1974

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G. C. Patel
Chicago Pneumatic Equipment Co.

J. Piersimoni
Chicago Pneumatic Equipment Co.

DEVELOPMENT REFINEMENT OF THE HIGH PERFORMANCE ROTARY OIL FLOODED COMPRESSORS

Dr. Ghanshyam C. Patel, Lead Project Engineer
Chicago Pneumatic Equipment Company, Franklin, Pennsylvania

Mr. Joseph Piersimoni, Manager, R & D Department
Chicago Pneumatic Equipment Company, Franklin, Pennsylvania

1. INTRODUCTION

The present state of the positive displacement rotary compressors has been achieved by many development efforts in the past thirty years. These efforts have significantly reduced the major losses in the compressors, leaving only a few areas possible for significant improvements. These few losses, like friction losses, compression heat losses, leakage losses, etc., are also very difficult to reduce further by a significant amount. Consequently, the development engineers are faced with a problem of evaluating the design modifications having minor improvements. The recent energy crisis has given further impetus to these developments by demanding more efficient compressors. These circumstances provide challenging opportunities for engineers to reevaluate all design modifications even having small magnitude (0.5 to 5.0%) improvements.

Refinements in the experimental development testing procedures of the compressors are needed for the accurate measurements of the small magnitude improvements. It is extremely difficult to assess accurately such small changes in the performance by experimental methods due to many variables. This paper describes some solutions by identifying the variables and the ways to reduce the effect of the variables on the compressor's performance.

The variables are divided into five major groups; such as, air end, remaining compressor package, input parameters, instrumentations, and testing personnel variables.

A criteria for economical evaluation of the compressor improvement is also described. Other sections in this paper are on conclusions, recommendations, and useful references.

2. COMPRESSOR VARIABLES

There are mainly two types of performance testing; one, for short term (say less than 200 hours) performance evaluation of the compressor and, two, for long term (say greater than 500 hours) endurance testing. Ideally, the engineer would like to know the effect of the design modification for both the short term and long term performance on the compressors. Fortunately, most of the modifications can be evaluated either by the short term or long term performance tests. The short term tests are mainly repeatedly conducted either for measuring relative change in the compressor's performance due to the design modifications or production testing of the compressors. The variables affecting both the short term and long term performance can be divided into the following groups, and they are discussed below in order.

2.1. Variables in the Air End.
2.2. Variables in the Remaining Package.
2.3. Variables in the Input Parameters.
2.4. Variables in the Instrumentations.
2.5. Variables from the Testing Personnel.

2.1. Air End Variables

The air end variables are isolated from the remaining package because the air end is the heart of the compressor and requires significant development efforts by the compressor manufacturers. The variables in the air end can be introduced by the following factors.

2.1.1. Manufacturing Tolerances of the Parts.
2.1.2. Assembly Variations.
2.1.3. New Design Modifications in the Air End.
2.1.4. Wear and Tear on the Air End.

The manufacturing tolerances for rotors, housing, seals, etc., produces variations in the critical clearances of the air end which in turn produces the performance variations. There are also some critical clearances for the air end which can be adjusted by assembling rotors, seals, housing, etc., with help of shims or other means.

Thus, assembly of the air end can produce the variations in the performance. The performance variations by actual new design modifications are obvious and need not be explained here. The performance variation of the air end with respect to time is also an important factor, since this is so gradual that in many situations the design engineer is tempted to ignore it. However, all care should be taken for evaluating the air end to eliminate resulting errors.

Thus, it is desirable to keep above variables constant during evaluation of one of the other variables. These can be easily achieved by using the same air end and understanding its wear characteristics.

2.2. Remaining Package Variables

The remaining package consist of the complete compressor system except the air end. This includes variables introduced by the performance of the following subsystems of the compressor package.

2.2.1. Inlet Passage.
2.2.2. Discharge Passage.
2.2.3. Oil System.
2.2.4. Heat Exchangers.
2.2.5. Prime Mover System.
2.2.6. Controls.

The pressure losses in the inlet passage due to filter, valve, elbows, etc., causes drop in the compressor performance. Any changes in these pressure losses produces variations in the compressor's performance. Variations can be caused by plugging of the inlet filter, deposits in the passage, etc. Similarly, variations in the discharge passage pressure losses causes the changes in the performance. The oil system includes oil receiver, oil separator, oil filters, oil pump, oil coolers, etc. Many compressors are sensitive to the oil injection temperature, pressure, and flow and any changes in them results in variations in the performance. Performance of the heat exchangers (oil cooler and air aftercooler) affects the flow delivery of the compressor. Changes in the inlet humidity of the air coupled with the changes in the aftercooler efficiency provides variations in the performance. Efficiency of the prime movers also affects the performance; and hence, these variables should be eliminated by measuring power and speed input to the compressor directly. The controls in the compressor require some power (electrical and/or pneumatic) and all attempts should be made to eliminate variations caused by the controls. Many controls require the air flow through orifices and should be carefully controlled to avoid variations in the performance.

These variables produces significant variations for long term performance testing and are very difficult to eliminate. For short term testing, like air end development, the following method is recommended to eliminate variations from the package by measuring the performance flange (inlet) to flange (discharge) for the air end.

--Inlet restrictions should be removed and inlet pressure should be measured just upstream of the inlet flange.

--Discharge pressure should be measured just downstream of the discharge flange.

--Power input to the air end shaft should be measured directly.

--While measuring delivery, the flow loss of the control system, flow loss of the oil returned line (scavenging line) from the separator, etc., should be regained or stopped.

--Oil flow, temperature, and pressure to the compressor should be kept constant to eliminate effect of the oil system.

--Aftercooler should be eliminated to reduce variations in the flow due to inlet humidity changes.

2.3. Input Parameters Variables

The input parameters to the compressor play very important parts in the performance of the compressor. The input independent variables for most of the compressors are listed below.

2.3.1. Inlet Air (or Gas) Temperature.
2.3.2. Inlet Air (or Gas) Pressure.
2.3.3. Inlet Air (or Gas) Humidity.
2.3.4. Cooling Water (or Fluid) Temperature.
2.3.5. Cooling Water (or Fluid) Pressure.
2.3.6. Input to the Prime Mover.
2.3.7. Discharge Pressure.

It is imperative for accurate testing to keep these variables constant or eliminate their effect by other means. The first approach of keeping the variables constant is a more accurate method, but requires higher cost. Even in many applications it is almost impossible to control the input parameters; for example, compressors in the large production plant or customer's location. This creates the need for understanding the effect of the input variables on the performance so that the variations can be eliminated by appropriate corrections. Such simple empirical correction methods are described here. The correction steps described in the previous section 2.2 can be used for eliminating variables 2.3.4, 2.3.5, and 2.3.6; hence, further discussion is omitted here.

2.3.1.1. Inlet Gas Temperature Effect on Flow

The inlet gas temperature has significant effect on the flow delivery of the compressor. Figure 1 shows the compressor flow at the inlet conditions for different air inlet temperatures, at 100 psig discharge and 3550 RPM for oil flooded rotary Singl/Screw (Zimmern) Compressor.

The performance characteristics of the Singl/Screw compressors are similar to that of rotary twin screw and rotary vane compressors, with some variations in the magnitude of compressor losses. The variation in the flow described above is due to the heat transfer between air and oil in the intake passages and compressor cavities, before compression begins. If this effect is ignored, the compressors tested at higher intake air temperature will show better performance and the poor performance at the lower temperatures. This effect is not considered in present ASME code or other literature (to the best of our knowledge); and when the same compressor is tested according to the code in the summer and winter, it shows different performance. The following simple empirical equation is recommended to correct performance at standard temperature condition, say, 70°F.

$$Qs = Q_t \, (1 + \alpha (530/T_i - 1)) \, . \, . \, . \, . \, . \, . \quad (1)$$

Where: Qs - Flow at standard temperature condition, (CFM).
Q_t - Flow at test condition, (CFM).
T_i - Absolute intake air temperature, (°R).
α - Inlet heat transfer coefficient, (nondimensional).

The flow at test condition is computed by ASME low pressure nozzle code. Compressors having lower inlet heat transfer would have a low value for α in the above equation. The value of α is zero when no heat is transferred to the inlet air before compression begins; and hence, capacity at inlet conditions remains constant as depicted in Figure 1. On the other hand, for maximum heat transfer the value of α will be unity and the actual mass flow will be constant regardless of inlet temperature; and capacity at inlet condition will vary inversly with the inlet temperature. Most rotary compressors have a value of α from 0.4 to 0.9 depending on the inlet heat transfer losses. Figure 1 describes how data taken under non-standard conditions agrees with the theoretical curve for the compressor having $\alpha = 0.55$. Under similar conditions when compressor sizes were increased 25% and 100% the value of the coefficient obtained was 0.50 and 0.49 respectively. Thus, for this family of the compressors $\alpha = 0.51$ can be used. An error in the coefficient affects the standard flow, Qs, by a minor amount since it is of secondary order. Thus, equation (1) suggested here can be extremely useful in converting data taken at non-standard inlet temperatures to the standard inlet temperature and vice versa. Observe that equation (1) does not only convert (using perfect gas law) the actual mass flow at non-standard inlet condition to the standard condition, but also adjusts the mass flow to eliminate variations caused by the inlet heat transfer effect.

2.3.1.2. Inlet Gas Temperature Effect on Power

This is also another phenomenon where total power into the compressor is affected by the inlet temperature. Figure 1 shows these variations. This variation may be caused by changes in the internal clearances, viscous friction, inlet heating effect, etc. For some rotary compressors, this effect is negligible. The following empirical equation is suggested to correct the power.

$$HPs = HP_t \, (1 + \gamma (530/T_i - 1)) \, . \, . \, . \, . \, . \quad (2)$$

Where: HPs - Power at standard condition, (HP).
HP_t - Power at test condition, (HP).
T_i - Inlet temperature (°R).
γ - Empirical coefficient (nondimensional).

The value of coefficient γ should be experimentally found for the compressor. The figure 1 depicts the experimental data and theoretical curve for $\gamma = -0.19$ for the rotary oil flooded compressor. The negative value of the coefficient indicates decreasing power with increasing temperature. When compressor size was increased to 25% and 100% the value of the coefficient obtained were -0.24 and -0.23 under similar conditions. As mentioned earlier, this effect is very small for some compressors.

2.3.2. Inlet Gas Pressure Effect on Power

This effect is well known by most of the engineers and described in the ASME code book and other literature. However, it will be advisable to verify the theoretical isentropic corrections described in the code book for the compressor being under test since the theoretical values may differ from the experimental values. The following empirical equation is suggested for its simplicity and accuracy.

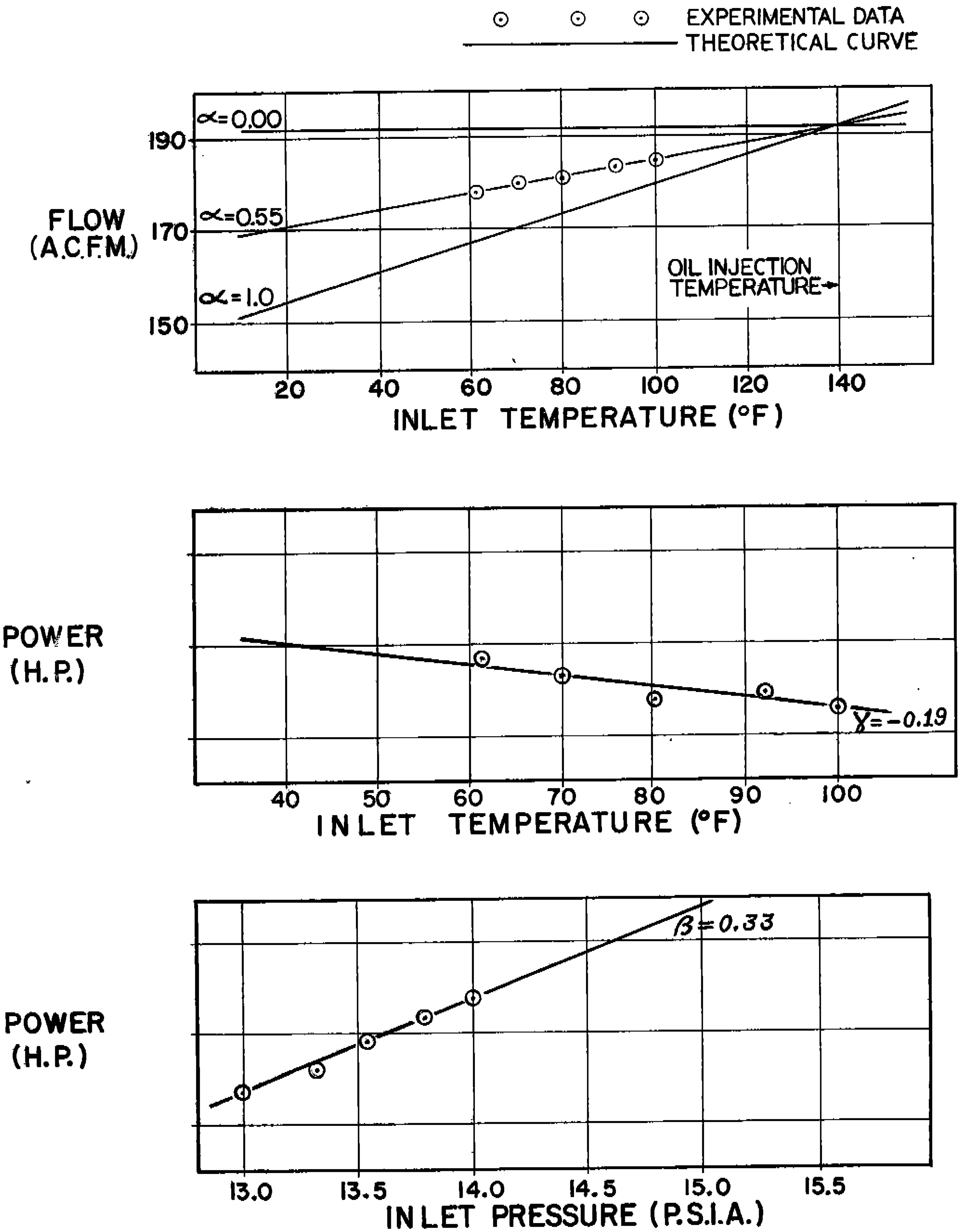

FIGURE 1. VARIATIONS OF THE INPUT PARAMETERS FOR THE ROTARY OIL FLOODED COMPRESSOR.

$HPs = HP_t\ (1 + \beta\ (Ps/P_t - 1))$. . . . . (3)

Where: HPs - Power at standard inlet pressure, (HP).
HP_t - Power measured at test inlet pressure, (HP).
Ps - Standard inlet pressure (14.7 psia), (PSIA).
P_t - Inlet pressure for test condition, (PSIA).
β - Empirical coefficient (non-dimensional).

The value of β was found as 0.33 for 100 PSIG discharge pressure, oil flooded compressor. When compressor size was increased 25% and 100%, the value of coefficient found was 0.34 and 0.30 respectively for similar conditions.

Whenever inlet pressure corrections are made according to ASME code book, the power changes due to both the inlet and discharge pressures (absolute) should be made. With equation (3) only one correction is required for change in the barometric pressure.

When equations (2) and (3) are combined, the following relationship is obtained for correcting the power for changes in the inlet pressure and temperature.

$HPs = HP_t\ (1 + \beta\ (Ps/P_t - 1))\ (1 + \gamma\ (530/T_i - 1))$. . . . .(4)

2.3.3. Inlet Gas Humidity Effect

The humidity of inlet air affects the flow and power of the compressor. The correction method is described in the ASME code book. The magnitude of the correction depends on the inlet humidity, discharge pressure, cooling during compression, and after cooling, etc. For air end testing, the aftercooler should be eliminated to reduce flow variations from this factor. The package temperatures are normally kept higher in most of the compressors to avoid condensation of the moisture. The empirical equations are not suggested here due to inadequate test data.

2.3.7. Discharge Pressure Effect on Power

For many rotary compressors having a fixed compression ratio, it was found that power changes almost linearly with changes in the discharge pressure. This leads into an interesting and useful empirical rule for obtaining the power at any required discharge pressure as given by the following equation.

$HP_r = HP_d\ (1 + \delta\ (Pr/P_d - 1))$. . . . (5)

Where: HP_r - Power required at discharge pressure, Pr., (HP).
HP_d - Required power at design pressure, Pd., (HP).
Pr - Discharge pressure at which power is required, (PSIG).
P_d - Design discharge pressure, (PSIG).
δ - Coefficient for correcting power, (dimensionless).

For many rotary oil flooded compressors having the design discharge pressure of 100 PSIG, the approximate value of the coefficient δ was found to be 0.5. This value 0.5 also means that the approximate power change is 1/2 of 1 percent for every 1 PSI change in the discharge pressure from 100 PSIG. Hence, approximate power required for the compressors designed at 100 PSIG is 90, 95, 100, 105, and 110 percent of the power required at 100 PSIG for the discharge pressure of 80, 90, 100, 110, and 120 PSIG respectively.

For experimental laboratory testing, it is recommended to have an enviromental chamber consisting of a radiator, fan, and throttle valve to hold the inlet pressure and temperature constant. A more sophisticated chamber is required to hold the humidity constant. Using such experimental setup, the empirical coefficients described in this section can be obtained for the specific compressors. Then proper corrections as described in this section can reduce effect of the input variables and more accurate testing can be achieved.

2.4. Variables in the Instrumentation

Variables that occur in the instrumentation have a direct effect on the compressor's performance through accuracy and repeatability of the data. Instrumentation used in performance testing measures the following parameters and also describes each type of measurement and the steps taken to reduce the variation.

2.4.1. Pressure.
2.4.2. Temperature.
2.4.3. Flow (liquid and gas).
2.4.4. Speed.
2.4.5. Shaft Horsepower.

Each of these parameters will have an effect on the measurement of the total performance. In order to minimize deviation in each of the measurements, instruments with a high degree of accuracy and repeatability must be used and should be periodically calibrated.

2.4.1. Pressure

When measuring pressure less than 10 PSIG or a vacuum, a manometer filled with water, mercury, or similar liquid is used. For higher pressures, a Bourdon

Gage is used. Selection of the pressure scale is based on measuring in the center third of the gage, which gives the highest accuracy. A single gage can be used on a test stand to measure multiple points. This is accomplished by installing a shut-off valve in a line between the pressure point and a common manifold attached to the gage. In addition to reducing the number of gages required, the method permits measuring with near absolute accuracy, pressure differentials between one point and another.

2.4.2. Temperature

The technology of temperature measurement has progressed more rapidly than any of the other parameters. This is particularly true with regard to thermocouple thermometers and digital potentiometers. Using copper-constantan thermocouple, system accuracy of 0.5 degrees F from 0-300 degrees F can be held.

Variation, therefore, does not occur in the thermocouple itself, but how and where it is installed in the system. An error will occur if the thermocouple is not adequately immersed in the gas or liquid stream. Also, radiation of heat to or from the thermocouple will cause an error in the measurement.

Thermocouple can be made with the measuring junction insulated from the sheath to reduce this radiation error. Because thermocouple can be made small and flexible, they are capable of being installed in relatively inaccessible spaces. Thermocouple will also permit, in conjunction with a digital indicator and multiple position switch, obtaining a large number of reading in only a few minutes. This reduces the chance of the operating condition changing before all the data is recorded.

2.4.3. Flow

Measurement of liquid flow is currently being accomplished by turbine-type flowmeters. Advantage of this over the gear type is instant readout and the turbine meter is small and compact and can be installed into the system without requiring additional piping or creating pressure differentials.

The gas flow is measured by the ASME Power Test Code. This method measures gas flow through a low pressure nozzle and converts the compressor output to the inlet conditions. In order to obtain accurate and repeatable data, these inlet parameters (pressure, temperature, and humidity) must be held constant.

2.4.4. Speed

The compressor speed is generally measured by means of a hand tachometer or a strobe light. Accuracy of these indicators are 1/2 to 1%. To obtain greater accuracy, a 60 teeth gear is installed on the shaft and a magnetic pickup is placed in a stationary position adjacent to the gear face and connected to frequency counter. This gives a readout in revolutions per minute with an accuracy of less than 1 RPM at any speed. This system also provides for measuring the speed over a very short (1 second) duration or a longer duration (10 to 100 seconds).

2.4.5. Shaft Horsepower

The measurement of shaft horsepower of rotating machines is accomplished either by direct or indirect methods. The indirect method measures the horsepower by metering the electrical input to the motor and correcting for motor efficiency and transmission losses. Accuracy of this method can be ±5%. By utilizing system calibration, accuracy can be improved to 2%.

There are several direct methods of measurement, the cradled dynometer, the brake, and the torquemeter having accuracy of ±1% and ±1/4% respectively. In addition to high accuracy, the torquemeter is very versatile. Its compactness permits use in restricted spaces and its broad range of torque output permits its use on several sizes of motors. Correct speed indication is essential for accuracy.

The instrumentation described here can be obtained in varying degrees of accuracy and repeatability. It is possible to obtain transducers, gages, meters, or indicators with accuracies of 1/10%, but the cost in some instances is prohibitive. It is interesting to note that instrumentation technology is advancing so rapidly that higher accuracy can be obtained at little or no extra cost.

2.5. Variables from the Testing Personnel

The effect of variables by the testing personnel will affect all of the preceding variables.

Variations in the air end will occur when different testers do the assembly. The measurement of critical clearances is subject to human factors, which in turn produces variations in the compressor performance.

These factors are critical in the area of instrumentation. Variations will occur if precise testing procedures are not followed. Reading gages, manometers, and meters are subject to deviation from one tester to the next.

To avoid some of these variations, test programs are scheduled so that one tester will follow a test series through completion. Personnel are taught to read meters from a specified position, utilizing mirror scales for proper alignment. Test data is recorded by several testers at the same time and compared to determine the amount of variation that exist and these areas are rechecked to minimize these deviations.

3. ECONOMICAL EVALUATION OF THE MODIFICATIONS

In the previous sections the importance and the ways of the compressor testing under standard conditions are described for evaluation of modifications providing small magnitude (0.5 to 5%) of improvements. Since major improvements have already been made in the air ends, the authors believe that most of the new modifications will be of small mganitude. However, it is vitally important to evaluate these modifications with respect to the magnitude of improvement and its cost.

The improvements in the performance can mainly provide savings in the following costs for the customers.

--Initial cost of the compressor.
--Operating cost of the compressor.
--Maintenance cost of the compressor.

The above cost savings should be converted to the present value savings and summed together to obtain total present value savings. The total present value savings then should be compared with the cost of the modification and decision should be compared with the cost of the modification and decision should be made for providing the modification on the compressor package.

For example, let us consider the improvement in the air end which provides 1% performance improvement for 40HP compressor. Assume that the improvement does not affect the maintenance cost. Also assume that the improvement costs $50 more including factory cost and other appropriate overhead expenses. Consider initial package cost to the customer as $5,000 and approximate operating cost as $5,000 also. The operating cost for every customer may be higher or lower than $5,000 depending on the use and local power rate. (The operating cost for 40HP, 8,760 hours/year and $0.02/KWH is $5,240/year). Assume 20% as profit rate before taxes for the customer's plant. The present value factors for the first, second, third, fourth, etc., year will be 1, 0.8, 0.64, 0.51, etc., respectively.

The improvement provides 1% more flow; and hence, 1% saving in initial cost/CFM and also 1% savings in the operating cost every year. The present value savings for say three years can be obtained from Cost Savings Table 1.

Thus, for investing $50 in the present year the customer can save $200 in three years which has present value as $172, and the improvement is worthwhile for the customer. This logic is, however, difficult to sell to all customers but can be very useful to the design engineers for convincing management and other technical personnel.

Thus, importance of the small improvements can be justified by coupling technical information with the economical aspect of the modifications.

4. CONCLUSIONS AND RECOMMENDATIONS

The problem of accurate experimental evaluation of the compressor is extremely difficult, and it must be resolved by the engineers envolved in the development of the high performance compressors. Enough care should be taken to keep constant the variables described in this paper for accurate testing, otherwise, wrong conclusions can be reached. Effect of some of the variables are so subtle that it is often ignored by many engineers and it is also not described in the ASME codes or literature.

Coupled with the technical improvements, economical analysis of the modifications in the compressor as described here provides a very useful evaluation tool for the engineers. The small magnitude improvements are worth considering.

5. ACKNOWLEDGEMENTS

The authors are grateful to many staff members of the Engineering Department of Chicago Pneumatic Equipment Company for their assistance in constructing the test equipments, conducting the tests, computing the test data and for providing other help for this investigation.

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COST SAVINGS

Table 1

Year	Cost Savings Initial	Cost Savings Operating	Total Savings	Pres. Value Factor	Pres. Value Savings
1	$50	$50	$100	1.00	$100
2	00	50	50	0.80	40
3	00	50	50	0.64	32
			$200		$172