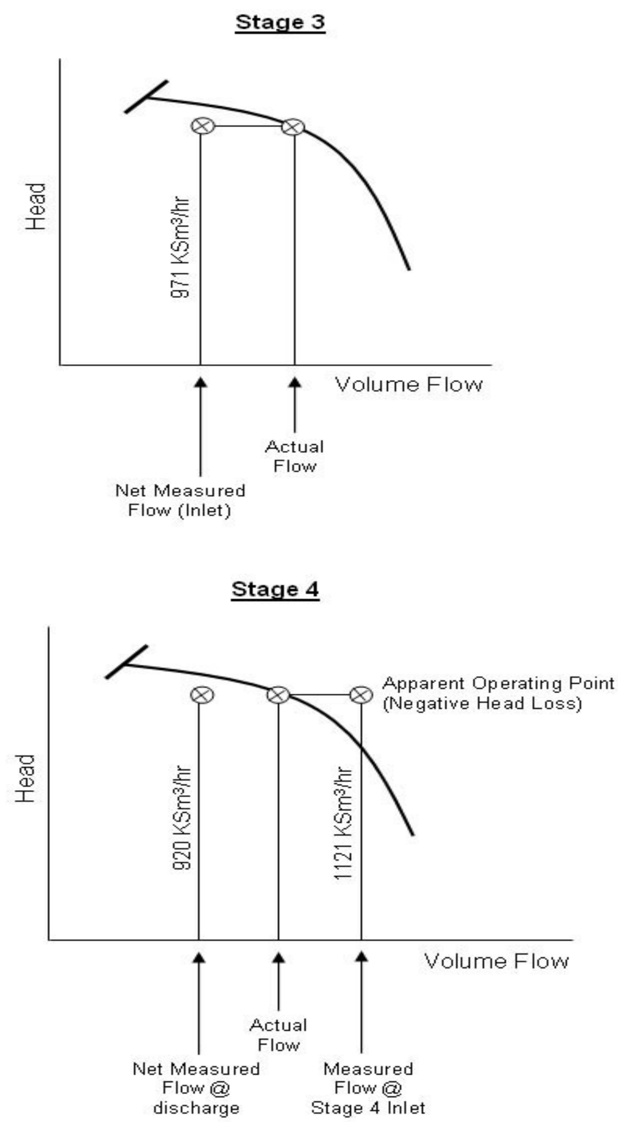
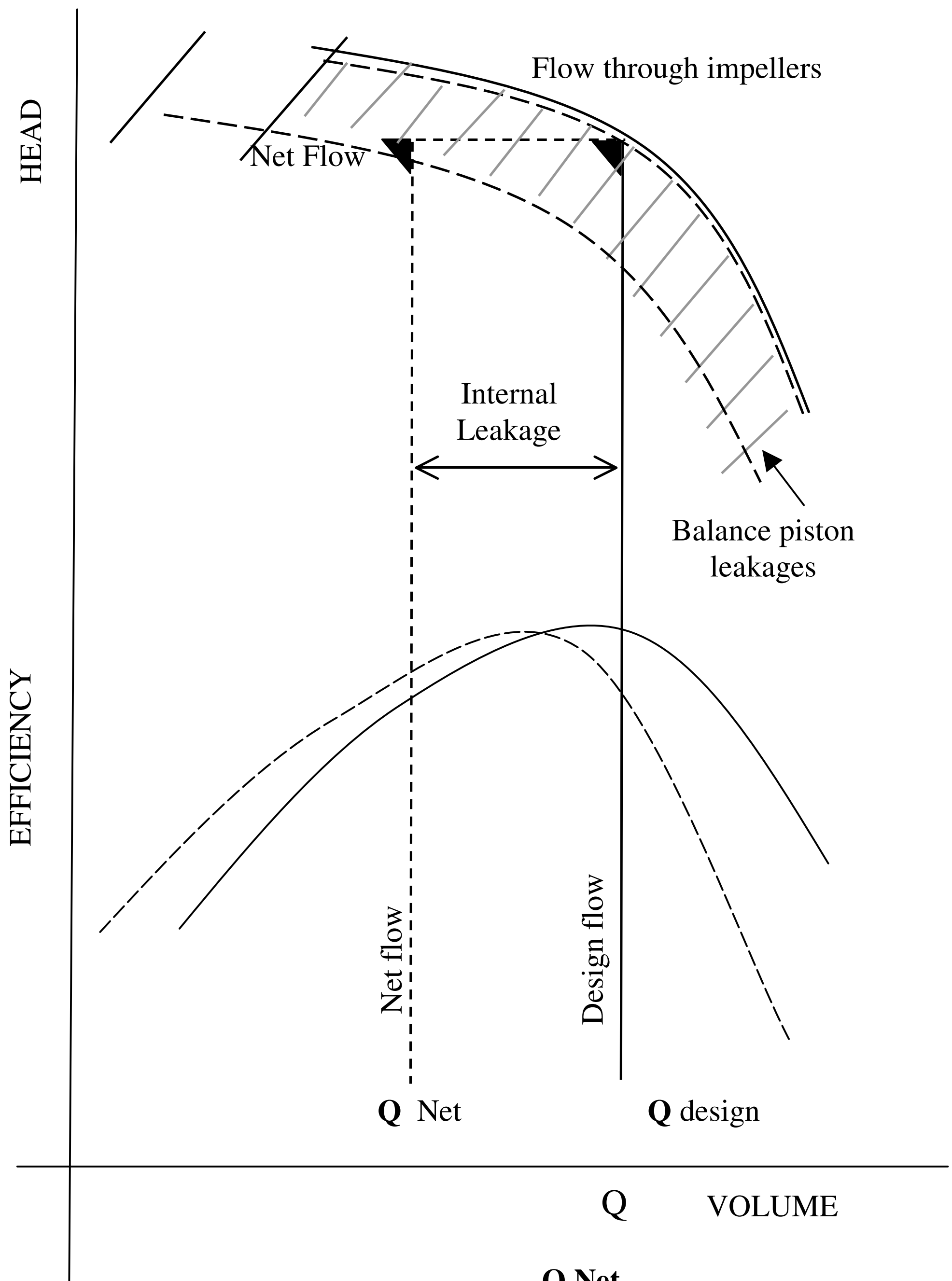


Figure 4

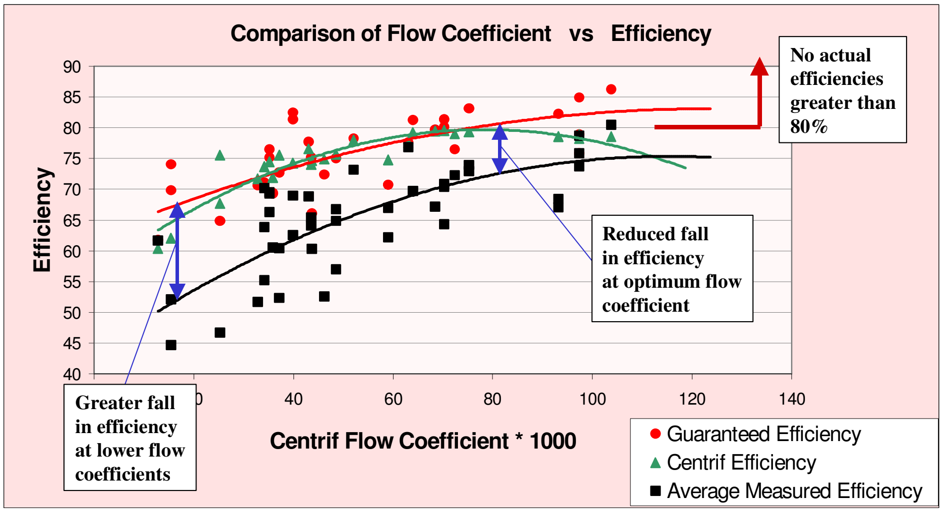
Explanation of Volumetric Efficiency



Volumetric efficiency $\dfrac{\textbf{Q Net}}{\text{Q design}}$

Figure 5

As Designed vs. Actual Efficiencies



The Relationship between Flow Coefficient and Efficiency and Performance Retention

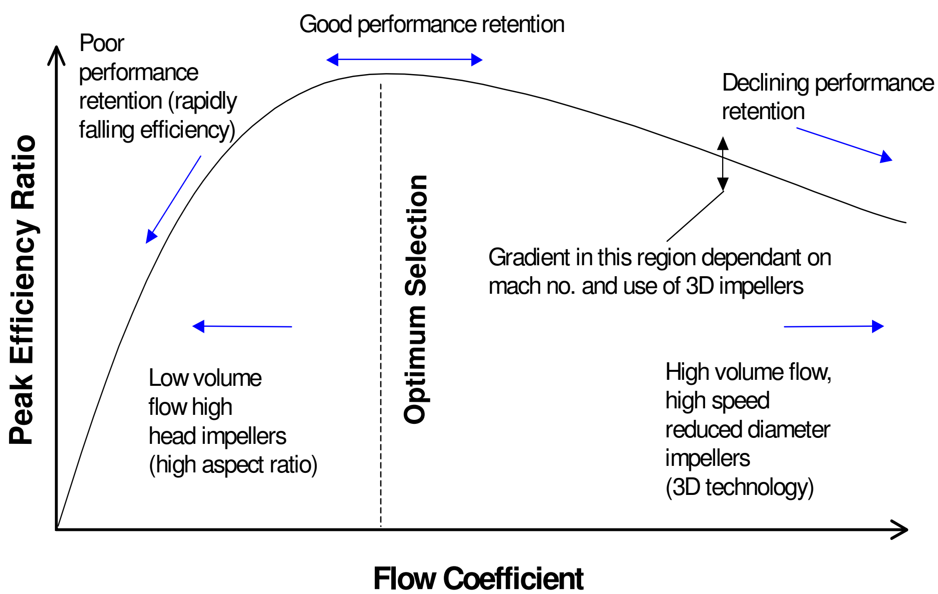


Figure 6

References

1. API 617, American Petroleum Institute Centrifugal Compressor for General Refining Service
2. ASME Power Test Code 10
3. GPA Hand Book Volume I, Engineering data book – FPS Volume 1
4. Dr M S Akhtar , Selection and optimization of centrifugal compressors for oil and gas applications, I MECH E paper, 1993