

# MEASURING THE THERMODYNAMIC PERFORMANCE OF MULTISTAGE COMPRESSORS OPERATING ON MIXED HYDROCARBON GASES

by

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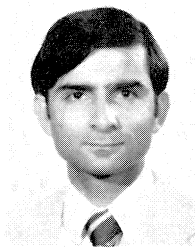
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## ABSTRACT

Performance measurement of multistage centrifugal compressors is often required to assess their current operating condition, e.g., to discover fouling or to gauge the operation of the compressor in relation to the overall process, especially when compressing mixed hydrocarbon gases. Generally, such field testing has not been successful in the sense of obtaining accurate and reproducible results.

This paper describes the development and validation of a unified testing and data analysis procedure for field evaluation of centrifugal compressors. Experimental aspects included developing instrumentation to obtain accurate measurements of pressure, temperature, and flow rate at the suction and discharge locations of individual stages. Sampling of the gas streams and the subsequent compositional analyses were given particular attention. Equation of state computer codes were prepared to convert measured compositions to thermodynamic properties as a function of pressure and temperature. Finally, procedures were developed and reduced to computer codes for

calculating stage-by-stage the work input to the gas and the compressor efficiency. Experience has demonstrated the validity of this approach to performance testing and it is now being adapted to on-line, computer-based monitoring.

## INTRODUCTION

Centrifugal compressors in petrochemical plant service normally operate under conditions that are determined by the demands of the overall process of which the compressors form a part. In general, these operating conditions may not be those which coincide with the compressor design point or at which "optimal" performance of the compressor in a thermodynamic sense may be expected. Nevertheless, a significant energy savings may result if, within the limitations of the overall process parameters, the compressor could be operated in a more efficient manner. Another operating problem is that over a period of time a compressor's performance characteristics may deteriorate due to fouling, seal leakage, etc. It would be advantageous to be able to follow these changes in time. This in turn requires a method for field evaluation of the machine's thermodynamic performance.

Three types of effort are involved:

1. acquisition of the pressure, temperature, flow rate, and gas samples from the suction and discharge sides of each stage of compression,
2. gas compositional analysis, and
3. calculation of head, power input, and compression efficiency based upon the acquired test data.

In this paper, except where otherwise noted, a "stage" of compression may include multiple impellers between its suction and discharge ends with intercooling being present between stages.

This paper describes the techniques developed in each of the above areas to measure the thermodynamic performance of the process gas compressor train in a large olefin plant. A more complete discussion of item (3) is reported in a related paper [1].

## EXPERIMENTAL ASPECTS

### *Instrumentation*

At the outset, a program was undertaken to assess the significance to the overall inaccuracy in the evaluation parameters of individual inaccuracies of each measured quantity in a performance test. This study raised some doubt about the

suitability of using the plant's process instrumentation for evaluating compressor performance. For this reason it was deemed necessary to utilize special instrumentation in the form of probes which could traverse the compressor ducts radially, and whose pressure and temperature transducers were carefully calibrated before the tests.

A pair of such probes was used to obtain stagnation and static pressures and stagnation temperatures at various radial positions in the suction and discharge ducts of a four-stage gas compressor train. A detailed description of the probe and its insertion device appears in the Appendix. Basically, the sensor end of the probe is wedge-shaped and has ports which are connected, through internal passages, to a static and differential pressure transducer at the end that remains outside the duct. The probe tip also has an embedded thermocouple for temperature measurement. Access for the probes is gained through gate valves at existing process instrumentation taps and an insertion device is used to position the probe in the ducts while maintaining a gas tight seal. This type of probe can provide an accurate measurement of the velocity pressure for the purpose of estimating the mass flow rate in a duct only if it is inserted at a point where the flow is fully developed, i.e., well downstream of such disturbances as elbows.

Gas samples could be obtained through the static pressure ports of the probe using sample lines heated by a steam tracer and a sample flask heated to approximately the gas stream temperature. This heating prevented condensation of any of the higher molecular weight constituents. The gas was allowed to bleed to the atmosphere through the entire sampling system for a short time to assure obtaining a representative sample. It was deemed necessary that gas analysis be performed off-line using a mass spectrometer (180° sector-magnetic) and results reported on a dry basis.

#### Evaluation of Pressure and Temperature Data

Data were acquired from the suction and discharge sides of all four stages of the above mentioned compressor train. However, experiments to study the effect of radial traverses on measured pressure, temperature, and gas composition were restricted to the second stage because of easier access at that location. The experiments resulted in the following general observations:

- The static pressure and temperature of the gas stream remained fairly constant across the radius of the compressor ducts (maximum pipe Reynolds number =  $2 \times 10^6$ ).
- Pressure and temperature data from the probe were in good agreement with those reported by the plant instrumentation except in some random cases. (It may be noted that the plant instrumentation was calibrated prior to the experiments). From this observation it may be concluded that the probe may be used initially to provide a laboratory calibrated means to establish the accuracy of the plant instrumentation.

The velocity profile of the gas flow in the duct was determined from the differential (velocity) pressure obtained as a function of radial position. The gas velocity,  $v(r)$ , as a function of the radial position,  $r$ , was obtained by using

$$v(r) = \sqrt{\frac{2\Delta P(r)}{\rho}} \quad (1)$$

where  $\Delta P(r) = P_0(r) - P(r)$   
 $P_0$  = stagnation pressure  
 $P$  = static pressure  
 $\rho$  = gas density.

It may be noted that the quantity  $\Delta P(r)$  is directly measured by the differential pressure transducer of the probe. The gas density is obtained from the computer program that calculates the gas properties — further discussion will follow in a subsequent section.

The velocity profile for the suction side of the second stage, for example, is shown in Figure 1. It shows good repeatability of the acquired data from three tests. It is quite apparent that the velocity profile is indicative of a well mixed turbulent flow condition. The Reynolds number for the flow condition was estimated to be  $2 \times 10^6$  — a further indication of its turbulent nature. This also explains why the static pressure and temperature were noted to be fairly constant in the radial direction.

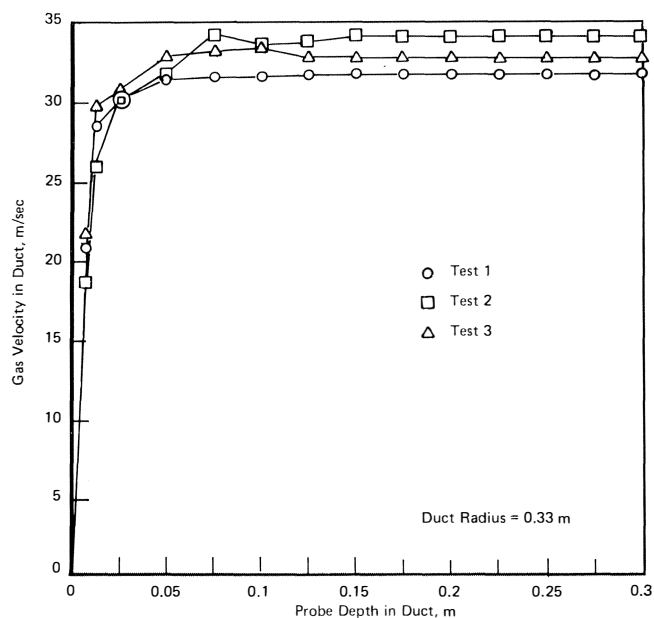


Figure 1. Gas Velocity Profile Across Second Stage Suction Duct.

The velocity profile was integrated over the duct radius to result in a mean velocity  $\bar{v}$  of the flow and was used to determine the flow rate through the duct. The mass flow rates predicted by the probe measurements were noted to be consistently higher than the corresponding ones indicated by the process computer based on orifice flow meters. This discrepancy existed because the calculations performed by the process computer in determining the flow were based upon the measured differential pressure across the flow meter and the design values (rather than actual operating values) of static temperature, pressure, and molecular weight of the process gas. It was shown that if the mass flow rate  $w_{pc}$  as reported by the process computer was corrected by the following relation

$$w_{\text{actual}} = \left( \frac{\rho_{\text{actual}}}{\rho_{\text{design}}} \right) w_{pc} \quad (2)$$

then good agreement (within one percent) was obtained between the corrected values and those obtained from the probe measurement. (Note that density is a function of pressure, temperature and molecular weight and hence the simplified form of equation (2)).

#### Gas Sample Analysis

Several types of experiments were performed to find the relative influence of various sampling parameters on measured gas composition. One of the first was done to find the short time variation of composition. In this case three samples were taken 15 minutes apart with the probe at a fixed location in the compressor duct. Table 1 shows the mean and standard deviation for the components in the three test samples. As may be seen, no significant variation in composition occurred over the half hour time period. This test was conducted primarily to ensure that the composition would remain stable while the influence of parameters other than time could be studied. Needless to say, the pressure and temperature of the gas over the duration of the experiments were monitored to determine if operating conditions also remained stable.

TABLE 1. MEAN AND STANDARD DEVIATION OF GAS SAMPLE COMPONENTS TAKEN OVER A 30 MINUTE PERIOD

Component #	Component Molecular Weight	Component Mole Percent	
		Mean (3 Tests)	Standard Deviation
1	2.01	11.62	0.06
2	16.04	24.30	0.06
3	26.03	0.32	0.01
4	28.05	26.77	0.12
5	30.07	6.97	0.10
6	40.06	0.17	0.01
7	42.08	12.71	0.04
8	44.09	0.58	0.01
9	54.09	3.10	0.01
10	54.10	4.39	0.03
11	58.12	0.43	0.01
12	66.12	0.80	0.01
13	68.12	2.07	0.04
14	70.13	1.27	0.02
15	72.15	0.24	0.01
16	80.14	0.47	0.03
17	84.16	0.26	0.01
18	86.17	0.29	0.00
19	88.12	0.17	0.01
20	78.11	1.90	0.08
21	92.14	0.70	0.03

Gas samples were taken from both the suction and discharge side on one of the stages; their analyses revealed them to be identical in composition as may be expected. An extensive series of tests was conducted to determine whether radial sampling location in the duct influenced measured composition. A probe was located 0.23m (9 in) from the wall of the 0.6m (23.5 in) duct on the second stage of the compressor. Samples were taken every 0.025m (1.0 in) as the probe was moved

outward toward the wall over a time period of 19 minutes. Columns (3) and (4) of Table 2 show the mean and standard deviation for the nine tests and Figure 2 shows the radial variation for three arbitrarily selected components having a low, a medium, and a high (relatively speaking) molecular weight. It may be seen that no systematic variation was found with radial location. Apparently, traverses for gas sampling purposes are not required in such a well-stirred mixture.

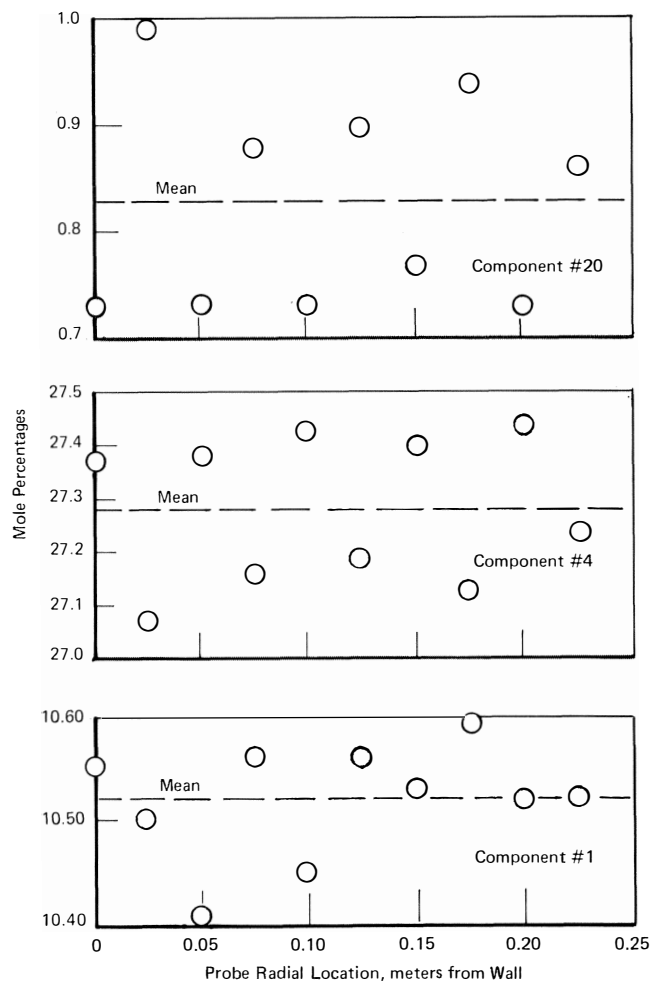


Figure 2. Gas Component Variation Across Duct Cross-Section.

The final test of sampling parameters was to determine whether in fact the probe was essential. One sample was taken directly from the process instrumentation wall tap. The analysis of the sample is shown in column (5) of Table 2. This analysis is compared with the analysis of the gas sample taken with the probe located just inside the duct. Column (6) shows the difference between these two analyses. It can be seen that in almost every instance differences in composition of the gas taken by these two methods are greater than the standard deviation of the nine tests at various radial locations — (column (4)). Generally, the sample from the wall tap is leaner in higher molecular weight components than is the sample from the probe. From this test, it is concluded that the wall tap sample which comes from the boundary layer in the duct is not a representative sample. A sample probe that projects into the gas stream at least 0.025 m (1.0 in) is necessary.

TABLE 2. GAS ANALYSIS RESULTS SHOWING EFFECT OF RADIAL LOCATION OF SAMPLE POINT

(1)	(2)	(3)	(4)	(5)	(6)
Component #	Molecular Weight	Data Mean (9 Tests)	Standard Deviation	Wall Tap (1 Test)	Difference*
1	2.01	10.52	0.06	10.64	0.11
2	16.04	23.71	0.13	23.92	0.27
3	26.03	0.24	0.01	0.21	-0.04
4	28.05	27.28	0.14	27.47	0.23
5	30.07	7.35	0.03	7.43	-0.09
6	40.06	0.17	0.01	0.17	0.00
7	42.08	14.58	0.06	14.62	0.06
8	44.09	0.77	0.01	0.78	0.03
9	54.09	3.88	0.03	3.81	-0.07
10	54.10	6.00	0.04	5.91	-0.09
11	58.12	0.35	0.01	0.37	0.02
12	66.10	0.74	0.03	0.69	-0.06
13	68.12	1.68	0.08	1.52	-0.03
14	70.13	1.22	0.04	1.14	-0.07
15	72.15	0.07	0.00	0.07	0.00
16	80.14	0.19	0.03	0.19	-0.03
17	84.16	0.15	0.01	0.13	-0.02
18	86.17	0.14	0.01	0.14	0.00
19	88.12	0.01	0.00	0.00	0.00
20	78.11	0.83	0.10	0.67	-0.19
21	92.14	0.10	0.02	0.07	-0.04

\* Data in this column are the difference between wall tap data and data from probe located just inside the duct wall.

## ANALYTICAL DEVELOPMENT

### Characterization of Compression Path

The method developed by Shultz [2] for representing the compression of real gases has been in use since its acceptance in the ASME Power Test Codes PTC-10 [3]. For the sake of discussion, a brief summary of this method is presented below. Consider a multi-impeller, adiabatic stage of compression of a real homogeneous gas. This compression may be represented on a P-V diagram as shown in Figure 3 where the state of the gas is known at the suction (inlet) and discharge ends.

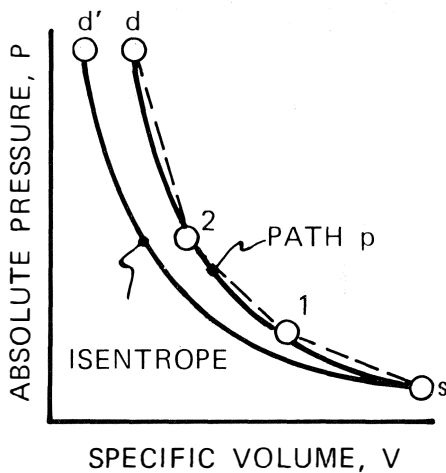


Figure 3. Pressure Volume Diagram for Gas Compression in a Centrifugal Compressor [2].

Let path  $p$  be a continuous curve connecting points  $s$  and  $d$ , and let curve  $sd'$  represent an isentropic path to the same discharge pressure. For a steady flow, adiabatic process, the net shaft work input  $W$  is given by

$$W = H_d - H_s + \frac{v_d^2 - v_s^2}{2g} \quad (3)$$

where  $H$  = gas enthalpy  
 $v$  = gas velocity.

Along path  $p$ , let  $W_p$  and  $Q_p$  be the net reversible mechanical work input and net reversible heat input, respectively. Then

$$W_p = \int_{P_s}^{P_d} V dP \quad (4)$$

and

$$\Delta H = H_d - H_s = Q_p + W_p \quad (5)$$

One measure of compressor efficiency can be defined by

$$\begin{aligned} \eta_p &= \frac{W_p}{W_p + Q_p} = \frac{W_p}{H_d - H_s} = \frac{\int_{P_s}^{P_d} V dP}{H_d - H_s} \\ &= \frac{\text{Reversible Work Input}}{\text{Enthalpy Rise}} \end{aligned} \quad (6)$$

If  $\eta_p$  is assumed to be a constant along p, then p is defined to be a path with a governing equation

$$\eta_p = V \frac{dP}{dH} \quad (7)$$

where  $\eta_p$  is that constant for which p passes through points s and d. It may be noted that along the isentropic path  $dS = 0$ . Because

$$dH = TdS + Vdp \quad (8)$$

it follows that  $\eta_p = 1$  along the isentrope — an expected conclusion.

*The only assumption inherent in the preceding analysis has been that of a constant efficiency along the compression path;* this appears to be a valid and logical assumption. However, if the state of the gas at intermediate points 1, 2 (i.e., between adjacent impellers) is not known, then it is further assumed that all impellers between points s and d have the same efficiency  $\eta_p$ . In order to determine the constant efficiency path, it can be shown that any one of the following equations must be integrated:

$$\frac{P}{T} \frac{dT}{dP} = m \quad (8a)$$

$$\frac{P}{V} \frac{dV}{dP} = -\frac{1}{n} \quad (8b)$$

$$\frac{P}{Z} \frac{dZ}{dP} = \frac{n-1}{n} - m \quad (8c)$$

where m and n are variables and are a function of the state of the gas along the compression path.

To integrate equations (8) directly would be tempting. However, it does not appear from the literature to have been done because the integration would require an equation of state representation of the gas medium as described later in this paper, which has not been generally available until recently. Reference [1] describes in detail how the above equations are combined with an equation of state representation of a hydrocarbon gas mixture. This permits a numerical integration of the equations and, therefore, the effect of the variation of m and n on the performance parameters can be examined.

The common assumption has been that m and n are constants. In that case, for example, equation (8b) appears in the familiar form

$$PV^n = \text{constant} \quad (9)$$

and n is then referred to as the polytropic volume exponent. In reference [1] it is further shown that the assumption of  $n = \text{constant}$  is generally valid for most practical applications. Now equation (9) can be used as the path equation and n,  $W_p$ , and  $\eta_p$  can be determined as

$$n = \frac{\log(P_d/P_s)}{\log(V_s/V_d)} \quad (10a)$$

$$W_p = f\left(\frac{n}{n-1}\right) [P_d V_d - P_s V_s] \quad (10b)$$

$$\eta_p = \frac{W_p}{H_d - H_s} \quad (10c)$$

In equation (10b), f is the polytropic head factor which is used therein to compensate for the variation of n along the compression path. It is calculated as follows:

$$f = \frac{H_{d_{is}} - H_s}{\frac{n_{is}}{(n_{is} - 1)} (P_d V_{d_{is}} - P_s V_s)} \quad (11a)$$

$$n_{is} = \frac{\log(P_d/P_s)}{\log(V_s/V_{d_{is}})} \quad (11b)$$

where  $H_{d_{is}}$  and  $V_{d_{is}}$  are the enthalpy and volume, respectively, of the gas at the discharge assuming an isentropic compression.

#### Thermodynamic Properties of Gas Mixtures

Equations (10) and (11) can now be used for evaluating the performance of a centrifugal compressor. However, it is necessary to calculate the enthalpy change ( $H_d - H_s$ ) incurred by the gas to determine the efficiency of compression. If the gas being compressed is a pure (single) entity, then it is a relatively straightforward procedure to determine  $\Delta H$  from, say, a Mollier diagram of the gas. However, this presents a difficulty for the case of hydrocarbon gas mixtures [3]. A method outlined in reference [4] utilizes a tabular procedure to obtain enthalpy, entropy, and specific heat of gas mixtures based on ideal gas properties of the constituents. Sources of concern are the need to interpolate from tables of ideal gas properties and the accuracy of the pseudo critical pressure and temperature corrections applied to the results. Moreover, automation of the procedure via a computer is complicated by the need to store a vast amount of data. The overall results, therefore, cannot be expected to be very accurate.

In the present work, an equation of state representation of the gas mixture (amenable to computer manipulation) was developed to calculate accurately the detailed thermodynamic properties required for the performance computation. Details will not be given here, but a data base consisting of pure component properties forms part of the overall computation scheme and the parameters supporting the equation of state have been verified experimentally.

The performance and properties evaluation relations were reduced to computer codes. Basically, input to this computer program for a compression stage consists of suction and discharge pressures and temperatures, the volume flow rate, and a gas composition analysis. The program is designed to obtain the required information from the user in an interactive manner.

#### APPLICATION OF TECHNIQUE

Figure 4 shows a typical input and output of the evaluation computer program; it is self-explanatory. For the purpose of the discussion that follows, the results of the computer program for the first and fourth stages of the subject compressor are tabulated in columns (3) and (6) of Table 3. It would be of interest to compare them with results obtained by other means such as:

1. Measured torque or horsepower into the compressor stage. This instrumentation is currently under preparation prior to installation on the compressor. Once it is operating, it will provide an accurate check of the analysis technique that has been developed.
2. The 'N' method of calculation. This method requires an estimate of the polytropic efficiency which might be

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THE THERMODYNAMIC PERFORMANCE EVALUATION PROGRAM FOR A COMPRESSOR
STAGE REQUIRES THE FOLLOWING INFORMATION:
1. AVERAGE SUCTION AND DISCHARGE PRESSURES
2. AVERAGE SUCTION AND DISCHARGE TEMPERATURES
3. AVERAGE PRESSURE FROM DR FLOW CELL ALONG WITH A CALIBRATION FACTOR
   TO CONVERT THE PRESSURE TO VOLUME FLOW RATE
4. GAS ANALYSIS

THE PROGRAM WILL PROMPT YOU FOR THE ABOVE INFORMATION:
COMPRESSOR IDENTIFICATION (6 ALPHANUMERIC CHARACTER): KOL300
STAGE NUMBER (IF ANY): 2
AVERAGE SUCTION PRESSURE (PSIG): 145.69
AVERAGE DISCHARGE PRESSURE (PSIG): 129.89
AVERAGE SUCTION TEMPERATURE (DEG F): 196.2
AVERAGE DISCHARGE TEMPERATURE (DEG F): 205.45
AVERAGE DR-CELL PRESSURE (PSI): 12.5
IS DR-FLOW ON (SUCTION) OR DISCHARGE (D) (DIBETS):
DR-CELL CALIBRATION FACTOR (FT CUBED/SECOND/PSI): 20

THE GAS ANALYSIS CAN BE INPUT TO THE PROGRAM BY ENTERING THE DRY MOLE
PERCENT ALONGSIDE THE GAS CONSTITUENTS AS THEY APPEAR BELOW. ALTERNATIVELY,
IF A PROPERLY CREATED CMS FILE CONTAINING THE GAS ANALYSIS INFORMATION
EXISTS THEN IT CAN BE USED BY ENTERING THE NAME OF THE FILE WHEN
PROMPTED TO DO SO.

DO YOU WISH TO ENTER YOUR GAS ANALYSIS FROM A CMS FILE?
NAME OF CMS FILE CONTAINING GAS ANALYSIS: GAS DATA

COMPRESSOR: KOL300
COMPRESSION (PROCESS) STAGE NO.: 2

          STAGE
          SUCTION  DISCHARGE

AVERAGE PRESSURE (PSIA)          60.29          144.54
AVERAGE TEMPERATURE (DEG F)      94.20          205.45
AVERAGE INTEGRATED DELTA PRESSURE (PSI) 12.5000
CALCULATED WEIGHT FLOW (LB/HR)    28241.7
MOLECULAR WEIGHT                   30.11
COMPRESSIBILITY FACTOR             0.971
GAS DENSITY (LB/FT**3)            0.3199
ENTHALPY (BTU/LB)                 177.28
ENTROPY (BTU/LB*F)                1.828
HEAT CAPACITY AT CONST PRESS (BTU/MOLE*F) 12.727
HEAT CAPACITY RATIO (C/P)        1.2104

SUMMARY OF RESULTS:
ISENTPROPIC EFFICIENCY             0.705
POLYTROPIC EFFICIENCY             0.742
POLYTROPIC HEAD (FT)              26345
GAS HORSEPOWER                    5067

DO YOU WISH TO RUN ANY MORE DATA?
-1
    
```

Figure 4. Typical Input/Output of Performance Evaluation Computer Program.

obtained from the manufacturer. Even with a clean original machine, the manufacturer's best estimate of the polytropic efficiency follows a shop test on a gas most certainly different from the mixed hydrocarbon gases used in the present application. In fact, it is the compression efficiency that is likely to degrade over time and must be determined (as in the present work) rather than assumed (as in the 'N' method). Clearly, the 'N' method of calculation cannot provide an independent check for the present technique.

Having evaluated the performance of the compressor, the question that is sometimes asked is "How does the present performance relate to the 'design' performance?" In order to attempt an answer to this question, information was obtained from the design data sheet for this compressor and placed into columns (1) and (4) of Table 3. It may be noted that the 'N' method of calculation was used at the time of designing the compressor utilizing an assumed efficiency of 0.758 and 0.718 for the first and fourth stages, respectively.

Because the computation methods in the present work differ from the 'N' method, columns (1) and (4) of Table 3 cannot be compared directly with columns (3) and (6). In order to permit a comparison, the 'design' pressure and temperature data along with the 'design' gas composition were used as input to the performance evaluation program of the present work and the results tabulated in columns (2) and (5) of Table 3.

As may be seen, there exists a discrepancy between columns (1) and (2) and (5) and (6) for the following parameters: compressibility factor, heat capacity ratio, polytropic head, and efficiency. Further, this discrepancy increases as one goes from

stage 1 to stage 4. In the case of stage 4, the heat capacity ratio increases as one goes from suction to discharge which is contrary to expectation. These discrepancies can be explained on the basis that there is a mismatch between the 'design' pressure/temperature data and the corresponding gas composition. In particular, it is suspected that the 'design' discharge temperatures were predicted to be lower than they should have been. This has been observed to be the case when the compressors are operated for the first time in the field. This will be documented in an example later. It should be mentioned, however, that the calculated head via the 'N' method and the one calculated by the present analysis will be in reasonable agreement as long as the compressor is actually operating near the nominal efficiency indicated by the manufacturer.

Primarily for the reason that all input data for a performance calculation must be consistently obtained, it is not practical to compare present compressor performance with 'design' performance. Rather, performance changes must be determined from similarly run performance tests. However, comparisons of such tests would require that the results be normalized in some manner. This aspect of the work is presently under investigation.

Finally, it may be noted that the present computer program can aid in the selection of a compressor stage. This selection process usually requires the calculation of the discharge temperature after the choice of a compressor stage has been made to ensure that it falls within a specified range. The program can be run iteratively with different values of the discharge temperature (see Figure 4) until the calculated efficiency corresponds to the desired chosen nominal efficiency. This approach can produce a closer match between predicted discharge temperatures and those measured in the field after the compressor is installed. An example appears in Figure 5 which shows the discharge temperatures for a newly installed five-stage compressor obtained via a) field measurement, b) predicted via the 'N' method and c) predicted via the present technique. In each of the latter two cases, the prediction was accomplished using the stage efficiency provided by the manufacturer shown in the inset on the Figure. The inset also includes the *actual efficiency* calculated by the present analysis from measured conditions. It may be seen that the discharge

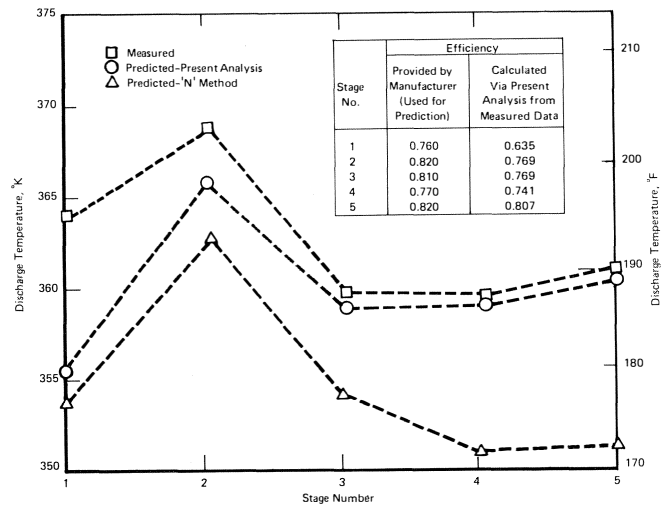


Figure 5. Measured and Predicted Discharge Temperatures for a New Five-Stage Centrifugal Compressor.

TABLE 3. COMPARISON OF PERFORMANCE PARAMETERS

		STAGE 1			STAGE 4		
		(1)	(2)	(3)	(4)	(5)	(6)
		'Design' Data Sheet	'Design' Data Current Program	Current Data Current Program	'Design' Data Sheet	'Design' Data Current Program	Current Data Current Program
Pressure kPa (psia)	Suction	155.82 (22.6)	155.82 (22.6)	180.71 (26.21)	1630.6 (236.5)	1630.6 (236.5)	1827.0 (265.0)
	Discharge	362.65 (52.6)	362.65 (52.6)	409.88 (59.45)	3860.9 (560.0)	3860.9 (560.0)	4103.6 (595.2)
Temperature °K (°F)	Suction	310.9 (100.0)	310.9 (100.0)	307.9 (94.5)	297.1 (75.0)	297.1 (75.0)	291.9 (65.71)
	Discharge	360.9 (190.0)	360.9 (190.0)	363.5 (194.72)	343.2 (158.0)	343.2 (158.0)	357.0 (182.9)
Molecular Weight		30.94	30.95	30.60	27.63	27.69	25.85
Compressibility Factor	Suction	0.991	0.989	0.987	0.916	0.891	0.890
	Discharge	0.986	0.984	0.982	0.830	0.836	0.871
Enthalpy kJ·kg <sup>-1</sup> (Btu·lbm <sup>-1</sup> )	Suction	NA NA	408.4 (175.6)	411.9 (177.1)	NA NA	390.3 (167.8)	399.9 (171.9)
	Discharge	NA NA	495.2 (212.9)	511.0 (219.7)	NA NA	447.0 (192.2)	503.1 (216.3)
Heat Capacity Ratio Cp/Cv	Suction	1.161	1.197	1.1966	1.147	1.3203	1.3422
	Discharge	1.148	1.180	1.1767	1.126	1.3794	1.3399
Polytropic Head kJ·kg <sup>-1</sup> (ft·lbf/lbm <sup>-1</sup> )		76.01 (25430)	77.51 (25931)	73.33 (24533)	75.62 (25300)	69.77 (23341)	66.97 (22405)
Polytropic Efficiency		.758	0.896	0.740	.718	1.230	0.649

temperatures predicted by the present analysis are much closer to the measured ones than those predicted by the 'N' method. The one exception is the first stage and the discrepancy occurs because the actual operating efficiency of that stage is significantly lower than that predicted by the manufacturer based upon test stand data; therefore, the measured temperature is higher than predicted.

## SUMMARY

A complete experimental and calculational procedure has been described for measuring the thermodynamic performance of centrifugal compressors in chemical plant service. These procedures have to be different from those used in shop tests because:

1. the existing instrumentation used for process control is generally not sufficiently accurate so must be supplemented, and
2. rather than dealing with a single component gas whose thermodynamic properties are well characterized, one is often compressing gas mixtures whose compositions are not even well known.

From the experimental standpoint a pitot tube measuring system was developed which could be inserted safely into a pressured duct and which made use of laboratory grade transducers to measure local gas stream temperature and static

pressure as well as dynamic pressure for gas stream velocity determination. The probe also permitted withdrawing gas samples directly from the main gas stream to heated sample flasks thereby avoiding condensation of high molecular weight components.

Computationally, equation of state computer codes were developed which permitted determination of gas enthalpy changes from suction to discharge of each stage using measured pressures, temperature, and gas compositions. This combined experimental-computational procedure allows realistic estimates to be made on a stage-by-stage basis of shaft work input and compressor efficiency. By means of several examples it was shown that this newly developed capability is useful not only in checking a manufacturer's prediction of compressor performance in actual service conditions, but also in monitoring performance in time for the purposes of adjusting process conditions for most economical operation and of noting equipment degradation due to fouling or seal leakage.

## ACKNOWLEDGEMENT

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## APPENDIX

### DETAILS OF PROBE AND ITS INSERTION DEVICE

Figure A-1 shows the probe and its insertion device, and a schematic representation of the probe appears in Figure A-2. The probes which are commercially available have an outside diameter of 0.0095 m (3/8 in) and are about 1.07 m (42 in) long. The sensor end of the probe is wedge-shaped and has ports which are connected through internal passages to a static and differential pressure transducer at the end that remains outside the duct. The probe tip also has an embedded thermocouple for temperature measurement. The insertion device is used to position the probe in the ducts. This insertion device and probe were hydrostatically tested to 17.2 MPa (2500 psi) and operated under pressurized gas at that same pressure. Gas samples can be obtained through the static pressure ports of the probe using sample lines heated by a steam tracer and a heated sample flask to prevent condensation of high molecular weight constituents.

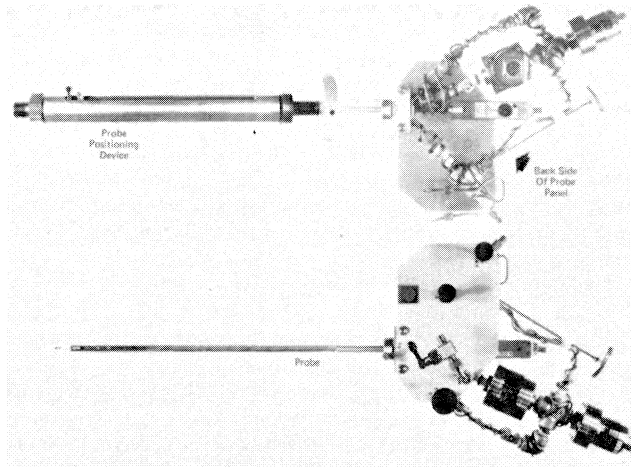


Figure A-1. Pressure, Temperature and Gas Sampling Probe and Its Insertion Device.

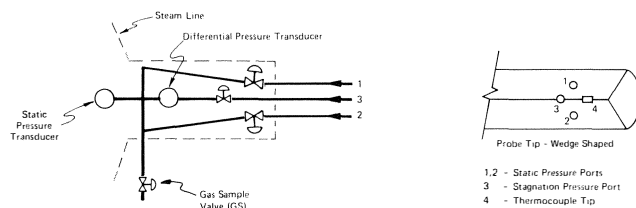


Figure A-2. Schematic Representation of Pressure, Temperature and Gas Sampling Probe.

The probe tip along with the external tube and internal passages is essentially a pitot tube. The structural integrity of the long probe was assured to withstand safely any drag forces arising from the flow field to which it would be subjected. The probe is extended into the duct in a cantilever mode and it is essential to ensure that its natural frequencies of lateral vibration were not in the neighborhood of those induced by vortex shedding. The natural frequencies were determined theoretically as well as experimentally. In the former case, the probe was treated as a cantilevered tube with the contribution from the internal tubes also being considered. In the experimental approach, the probe was mounted in a manner that would simulate the actual test set-up and the probe was set into a lateral motion which was measured and then transformed to the frequency domain using a real time analyzer. It was determined that the Karman vortex shed frequencies were at least an order of magnitude higher than the first natural frequency of the tube. Consequently, any significant flow induced lateral vibration of the probe would not occur.

The static and differential pressure transducers attached to the end of the probe were individually calibrated in the laboratory prior to their use in the field. The embedded thermocouple in the probe tip (copper-constantan Type T) was also suitably calibrated. The differential (velocity) pressure indicated by the pressure transducers was multiplied by a calibration factor to correct for the wedge shape of the tip; these factors were calculated from those provided by the probe manufacturer. It may be noted that the dashed line in Figure A-2 represents a steam line which is used to heat the internal passages of the probe in order to prevent condensation of the gas constituents as mentioned earlier; the lines are heated only when the probe is used to obtain gas samples.

### NOMENCLATURE

- $f$  = Polytropic head factor, see equation (11)  
 $g$  = Acceleration of gravity,  $\text{m}\cdot\text{s}^{-2}$  ( $\text{ft}\cdot\text{s}^{-2}$ )  
 $k$  = Ratio of specific heat capacities ( $C_p/C_v$ )  
 $m$  = See defining equation (8a)  
 $n$  = Polytropic exponent (see also defining equation (8b))  
 $p$  = Path followed by gas in compression process  
 $r$  = Variable in radial direction of compressor duct, m(ft)  
 $v$  = Gas velocity,  $\text{m}\cdot\text{s}^{-1}$  ( $\text{ft}\cdot\text{s}^{-1}$ )  
 $w$  = Mass flow rate,  $\text{kg}\cdot\text{s}^{-1}$  ( $\text{lb}\cdot\text{hr}^{-1}$ )  
 $C_p$  = Specific heat capacity at constant pressure,  $\text{kJ}\cdot\text{kg}^{-1}\cdot^\circ\text{K}^{-1}$  ( $\text{Btu}\cdot\text{lbm}^{-1}\cdot^\circ\text{R}^{-1}$ )  
 $C_v$  = Specific heat capacity at constant volume,  $\text{kJ}\cdot\text{kg}^{-1}\cdot^\circ\text{K}^{-1}$  ( $\text{Btu}\cdot\text{lbm}^{-1}\cdot^\circ\text{R}^{-1}$ )  
 $H$  = Gas enthalpy,  $\text{kJ}\cdot\text{kg}^{-1}$  ( $\text{Btu}\cdot\text{lbm}^{-1}$ )  
 $P$  = Absolute pressure, Pa ( $\text{lb}\cdot\text{in}^{-2}$ )  
 $Q_p$  = Net reversible heat input,  $\text{kJ}\cdot\text{kg}^{-1}$  ( $\text{Btu}\cdot\text{lbm}^{-1}$ )  
 $R$  = Gas constant,  $\text{kJ}\cdot\text{kg}^{-1}\cdot^\circ\text{K}^{-1}$  ( $\text{ft}^2\cdot\text{s}^{-2}\cdot^\circ\text{R}^{-1}$ )  
 $S$  = Gas entropy,  $\text{kJ}\cdot\text{kg}^{-1}\cdot^\circ\text{K}^{-1}$  ( $\text{Btu}\cdot\text{lbm}^{-1}\cdot^\circ\text{R}^{-1}$ )  
 $T$  = Absolute temperature,  $^\circ\text{K}$  ( $^\circ\text{R}$ )  
 $V$  = Specific volume,  $\text{m}^3\cdot\text{kg}^{-1}$  ( $\text{ft}^3\cdot\text{lbm}^{-1}$ )  
 $W$  = Net shaft work in steady flow adiabatic process,  $\text{kJ}\cdot\text{kg}^{-1}$  ( $\text{Btu}\cdot\text{lbm}^{-1}$ )  
 $W_p$  = Net reversible mechanical work input,  $\text{kJ}\cdot\text{kg}^{-1}$  ( $\text{Btu}\cdot\text{lbm}^{-1}$ )  
 $Z$  = Compressibility factor  
 $\eta_p$  = Efficiency along path  $p$   
 $\rho$  = Gas density,  $\text{kg}\cdot\text{m}^{-3}$  ( $\text{lbm}\cdot\text{ft}^{-3}$ )



*Subscripts*

s = Suction

d = Discharge

is = Isentropic

p = Along path p — see Figure 3

0 = Stagnation

