MEASURE POWER BY THE SIMPLEST METHOD AND USE FOR ONSTREAM COMPUTER MONITORING

by

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ABSTRACT

Determining brake horsepower by means other than calculations of compressor performance for compression of hydrocarbon mixtures is usually the more accurate procedure for field testing. Various means of determining this power are discussed including torquemeters, electric motors and steam turbine drivers calculating procedures. The complexity of the compressor calculations which leads to inaccurate results is discussed. This includes the problems of obtaining accurate test data along with the difficulty of obtaining accurate gas constant data. Lastly, on-stream continuous performance monitoring by computer is discussed. The principle benefit of computer performance monitoring is in determining the condition of the machine rather than obtaining high accuracy test data to verify guarantees. Indications of compressor fouling and possible internal mechanical problems can be detected. Performance calculation procedures along with the algorithms used in the computer are also defined.

INTRODUCTION

Accurate compressor field testing at most existing petrochemical plants is very difficult to achieve because of many factors, the major ones which are listed below:

- Short runs of straight pipe for pressure and temperature taps.
- Short meter straight pipe length runs for good flow measurements.
- Insufficiently precise and calibrated instruments.
- Achieving good gas samples and obtaining accurate gas analysis.
- Questionable gas constants for hydrocarbon gas components.
- Inaccuracies associated with computing methods for non-ideal gas behavior during compression.
- Mollier charts not available for gas mixtures.

In spite of these difficulties, the accurate determination of compressor performance is critical to optimizing energy costs in refineries and chemical plants. A typical 30,000 horsepower steam turbine driven centrifugal compressor will cost approximately $300,000 per year for additional steam costs for each one percent of efficiency loss. These numbers are based on approximate costs for 1500 psi steam. Therefore, it is very important to develop approaches to check the thermodynamic performance of machines in the field.

DESIGN FIELD PIPING ARRANGEMENT FOR PERFORMANCE TESTING

Several of the problems above can be eliminated or significantly improved by giving special attention to design layout for new installations. In most cases, inlet and discharge compressor and turbine piping can be arranged with pressure and temperature taps provided as required by the ASME power test codes with gas sample taps also provided at correct locations. Proper lengths of straight runs of pipe for flow measurements are usually more difficult to engineer into the new plant with pressures of plant investment, cost and construction schedules, but if code requirements cannot be met completely, a compromise can usually be provided. In addition, for new plants, provide some of the newer state of the art items such as torquemeters on steam turbine and gas turbine drivers, and supply impeller stage taps on centrifugal compressor casings, especially for dirty gas service and sidestream machines. For most installations already in operation, the instrumentation locations will not be as ideal as a specially designed piping setup and therefore may cause some additional inaccuracies in the test results. It is important to provide good instrumentation so as to minimize these errors.

NO ACCURATE WAY TO CALCULATE FOR REAL GAS MIXTURES

The accurate way to calculate isentropic work for compressors for all gases employs the enthalpy method:

$$W_s = (H_2' - H_1)/J$$

where W_s is isentropic work

H_2' is enthalpy, BTU per lb, at the discharge pressure and entropy corresponding to inlet conditions

H_1 is enthalpy, BTU per lb, at inlet conditions

J is the mechanical equivalent of head, ft-lb per BTU, 778.16.

Compressor performance is usually always expressed in polytropic parameters rather than the isentropic. For perfect gases, polytropic work for compressors can be accurately calculated from measured pressures and temperatures using the so called "n" method, with "n" defined as the polytropic exponent. For real gases for which Mollier charts are available, the polytropic work, W_p, can be accurately calculated using the same "n" method as for perfect gases, but multiplying the result by a "f" factor (polytropic work factor). This factor corrects for a varying "n" during the compression and is calculated from Mollier chart gas data.

For real gases which are multicomponent hydrocarbon mixtures, for which Mollier charts are not available, the "f" factor cannot be evaluated. This factor is then assumed to be unity which introduces inaccuracies into the calculation. Fortunately, this factor is usually small, in the range of a few percentage points.

Calculating power from the compressor test data without Mollier charts for a real gas provides the same type of inaccuracies. Enthalpy must be calculated from the mean specific heat values of the gas mixture and as stated in the ASME PTC-10, it can be extremely in error.

OBTAIN SHAFT POWER FROM DRIVER DATA

Because of the inaccuracies associated with the compressor calculations method, gas samples and gas analysis, alternate methods of performance evaluation are desirable. For a non-condensing steam turbine, the shaft power can be calculated with very good accuracy. If the pressure, temperatures and flows are measured accurately, the calculated power for large turbines should be within one-half of one percent accuracy. A review of the calculation procedure follows:

- Measure T_1, P_1, T_2, P_2, speed and steam weight flow. From Mollier diagram or steam tables determine:

  $H_1 - H_2$ Enthalpy Delta

  $H_1 - H_2'$ Enthalpy Isentropic Delta

- Wheel Efficiency, Isentropic =

$$\frac{H_1 - H_2}{H_1 - H_2'}$$

- Shaft Horsepower =

$$\frac{(H_1 - H_2)\ \text{WT Flow}}{2544} - \text{Mechanical losses}$$

where WT Flow is in lbs/min.

Unfortunately for a condensing turbine, there is no easy way of calculating efficiency or horsepower due to the difficulties associated with determining the amount of moisture in the exhaust. A torquemeter is the ideal solution in this case. If no other way exists, use the vendor's performance and correction curves, starting with the accurate test data. Our experience would indicate that the majority of impulse turbine vendor performance curves are fairly accurate, within one or two percent, even after many years of operation when no fouling is indicated. Wheel chamber pressure should be kept to provide a means of indicating internal turbine fouling. For an extraction-condensing steam turbine, the same power calculation difficulty exists. However, the performance of the front end should be checked periodically as outlined above for noncondensing steam turbines. These checks can usually spot problems either within the turbine or the driven equipment. The vendor curves can be used with the same accuracy as stated above.

For motor drivers the measurement of shaft power is simpler and more accurate than steam turbine calculations. Volts and amperes or watts can be measured very accurately, and with the motor efficiency curve, the output can be determined within half of one percent. If a speed increasing or decreasing gear is used, the shaft power after the gear loss is subtracted will have an accuracy of better than one percent. Most large motors run about 96% efficient while gears are close to 98%.

CONTINUOUS ONSTREAM MONITORING FOR MACHINE CONDITION

The need to maintain the best efficiency to minimize energy costs provides the incentive for an early warning monitoring system to indicate deviations from the "as new" thermodynamic performance condition. This on-stream monitoring does not require the high accuracy measurements necessary to evaluate guarantee points, such as vendor shop tests or field acceptance tests. However, the important parameter is any change from normal which could indicate fouling or mechanical problems such as eroding internal clearances, impeller wear, leakage bypasses and blockages.

A continuous onstream computer monitoring system where torquemeters are installed is described below.

For each compression stage, fix molecular weight MW; compressibility at inlet and discharge, Z_1 and Z_2; mean specific heat, C_p and gas "k" value from latest gas sample analysis. The MW can be calculated from the density meters; however, confidence must be established in the accuracy of the installed meter before using for performance calculations. The computer constants listed above are updated when new samples are taken.

Measure P_1, T_1, P_2, T_2, speed and flow.

Calculate

$$n = \frac{\ln\left(P_2/P_1\right)}{\ln\left(\dfrac{T_1 Z_1 P_2}{T_2 Z_2 P_1}\right)} \qquad \text{where} \quad P = \text{psia},\ T = {}^\circ\,\text{Rankine},\ \text{Head} = \text{ft-lbf/lbm}$$

Calculate Head $= W_p = R \times T_1 \times b \times Z$ avg.

$$= \frac{1545}{MW} \times Z\,\text{avg.} \times T_1 \times \frac{n}{n-1}\left[\left(P_2/P_1\right)^{\frac{n-1}{n}} - 1\right]$$

Calculate theoretical polytropic power =

$$P_p = \frac{\text{Head} \times \text{WT Flow}}{33000}$$

Calculate shaft power, SHP, from torquemeter when available, or perform steam turbine performance calculation from turbine vendor curves.

Calculate power from compressor data:

$[\text{WT Flow} \times C_p \times \Delta T/42.41]$ + Mech Losses

Calculate Efficiency

$$\eta_{poly} = \frac{\text{Theoretical HP}}{\text{SHP}} = \frac{P_p}{\text{SHP}}$$

For the example indicated, data are continuously monitored by the computer except for fixed constants previously indicated. The computer can now continuously update machine performance with display and alarms for operator action.

RESULTS SHOW COMPRESSION CALCULATIONS DO NOT CHECK

Of the various ways to evaluate performance, the compressor calculations are the most difficult to make and seldom check the other methods.

The torquemeter and the turbine vendor curves (extraction-condensing turbines) check very well and are consistent; however, the compressor power calculations vary considerably. A good comparison factor is shaft power divided by 1000 pounds per hour. This parameter is a measure of efficiency and changes will indicate performance deficiencies. The computer also can use this factor to compare the various methods. If the torquemeter is used as the base, the ratio of the other methods to the torquemeter provides a monitor of the methodology.

If the calculations indicate signs of fouling, the individual impeller casing taps can be used to determine where the fouling is occurring and the flushing may be able to be localized to the fouled locations.

CONCLUSIONS

A means of continuous thermodynamic performance monitoring is highly desirable as a way of checking the health of centrifugal compressors and their drivers. The calculation method does not have to be of super accurate ASME code quality, but it should be consistent so that changes in performance can be detected.

BIBLIOGRAPHY

1. Schultz, J. M., "The Polytropic Analysis of Centrifugal Compressors," ASME Paper 60-WA-296.
2. ASME — Power Test Code, Compressors and Exhausters, PTC-10-1965.
3. *Compressed Air and Gas Data* (Ingersoll-Rand Company).