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PERFORMANCE ASSESSMENT OF GAS TURBINE DRIVEN CENTRIFUGAL COMPRESSOR TRAINS DERIVED FROM FIELD TESTING

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ABSTRACT

Field testing of gas turbine driven centrifugal compressor units at operational production facilities can provide valuable information about the condition of the equipment. In this paper, a case study is presented based on field testing performed on the compressors and gas turbine drivers of six units at a gas compression station. Since the compression units at this station are two sets of three identical machines, this case provides a unique opportunity to study both the advantages and some of the challenges inherent in this type of testing. For the compressors, comparisons are presented of the measured values of head rise, required power, and thermodynamic efficiency. For the gas turbines, performance comparisons are presented in terms of delivered power and heat rate. Calculations of the effects of ambient temperature on power and heat rate of the gas turbines are also presented. Some of the challenges of field testing are addressed, with the effects of fluctuations in gas composition on the experimental uncertainty in the measured performance treated in some detail. The topic of how field testing can help to identify degradation of the capability of compressors and gas turbines is also addressed. It is also shown how models derived from the performance tests can be used to forecast when factors such as high ambient temperatures and changes in the composition of the process gas can create situations where the compression units may not be able to deliver the gas flow rates and pressure levels required by the process.

INTRODUCTION

Performance testing of gas turbine driven compression units in the field can be used to address a number of practical issues related to the operation and maintenance of this equipment. One such issue is being able to identify performance degradation in aging equipment. For a gas compressor, examples of measureable effects of degradation would be losses in head rise, flow range, and efficiency. For the gas turbine, an important measurable effect of degradation is a reduction in the power produced to drive the compressor. Another issue is an ability to accurately predict of the capability of compression units to cope with changing process requirements. Such changes can include variation in the mole weight of the gas being handled due to such factors as well depletion, seasonal fluctuations, and switching sources from which the process gas is drawn. In some instances, an operator may also have a need to assess whether existing equipment can provide the compression capability needed to meet demands of increased production. In addition, the power output available from a gas turbine is affected by the atmospheric conditions in which it operates. In particular, power output falls off as the ambient temperature rises. This effect is felt particularly strongly in the Arabian Gulf region where temperatures in the summer months can exceed 50 °C.

An up-to-date and accurate picture of the performance capabilities of compression units in service can be highly beneficial to users of this equipment. Such information can be used to make optimal use of their compression assets, and help operators to avoid situations where gas must be flared because equipment cannot process all of the available supply. In the longer term, information derived from field testing can assist companies in making decisions about whether to upgrade or replace compression equipment.

This paper discusses practices for carrying out performance testing of gas turbine-driven compression units in the field and techniques for interpretation of the test data so obtained. The discussion is centered around a case study involving testing of six units which comprise the compression trains at a gas booster station in Kuwait. The testing was performed while the station was engaged in active production, and was carried out in a manner that caused a minimum of disruption to either production or to other operations at the facility.

The basic principles and practices of turbomachinery performance testing are well documented in industry standards such as ASME PTC-10 (1997) and ASME PTC-22 (2005). Guidelines and technical papers focused specifically on performing such tests in the field include those by Brun & Nored (2006) and Brun & Kurz (2001). The testing described in this paper was carried out in accordance with the recommendations of these publications to the extent possible. In the interpretation of the experimental data, established principles of turbomachinery performance and dynamic similitude were also employed. These principles can be found in the literature in such works at those by Shephard (1956), Dixon (1998), and Aungier (2000).

However, there are sometimes practical considerations at operating production facilities that require compromises to be made during the test program. In addition, there are some sources of uncertainty encountered that are not found under the more highly controlled conditions of the research laboratory or the factory acceptance test. This paper addresses some of these compromises and uncertainties and demonstrates that in spite of them, performance characteristics can be measured in the field that will be highly valuable to the owners and operators of gas turbine driven compression equipment.

EQUIPMENT TESTED

The equipment tested consisted of six gas turbine driven natural gas compression units located at a booster station belonging to the Kuwait Oil Company. The arrangement of equipment at the site is shown schematically in Figure 1. The compression equipment is arranged in three compression trains. Each train consists of a Low Pressure (LP) Compressor and a High Pressure (HP) Compressor, each having its own gas turbine driver. The gas compressed by the LP units is supplied from a gas network. The gas compressed by the HP units consists of gas from the LP compressors and also gas from other streams which are already at pressures high enough to be fed directly to the HP units. In terms of mass flow, a substantially larger amount of gas is processed by the HP units than the LP units. The compositions, and hence the mole weights, of the LP gas and the HP gas fluctuates somewhat during normal operation. The gas turbines are fueled by natural gas fed from a separate set of lines. The composition of the fuel gas also fluctuates.

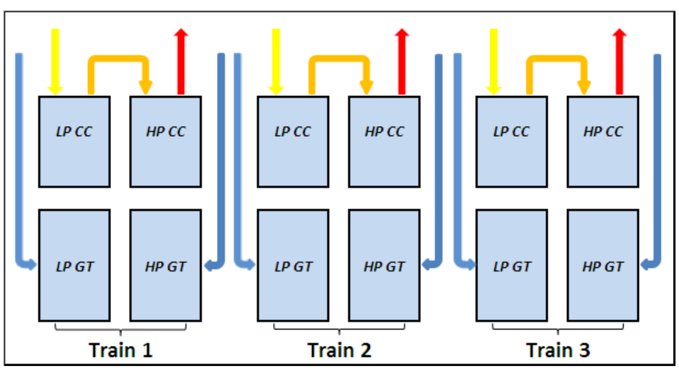

Figure 1. Arrangement of Compression Units at the Gas Booster Station Where Testing was Performed

TESTING PROCEDURES & INSTRUMENTATION

The procedures employed in the test program were guided by published industry standards for turbomachinery testing. These included the relevant ASME compressor and gas turbine test codes (ASME PTC-10 (1997) and ASME PTC-22 (2005)), and the GMRC guideline on performance testing in the field (Brun & Nored (2006)). During the test program described here, the relevant parts of these standards were adhered to as closely as practical.

In accordance with the project test plan, performance measurements were taken on each unit at five test points. For four of these points, the compressor was run at 100 percent of its normal operating speed and the load on the compressor was varied in order to produce different operating conditions. The primary control used for adjusting the load was the compressor recycle valve. The purpose of the fifth test point was to measure the maximum power capability of the gas turbine. For this test point, the compressor speed was increased to the maximum allowable value. Then, the compressor load was increased in an attempt to reach a condition requiring the maximum gas turbine power. The maximum power condition was indicated by the turbine exhaust gas temperature attaining its maximum allowable value. The maximum power condition was achieved for all three of the LP drivers. For the HP units however, constraints in the system that did not involve the compression units themselves prevented the maximum power condition from being reached. Specifically, maximum allowable values of pressures and velocities in components downstream of the compressor were reached while there was still some exhaust gas temperature margin remaining.

Each time the operating condition was changed, time was allowed for the pressures and temperatures to stabilize. This stabilization period is particularly important for temperature, as some time is required for heat transfer to occur that brings such items as thermowells and compressor casings to the same temperature as the flowing gas. Once equilibrium was reached, readings were then taken using the data acquisition system.

The testing standards referenced in this paper generally recommend taking four or more readings at each operating condition, spaced at intervals of at least 10 minutes. Adherence to this recommendation means that an equilibrium condition must be held for at least 30 minutes. During these tests, the interval between readings was generally shortened to approximately seven minutes, allowing four readings to be taken in about 20 minutes. This was done for two reasons. Firstly, due to operational considerations at the compressor station, it was often not possible to hold the unit under test at an operating point for a longer period. This was particularly true for operating conditions requiring a large amount of gas to be diverted to the unit under test. Secondly, the shorter measurement time allowed test activities to be completed on one unit per working day. This modification of the test procedure was felt to be a reasonable compromise between maximum accuracy and practicality given the overall goals of this particular test program.

Instrumentation

The schematic placement of measuring instrumentation on the driven compressors and gas turbines is shown in Figure 2 and Figure 3. Calibrated laboratory grade transducers were used for measuring the temperatures and pressures required to obtain machine performance. The temperature transducers used were of the Resistance Temperature Detector (RTD) type. For making temperature measurements on the process gas and the fuel gas, these were placed in available thermowells located at appropriate locations on the piping. Gas turbine air inlet temperature was also measured using an RTD; this was placed close to the inlet filter in an area shaded from the sun.

Fuel and process gas pressures were measured with piezoelectric style transducers. Sensitive differential pressure gauges were used to measure the pressure differences across the orifice plates used in making flow measurements. Pressure transducers were connected in parallel with those that are part of the permanently installed instrumentation. A hand held weather station was employed to monitor barometric pressure and humidity throughout the tests.

The temperature and pressure data was collected using a custom-built automated data acquisition system. This data was stored on a laptop computer for later post processing. Other unit operating parameters which were necessary for calculating performance were collected from the unit monitoring and control systems. This included such items as compressor rotating speed and turbine exhaust temperature.

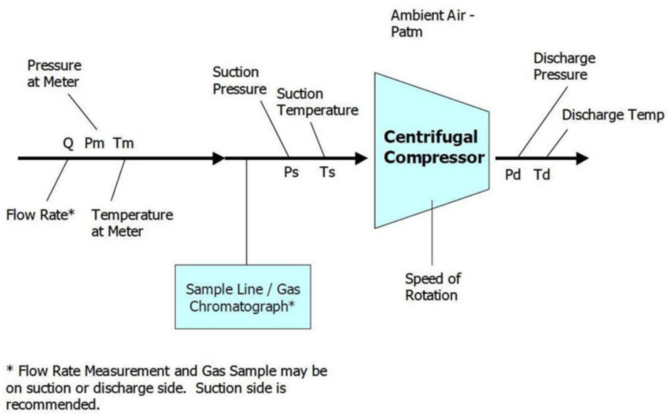

Figure 2. Schematic of Placement of Test Instrumentation on the Driven Compressor. (Excerpted from Brun & Nored, 2006)

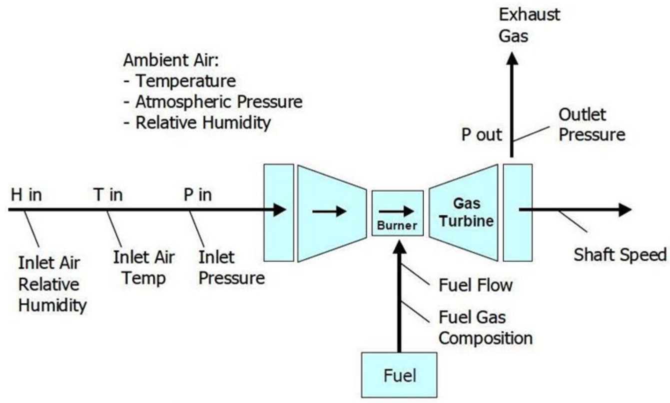

Figure 3. Schematic of placement of test instrumentation on the gas turbine driver. (Excerpted from Brun & Nored, 2006)

Gas Properties

Thermodynamic properties of the process gas and the fuel gas are essential for the data reduction process. These are required for calculating the parameters describing compressor performance, including head rise, power input, and efficiency. Gas properties are also required for calculating gas flow rates from the pressure and temperature measurements taken on the flow meters. In addition, the lower heating value of the gas (LHV) is required in order to determine how much fuel energy is being supplied to the engine per unit time.

Gas properties were obtained by taking samples of both process gas and fuel gas and having the compositions of the samples analyzed in a laboratory. The lab analysis provided the molecular composition of the gas and also the LHV. During the data reduction process, the gas compositions were input into a thermodynamic properties software package in order to calculate the state point variables needed for making performance calculations.

Two samples each of process gas and fuel gas were taken during the testing of each of the six units. The collection of the samples occurred close to the beginning and end of the test period on each unit. This sampling interval resulted in 24 samples being collected and analyzed during the test program. While it was recognized that fluctuations in gas composition could occur in much shorter time intervals, the number of samples taken was the maximum that it was considered practical to analyze for this program. It is perhaps possible that a portable gas chromatograph could be employed to analyze the composition much more frequently in future testing if more precise knowledge of gas composition is deemed necessary.

RESULTS AND DISCUSSION

Compressor Performance

The data taken on the compressors was reduced to the form of non-dimensional coefficients. Expressing the results in this form allows performance to be compared on a common basis, compensating for small variations of such quantities as inlet pressure and temperature, gas composition, and rotating speed.

The coefficients employed are flow coefficient;

$$\varphi = \frac{Q}{\pi U_2 R_2^2}$$

isentropic head coefficient;

$$\psi = \frac{\Delta H_{isentropic}}{U_2^2}$$

power coefficient;

$$\overline{P} = \frac{P}{\pi \rho U_2^3 R_2^2}$$

and isentropic efficiency.

$$\eta = \frac{\Delta H_{isentropic}}{\Delta H_{actual}}$$

Figure 4 through Figure 6 show plots of the variation of isentropic head rise, power required, and efficiency with volume flow for the LP compressors. Figure 7 through Figure 9 show the corresponding information for the HP compressors. For purposes of this paper, compressor performance results have been normalized to values at an operating point supplied on the original compressor data sheets, referred to here as the "specified point". The specified point is represented by a yellow diamond symbol on each plot. The solid red lines in the plots are curve fits to the combined performance data for each set of compressors. These fitted curves provide a useful reference against which performance of the individual compressors can be compared. Also, these curves can serve as representations of typical unit performance. Such representative curves are useful in making predictions of the effects of changes in such quantities as gas composition, and inlet pressure and temperature.

As seen in the figures, the flow range covered during the LP tests was fairly narrow. The largest variation of volume flow is seen for Unit 3, for which the flow ranges between about 90 percent and 103 percent of that at the specified point. However, the narrow flow range covered was not a consequence of flow limitations inherent in the compressors themselves (such as occurrence of surge or choke). Rather, the flow range of the tests was limited by such factors as the amount that the compressor load could be varied, allowable suction and discharge pressures in the system, and occasionally, by the amount of gas available in the system at the time of the testing. Nevertheless, the flow range covered in the testing is representative of that within which the LP units operate almost all of the time.

Comparison of the head rise, power required, and efficiency data with their respective curve fits shows that the performance levels of the three LP units match each other quite closely. This consistency indicates that none of the LP compressors has suffered a noticeably larger amount of performance degradation than the others. The measured head rise levels of the individual compressors seen in Figure 4 generally lie within +/- 2 percent of the curve fit to the combined data, with the largest departure for an individual point being about 4 percent. For the compressor power (Figure 5), most of the individual measurements fall within about 1½ percent of the curve fit. Similarly, most of the efficiency data in Figure 6 fall within 1½ percent of the curve fit, except for a few outliers. Some idea of the repeatability that can be achieved in these measurements can be gleaned from the consistency seen in the data points from LP Unit 1, which are clustered in the neighborhood of 96 percent of the flow at the specified point.

In comparing the curve fit of the field measurements to the performance given for the specified point on the original compressor data sheets, a shortfall is evident. At the corresponding flow point, the head rise falls about 1 percent lower, and that the required power 4 percent higher than the values given for the specified point. Consequently, the measurements show about a 5 percent shortfall in efficiency. However, it could not be determined from the records available for these machines whether they had performed at the levels indicated by the specified point even when they were new. Therefore, the shortfalls may be interpreted as something of a "worst case" scenario for the amount of degradation that has taken place in the many years that these machines have operated. Indeed, one of the reasons for the field test of these machines was to provide a new benchmark level for the performance.

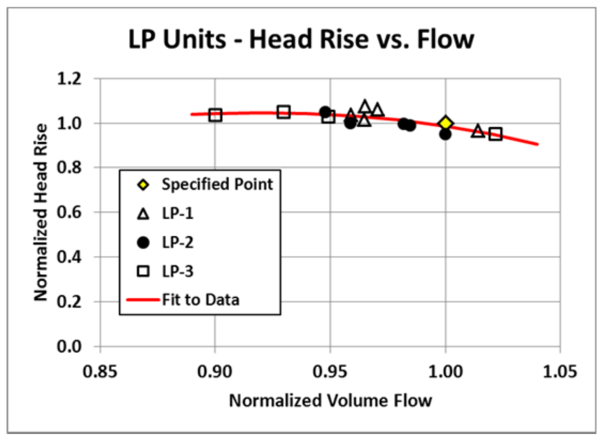

Figure 4. Measured Head Rise vs. Volume Flow Rate for LP Compressors

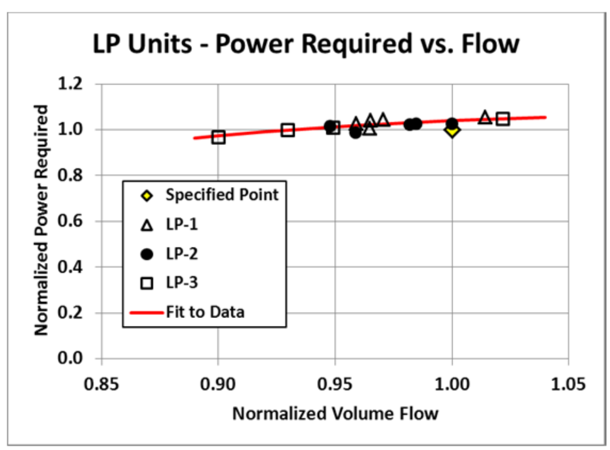

Figure 5. Measured Values of Power Required for Process Gas Compression vs. Volume Flow Rate for LP Compressors

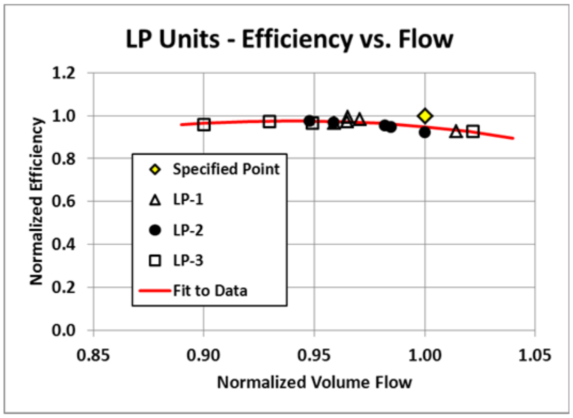

Figure 6. Efficiency vs. Volume Flow Rate for LP Compressors Calculated from Measured Head and Power Characteristics

Inspection of the performance plots for the HP machines (Figure 7 through Figure 9) shows that it was possible to achieve a larger range of flow rates than it had been with the LP units. For the HP units the flow range spanned by the combined data set covers from 72 to 110 percent of the flow rate at the specified point. The curve fit to the combined field data shows a head rise that is about 7 percent short of that given in the data sheets for the specified operating point. However, the power required at this operating point is lower by about 5 percent. The lower power required partially offsets the shortfall in head rise so that the efficiency at the specified point is only about 1½ percent lower than quoted on the data sheets.

It is also seen that the scatter in the measured data is larger than that which was seen for the LP units. The majority of the head rise data points lie within +/- 7 percent of the curve fit. The scatter in the power data is somewhat smaller, with most points lying within +/- 3 percent of the curve fit. Most of the efficiency data lie within a band of +/- 5 percent, with a few outliers having a scatter of more than 10 percent. In reviewing the raw data taken on the compressors it was found that it had been more difficult to hold the HP units at a constant operating condition. It is expected that at least some of the increased scatter in the HP data is attributable to this factor.

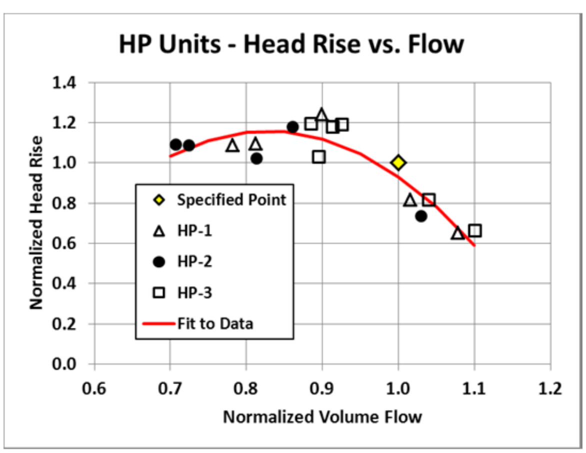

Figure 7. Measured Head Rise vs. Volume Flow Rate for HP Compressors

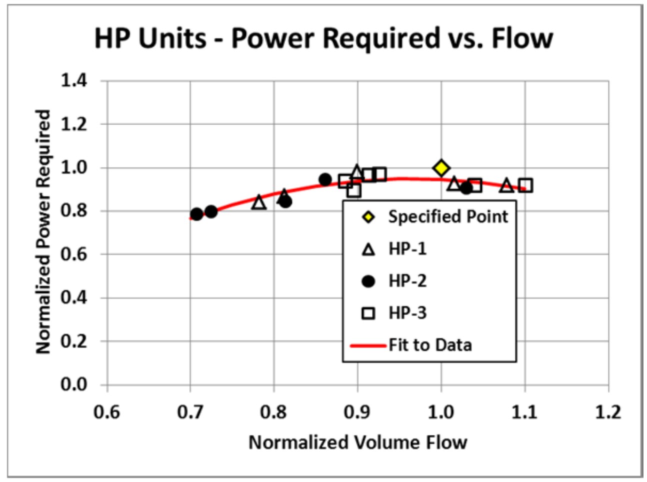

Figure 8. Measured Values of Power Required for Process Gas Compression vs. Volume Flow Rate for HP Compressors

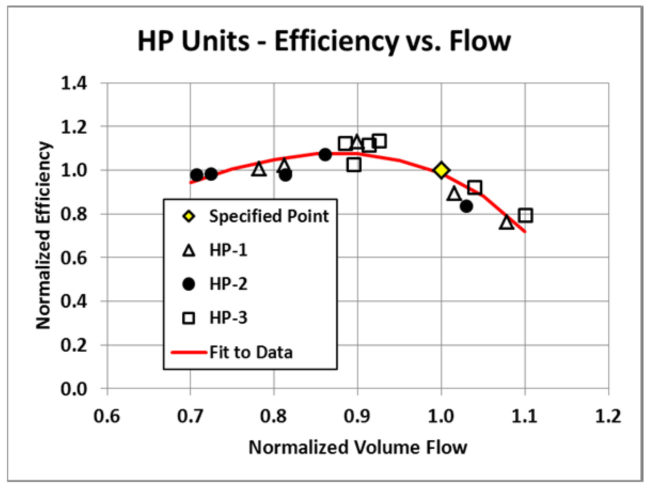

Figure 9. Efficiency vs. Volume Flow Rate for HP Compressors Calculated From Measured Head and Power Characteristics

Gas Turbine Power and Heat Rate

The performance of the gas turbine drivers was evaluated in terms of their power capability and their heat rate. The heat rate is a quantity defined as the ratio of the rate that fuel energy is delivered to the engine and the power output. The formula for heat rate is given by

$$HR = \frac{\dot{E}_{fuel}}{P} = \frac{\dot{m}_{fuel} \times LHV}{P}$$

The power level used in the gas turbine evaluation is the thermodynamic power delivered to the gas in the process compressor. This quantity is, for all practical purposes, equal to the shaft power delivered by the turbine. (The thermodynamic power will differ from the shaft power by a small amount due to mechanical losses and other minor parasitic losses that absorb power without imparting energy to the process gas.)

The maximum delivered power and heat rate evaluated at maximum power for the three LP units are shown in Table 1. These quantities are shown for operation of the engines at ambient temperatures of 15 °C and 50 °C. For purposes of this paper, the power and heat rate values have been normalized by the average values of these quantities obtained at 15 °C. The full power tests on the three LP drivers were performed on dry days during the winter months at ambient temperatures in the range of 25-27 °C. Correction curves provided by the engine manufacturer were used to calculate the power and heat rate values expected to be obtained at the two temperatures of interest.

The Table shows that the variation in power and heat rate among the engines is quite small. Less than 1 percent variation from the mean is seen for the two quantities at each temperature condition. It is also seen that an increase of the temperature from 15 °C to 50 °C (a temperature that is easily reached in the Arabian Gulf region during the summer) results in a 22% reduction in available driver power. This loss of power is accompanied by an increase in heat rate of more than 6%.

Maximum power tests were attempted for the HP gas turbines at this booster station. However, the HP tests were also performed with relatively cool ambient temperatures. Under these conditions, not enough load could be applied to the gas compressors to extract maximum power from the drivers. (This load limitation was due to constraints on pressures and flow rates in the HP compression system imposed by components other than the gas compressors themselves.) Nevertheless, the measurements taken at the highest power level obtained showed that the three HP drivers were giving quite similar power levels at a given exhaust temperature and that there is a substantial power margin available.

Table 1. Comparison of Corrected Power and Heat Rate for the LP Unit Gas Turbine Drivers

	Ambient Temp. 15 deg. C		Ambient Temp. 50 deg. C	
Gas Turbine	Normalized Power	Normalized Heat Rate	Normalized Power	Normalized Heat Rate
LP Unit 1	1.005	0.997	0.785	1.060
LP Unit 2	1.002	1.005	0.782	1.070
LP Unit 3	0.993	0.998	0.776	1.062

Quantifying Effects of Changes in Gas Mole Weight and Ambient Temperature on Compression Unit Performance

One of the goals of carrying out the performance testing was to be able to accurately predict effects of changes in process gas conditions and of atmospheric conditions on the production capacity of the compression units. Two factors that are of particular interest in terms of their effect on production capacity are the temperature of the inlet air to the gas turbines, and the molecular weight of the process gas. The air temperature affects the power that the gas turbines can produce as discussed in the previous section, while the mole weight of the gas being handled affects the power needed to drive the process compressors.

The power required to drive a compressor is given by the equation;

$$P = \pi \rho U_2^3 R_2^2 \times \overline{P}$$

This equation shows that the power required increases with the cube of the impeller speed. There is a linear dependence on the gas density. The gas density term, includes the effect of gas mole weight, and also the effects of temperature, pressure, and compressibility of the gas. These effects are seen in the equation of state;

$$\rho = \frac{pMW}{Z\Re T}$$

For a given compressor, the values of the non-dimensional power coefficient can be obtained from curves of measured compressor performance, such as those obtained from the field measurements performed on the LP and HP units.

An example of a practical application is shown in Figure 10 and Figure 11. This example addresses the capability of the LP units to process gas having two different mole weights for a range of ambient temperatures. The heavier gas, having a mole weight of 29.1, corresponds to the composition of the gas for which the LP compressors were originally designed. The lighter mole weight, 26.0, is typical of the gas being handled by these units at the present time. The green and red horizontal lines on each plot indicate the maximum power that is available from the gas turbines when operating with inlet air temperatures of 15°C and 50°C, respectively. Also, on each plot are curves showing the power required by the compressor at two operating speeds. The lines labeled “100% N” correspond to operation of the compressor at 100 percent of its design operating speed. The lines labeled “105% N” correspond to operation at 105 percent of the design speed, which is the maximum continuous operating speed permitted for these machines. The volume flow rates in these figures have been normalized to those at the specified point for the LP compressors used earlier in presenting the compressor performance measurements.

Figure 10 shows that high ambient temperatures strongly restrict the compression capability with the 29.1 mole weight gas. It is seen that when the compressor is operated at 100 percent of design speed, the driver power required exceeds that available unless the volume flow is reduced to about 88 percent of that at the specified point. At cooler temperatures a wider range of operation becomes possible. At sufficiently cool temperatures, if additional head rise is needed, there is power available to run the compressors at the maximum continuous operating speed. However, even on a 15°C day operation at 105 percent speed will require more power than is available at flows above about 106 percent of the reference value.

Figure 11 shows that when the mole weight is reduced more operating flexibility is available. Even on a 50°C day, the power available exceeds that required up until about the 105 percent flow point. Also, not surprisingly, it is seen that operation at maximum continuous operating speed with flows at or above that of the specified point will be possible for a wider range of temperatures. It should, however, be noted that at a given rotating speed, a gas of a lower mole weight will undergo a smaller pressure rise. Therefore, compressor speed may need to be increased if the process requires a certain minimum discharge pressure.

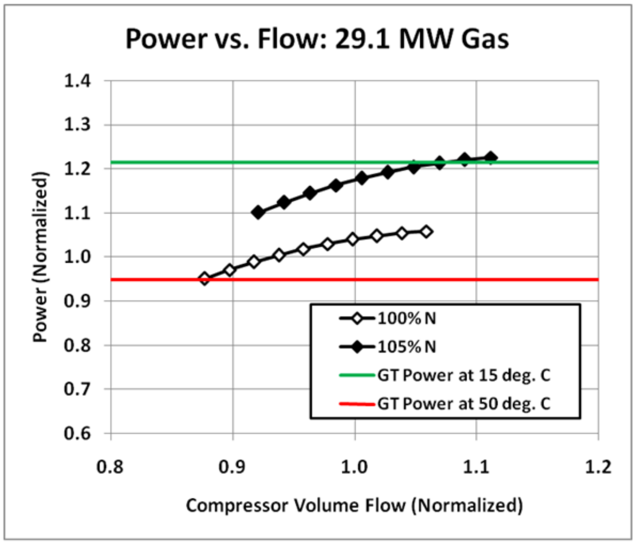

Figure 10. LP Compressor Power Requirements with 29.1 MW Process Gas Compared to Available Gas Turbine Power at 15 and 50 °C

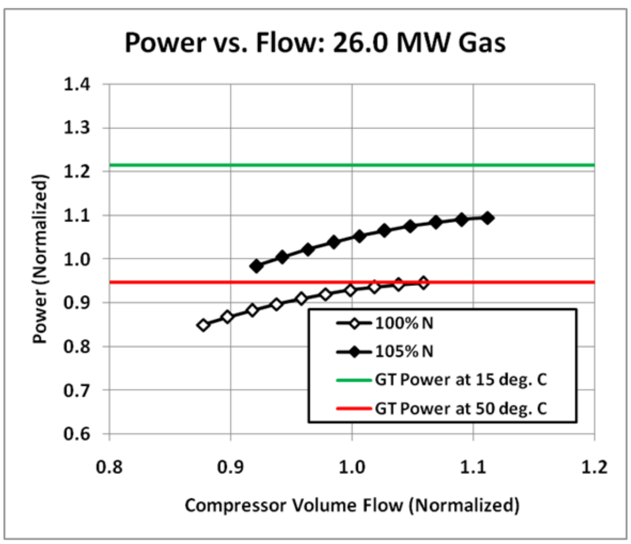

Figure 11. LP Compressor Power Requirements with 26.0 MW Process Gas Compared to Available Gas Turbine power at 15 and 50 °C.

DISCUSSION: EFFECT OF MOLE WEIGHT UNCERATAINTY ON PERFORMANCE MEASUREMENTS

Fluctuations in gas composition introduce a source of uncertainty in field testing which is not normally a factor in the more controlled setting of the laboratory or factory test stand. This source of uncertainty occurs because models based on a specified gas composition are used to calculate the thermodynamic state of the fluid from the measured temperature and pressure data. These thermodynamic state variables including density, enthalpy, and entropy, are in turn used in calculating performance. In addition, the composition affects the energy content of the gas, namely the lower heating value (LHV). The LHV is an important part of determining the flux of fuel energy to the engine used in calculating the gas turbine heat rate.

In the field testing discussed in this paper, two samples each of process gas and fuel gas were obtained from each unit. The samples were taken near the start and near the end of testing on each compression unit. The samples received laboratory analysis to obtain their compositions and heating values. However, in general, the compositions of the gas streams can fluctuate on a much shorter time scale than the gas sampling interval. Therefore, the precise compositions of the fuel and process gas at any given measurement point cannot be determined.

As an example of the variation in gas properties seen in these tests, Figure 12 shows a plot of normalized values of the mole weight and lower heating value for 19 fuel gas samples. These samples were collected during the testing of ten gas turbines, including the six units discussed in this paper and four others at a neighboring booster station. The plot shows that while mole weights and heating values for most of the samples are clustered in a fairly narrow range, there were a few examples of larger variation, with the biggest outlier having a mole weight about 20 percent higher than most of the other samples.

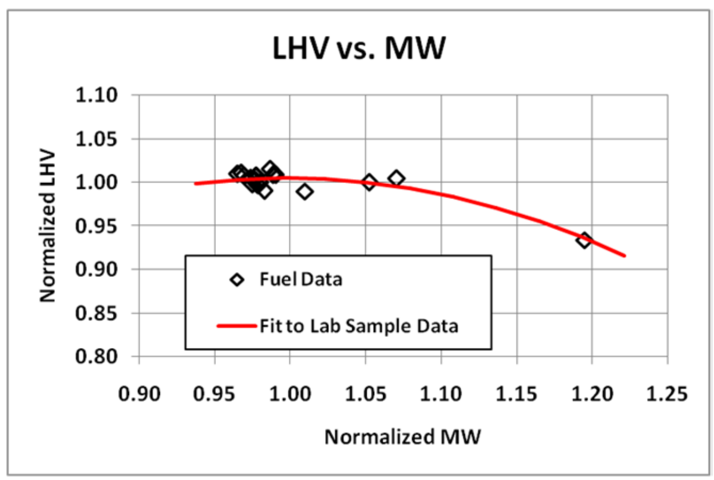

Figure 12. Variation of Mole Weight and Lower Heating Value of 19 Fuel Gas Samples

Estimates of the effect on performance related variables caused by uncertainties in the gas composition can be carried out using the method of Kline & McClintock (1953). (See also Brun & Kurz (2001), Brun & Nored (2006), and ASME PTC 19.1 (1990).) While this method is applicable to any of the performance related quantities, it is illustrated here using the measurement of flow rate and fuel energy flux as examples.

Effect of Gas Composition on Flow Measurement

The measurement of the flow rates of both process gas and fuel gas was carried out using orifice style flow meters. Referring to the ASME standard MFC-3M-1989 (1990), the volume flow rate, Q, through an orifice meter is given by

$$Q = K\sqrt{\frac{\Delta p}{\rho}}$$

where Δp is the measured pressure drop across the orifice and ρ is the density of the fluid passing through it. For purposes of this discussion, the quantity K can be taken to be constant for an orifice of a given geometry. Also, for simplicity, the gas is treated as an ideal gas ($Z = 1$), although effects of compressibility can be easily incorporated if desired. Introducing the ideal gas equation of state, the expression for the volume flow

$$Q = K\sqrt{\frac{\Delta p}{p}\frac{\Re T}{MW}}$$

This equation illustrates that the uncertainty in the gas composition is embodied in the uncertainty in the gas mole weight. Applying the Kline & McClintock method, it can be shown that the fractional uncertainty in the volume flow rate due to uncertainty in the mole weight is given by

$$\frac{\delta Q_{MW}}{Q} = -\frac{1}{2}\frac{\delta MW}{MW}$$

Therefore, if for example the uncertainty in the mole weight of the gas is estimated to be +/- 4%, this will contribute a +/- 2% uncertainty in the measured flow rate.

Effect on Uncertainty of Fuel Energy Flux

The rate at which fuel energy is supplied to the engine, or fuel energy flux, is equal to the product of the mass flow rate of the fuel (density multiplied by volume flow) and the heating value of the fuel. Since fuel flow rate is measured using an orifice flow meter like that used for the process gas, the orifice flow equation used in the previous section applies, and it can be shown that the fuel energy flux is given by

$$\dot{E}_{fuel} = K\sqrt{\Delta p \frac{p}{\Re T}} \times LHV\sqrt{MW}$$

Inspection of the resulting equation shows that the effect of uncertainty in the gas composition is embodied in the product of the fuel heating value and the square root of the mole weight ($LHV\sqrt{MW}$). Figure 13 shows this quantity plotted for the fuel samples from Figure 12. This plot shows that even for the rather large span of mole weights for these samples, the ($LHV\sqrt{MW}$) term remains within about +/- 3 percent of a constant value. What this illustrates is that uncertainties in mole weight are partially compensated for by the way that heating value changes with mole weight for these gas mixtures.

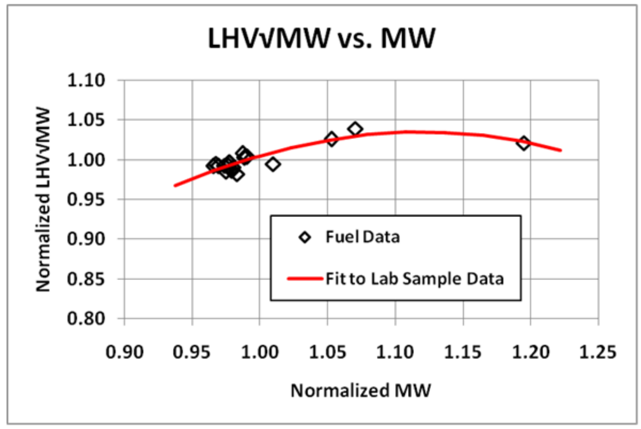

Figure 13. Variation of the $LHV\sqrt{MW}$ Parameter Which is Proportional to Fuel Energy Flux for a Given Pressure Drop Across an Orifice

CONCLUSIONS

This paper has demonstrated that valuable assessments of the performance capabilities of gas turbine driven compression units can be obtained from field measurements carried out at operating production facilities. The points made have been illustrated with a case study comprising measurements made at a gas booster facility in Kuwait operating six compression units. These consist of three identical units used for LP compression, and three identical units for HP compression. This afforded a unique opportunity to make comparisons between machines and to evaluate the quality of the measurements.

For the driven compressors, performance was evaluated using measured values of head rise, shaft power required, and efficiency. Comparisons of this data were made from machine to machine, and also against original performance specifications. For the gas turbines, performance was assessed through measurements of power and heat rate. Measurements of power and heat rate were made at the condition when the maximum allowable exhaust gas temperature was reached when system conditions allowed running at this high power setting. Even when maximum power could not be reached, valuable comparisons could still be made at power levels typical of those at which these units usually operate.

It has also been shown how compressor performance derived from field tests can be used to predict the compressor power required when operating parameters such as gas mole weight and compressor rotating speed are changed. Used in conjunction with calculations of the drop off in driver power that occurs with an increase in ambient temperature, accurate predictions can be made of limitations on production that may occur, especially during the summer. The improved information about compression capability based on the actual current state of the equipment can be used to assist in making operational decisions that can result in benefits such as optimal use of assets and avoidance of undesirable flaring.

It has also been shown how fluctuations in gas composition introduce an additional source of uncertainty when making performance measurements in the field. This, however, did not appear to be a significant detriment to the value of the measurements obtained in the present case study. However, in some instances it may be desirable to reduce this source of uncertainty. An example of such an instance would be one in which mole weight fluctuations are expected to be particularly large. In such cases, more frequent sampling of gas properties, perhaps making use of a portable on-line gas chromatograph would be recommended.

NOMENCLATURE

$\dot{E}_{fuel}$	= fuel energy flux to gas turbine
HR	= gas turbine heat rate
LHV	= fuel lower heating value
MW	= molecular weight (fuel or process gas)
p	= pressure
P	= power (compressor or gas turbine)
$\overline{P}$	= compressor power coefficient
R_2	= compressor impeller radius
$\Re$	= Universal Gas Constant
T	= temperature
U_2	= compressor impeller tip speed
Z	= gas compressibility factor
$\Delta H_{isentropic}$	= isentropic enthalpy change
ΔH_{actual}	= actual enthalpy change
φ	= compressor flow coefficient
η	= compressor efficiency
ψ	= compressor head coefficient
ρ	= gas density (process gas or fuel)

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