AXIAL AIR COMPRESSORS—MAINTAINING PEAK EFFICIENCY

by

M. Theodore Gresh

Service Engineer

Marc J. Sassos

Service Engineer

and

Andrew Watson

Senior Product Engineer

Elliott Company

Jeannette, Pennsylvania

M. Theodore Gresh has been with Elliott Company in Jeannette, Pennsylvania, since 1975, initially working on the mechanical and aerodynamic design of centrifugal compressors. He is currently a member of Elliott's service organization and travels to field installations to investigate warranty claims, conduct field performance tests, and troubleshoot various user problems. He is a graduate of the University of Pittsburgh with a B.S. degree in Aerospace Engineering, and he is a registered Professional Engineer in the State of Pennsylvania. Mr. Gresh has published several documents regarding the performance of compressors.

Marc Sassos is a Service Engineer for Elliott Company in Jeannette, Pennsylvania. His responsibilities include technical support for Elliott's field service facilities worldwide, in addition to field supervision during installation, commissioning, maintenance, and troubleshooting of centrifugal and axial compressors and their associated systems.

Mr. Sassos joined Elliott after receiving a B.S.M.E.T. from the University of Pittsburgh. He has held various staff engineering positions during the past 15 years, which have included Test Engineer in the Donora plant and Field Service Engineer in the Houston field office, before joining the Technical Services Department in Jeannette in 1980. He was also a coauthor for a paper on lubrication systems for the 13th Turbomachinery Symposium.

Andy Watson is Senior Design Engineer for axial and single stage centrifugal compressors with Elliott Company, Jeannette, Pennsylvania. He is responsible for all technical aspects of these compressors, including aerodynamic performance, rotordynamics, and mechanical design. In addition, he assists Sales/Marketing in pre-contract selection and technical data, works with Manufacturing and Test Engineering on shop order activities, and consults with the Elliott Service organization on technical matters concerning units in the field. Mr. Watson has been with Elliott Company since 1981, and prior to that he had nine years experience in the design and development of turbomachinery. He has a BSME from Pennsylvania State University and an MSME from Rensselaer Polytechnic Institute. He is a member of ASME.

ABSTRACT

Axial air compressors have seen increased service in various applications over the years, mainly because of the axial compressor's high operating efficiency, which can be 85 to 90 plus percent. The design of the aerodynamic components, however, makes the machine very sensitive to fouling. This can reduce operating efficiency by 10 points or more, reducing the compressor output and adding to the plant utility load. It is advantageous to monitor compressor performance on a regular basis, and to clean the machine and/or perform other maintenance when calculations indicate the need is economically justified.

INTRODUCTION

The operating characteristics of the axial compressor are dependent on the lift and drag coefficients of the cascade of airfoil blades. Because only a low pressure rise is available from a single axial stage, high speeds are utilized to maximize its effectiveness. Mach numbers are high, so any change in the blade profile due to dirt buildup or fouling can decrease the efficiency and flow capacity.

Surge is especially damaging to an axial compressor because the cyclic reversal of mass flow can cause high bending stresses on the thin blades. These reverse bending stresses could be high enough to fatigue the blades and eventually lead to blade failure. In addition to blade stress problems, surge can also lead to extremely high casing temperatures. During surge, discharge gas is forced back through the compressor, then recompressed. The compressor recompresses the heated gas, and temperatures rise quickly, causing the thin blades to grow more rapidly than the casing, eventually resulting in a rub. As a minimum, the rub will increase tip clearances, resulting in reduced efficiency. The extreme temperatures generated during surge can also cause the casing to warp, further degrading the performance.

At reduced speeds (70 percent of design or lower), rotating stall is more apt to occur. However, since rotating stall is only localized and temporary, the overall system is stable, and steady through flow occurs, keeping the blades cool. Since energy levels are lower, blade fatigue is not a serious concern, unless the speed

coincides with a blade resonant frequency. Even so, it is wise to minimize operation at this point and ensure that the blowoff valve is open when operating below minimum design speed.

Adjustable vanes can be used to extend the useful operating range of any compressor (Figure 1). Axial compressors are very sensitive to the angle of attack and have a very short operating range. To compensate for this, axial compressors require variable stator vanes or variable speed to obtain a practical operating envelope. Adjustable vanes are typically only needed on the first several stages of axial compressors to obtain the desired operating range. Adjustable vanes can compensate for the loss in flow due to blade fouling or increased tip clearance; however, the limit of adjustment will eventually be reached, and the deficiency will require correction to continue operating at desired conditions [1].
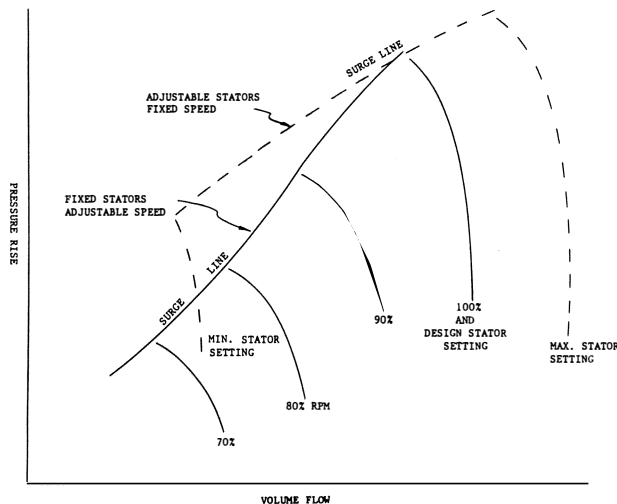

Figure 1. Axial Compressor Performance Map Showing Effect of Adjustable Stator Vanes. Adjustable stator vanes are typically used on the first several stages of axial compressors to extend the useful operating range [1].

FIELD PERFORMANCE TESTING

Field performance testing of axial air compressors is very straightforward and can be easily conducted once the proper equipment is in place. Before attempting a performance test, review the following check list and be certain all the required data can be obtained. As a reference, see the instruction book for design conditions.

- Pressure and temperature at each flange
- Stator vane setting
- Mass flowrate
- Equipment speed
- Driver power
- Compressor and driver mechanical losses

Field performance test procedures should be in accordance with ASME PTC10-1974 Compressors and Exhausters within practical limits. Note that the method outlined here is for routine performance evaluation only, and is not necessarily sufficient for an OEM acceptance test. If an acceptance test is to be performed, details should first be worked out with the manufacturer.

Offline testing is the safest and most practical method of verifying compressor performance and inlet nozzle flow calibration, or confirming vendor performance predictions. This testing can be accomplished by isolating the compressor from the process during installation/commissioning, or during a scheduled unit overhaul (Figure 2).
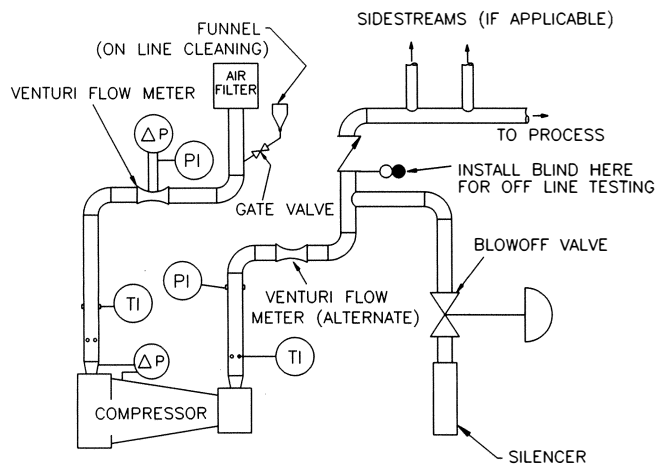

Figure 2. Typical Performance Test Setup Showing Instrumentation Requirements. Preferred location for the flow element is the suction line. Check for proper straight run of piping for compressor inlet and instrument locations as indicated. Note funnel port for online cleaning.

This differs from online testing in that compressor operating points can be varied freely, with no concerns for process upsets. Offline testing allows baseline data to be easily established for future reference.

The compressor can be isolated by closing a discharge block valve (if available) or the slide valves if the process piping configuration permits. Alternately, a blind can be installed at some convenient location in the compressor discharge piping, possibly at the air preheater (if available). Note that the blind must be located in the piping such that all compressor flow will be accurately measured. Note also that any isolation device must be located downstream of the compressor blowoff (antisurge) valve (Figure 2). The process check valve should not be used for isolation as it will leak, and the compressor discharge pressure could damage the check valve assembly during a prolonged test.

Plant piping and vessel configurations may require that the compressor be completely isolated, and that flow be trimmed by using the blowoff valve—this is acceptable. The above comments regarding blinding and isolating the process apply. Actually, utilizing the blowoff valve affords a great degree of control not possible when the machine is online and the process onspec. This is one advantage of testing the machine offline.

This is also a good opportunity to assess the integrity of the blowoff valve. The blowoff valve should not leak when in the full closed position. If leakage occurs, it suggests a possible problem with the valve seat or disc. Since this valve will normally be operating in the full closed position, any leakage represents wasted hp.

Online test data will provide test points, one point at a time, at different locations on the compressor performance curve. The offline test can provide the base line information necessary for trending of the online data.

Instrumentation

Temperature and pressure accuracy are crucial parts of performance testing. Very small errors in these readings can result in significant efficiency or power calculation errors.

General instrumentation requirements are shown in Figure 2. All temperature and pressure indicators must be dual, i.e., a minimum of two independent instruments per location.

Ideally, such as under development testing conditions, the static pressure tap hole should be very small (approximately 1/5) compared to the boundary layer thickness [2]. But on a more practical note, the static pressure connection should have a pressure tap hole 0.125 in in diameter, but no greater than 0.5 in, deburred and smooth on its inside edge with a sharp corner. A small hole may collect dirt or condensate, but it will provide better accuracy. Note that the hole must be deburred but have a sharp corner on the measurement side. Take care that the deburring procedure does not round the edges, as this will give erroneous readings.

If possible, all pressures below 20 psig should be measured with a vertical-type fluid manometer of single or double-leg design. Manometer fluids must be chemically stable when in contact with air, and the specific gravity should be measured before and after the test. If safety regulations do not permit the use of manometers, calibrated pressure transducers of the proper range should be used.

All pressures above 20 psig at compressor flanges and flowmeter devices should be measured using pressure transducers or quality pressure gages, 6.0 in or larger in diameter, with a 0.25 percent sensitivity, and a maximum error of 0.5 percent full scale.

Pressure readings during testing should be at midrange of the instrument or greater. Note that instrument error is percent of full scale of the device being used. Do not use a pressure transducer designed for 1,500 psi for a 10 psi measurement. The error of the reading will be greater than the pressure being measured. Pressure gages should be mounted on a vibration-free local panel, connected with pressure lines of at least 1/2 in ID tubing; lines must continually slope down toward the unit to automatically drain any condensate. Block vent valves should be mounted at the gages to facilitate their inplace calibration. Calibration using a certified dead weight tester is preferred.

Temperatures should be measured using a thermocouple or RTD system having a sensitivity and readability of 0.5°F and an accuracy within 1.0°F. Care should be taken to avoid intermediate T-C junctions at terminals and switch boxes with a thermocouple system. Glass-stem thermometers are unacceptable for safety reasons. The temperature-sensing portion of the probe must be immersed into the flow to a depth of 1/3 to 1/2 the pipe diameter. The temperature-sensing elements should be in intimate thermal contact if using wells, utilizing a suitable heat transfer filling medium, such as graphite paste. Stem conduction errors can be further minimized by wrapping the stem and well with fiberglass or wool insulation.

Speed should be determined utilizing two independent systems, one being the compressor key-phaser with calibrated digital readout with 0.25 percent or better system accuracy.

Pressure taps should be spaced 90 degrees apart. On horizontal runs of pipe, pressure taps must be in the upper half of the pipe only.

Flowrates derived from the process flow indicator should be checked by direct computation of mass flowrates through the metering device. For this reason, metering device upstream temperature, upstream pressure, and differential pressure must also be recorded.

In order to minimize pressure losses, it is recommended that a Venturi be utilized in the inlet piping/duct. The Venturi should be the Herschel type, designed per ASME PTC 19.5 and/or ISO 5167.

Location of the Venturi in the inlet duct also allows true measurement of the compressor inlet flow. Although a flowmeter in the discharge line accurately measures the discharge flow, it does not measure the flow that leaks out of the shaft end seals or thrust vent. Also, placement of a discharge Venturi must be ahead of any sidestreams or blowoff connections (Figure 2), or gross performance calculation errors will result.

Instrument Calibration

In general, all instruments used for the measurement of temperature, pressure, flow, and speed should be calibrated by comparison with appropriate standards before the test. General recommendations for calibration procedures are outlined below. A comprehensive log book should be maintained for all calibrations. Pressure transducer and pressure gage calibration should state actual dead weight pressure and indicated value.

Pressure gages and transducers should be check calibrated against dead weight standards throughout the range. Calibration using both increasing and decreasing pressure signals should be done to check for hysteresis. Any instruments not within 0.5 percent error of full scale should not be used. Pressure gage needles should not be changed or adjusted. The gages must have a readable sensitivity to 0.25 percent.

Check on the accuracy of the thermocouple and pressure transducer system (lead wires, reference junctions, readout devices) for each device. One method of accomplishing this is to read voltage output of the thermocouple locally, and then compare this to the remote reading of thermocouple output.

It is also recommended that the accuracy of the thermocouple itself be checked by subjecting it to varying temperatures and comparing its output to a reference standard. The thermocouple should be checked throughout its operating temperature range. The thermocouple system should have a readable sensitivity to 0.5°F and an accuracy within 1.0°F.

Calibration of the flowmeter differential pressure transmitter can be verified by impressing a known differential pressure at the instrument and measuring the output. Finally, an overall system check can be made by impressing a differential pressure across the transmitter and reading the final control room output. The flow element should be removed, and its dimensions should be checked, recorded, and compared to design criteria.

The tolerance for the measurement of compressor speed should not exceed 0.25 percent. Use of two independent instruments, one to provide a check on the other, is recommended. Electronic tachometers must be checked.

Stator Vanes

Adjustable stator vanes directly affect the compressor overall head and efficiency. They can contribute significant error and confusion to performance audits if not properly adjusted and calibrated prior to data collection. This may require vendor assembly and adjustment information. The linkage assemblies also must be inspected, and mechanical integrity established, to assure proper tracking of the vane assemblies.

Axial compressors typically utilize adjustable stator (stationary) blades in the first several stages to facilitate regulation of the unit throughput to meet process requirements. This feature also conserves hp by reducing compressor load requirements, and extends the operating range of the machine.

Therefore, stator vane position is very important when conducting a field performance audit or when merely trending compressor performance. Compressor manufacturers provide performance curves based upon stator vane position (Figure 1). Stator vane position must be determined accurately in order to compare field performance data with the predicted performance curves.

Stator vane position is typically designated in degrees. This position is based upon the neutral position of the first vane (prewhirl) with respect to some reference (usually the rotor centerline). Thus, zero degrees stator blade position is actually the neutral relative position of the vane, and is typically near midtravel of the variable stator assembly. Vane movement is then either

greater than or less than the neutral position. As shown in Figure 3, a stator blade position of -20 degrees means that the blade has moved towards the reference centerline from the neutral position a total of 20 degrees, and is negative because the relative angular displacement decreases. Note as well that the vane position now results in an increase in unit throughput due to a change in the blade angle-of-attack of the downstream blading. Conversely, a vane position of +30 suggests that the stator vane has moved away from the neutral position 30 degrees, with a resulting decrease in compressor flow.
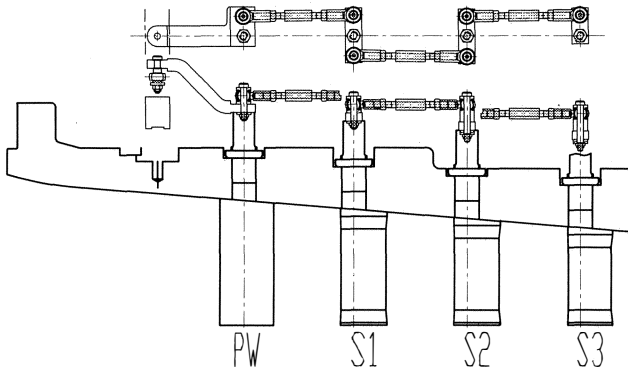

Figure 3. Stator Vane Mechanism. Inspect and adjust per vendor specifications prior to offline testing, and during every normal maintenance overhaul.

Prior to conducting a rigorous performance test, the variable stator vane control assembly must be inspected and calibrated. Worn components or improper calibration of the assembly can lead to errors in performance results.

It is essential that all linkages and control rings be free of wear. Loose linkages can contribute to control system instability, and to a lesser extent, performance problems. Any suspicious components should be replaced. The servo motor(s) should be inspected and overhauled, if necessary.

The control system should be calibrated per the manufacturer's recommendations and assembly drawings. First, verify that the stroke of the servo motor agrees with the stator vane scale; adjust as necessary. Next, the stator vane assembly must be calibrated. This usually involves clamping the assembly (using a special fixture) and setting the blade angles by adjusting the linkages or control rings. After this portion of the procedure is complete, the scale indicator should be repositioned so as to accurately measure, full open, full closed, and neutral positions. The indicator itself should allow measurement of the variable stator vane position within 1/32 in.

Data Collection

Test point readings should not be taken until such time as the compressor is shown to be in equilibrium. Equilibrium is defined as the condition in which the discharge temperature does not vary more than 1.0°F over a five minute period at constant inlet temperature.

Upon achieving equilibrium, three complete scans of data readings per data point are taken over a 20 to 30 minute period and averaged for calculations.

It is recommended that a minimum of three data points at full close, zero degrees, and full open guidevane settings be taken to establish the performance curve shape. The first point for each setting should be the surge point. The pressure ratio should then be reduced until there is no significant increase in flow with decreas-

ing pressure ratio; this will be the second data point. The third data point should be taken at the midpoint between the first and second point flows (Figure 4).
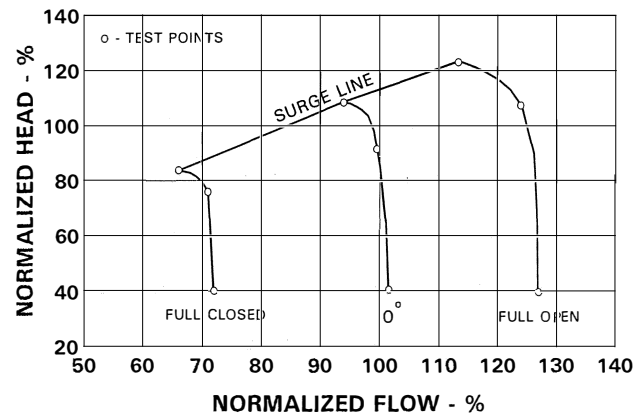

Figure 4. Data Collection, Offline Testing. Three data points should be obtained at full close, zero, and full open guidevane positions to establish the overall performance curve.

Calculation Procedures

Since air follows the perfect gas laws for the range of operation, equations are simple, and hand calculations are very accurate and reliable. Unless an acceptance test is being conducted, the effects of humidity (one to two percent) are negligible and can be ignored. The necessary equations, along with examples, are shown below.

A final check of overall test accuracy should be made by performing a power balance. Measured driver-delivered power minus gear (if applicable) power losses can be compared to calculated compressor-absorbed power. The overall accuracy of the test is no better than the accuracy indicated by this difference.

Head

$$H_{ad} = RT_1 \left(\frac{k}{k-1} \right) \left[\left(\frac{P_2}{P_1} \right)^{(k-1)/k} - 1 \right] \qquad (1)$$

where

H_{ad} = Adiabatic Head, Feet

k = c_p/c_v

Efficiency

$$\eta_{ad} = \frac{T_1 [(P_2/P_1)^{(k-1)/k} - 1]}{T_2 - T_1} \qquad (2)$$

Flow

Mass flow is best determined by using the process flowmeter. The flowrate is adjusted to account for variations from meter design flow conditions. Mass flow is best checked by direct calculation from flowmeter upstream conditions and differential pressure.

Whatever procedure used must accurately measure the true throughput of the compressor. Unaccounted sidestreams not measured by the flow element will directly impact performance calculations. Following are basic flow measurement equations [1, 3].

Square-edged Orifices

$$\dot{M} = (5.983)\,(K)\,(d^2)\,(Fa)\,(Y)\,\sqrt{\frac{h_w}{v_1}} \qquad (3)$$

Flow Nozzles and Venturi Tubes

$$\dot{M} = (5.983)\,(C)\,(E)\,(d^2)\,(Fa)\,(Ya)\,\sqrt{\frac{h_w}{v_1}} \qquad (4)$$

At many installations, flow is monitored in percent of the meter design rate. For a quick check, use the reading on this "root" gauge to determine the approximate flowrate. Be sure to use the correction factors shown below to obtain the best accuracy.

$$Q = (\frac{\%}{100}) \times Q_D \qquad (5)$$

where

Q_D = 100% design meter flow rate, SCFM

% = Meter reading, percent

Since this 100 percent flow was calculated for the design condition of the flowmeter, the flow must be corrected for the actual conditions at the flowmeter:

$$Q_c = Q \times \sqrt{\frac{P_A}{P_D} \times \frac{T_D}{T_A}} \qquad (6)$$

where

Q_c = Corrected flow, scfm

A = Actual condition at flow meter

D = Design condition of flow meter

Also, the compressor characteristics are dependent on actual inlet flow, so the standardized flow must be corrected to inlet conditions.

$$Q_1 = Q_c \times \frac{P_S}{P_1} \times \frac{T_1}{T_S} \qquad (7)$$

where

S = Standard conditions

1 = Inlet conditions

Casing Nozzle Meter

The flow in most compressors is generally accelerated somewhat before entering the first stage of compression. This can be equated to a Venturi effect and used to monitor flowrate. The compressor inlet can be calibrated during performance testing. The results can then be plotted for various suction temperatures (Figure 5).

The onsite inlet piping must have a sufficient amount of straight run prior to the compressor inlet to remove the effects of upstream valves, filters, elbows, etc. If not, the nozzle meter will not provide accurate flow data. For detailed information, refer to guidelines on upstream piping requirements for compressors and flowmeters [1, 3].
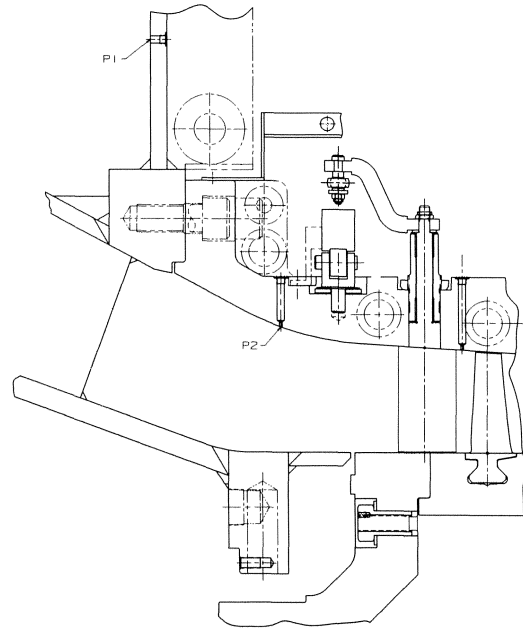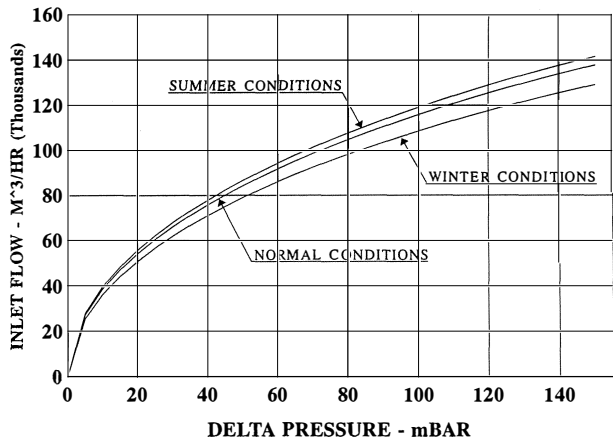

Figure 5. Inlet Calibration Curve. The compressor inlet can be calibrated and the results plotted for various suction temperature [1].

Work

$$\text{Work input, } W = \frac{H}{\eta} \quad \frac{Ft-lb_f}{lb_m} \qquad (8)$$

Gas Horsepower

$$GHP = \frac{(h_2-h_1)\dot{M}}{42.416} \qquad (9)$$

or

$$GHP_{ad} = \frac{H_{ad} \times Q_1 \times 144 \times P_1}{\eta_{ad} \times T_1 \times R \times 33000} \qquad (10)$$

or

$$GHP_{ad} = \left(\frac{k}{k-1}\right) \left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} -1\right] \times \left(\frac{144 \times P_1}{\eta_{ad} \times 33000}\right) \times Q_1 \quad (11)$$

Shaft Horsepower

Compressor shaft hp is best determined by adding the calculated gas hp to standard values of bearing and seal mechanical power losses. If the proper instrumentation is in place to determine oil temperature rise and flowrate, Equation 12 can be used to determine shaft hp.

SHP = GHP + Bearing and Seal Horsepower

$$\quad (12)$$

$$= \text{GHP} + (\text{GPM} \times \Delta T/12.6), \text{ for light turbine oil}$$

Example Calculation

Test Data

$$P_1 = 14.5 \text{ psig} \quad T_1 = 56°F = 516°R$$
$$P_2 = 54.5 \text{ psig} \quad T_2 = 349°F = 809°R$$
$$\text{Inlet } \Delta P = 40 \text{ in. } H_2O \quad N = 5804 \text{ rpm}$$

Head

$$H_{ad} = RT_1 \left(\frac{k}{k-1}\right) \left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} -1\right]$$

$$= \frac{1545}{28.97} \times 516 \left(\frac{1.4}{1.4-1}\right) \left[\left(\frac{54.5}{14.5}\right)^{(1.4-1)/1.4} -1\right]$$

$$= 53.33 \times 516 \times 3.5 \, (3.76^{.286} - 1)$$

$$= 44353 \text{ ft-lb/lb}$$

Efficiency

$$\eta_{ad} = \frac{T_1 [(P_2/P_1)^{(k-1)/k} -1]}{T_2 - T_1}$$

$$= \frac{516 [(54.5/14.5)^{(1.4-1)/1.4} -1]}{809 - 516}$$

$$= \frac{516 \times .46}{293}$$

$$= .81$$

Horsepower

Inlet flow for this compressor is easy to check since it has a calibrated inlet.

$$Q_1 = 92000 \text{ ICFM (From calibration curve, Figure 5.)}$$

$$GHP = \frac{H_{ad} \times Q \times 144 \times P_1}{\eta_{ad} \times T_1 \times R \times 33000}$$

$$= \frac{44353 \times 92000 \times 144 \times 14.5}{.81 \times 516 \times 53.33 \times 33000}$$

$$= 11583$$

Power Balance

Motor Power

$$BHP = E \times I \times \eta \times P.F. \times \sqrt{3} \ / 746$$

$$= 13800 \times 442 \times .95 \times .91 \times 1.732/746$$

$$= 12244 \text{ hp}$$

Mechanical Losses

75	compressor bearings
35	motor bearings
235	gear
345	total

$$TestError = \left[\frac{(Driver\,Power-Losses)}{GHP} -1\right] \times 100\%$$

$$= \left[\frac{(12244-345)}{11588} -1\right] 100 \text{ percent}$$

$$= 2.7$$

MAINTAINING PERFORMANCE

An axial compressor is designed for minimum maintenance. With proper lubrication, inlet air filtration, and operation, an axial compressor will have a long, useful life.

The lubrication requirements are similar to all turbomachinery using hydrodynamic bearings. Proper temperature, grade, quantity, and cleanliness of the lubricating fluid are a few of these requirements. Inlet air filtration is important to all types of blowers and compressors, but is especially important to axial compressors, since the rotating blades are thin airfoils moving at high speeds, and large particles would quickly take their toll in erosion. Erosion in axial compressors is critical because the base of the vanes become thinner and/or notched, leading to cracks and ultimately to catastrophic failure. Note that an axial rotor blade failure will damage both stator vanes, along with other rotating blades. It is a good reason to have spare stator vanes, along with a spare rotor, on hand in order to get the unit back online in the unlikely event of blade failure.

Proper operation of the compressor means maintaining the speed, flow, and head within the envelope set forth by the compressor manufacturer. This is accomplished by a control system that senses these parameters and changes the variable vane setting and/or speed to give the flow and pressure required by the process. When the combination of flow and head exceed the limits set by the compressor manufacturer, the control system opens a blowoff valve to bring the compressor back into a safe operating range, while providing the process with only the flow it requires. The operating envelope is defined by a surge control line, which is the surgeline plus a margin (usually surge flow plus 10 percent), the maximum and minimum variable stator positions (if so equipped),

and the maximum and minimum speed. On many axial compressor applications (i.e., FCC service) the discharge pressure requirement is fairly constant over a large flow range, and the speed is fixed, in many cases by an electric generator that locks in the speed. Variable speed control is normally used for systems where the discharge pressure varies proportional to the flow.

Even with proper inlet air filtration, vapors, fumes, and fine particles will get through and collect on the stator vanes and rotor blades. The amount and speed of this buildup depends on the type and amount of vapors, fumes, and fine particles ingested. This is not a severe threat to the compressor, provided the buildup is not corrosive and does not begin to attack the blade material. If the user intends to operate the compressor in a humid environment that contains acid forming gases, such as CL_2, SO_x, NO_x, etc., the user should have the compressor manufacturer coat the vanes and blades (at least the first three stages) with a protective coating. Buildup on the vanes and blades creates blockage that decreases the flow. The buildup also changes the blade profile, which decreases the lift of the airfoil and, in turn, decreases the head and efficiency of the compressor. This is synonymous with "icing" on an aircraft wing. The increase in power and decrease in maximum flow is imperceivable at first, but over a period of time, may effect operation depending on the nature of the buildup. Without a periodic tracking of compressor parameters, the first noticeable symptom may be that the compressor cannot supply the flow required by the process.

At this point, the unit may simply be in desperate need of cleaning. The limited operation of the compressor and increased power usage cost the user money, and failure to recognize and properly react to the problem will only multiply these costs. To keep the problem from getting this far, a preventative maintenance program should be instituted to clean the compressor blades on a regular basis.

Trending Performance

Trending performance data can help determine when the compressor should be cleaned or shut down for maintenance in advance, allowing for proper control and planning of work schedules. Procedures for trending aerodynamic performance can consist merely of monitoring parameters such as discharge temperature, vane position or speed, head, and/or efficiency. It is important to note that these parameters vary with changes in the operating point, and considerations for this variance must be made.

Tracking of compressor performance to determine when cleaning is needed has the added benefit of allowing the user to know the past and present condition of the compressor. A trend analysis of this data can be very useful in ascertaining the cause of performance problems other than buildup on blades.

A form that can be used for tracking these parameters is shown in Table 1. Data should be logged at the beginning of each shift, along with other regularly collected data. This log will prove invaluable when troubleshooting performance-related difficulties.

Abbreviated Parameters

In order to monitor the performance of a compressor, the best thing to do is to monitor head and efficiency *vs* flow and vane setting, and compare predicted and previous test data on a continuous day-to-day basis. This will give the up-to-date performance and trend information needed to predict when a maintenance shutdown is required for performance reasons and/or to help troubleshoot aerodynamic problems. Since a thorough test is not always practical on a continuous basis, some abbreviated parameters or methods of calculation are demonstrated here. When using these methods and procedures, remember that they are approximations and are only meant to monitor trends, not to replace the performance test described previously.

Table 1. Trending Performance. Recording these data will prove invaluable for anticipating performance related maintenance requirements and subsequent troubleshooting.

P1			
P2			
T1			
T2			
Delta P			
% Flow			
Speed			
Vane Position			
Time			
Date			

• *Pressure Ratio.* A close look at the equations for calculating head (Equation 1) will show that the primary variable in the equation is pressure, P_2 and P_1. Therefore, monitoring pressure ratio will give trends similar to monitoring head. Also, the plot of pressure ratio, r_p *vs* flow, Q will be similar to head *vs* flow, Q.

$$r_p = \frac{P_2}{P_1} \qquad (13)$$

• *Temperature Rise.* Compressor efficiency is defined as useful work done on the gas (head), divided by the total work. Since total work is directly related to enthalpy, which in turn is related to temperature, monitoring temperature rise will be an indication of total work input. If pressure ratio (r_p) goes down and/or temperature rise goes up (for a given flow, stator vane setting and speed), then this is an indication that the efficiency of the compressor has gone down.

$$\Delta T = T_2 - T_1 \qquad (14)$$

• *Root Reading.* Flowrate through the flow element is primarily a function of ΔP. Noting the change in ΔP or "root" reading will indicate changes in flowrate. If available, track the inlet casing differential pressure.

$$\Delta P = P_2 - P_1 \qquad (15)$$

Online Cleaning Procedures

Fouling of some degree occurs, no matter how meticulously inlet air filters are maintained. Moisture, with the slightest amount of an oil film commonly found around any machinery, assures that dust particles are well attached to the compressor blades.

As mentioned before, a preventative maintenance program should be established for cleaning an axial compressor. This program could be cleaning the compressor periodically whether it

needs it or not. Cleaning it periodically, e.g., once per month, is acceptable. If the flow as measured by the pressure differential across the inlet, or a flow measuring device in the process piping, shows an increase of three percent or more when the unit is cleaned, then the time period for cleaning should be decreased. If the change is less than one percent in flow, then the time period can be increased.

Organic Abrasives

If, at any time during operation, there is a drop in flow capability for a given head, vane setting, and speed, it may be best to try cleaning the compressor prior to other troubleshooting procedures. Cleaning the compressor blades is a fairly simple online operation. The usual method, called "ricing," consists of adding uncooked rice or crushed walnut shells to the inlet. Of course, the minute amount of rice or walnut shell particles must be compatible with the process and other related hardware. These particles are hard enough to remove most of the buildup on the vanes and blades but soft enough so that they do not pit (erode) the stainless steel. With this method of ricing, add about 1.0 lb of rice or walnut shells a handful at a time until the flow is back to normal. A recommended step-by-step procedure for "ricing" follows below. As the rice is added, the flow should be monitored. Normally, the pressure drop across the inlet casing is used. When the flow stops increasing, the unit is clean and the maximum flow is restored.

Depending on the process the compressor is in and the extent of contamination permitted to the process, the compressor can be cleaned with air going to the process or through the atmospheric blowoff valve. Proceed as follows:

Step 1. Locate a suitable piping or instrument connection at the inlet pipe as close as possible to the compressor inlet flange where a funnel with a 1/2 in ID spout or section of 1/2 in diameter tubing can be introduced (Figure 2). This connection will be used to introduce the abrasive material and, therefore, should preferably be located at the top centerline of a horizontal piping run. This will allow the abrasive to fall into the air stream and prevent any accumulation of material where it may be pulled into the machine as one large mass.

If no suitable connection is available, one of the second stage filter elements may be removed to allow the introduction of material. When using this method, be certain that personnel entering the filter house have no loose clothing, rags, pens, etc., which may be sucked into the compressor suction.

If suction pressure is above atmospheric pressure, an ejector may be used to inject the material.

Step 2. Position an operator where he can monitor the air flow and discharge temperature through the machine. Note the air flow and discharge temperature through the machine prior to introducing any foreign material. Vibration should be monitored for any abnormality during the cleaning. If any vibration increases suddenly, suspend the cleaning immediately until the source of the vibration is determined and corrected.

Step 3. Feed the abrasive material into the funnel, or insert the free end of the flexible tubing into the container of abrasive, to draw the material into the suction line at a rate not to exceed one pound per minute (for an air flowrate of 300,000 cfm). The largest dimension of the abrasive should be no greater than 1/8 in, or 1/3 the size of the smallest passageway in the compressor—whichever is smaller.

Step 4. Stop introducing the abrasive after 5.0 lbs and check the difference in air flow and discharge temperature through the machine (this assumes that speed, discharge pressure, and stator vane setting have remained constant). Note the variation in flow and temperature from the flow noted in Step 2.

Repeat Steps 2, 3, and 4 until no further upward change in air flow is noted through the machine, and a decrease in discharge temperature is observed.

Depending on the internal cleanliness of the machine, the air flowrate may change rapidly with the introduction of the abrasive (walnut shells or uncooked rice). Therefore, if the machine is online to the process, care must be taken that a process upset does not occur due to large changes in air flowrates.

When introducing the abrasive into the air flow stream, be certain to introduce the material in a steady flow. It is important that the abrasive not enter the air stream as a large mass, as this will cause damage to the blading.

Proper preparation by the people introducing the material will avoid problems. It is suggested that a "dry run" be made prior to the actual cleaning to determine the funnel size best suited to the abrasive particle size used. This will prevent plugging of the funnel or "batch" introduction of abrasive material and help to establish a better understanding of exactly what is required.

Liquid Wash

The only exception to ricing is when the blades are coated for corrosion protection. Coated blades are more resistant to buildup, but over a period of time they still can become fouled. In this case, the rice or walnut shells may be harder than the coating and erode it. To clean coated blades, a liquid spray is used. This liquid is an aqueous cleaning solution designed to clean the vanes and blades without damaging the coating. The procedure is usually automated to spray small amounts periodically into the compressor. The spray nozzles are located in the compressor inlet and generally use compressed air to atomize the liquid to prevent liquid erosion. Coated blades with liquid cleaning systems are commonly used on gas turbines, but are also used on axial compressors in other services, particularly FCC.

Performance Check

Once the unit is cleaned, be sure to take a complete round of data and check the compressor performance (Table 1). If the compressor is not performing as it should, steps should be taken to determine the deficiency.

Ricing or liquid washing cannot remove all the foreign material on the blades. Cleaning will only remove the light surface material. Hard "baked on" material will remain that can only be removed by direct means. Surface residue, blade surface damage, or increased blade tip clearances are just some reasons that performance may not be "like new" after cleaning.

INLET AIR FILTERS

Inlet air filters are important in maintaining the internal cleanliness of axial air compressors. In addition to filter performance, placement and maintenance must be considered.

The filter that is selected should have a maximum nominal filtration rating of 10 microns. Note that catalyst fines downstream of the third stage separator in an FCC unit are smaller than 20 microns, with the majority of this distribution being smaller than 5.0 microns. Particulate this small poses a greater fouling concern than a mechanical (erosion) concern.

Rain/water baffles must be utilized to help minimize the ingress of moisture during operation. The air in refineries and chemical plants typically contain abnormally high concentrations of caustic compounds (e.g., H_2S and HCl); mixing with water, either free or due to humidity, can create an acid which will attack aerodynamic components. Pitting of blades can occur, which can impact aerodynamic performance, but more importantly can lead to cracks and eventual failure of the rotating blades.

The filter housing should be elevated to help minimize dirt and water ingestion. Although advantageous from a reliability stand-

point, this requirement imposes construction, inlet piping, and maintenance considerations. The inlet air filter housing should be mounted so as to provide easy and safe access by personnel for replacement of air filter elements.

The filter assembly should utilize both expendable primary (first stage, or prefilter) filter elements, and long-life secondary (second stage, or final filter) filter elements. This arrangement permits maintenance of the primary filter elements during operation of the machine without the possibility of foreign particle ingestion. Primary filter replacement can be performed, either manually or automatically, by a mechanical device controlled by filter differential pressure.

All gaskets should be checked for damage or leak paths and repaired at the earliest convenience, if necessary. This includes housing/assembly gaskets, in addition to filter gaskets themselves.

Filter elements should be serviced per vendor recommendations. Recommended service intervals are typically based upon the differential pressure across the elements (in of water).

CONCLUSION

Monitoring the performance of an axial air compressor is relatively easy. Straightforward and simple hand calculations provide accurate answers as long as proper precautions are made to assure good raw data has been obtained. The information obtained from these test data is the first step in a quality maintenance program that will assure continuous peak efficiency and long term reliability.

NOMENCLATURE

A Area, ft^2

a Speed of sound, ft/sec

BHP Brake or shaft horsepower

C Discharge coefficient

c_p Specific heat at constant pressure BTU/lb mole °R

c_v Specific heat at constant volume BTU/lb mole °R

D Pipe diameter, inches

d Throat, or impeller diameter, inches

E Voltage

E Velocity of approach factor

Eff Efficiency

Fa Orifice meter thermal expansion factor

g_c Gravitational constant

  32.2 $\frac{\text{ft-lb mass}}{\text{lb force-sec}^2}$

GHP Gas horsepower

GPM Gallons per minute

H Head $\frac{\text{ft-lb force}}{\text{lb mass}}$

HP Horsepower

h Enthalpy BTU/lb mass

h_w Differential pressure, inches water

I Amperage

K Orifice meter flow coefficient

k Adiabatic exponent (c_p/c_v)

MW Molecular weight

$\dot{M}$ Weight flow (lb/min)

M Mach number, V/a

N Speed, RPM

P Static pressure (psia)

PF Power factor

Q Flow rate ft^3/min

R Gas constant (1544/MW)

Re Reynolds number

r_p Pressure ratio (P_2/P_1)

SHP Shaft horsepower

T Absolute temperature (°Rankine = °F + 459.6)

t Temperature (°F)

U Tip speed, FPS

V Velocity (ft/sec)

v Specific volume (ft^3/lb mass)

W Work $\frac{\text{ft-lbs force}}{\text{lb mass}}$

Y Orifice meter expansion factor

Ya Adiabatic expansion factor

Z Compressibility factor

Greek Letters

β Throat (or orifice) to pipe diameter ratio

η Efficiency

ρ Density lb/ft^3

φ Flow coefficient

Subscripts

ad Adiabatic process (H_{ad})

S Standard conditions - usually 14.6 psia, 60°F, dry air

1 Inlet conditions (P_1) (Q_1) (t_1)

2 Discharge conditions (T_2) (P_2)

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