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Channel Resonant Errors on P-V Indicator Diagrams for Reciprocating Compressors

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ABSTRACT

The passage length between the cylinder pressure and the measuring transducer on commercial instrumentation will cause distorted pressure time traces. This is due to gas dynamic effects causing pressure pulsations in the indicator passage. Solutions have addressed the problem in both the time and frequency domains. The intention of this paper is to examine four of these methods in comparison to laboratory data. Then determine the most acceptable solution based upon accuracy and personal computer computational time. It was concluded that depending on the solution tools available to the analyst, both a time and frequency domain solution can satisfy the requirements mentioned above.

NOMENCLATURE

A = Cross sectional area of indicator passage (ft^2)
A_{ref} = Reference speed of sound at 14.7 psia and 60°F (ft/min)
C = Capacitance (g_cA/c^2)
c = Non-dimensional speed of sound (c_d / c_{ref})
c_d = Local speed of sound (ft/min)
D = Diameter of indicator passage (ft)
f = Darcy friction factor
g_c = Acceleration of gravity (ft × lbm / lbf × s^2)
i = $\sqrt{-1}$
k = Ratio of specific heats (Cp / Cv)
L = Inertance ($1/g_cA$)
l = Length of the indicator passage (ft)
n = Number of harmonics
P_d = Complex pressure at downstream conditions (psia)
P_{ref} = Reference pressure (14.7 psia)
P_u = Complex pressure at upstream conditions (psia)
Q = Complex flow rate (ft^3/min)
R = Resistance (fQ/g_cA^2D)
R_g = Gas Constant (ft × lbf/lbm × °R)
t = Time it takes gas to travel from upstream to downstream conditions (sec)
T_{ref} = Reference temperature (60°F)
u = Non dimensional gas velocity (u_d / u_{ref})
u_d = Local gas velocity (ft/min)
Z = Compressibility
Z_c = Characteristic Impedence
Γ = Complex propagation constant
w = Forcing function circular frequency (rad/sec)
ρ = Density (lbm/ft^3)
μ = Absolute viscosity (lbf × s/ft^2)
dx/dz = Speed of the characteristic wave

Subscripts

cyl = Conditions at the cylinder portion of the passage
trans = Conditions at the transducer portion of the passage

This paper also refers to compressor tests at low ratio, low speed conditions and high ratio, high speed conditions. The low ratio, low speed condition refers to tests at a pressure ratio of 1.6 and compressor speed of 1515 rpm. The high ratio, high speed condition refers to runs at a pressure ratio of 2.5 and compressor speed of 1770 rpm (see **LABORATORY DATA**).

Please note that the Y-axis (pressure) of the graphs included in this report are given in millivolts. This is the unnormalized values of the transducer readings and will not effect the results found.

INTRODUCTION

The dimensions of a reciprocating compressor cylinder and commercial dynamic pressure indicator test equipment result in the pickup usually being installed remote from the working pressure in the cylinder bore. The dynamic pressure measurement device is connected to the working cylinder volume by a gas passage of varying designs. Gas dynamics between the inside of the cylinder and the transducer diaphragm can produce significant error in the recorded dynamic pressure trace. This translates into a distorted pressure volume diagram, error in the indicated horsepower and error in diagnosing compressor performance problems. The source of this indicated horsepower error is sometimes referred to as channel resonance.

This paper will review four methods of determining and correcting for the effects of channel resonance error. These four methods are:

1. A simple model that considers only the time for a pressure wave to move from the cylinder working volume to the indicator diaphragm.
2. A simple dynamic model that accounts for one dimensional flow in a duct of constant cross section and is solved in the time domain by the method of characteristics as published by Bradley and Woollatt in 1968.
3. A detailed dynamic model that includes acoustic damping and uses a frequency domain pressure pulsation simulation as published by H. Kammin in an ASME Pipeline Engineering Symposium in 1989.
4. A detailed dynamic model that is solved in the time domain and that includes a detailed evaluation of the physics of a reciprocating compressor including near field gas dynamics.

The paper will comment on the method that offers a balance between acceptable accuracy and relative simplicity for efficient adaptation to a personal computer environment.

TESTING PROCEDURE

The laboratory tests were conducted at the Dresser-Rand, Closed Loop facility in Painted Post, New York. For a description of this facility see reference #1. The test vehicle was a high speed, low horsepower compressor pumping nitrogen in a single stage configuration. Two passages of 6 and 12 inch lengths were designed that would allow a Ashcroft K8 transducer to be placed into the end of the passage and secured into place. Another passage of minimal length was developed to record a "channel resonant free" signal. The transducer's electric output was transmitted to one of four channels on a Nicolet 4094 Oscilloscope. A timing trace was transmitted to a different channel for top dead center definition. Each test point was recorded onto floppy disk through a disk drive connected to the oscilloscope. A BASIC computer program was written to establish communication between the oscilloscope and an IBM PC via a RS232 cable. The test data was transferred to the PC in ASCII format for later analysis and manipulation.

LABORATORY DATA

The experimental dynamic pressure data that was collected to evaluate channel resonant correction procedures was measured at two pressure ratios (1.6, 2.5) and two compressor speeds (1515, 1770 rpm) as previously noted. For each condition pressure-time traces were measured at the minimal, 6 inch and 12 inch channel lengths. For the sake of brevity, this paper will consider only the 1.6 ratio, 1515 rpm test point and the 2.5 ratio, 1770 rpm point.

The compressor cylinder pressure time diagrams plotted on graphs 1 through 4 were all measured on the outer end of the double acting cylinder. As expected, channel resonance is greater at the 12 inch passage lengths than at the 6 inch passage lengths. Also, channel resonance is greater at the low ratio, low speed condition than the high ratio, high speed condition.

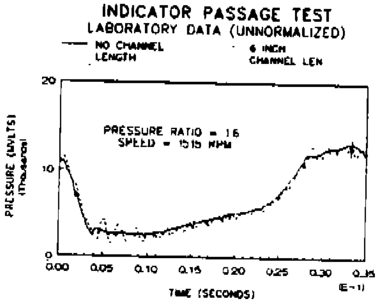
The laboratory data shows that indicator passage errors

a) can redefine the expansion and compression lines and,

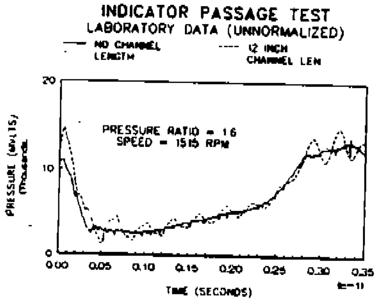
b) show substantial amounts of pressure pulsations on the suction and discharge events.

The effect these errors can have on compressor performance studies is important. The redefined expansion and compression lines will cause incorrect diagnosis of leakage and capacity calculations, while the pressure pulsations will cause an erroneous belief in valve dynamic problems. Both of these effects will produce an error in the indicator compressor horsepower measurement. The aim of this paper is to determine the best approach to correcting the indicator card for this channel resonant phenomena; the best approach being defined as that method that offers a balance between acceptable accuracy and relative simplicity for efficient adaptation to the personal computer.

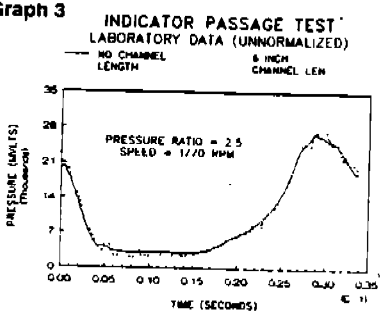
Graph 1



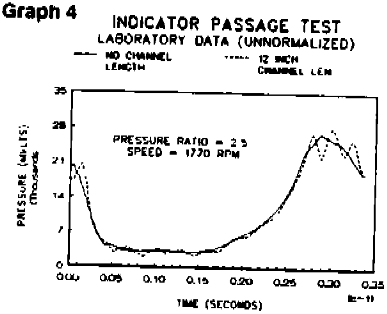
Graph 2



Graph 3



Graph 4



METHOD #1 : SIMPLE PHASE METHOD

Theory

The first correction method was recommended by SGA/PCRC in their 1984-10 report [2]. (SGA/PCRC has recently dropped this method of calculating indicator passage error as outlined in their 1990 test report [3]). It states that indicator passage error is a phase shift of the true indicator card or in other words the sole result of the time it takes for the gas to travel from the cylinder bore to the transducer diaphragm. This time is calculated as the length of the passage divided by the local speed of sound.

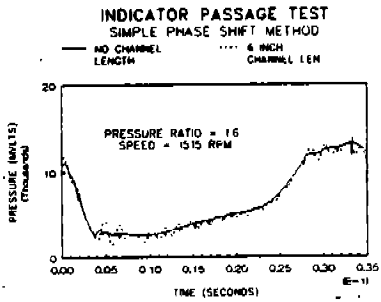
$$c = \sqrt{k g_c Z R_g T} \tag{1}$$

$$t = l / c \tag{2}$$

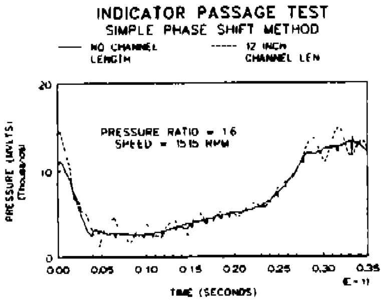
Observations

Graphs 5-8 show that this method did remove some of the phase shifting problems especially on the corrected 12 inch low ratio, low speed run. Unfortunately, phase shifting is only one part of the error caused by indicator passage lengths. Resonant effects are clearly present yet not corrected with this method. It is difficult to say if better results would be found at different conditions, i.e. shorter passage lengths, slower compressor speeds, different gases, etc... but it is certain this method cannot be used effectively in areas where pressure pulsations are present. If this method was assumed accurate, then faulty valve flutter diagnosis will occur. The idea of correcting phase shifting solely is an incomplete physical model. Other passageway errors must also be addressed.

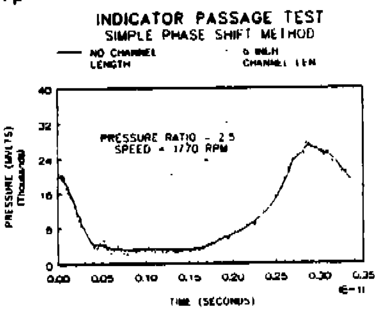
Graph 5



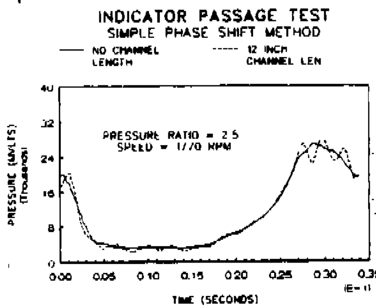
Graph 6



Graph 7



Graph 8



METHOD #2 : "SIMPLE" TIME DOMAIN SOLUTION

Theory

The next correction uses the method of characteristics to find a dynamic time domain solution that accounts for one dimensional flow in a duct of constant cross section[4]. The assumptions are;

1. frictionless passage
2. no substantial volume at the transducer
3. no pressure losses at the ends of the passage due to the finite gas velocity in the passage
4. the pressure wave characteristics have constant slope i.e. the gas velocity is much less than the sonic velocity.
5. homentropic flow

This method was introduced by Bradley and Woollatt in 1968 in which they concluded the preceding assumptions were reasonable based upon the scope of their testing and should not greatly affect accuracy.

The method of characteristics states;

$$c + \frac{k-1}{2} u \quad \text{is constant along} \quad \frac{dx}{dz} = u + c \tag{3}$$

and

$$c - \frac{k-1}{2} u \quad \text{is constant along} \quad \frac{dx}{dz} = u - c \tag{4}$$

Given assumption #4 is true, then the speed of the characteristics can be represented as the reference speed of sound A_{ref} and dz/dx = +/- 1.0. Assume that at time = 1, a pressure wave is at the transducer. Assume at time = 2, the wave reaches the cylinder. Finally, assume at time = 3, another pressure wave reaches the transducer again. Equations 3 and 4 gives;

$$C_{cyl(2)} + \frac{K-1}{2} U_{cyl(2)} = C_{trans(3)} + \frac{K-1}{2} U_{trans(3)} \tag{5}$$

and

$$C_{cyl(2)} - \frac{K-1}{2} U_{cyl(2)} = C_{trans(1)} - \frac{K-1}{2} U_{trans(1)} \tag{6}$$

Since the transducer end is considered closed, utrans(1) and utrans(3) are set equaled to zero. Adding equations (5) and (6) gives

$$C_{cyl}(2) = 0.5\,(C_{trans(1)} + C_{trans(3)}) \tag{7}$$

Using the isentropic change of state law;

$$c = \frac{C_d}{a_{ref}} = \frac{p}{p_{ref}}^{(k-1)/2k} \tag{8}$$

gives;

$$P_{cyl(2)} = \left\{ \frac{P_{trans(1)}^{(k-1)/2k} + P_{trans(3)}^{(k-1)/2k}}{2} \right\}^{2k/(k-1)} \tag{9}$$

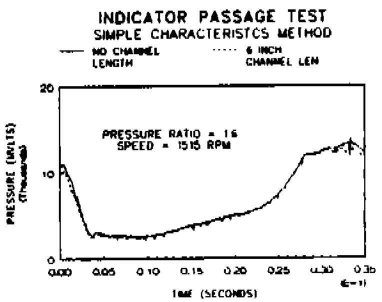
The program written to test this solution uses an array of pressure time points measured at the transducer for one full compressor cycle. The program uses a point at the transducer as time = 1. The passage length is divided by the reference speed of sound and multiplied by 2 to determine the pressure at time = 3. These two pressures are inputed into equation 9 and a value of the pressure at time = 2 is found. The new time is determined by simply adding on the passage length divided by the reference speed of sound to the original time = 1. This procedure is continued through the whole cycle.

Observations

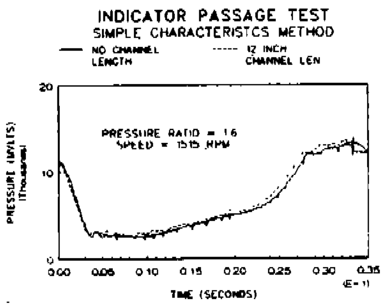
Graphs 9-12 present the results produced from this method. Acceptable accuracy was found at both the 6 and 12 inch lengths with some minor exceptions. This method successfully filtered the channel resonance found on the cards. A small error was found at the low ratio, low speed condition on the discharge event with the corrected pressure falling below the actual pressure. This can cause a small underestimation of compressor horsepower. For the 12 inch cards, the method did not accurately correct for the compression and expansion line phase shift although it reduced the magnitude of the original loss. A misrepresented expansion or compression line will cause erroneous capacity and horsepower calculations and lead one to believe that leakage is occurring in the compressor when it is not.

The condition that had the least favorable correction was the 12 inch, high speed, high ratio card. Both channel resonance and phase shifting was present. It may be advantageous to relax some of the assumptions for passageways of greater length and for compressors of higher speeds. This was not done for this current study.

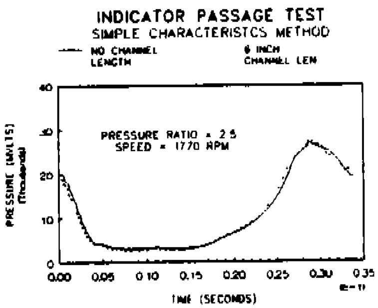
Graph 9



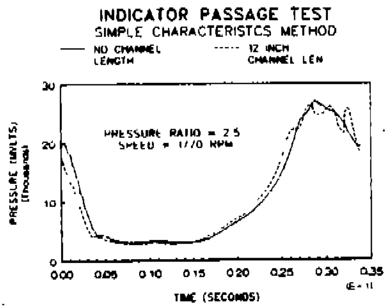
Graph 10



Graph 11



Graph 12



The simple time domain solution is a simple yet effective method to correct for channel resonance in short indicator passageways where resonant effects are limited. This correction requires minimal time for a solution in a PC environment.

METHOD #3 : FREQUENCY DOMAIN SOLUTION

Theory

This method is a frequency domain analysis that utilizes oscillatory forced vibrations for pressure pulsation simulation. The pressure and flow at either the upstream or downstream of the passage varies harmonically at each point in the system at the excitation frequency. The amplitude and phase of the pressures or flows change with position in the system but remain independent of time[6]. The governing equation used for this analysis is;

$$P_{cyl} = P_{trans}\cosh(\tau l) + Q_{trans}Zc\sinh(\tau l) \tag{10}$$

$$\tau = \sqrt{wC(-wL + iR)} \tag{11}$$

Since the downstream portion of the passage is a closed end, the transducer flow (Q_{trans}) is equal to zero, thereby reducing equation (10) to;

$$P_{cyl} = P_{trans}\cosh(\tau l) \tag{12}$$

The solution method generated solves equation 12 in a number of steps. The first step is to accurately represent the original pressure time card into a series of real and imaginary coefficients (a1,a2...an,b1,b2...bn). The indicator card can then be written as;

$$P(t) = P0 + \sum_{1}^{n} a_1\cos(wt) + a_2\cos(2wt) + \ldots a_n\cos(nwt)$$
$$+ \sum_{1}^{n} b_1\sin(wt) + b_2\sin(2wt) + \ldots b_n\sin(nwt) \tag{13}$$

Trial and error has shown that 20 or more harmonics is sufficient to accurately define the pressure time trace. Once the series is defined, the values for the pressures for each time step can be plugged into equation 12. Using complex number FORTRAN programming, a new value for the pressure at the cylinder bore is found. These pressures are represented as a series of real and imaginary coefficients. The final step is to generate a Fourier series to convert the coefficients into a pressure time equation for all crank angles.

Observations

The results found from using this method were good. Graphs 13-16 indicate that the solution was accurately filtered out channel resonant pressure pulsations. Similar to the previous time domain solution, the lower ratio, lower speed trace showed a corrected pressure line falling lower than the actual line. As mentioned before, this can cause a small underestimation of the indicated horsepower. Another problem is an inaccurate definition of the expansion and compression traces. Once again, this error can cause a misdiagnosis of leakage effects and capacity calculations.

It should be noted that the inaccuracies found with this particular set of data are small and that the remaining errors are insignificant compared to the cards without any type of correction.

Certain assumptions with this method could effect the accuracy of the results. One such assumption is the value of the resistance variable used in the imaginary portion of the complex function. For this particular application, a resistance value was used that is similar to the one discussed in Harlan Kammin's paper[5]. It is assumed the resistance in the passage is comprised of two parts; a part due to flow through the passage and the resistance due to the viscosity of the gas. It was reasoned that since the passage was a dead end channel, the resistance in the pipe due to gas flow was small or negligable. The equation used was;

$$R_{eq} = R_{flow} + \frac{W^2}{\rho\, g A C_d^{\,2}}\; \mu\; \frac{4}{3} + 1.6\,(k-1) \tag{14}$$

For this case, it is assumed that R_{flow} equals 0 and that the Prandtl number = 1.6.

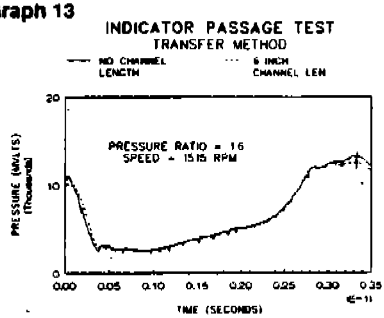
The mathematics of the solution will cause a greater amount of computational time (as compared to other methods) which should still be small with modern high speed personal computers. The downfalls associated with this method are that;

a) it will take some time to run on older, slower PC's especially if a large number of harmonics are specified and

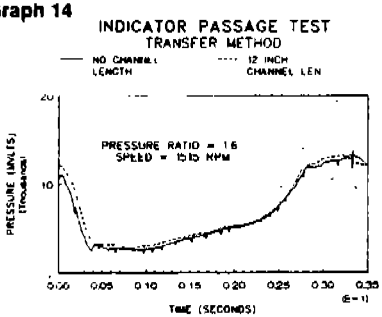
b) it requires complex number FORTRAN programming to work correctly.

It is felt that these downfalls are small and given the accuracy of the solution well worth the extra time.

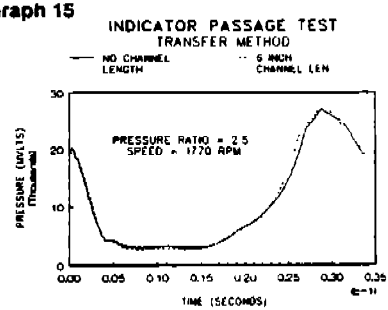
Graph 13



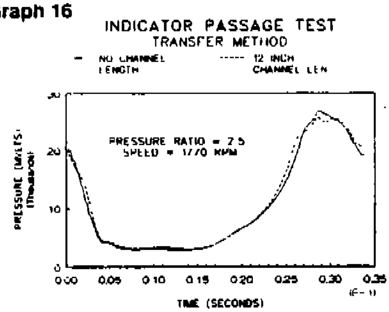
Graph 14



Graph 15



Graph 16



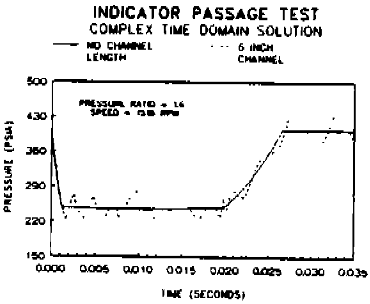
METHOD #4 : DETAILED TIME DOMAIN SOLUTION

The final method that will be addressed is a dynamic model that is solved in the time domain and includes a detailed evaluation of the physics of a reciprocating compressor near gas field conditions. This approach also uses the method of characteristics in which the magnitude of a pressure pulsation can be calculated by using the slopes of two intersecting characteristic pressure lines. The solution is adaptable for varying lengths, diameters and friction factors.

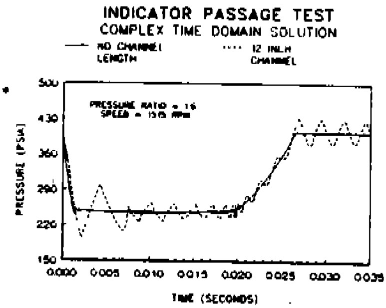
For this particular method, a test of its accuracy cannot be discussed since the routine is a subset of a large Dresser-Rand proprietary compressor cycle simulation program. Extracting this routine to run on a PC proved not to be an easy task. For the particular cases mentioned before, the graphs 17-20, show the comparisons between what the program thought a 6 and 12 inch card should look like and the actual measured card.

As with the previous frequency domain solution, the value used for the friction term has a large effect on the results. As evident from the graphs, the program calculated larger magnitudes of pressure pulsations for the tested channel lengths. This is a direct link to the damping factor used.

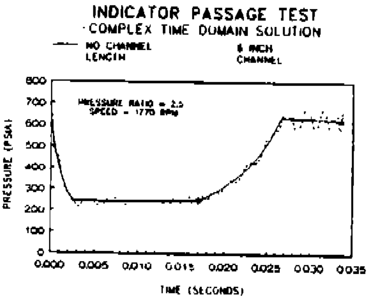
Graph 17



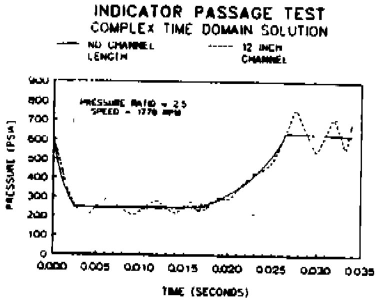
Graph 18



Graph 19



Graph 20



CONCLUSIONS

This paper discussed four different ways of calculating indicator passage error due to channel resonance. The methods looked at dealt with the problem in either the time or the frequency domain. Versions of both approaches offered acceptable accuracy while maintaining reasonable computational time. In cases where a high speed computer (AT or better) is not available, the simple time solution (Method #2) is recommended. The simplicity of the solution makes it extremely easy to program and relatively fast computational time on almost any computer. If a slightly more detailed solution is desired and a higher speed computer is available, the complex frequency domain solution (Method #3) is recommended. The test results showed that overall this method offered the best results. The detailed time domain solution (Method #4) is probably the most interesting because it is part of a compressor performance calculation package that determines all other variables present in compressor performance as well as indicator passage error. The emphasis with this method is to predict performance results based on sound principles of physics and not data-fit equations. Finally, the first method did not offer acceptable results because it was an oversimplified approach to the dynamics problem. The complex frequency solution dealt with the dynamics in grater detail and thus gave better results.

The four methods presented here give four ideas for correcting indicator passage error. Some feel the error's magnitude is less than the problem of correcting it. This is not the case. Even if horsepower numbers do not greatly change with the error, erroneous diagnosis can occur. It is quite evident from the laboratory data that the channel resonance appears as valve losses and leakage past valves. Substantial amounts of time and money can be saved if both the analyst and compressor owner can distinguish between what is real and what is error.

ACKNOWLEDGMENTS

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