FIELD TESTING OF CENTRIFUGAL COMPRESSORS

by

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ABSTRACT

Field testing of compressors may be carried out to monitor for performance shortfall in normal operation or to verify performance following an overhaul or revamp. Testing requires careful measurement of pressures, temperatures, flows, gas composition, speed and shaft power. Testing is discussed with reference to the procedures given in ASME Power Test Code 10. Some comparison is made of shop testing versus field testing and the latter is seen as complementary to the former. In most tests, shop as well as field, shaft power is determined by the heat balance method. This method is reviewed briefly in its application to single and multiple inlet compressors. An approach for calculating overall efficiency for multiple inlet machines, where internal temperature measurements are not available, is suggested. A few ways of plotting test results for ready comparison are set out. Finally, some shop and field test results are compared.

INTRODUCTION

With energy costs increasing as a percentage of total operating costs, it is important to maintain compressor performance at the highest practicable level. Performance falloff is usually caused by some degradation in the flow path. Also, performance change could be due to a change in process conditions. Field testing will show the performance level and possibly indicate the reason for performance shortfall.

FACTORS CAUSING PERFORMANCE FALLOFF

Compressor head and efficiency may be reduced by flow path deterioration brought about by one of the following:

- Fouling.
- Erosion.
- Corrosion.
- Foreign object damage.
- Enlarged clearances.
- Leakage around or through diaphragms.
- Leakage through recycle valve.

Labyrinth clearances can be opened up by rubs caused by unbalance or surge-induced vibration. Physically small machines with percentage-wise high clearances are most susceptible to this.

CONDUCT OF FIELD PERFORMANCE TESTS

Basic measurements are pressures and temperatures at the inlet and discharge flanges, mass flow, gas composition, compressor speed and shaft power input. The first purpose of the test is to determine what discharge pressure is produced and what shaft power is required for a particular gas for a given flow rate, suction pressure, temperature and speed.

In general, the testing should follow the precepts of ASME PTC 10-1974 [1]. Plant piping should have been laid out to accommodate flow meter runs and to meet placement requirements for pressure and temperature measurement stations. Reference [2] contains cogent advice on instrumentation and sampling.

A test will usually be run at existing process conditions. However, it is advantageous if the test can be run entirely on recycle which will allow running through the full volume range from near-stonewall to surge. The flow rate is regulated by the antisurge bypass valve which, of course, has to be large enough for maximum flow. Another advantage of testing with a closed recycle loop is that any condensate initially in the loop will condense out in the cooler at the discharge pressure and can be drained off to leave only dry gas.

In general, gas samples should be taken near the discharge if there is a possibility of trace condensate being present; only the vapor phase will exist at the discharge. Using the discharge gas composition, phase equilibrium calculations will show whether a liquid phase exists at the suction. Presence of liquid means that certain efficiency calculation procedures are inapplicable.

COMPUTATIONS

Pressures, temperatures, mass flow, gas composition and speed may be said to be direct measurements. Shaft horsepower preferably should be found by direct measurement. If the drive is by an electric motor, motor input measurement less motor losses ought to give a fairly accurate shaft horsepower (within a few percent). Torquemeters have been used in a few field installations; one manufacturer claims ± 1.5 percent for his horsepower readout [3]. However, no compressor manufacturer known uses a torquemeter for shop performance testing. In practice, shaft power is usually found by a heat balance, even in shop tests [4]. If the shaft power could be found by other means, the heat balance calculations would still be done as a check on the overall consistency of the data. The heat balance method is given in PTC 10, but is discussed briefly here.

Application of the First Law to the system shown in Figure 1 gives:

$$SHP = \frac{(W_i - W_{11})(h_d - h_i) + Q_R + Q_M}{2545} \qquad (1)$$

where SHP = shaft horsepower
W_i = inlet flange mass flow rate, lb/hr
W_{11} = inlet end seal leak to atmosphere, lb/hr
h_d = enthalpy of gas at discharge flange, btu/lb
h_i = enthalpy of gas at inlet flange, btu/lb
Q_R = heat loss from case outer surface area, btu/hr
Q_M = bearing and seal mechanical loss, btu/hr.

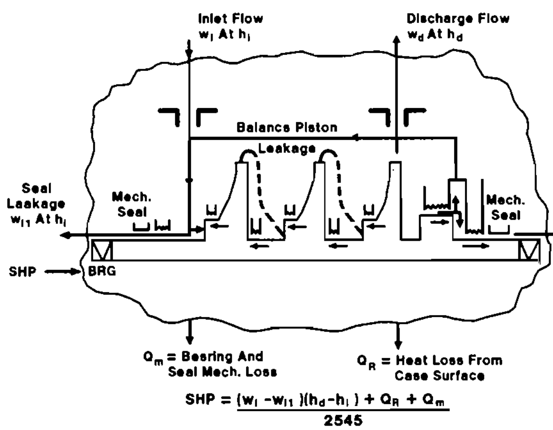

Figure 1. Heat Balance Diagram.

Inlet end seal leak to atmosphere, W_{11}, will normally be a small fraction of one percent and a theoretical calculation is usually acceptable. Q_R can be estimated from the case average surface temperature and area, and is usually negligible. Q_M is calculated from the quantity and temperature rise of the lube and seal oil. (Q_R and Q_M are smaller percentagewise in field tests than in most shop tests.) The flange enthalpies are calculated from ideal gas relationships or from equations of state. For most hydrocarbon gas mixtures, it is essential to use the latter.

The polytropic efficiency, which is the ratio of the polytropic work between the initial and final state points to the actual work (h_d - h_i) done on the gas, may be calculated for comparison to the manufacturer's value. From the user's point of view, a more meaningful overall efficiency would be the isentropic power divided by the shaft power, or:

$$\eta_{so} = \frac{(W_d)(h_d' - h_i)}{(W_i - W_{11})(h_d - h_i) + Q_R + Q_M} \qquad (2)$$

where η_{so} = overall isentropic efficiency
W_d = discharge flange mass flow rate, lb/hr
h'_d = enthalpy at discharge pressure at an entropy corresponding to inlet flange pressure and temperature, btu/lb.

For a compressor with multiple inlets, the heat balance method, of course, still applies. With two inlet flows, W_i and W_{ii}, discharge flow W_d, and shaft seal external leakages W_{11} and W_{12}, equation (1) becomes

$$SHP = \frac{(W_d + W_{12})(h_d) + (W_{11})(h_i) - (W_i h_i + W_{ii} h_{ii}) + Q_R + Q_M}{2545} \qquad (3)$$

The individual efficiency of each section cannot be found unless the first section internal discharge temperature is known, and this is normally not measured in the field. However, an overall isentropic efficiency can be calculated by summing the isentropic powers for the two sections (i.e., the sum of the products of mass flow times isentropic rise). The inlet enthalpy for the total flow entering the second section is that obtained by mixing the added flange flow at its measured temperature/enthalpy with the flow leaving the first section at an enthalpy based on isentropic compression from the first section inlet flange conditions (Figure 2).

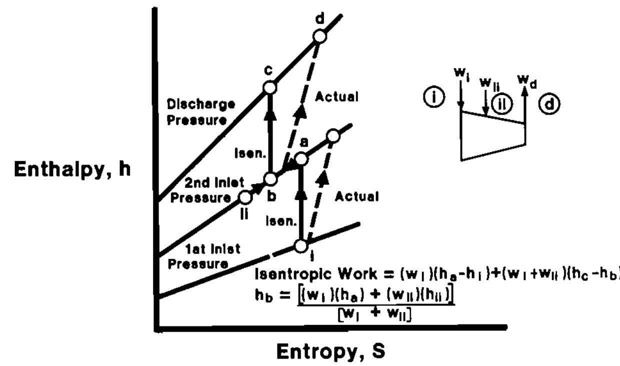

Figure 2. Isentropic Work Calculation for a Compressor with Two Inlets.

PTC 10 states that inlet gas temperature shall have a minimum of 20°F superheat. Many compressors have suction conditions at the dew point, i.e., zero superheat. Condensate does not of itself invalidate the heat balance method provided the enthalpy is defined. Sampling a wet stream is the problem. A solution is to calculate the vapor/liquid composition at inlet conditions using the composition of a sample taken off the discharge.

APPRAISAL OF RESULTS

Having calculated the performance at the conditions of the test, the results must be expressed in a form which permits comparison with the performance predicted or measured at some other conditions. A common form of the performance curve is that given in Figure 3. This plot is based on constant values of inlet pressure, inlet temperature, molecular weight and isentropic volume exponent. In a test, one or more of these will invariably have a different value, so that direct comparison to Figure 3 cannot be made. However, Figure 3 can be used without a change in scales by expressing inlet pressure, inlet temperature, compressibility factor and molecular weight as dimensionless ratios. A further modification is to convert the flow scale from a volume to a mass basis, a unit of greater utility to the process engineer. Figure 4 shows such a plot on a mass flow basis, with the volume flow parameter also indicated. Test data expressed in parametric form can be plotted directly on this curve. It will be valid for a fairly wide range of molecular weights, in excess of plus or minus ten percent of the reference value.

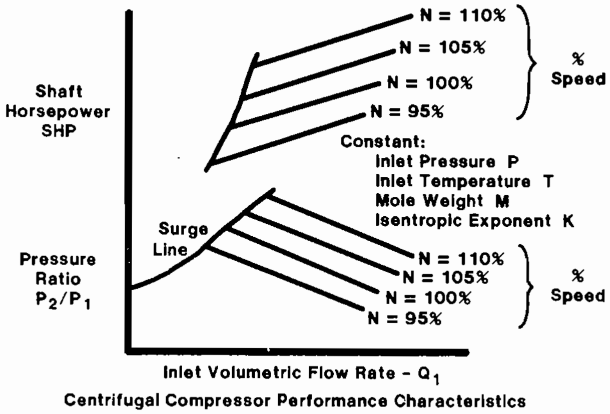

Figure 3. Centrifugal Compressor Performance Characteristics.

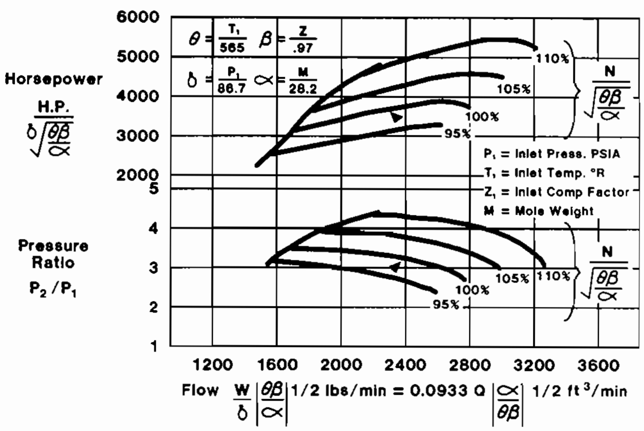

Figure 4. Performance Curves in Parametric Form.

Figure 5 shows the polytropic efficiency and the polytropic work coefficient, both plotted as a function of the load coefficient. There are three sets of curves. The higher ones were obtained in a shop test with low pressure nitrogen. The two lower ones were obtained in the field with high pressure natural gas on a second machine of design identical to the shop-tested unit. The lower performance of the field tested unit could be due to enlarged clearances, fouling, etc. Another possibility is that the field tested unit had a lower performance from the outset because of deviations in some critical design dimensions.

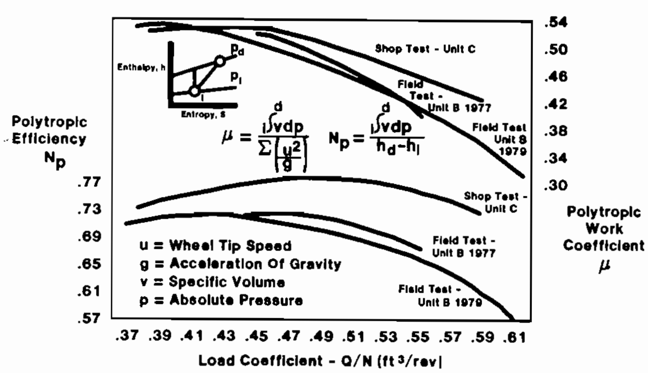

Figure 5. Comparison of Shop and Field Test Efficiencies and Work Coefficients.

Figure 6 shows plots of the polytropic head versus the compressor inlet volume flow for a given speed. The predicted head is compared to the actual head for three different machines built to the same drawings. One line is based on a shop test run on low pressure nitrogen while the other two are from field data on high pressure natural gas. In general, both shop and field tests show a faster-than-predicted drop off of head with increasing flow. The agreement between the shop test on Unit C and the field data on Unit A is quite good. Another point of interest is the indicated reduction in stability as tested compared to the predicted. The shop test surge point flow lies somewhat above the prediction, while the field test surge points occur at an even higher flow.

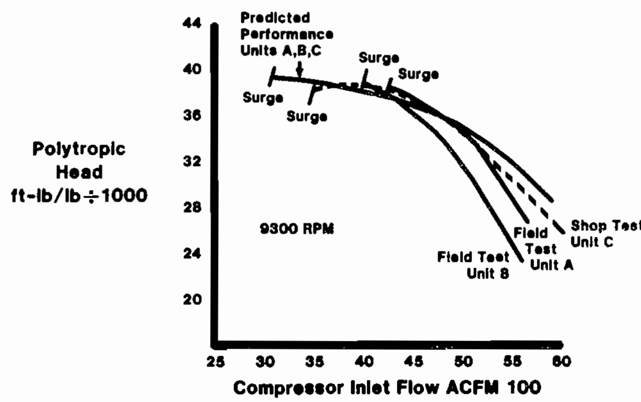

Figure 6. Comparison of Test and Predicted Heads.

FIELD VERSUS SHOP PERFORMANCE TESTING

It seems useful to discuss briefly the relation between shop and field testing. Shop performance tests are carried out at the purchaser's option to ensure compliance with the guarantee. Such tests are almost always PTC 10 Class II or Class III type, done with a substitute gas at operating conditions different from design [5]. The fundamental design parameters of suction-to-discharge-volume ratio, inlet volume-to-speed ratio, Mach number and Reynolds number must all be within certain percentages of the design values. The ability of such tests to predict design performance depends on the accuracy of the test, design gas properties, the method used to convert the test results, and the extent of deviation of test from design parameters.

A field performance guarantee test would be a Class I test, i.e. one which is made with design gas at close to design conditions. PTC 10 states "Class I tests should be used wherever feasible" and also "The accuracy of comparison between test and specified performance will be maximum." In practice, Class I tests are seldom done because considerations of safety, cost and power requirements do not allow them to be done in the shop. Shop tests offer several advantages over field tests, the foremost of which is that any performance shortfall found can be corrected before leaving the factory; it is by no means unusual for such tests to reveal shortfalls. Other advantages of shop testing are that the machine is known to be clean and in mint condition, data acquisition facilities are better, gas properties may be more accurate, and testing is free from processrelated constraints. If performance is critical and the alternative of building in large margins would unduly penalize efficiency, a shop peformance test should be done. A further consideration is that it is difficult to pursue most field performance problems with a manufacturer when there has been no shop performance test. The shop test provides a standard of comparison. Field performance testing is not a substitute for shop testing, but complementary to it.

CONCLUSIONS

Centrifugal compressor performance is found to fall off some with time in service in most installations. Careful testing allows the quantification of such fall off and plans for remedial action. It is feasible to make accurate tests in the field, if planned in the original installation. Field testing is not a substitute for shop testing but complementary to the same.

REFERENCES

1. *ASME (ANSI) Power Test Code, PTC 10 — 1974 for Compressors and Exhausters*, (New York: American Society of Mechanical Engineers).
2. Lock, J. A., "Techniques for More Accurate Centrifugal Compressor Performance Evaluation", ASME Petroleum Mechanical Engineering Conference, Dallas, Texas, September 16, 1974.
3. Cummings, J. M., "User's Experience with Instrumented Couplings for Continuous Measurement of Horsepower and Alignment on Large Turbo-Machinery Trains", ASME Paper 80-C2/DET-52.
4. Purvis, J. T., "Industrial Gas Turbine Performance Measurement", *Proceedings of the Third Turbomachinery Symposium*, Texas A&M University, College Station, Texas, 1974.
5. Sayyed, S., "Compressor Shop Tests: Do They Ensure Reliability?", *Hydrocarbon Processing*, April 1978.