Computer Program Development To Analyse Centrifugal Compressor Performance

by

Wan Abdul Halim bin Wan Rahim

(1516)

Dissertation submitted in partial fulfilment of

the requirements for the

Bachelor of Engineering (Hons)

(Mechanical Engineering)

MAY 2004

Universiti Teknologi PETRONAS
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CERTIFICATION OF APPROVAL

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Approved by,

(Dr. Chalilullah Rangkuti)

UNIVERSITI TEKNOLOGI PETRONAS
TRONOH, PERAK
MAY 2004

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

(WAN ABDUL HALIM BIN WAN RAHIM)

ABSTRACT

A centrifugal compressor, which is driven by a gas turbine, is one of the critical equipment used in oil and gas industry especially in the offshore platform (upstream operation) and need a regular performance monitoring. The compressor is used in a variety of applications including gas lift, gas injection, transmission, boosting and gas sales. The performance test of turbo-compressor requires the accurate determination of the efficiency, flow, head, power and losses. The objective of this project is to develop a computer program (TCCalc V1.0) that has the ability to calculate and analyse the centrifugal compressor performance characteristics.

This program provides an easy and useful ways to track or estimate centrifugal compressor performance from manufacturer supplied performance curves. Since the manual calculation using the individual head versus capacity curves to predict the overall performance of a multiple body tandem is a time consuming trial-and-error calculation, the development of this computer program is an alternative to overcome this problem. The study scopes of this project are the working principle of a compressor, thermodynamic behavior of the gases, performance tests, calculation methods, data analysis, performance curves and the computer program development itself.

The methodology used in this project including the literature review, plant visit, calculations and also the computer and software applications. At the end, the calculation results of the program have to be compared with the real operating data in order to test the accuracy of the program.

ACKNOWLEDGEMENT

Thanks to Allah for His kindness and mercy upon us giving me the opportunity to execute and accomplish this challenging project, TCCalc Development Program. Though quite a tough project but with the help, comments, praises and constructive criticism from these people, this project finished successfully.

I would like to take this opportunity to thank these people:

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ABBREVIATIONS AND NOMENCLATURES

A	Area
a	Speed of sound
BHP	Brake or shaft horsepower
C	Discharge coefficient
c_p	Specific heat at constant pressure
c_v	Specific heat at constant volume
D	Pipe diameter
d	Throat, or impeller diameter
Eff	Efficiency
GHP	Gas horsepower
H	Head
HP	Horsepower
h	Enthalpy
K	Flow meter coefficient
k	Adiabatic exponent
MW	Molecular weight
$\dot{M}$	Mass flow
N	Speed, RPM
n	Polytropic exponent
P	Static pressure
P_c	Critical pressure
P_r	Reduced pressure
P_T	Total pressure
PF	Power factor
PWR	Power
Q	Flow rate
q	Heat transfer
R	Gas constant
Re	Reynolds number
r_p	Pressure ratio (P_2/P_1)
s	Entropy
SHP	Shaft horsepower
T	Temperature
T_c	Critical temperature
T_R	Reduced temperature (T/T_c)
U	Tip speed
u	Internal energy
V	Velocity
v	Specific volume
W	Work
Y	Flow meter expansion factor
Y_a	Adiabatic expansion factor
Z	Compressibility factor
z	Vertical height

GREEK LETTERS

β	Throat (or orifice) to pipe diameter ratio
η	Efficiency
γ	Work coefficient
μ	Head coefficient
μ'	Absolute viscosity
ν'	Kinematic viscosity
ρ	Density
ϕ	Flow coefficient

SUBSCRIPTS

ad	Adiabatic process
p	Polytropic process
S	Standard conditions
1	Inlet conditions
2	Discharge conditions

CHAPTER 1

INTRODUCTION

1.1 BACKGROUND OF STUDY

A centrifugal compressor is a mechanical device that is designed to compress gas from one pressure to a higher pressure by either squeezing the same gas volume into a smaller volume or accelerating the gas and rapidly decelerating the gas flow hence in the process increases the gas pressure. Gas compressors are commonly used in the oil and gas company for gas sales, gas lifting and gas injection. As one of the critical equipment, about 50 to 60% of the company's production depends on the availability of these machines [8]. Therefore, the performance and the reliability of these gas compressors are very crucial in order to avoid potential production impacts.

Currently, the manufacturer of the compressor provides the performance curve for the individual compressor. The curves are the predictions of compressor performance based upon the operating conditions of suction temperature, gas composition and pressure (either suction or discharge) are assumed to remain constant. So, if the curve is used to predict performance for other than these base conditions, some inaccuracy may occur. In order to plot a new performance curve and do the performance test, the company has to pay a lot of money to the compressor's manufacturer because the computer program used to calculate the performance is not included in the package. Hence, by doing the similar program (TCCalc V1.0) using Visual Basic as the programming software, the user can easily calculate the compressor performance in the user-friendly windows.

1.2 PROBLEM STATEMENT

Currently, most of the centrifugal compressor user especially oil and gas company engages with the third party company (commonly the compressor's manufacturer to do the field performance test and analysis on the site when a new compressor is installed. After that, the reliability engineers will monitor closely the compressor performance behavior in order to predict whether the compressor needs to be overhauled or not.

The maintenance strategy for a compressor is quite different than a gas turbine that has its own schedule of planned preventive maintenance. From the economic point of view, corrective maintenance strategy is profitable for this compressor rather than a planned preventive maintenance. The manual performance calculation with the individual head versus capacity curves to predict the overall performance of a multiple body tandem is a time-consuming trial-and-error calculation. Thus, the development of this computer program is an alternative to overcome this problem. So, there is a need for the reliability engineer to have a computer program to calculate and analyse the compressor performance base on the operating data in order to predict the compressor's behavior.

1.3 OBJECTIVE

The aim of this project is to develop a computer program that has an ability to calculate and analyse the centrifugal compressor's performance. As the centrifugal type gas compressor is commonly used in oil and gas industry for the gas sales, gas lifting and gas injection application in the offshore platforms, this computer program will concentrate only on the centrifugal type compressor.

1.4 SCOPE OF STUDY

The early part of this project concentrated more on the problem identification and understanding via a desk study and research. Besides to be able to use the Visual Basic software and Microsoft Excel worksheet in the development of this computer program, this project is basically also towards the identification and understanding of the working principle of a centrifugal compressor, thermodynamic behavior of the gases, performance tests, calculation methods, data analysis and also the performance curves.

In order to write the programming code, the manual calculations was done first with the verification by the supervisor. After accomplishing the manual calculations, all the equations, formulas and data will be transformed into the programming codes in the Visual Basic software. This software will perform the calculation and iteration of the programming codes by using the required operating input. Before that, the user interfaces have to be designed in order to simplify the usage. Then, comparing the results with the real operating data in the platforms or any related plants will test the accuracy and feasibility of the program.

The overall planned activities for this project are based on the Gantt chart as shown in Appendix 1.

CHAPTER 2

LITERATURE REVIEW

2.1 THEORY

Operation of a centrifugal compressor is based on basic principle of thermodynamic. During compression, work is done on a fluid to raise its pressure. As the gas enters the impeller, the velocity energy is added to the gas by rapidly rotating impeller. After leaving the impeller, the gas with high velocity enters a diffuser, which is a stationary component. The gas velocity slows down resulting in an additional pressure increase. After leaving the diffuser, the gas either exits the compressor or enters the next impeller for multi stage compression.

The energy added to the gases through rotating impeller referred to as 'head'. The compressor produces a certain head in order to accomplish the required pressure ratio. Total head produced by a compressor is a function of several factors such as impeller diameter, rotative speed and number of stages. Figure 2-1 below shows a typical centrifugal compressor for petroleum, chemical and gas service industries with the nomenclature (API Standard 617).

The amount of head required from a compressor to increase the gas pressure is a function of available compressor suction pressure, discharge pressure required by the process, suction gas temperature, gas composition, ratio of specific heats, compressibility factor and the efficiency. In order to understand the operation of the centrifugal compressor, the basic of gas law is used.

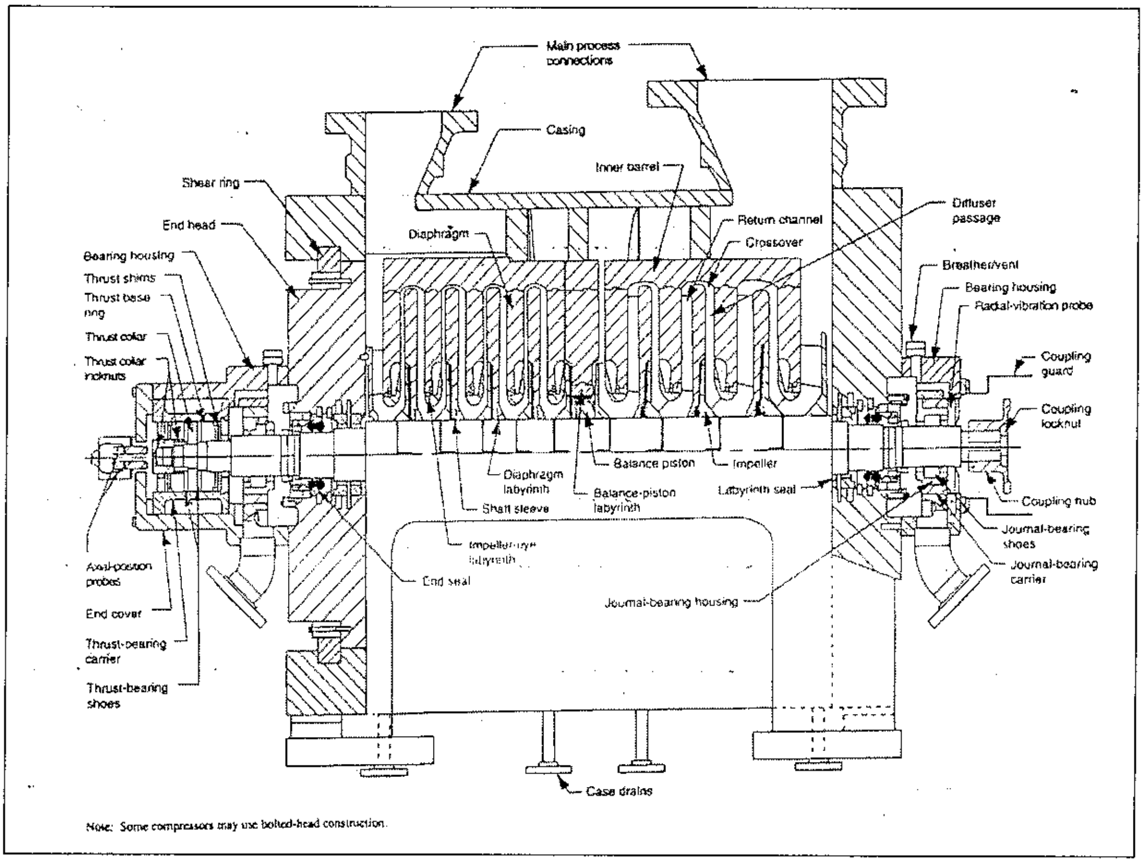

Figure 2-1: *Typical Centrifugal Compressor Nomenclature*

An adiabatic or isentropic process is defined as a process in which no heat transfer takes place. Temperature is not a constant here but there is no heat transferred into or out of the system. Compressor manufacturers normally use the term adiabatic to mean adiabatic isentropic (constant entropy). A polytropic process is a variable entropy process with heat transfer can take place. Adiabatic and polytropic process would be reversible if the efficiency were 100% (Lapina, Ronald P., 1982, p.21). Adiabatic process is a special case of the more general polytropic process:

$$Pv^n = \text{constant}.$$

where:

n = k (ratio of specific heats, C_p/C_v) for an adiabatic, isentropic process

n = 1 for an isothermal process (constant temperature)

n = 0 for an isobaric process (constant pressure)

n = ∞ for an isometric process (constant volume)

Familiarization with the basic components of the centrifugal compressor and an awareness of the available compressor arrangements are necessary for an optimum compressor estimate (Lapina, Ronald P., 1982, p.1). Centrifugal compressors are manufactured with two types of casings: the horizontally split casing and the vertically split or barrel casing.

2.2 CALCULATION FORMULA

Basically, all the formulas used in the gas compressor manual performance calculations are the thermodynamics formulas and taken from Leon Sapiro (1996) and Lapina, Ronald P. (1982). The sample of the performance calculation is shown in APPENDIX 2.

2.3 CENTRIFUGAL COMPRESSOR PERFORMANCE CURVE

In order to understand the performance curve of a compressor, let consider the simple model in Figure 2-2 below. Figure 2-2 shows a compressor taking gas at atmospheric pressure and discharging it into a receiver. The compressor and receiver are both initially at atmospheric pressure, since they are in equilibrium with their surroundings.

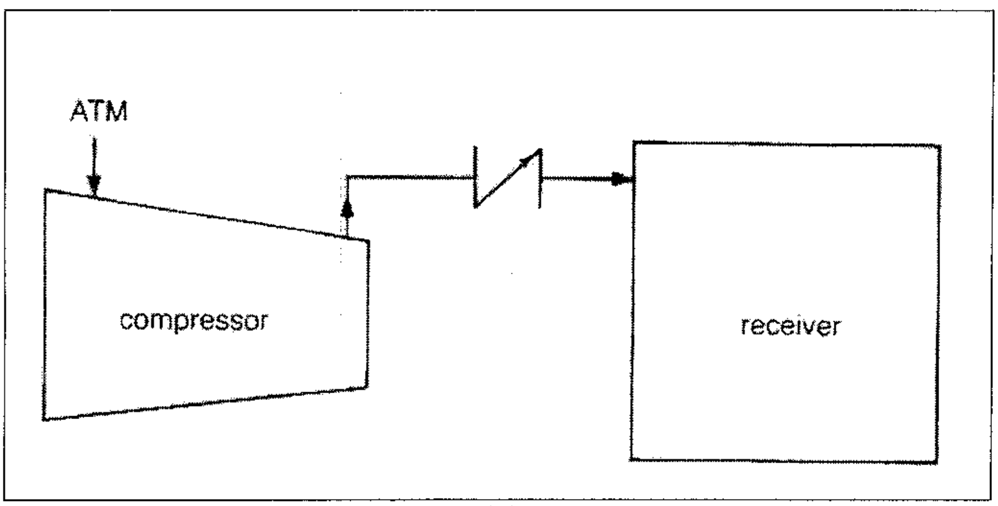

Figure 2-2: *Compressor delivering gas into a receiver*

When the compressor starts, the compressor inlet and outlet/ discharge pressures will be equal because of the prior equilibrium. Since the discharge pressure equals to the inlet pressure, there is no resistance to flow (with the assumption there is no pipe friction), and the pressure rise from inlet to discharge is zero.

As a result, the head produced by the compressor is zero. This point is plotted on Figure 2-3 as Point 1. As the mass of air in the receiver starts to increase, the pressure in the receiver will start to rise, providing some resistance. At first, the flow will drop very slightly. This point is shown on Figure 2-3 as Point 2 which is called the stonewall point, since the curve beyond this point is essentially a straight line.

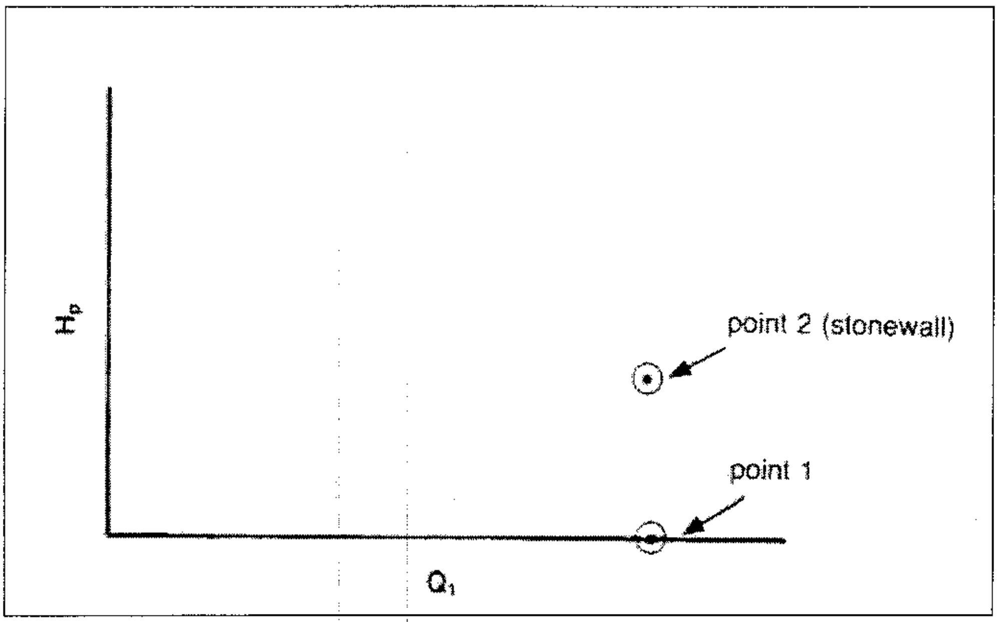

Figure 2-3: *Compressor volume flow with no resistance-stonewall*

As a mass, of air in the receiver continues to increase, the pressure also increases, providing greater pressure differential from inlet to discharge of the compressor and, at the same time, a greater resistance to flow. This means it is becoming more difficult to cram the air into the receiver. This is shown as Point 3 on Figure 2-4 and is a typical compressor operating point.

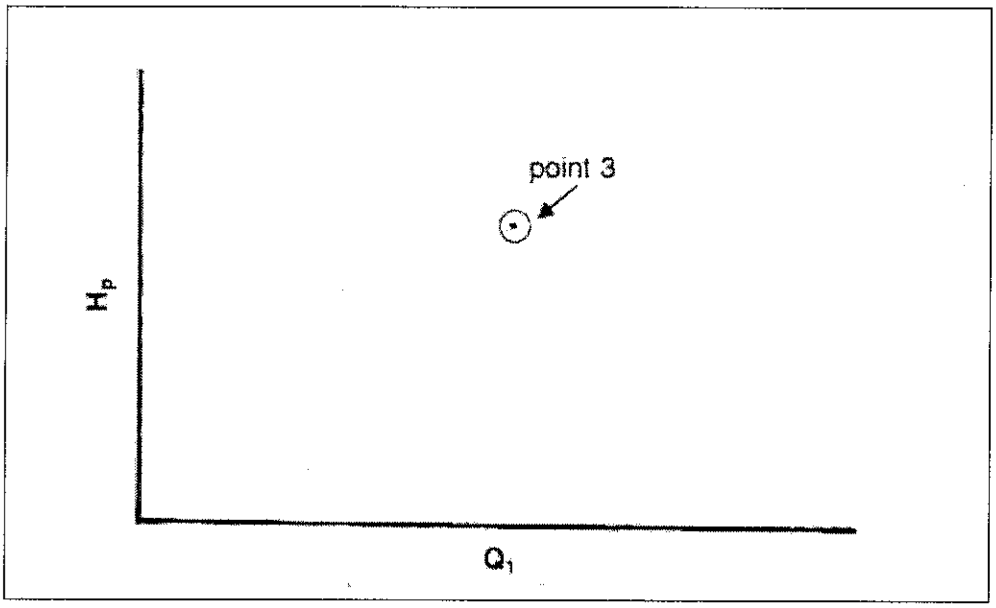

Figure 2-4: *Compressor volume flow with some resistance.*

As the mass of gas in the receiver increases further, a pressure is eventually obtained, above which the compressor cannot pump stably. This point, shown as Point 4 on Figure 2-5 is called the surge point.

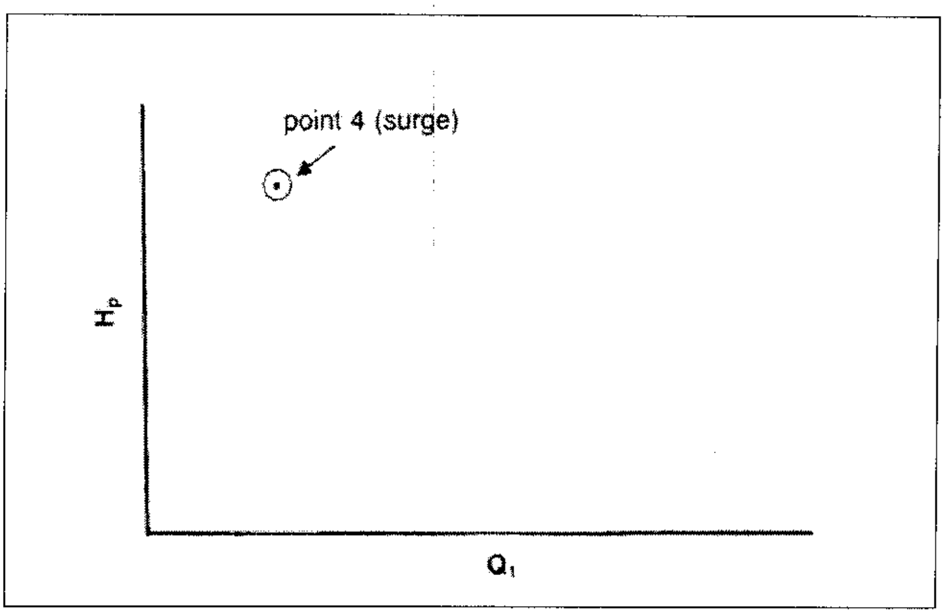

Figure 2-5: *Compressor volume flow at maximum resistance-surge*

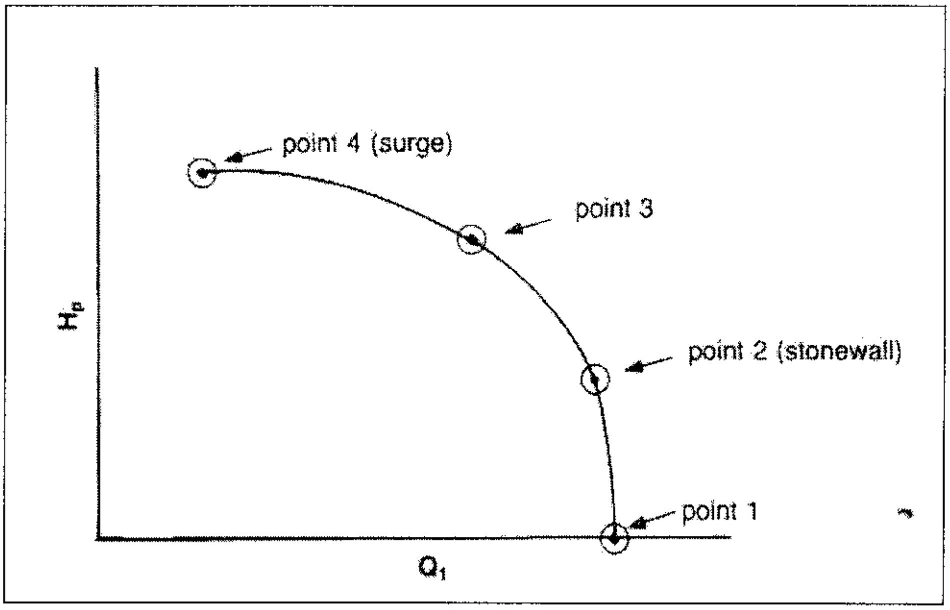

Figure 2-6: *The centrifugal compressor characteristic curve*

If all these points are plotted on a single curve and connected, the results as the centrifugal compressor characteristic curve shown in Figure 2-6. In the analysis, the three important points are the stonewall point (Point 2), the operating point (Point 3), and the surge point (Point 4). The software in this project will calculate the operating point of the compressor with the input given. Referring to Lapina, Ronald P., 1982, p.108, the stonewall point is the maximum stable compressor flow point. Consequently, it is the minimum head point under stable compressor operation. Flow increases beyond stonewall are minimal at best; operation beyond the stonewall point is unpredictable because of the vertical slope of the curve in this area. Manufacturers will usually stop their curve as far as they can reasonably predict performance.

The surge point is the minimum stable flow point and the highest head point. As the pressure in the receiver continues to increase, the volume flow rate into the receiver becomes less and less and therefore the velocity of the gas going through the compressor and into the piping continues to decrease. When the velocity becomes too slow, the compressor can no longer perform stably. Since the compressor pumping action is unstable, a flow reversal can result. With flow reversal, the volume in the receiver decreases and so does the pressure. The compressor can then, once again provide positive flow and will pump back into the receiver. This back-and-forth flow motion is called 'surging' and can be very damaging to the compressor. In severe cases, surging can cause destruction. While no one can predict exactly the location of the surge point because of manufacturing tolerances and several other factors, manufacturers have a fairly good handle on its location and will generally draw performance curves right to predicted surge. Precaution must be taken to prevent operation at the surge point. This is normally accomplished by recycling flow from the discharge back to the inlet of the compressor. A good rule of thumb is to start the recycle process when the flow through the compressor reaches 110% of the surge flow capacity. This approach essentially yields a 10% safety factor to allow for machining tolerances, instrumentation and valving lag time.

2.4 CURVE PARAMETERS

It is important for us to know the right curves with the right parameters for our reference in order to predict compressor operating condition. The most universal parameter to plot as the independent variable is the inlet, or actual, volume flow. There are several parameters that normally be plotted against inlet volume flow:

- Head (polytropic or adiabatic)
- Discharge pressure
- Power requirement
- Efficiency (polytropic or adiabatic)
- Pressure ratio
- Pressure rise
- Discharge temperature

Compressor manufacturers can furnish any combination of these performance curves. However, the most common are the first four.

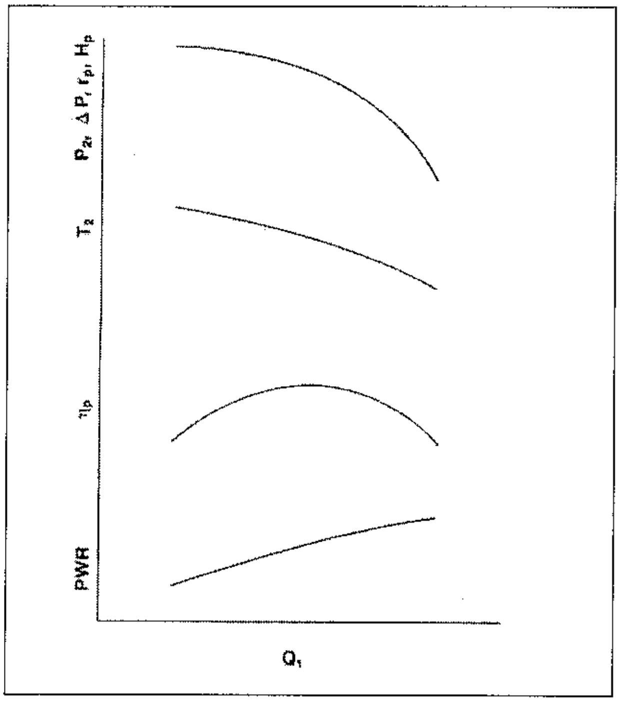

Figure 2-7: *Variations of several parameters with inlet volume flow.*

2.5 FAN LAWS

The head is proportional to the square of the mechanical tip speed:

$$H_p = \frac{\mu u^2}{\text{constant}}$$

The mechanical tip speed is directly proportional to the rotational speed through the following equations:

$$u = \frac{N\pi d}{720} \quad \text{in English system}$$

$$u = \frac{N\pi d}{6\times 10^4} \quad \text{in the metric system}$$

Where:

N = rotational speed, RPM

π = pi (3.1416)

d = diameter, in (English); mm (metric)

Therefore, the head is proportional to the square of the rotational speed:

$$H_p \propto N^2$$

and:

$$\frac{H_{p_n}}{H_{p_o}} = \left(\frac{N_n}{N_o}\right)^2 \quad \text{where: } n = \text{new}, \ o = \text{original} \qquad \text{..........................(1)}$$

The flow through an impeller, and therefore the flow through the compressor, is directly proportional to the rotational speed;

$$Q \propto N$$

and:

$$\frac{Q_n}{Q_o} = \frac{N_n}{N_o} \qquad \text{..........................(2)}$$

If the flow is allowed to increase proportionally to the speed and, at the same time, the head is allowed also to increase proportionally to the speed squared; the resultant power requirement will increase with the cube of the speed:

$$\frac{PWR_n}{PWR_o} = \left(\frac{N_n}{N_o}\right)^3 \qquad \text{..........................(3)}$$

Equation (1), (2), and (3) are known as the fan laws. They are used extensively in compressor performance calculations. For the single stage compressors, the fan law relationships are very accurate. As the number of stages increases, the accuracy of the fan laws deteriorates. We can generally assume that the fan laws will provide very good approximations up to $\pm 10\%$ of the rated speed (Lapina, Ronald P., 1982, p.114).

2.6 SYSTEM RESISTANCE

Consider a process whereby the gas entering the compressor is initially at atmospheric pressure. The gas goes through a system of piping, exchanger and vessels for extraction of a certain product and leaves the system at atmospheric pressure. For this type of process, the compressor is used only to elevate the gas pressure sufficiently to overcome pressure drops due to the piping, exchangers and vessels. These system pressure drops collectively are known as 'system resistance'. The system resistance, ΔP of a system normally varies with the square of the volume flow:

$$\Delta P \propto \Delta Q^2$$

Figure 2-8 shows a series of compressor curves for various rotational speeds. Superimpose is a system resistance curve. The only possible operating points for the compressor are those points where the compressor performance curves intersect the system resistance line. These are the only points that satisfy both the compressor and the system. Operation at other than these points under steady-state conditions is impossible. A lower flow has been obtained by decreasing the speed of the compressor. For constant speed drives, the same results could be obtained by throttling the inlet pressure. By a similar analysis, higher flows can be obtained by increasing the speed of the compressor. There are limits to speed increases. The American Petroleum Institute (API) requires that the compressor be capable of 105% of rated speed. This is the value that should be used for speed limitations. The low-end speed limitation is set by process requirements and a consideration of the critical speeds of the unit. A compressor should never be operated within 20% of any critical speed.

With constant-speed drives, flow increases above rated are usually impossible, since there are normally no ways to increase the suction pressure. Therefore, when purchasing constant-speed compressors, precaution must be taken to purchase sufficient polytropic head for the highest possible operating point. It would appear that it is impossible to surge a properly rated compressor operating in a closed system governed by system resistance. However, as the process operates over a period of time, it may begin to foul the system, thereby increasing the system pressure drop. If sufficient fouling occurs, it is entirely possible to drive the compressor into surge unless protected by recycle. With increased fouling, it also becomes increasingly more difficult to deliver rated capacity.

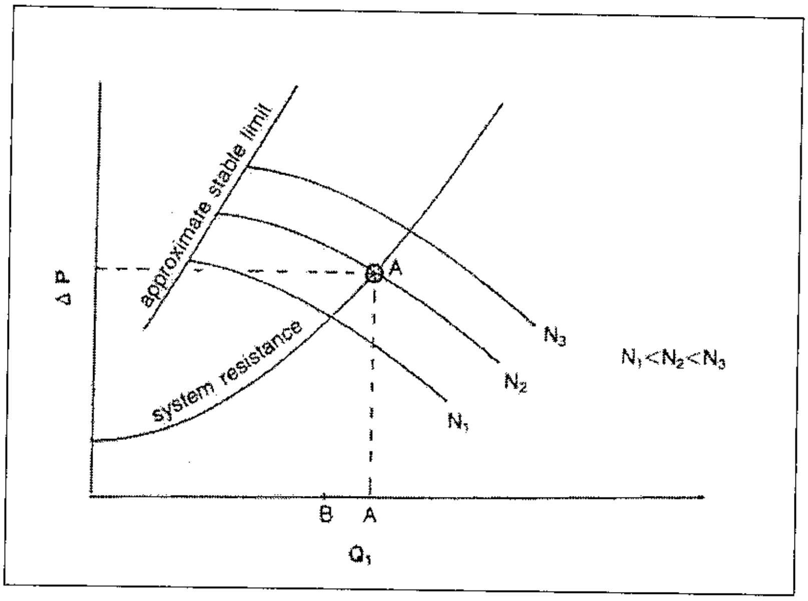

Figure 2-8: *Typical system resistance curve with compressor at rated flow.*

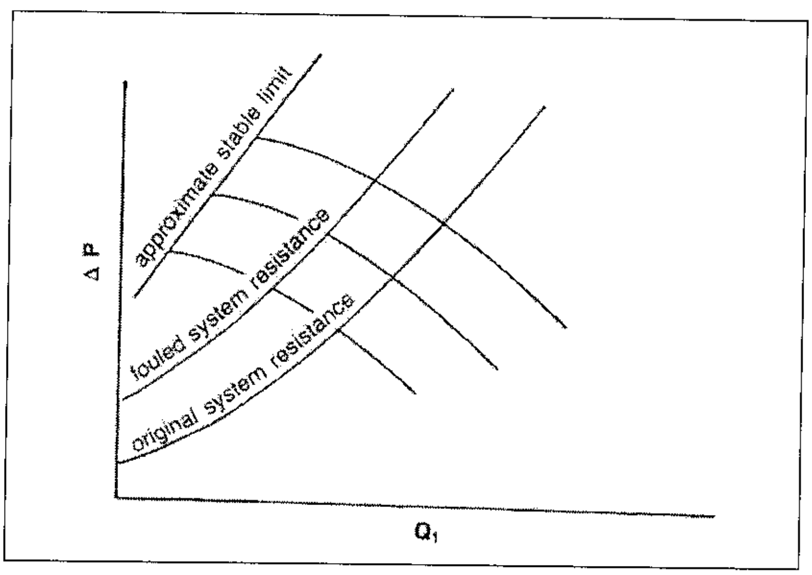

Figure 2-9: *Change in system resistance due to system fouling*

CHAPTER 3

METHODOLOGY

3.1 CALCULATION STEPS

When the appropriate formulas and equations needed are gathered from the literature reviews and some references, a systematic calculation approaches are used to obtain the final performance characteristics and properties of the compressor. Besides, the power input of the compressor's train taken from the gas turbine also can be determined. A sample of the calculation steps is shown in APPENDIX 2.

3.2 COMPUTER AND SOFTWARE APPLICATIONS

In order to complete the development of this project, a few computer programs and software are used initially for familiarization and training purpose especially on the Visual Basic 6.0 environment. Normally, before the program codes are developed, there is a need to come out with the program algorithm. However, while using the Visual Basic software, it is compulsory to design the user interface that can be synchronized with the input and output information needed in the calculation part.

Referring to Wallace Wang (1998), Writing a Visual Basic program requires nine steps:

1. Decide what you want the computer to do
2. Decide how the program will look on the screen (the appearance of the program is its user interface).

3. Draw the user interface using common parts such as windows, menus, and command buttons.
4. Define the name, color, size, and appearance of each user interface object
5. Write instructions in BASIC to make each part of the program do something.
6. Run the program to see if it works
7. Determine the bugs or errors when the program doesn't work perfectly
8. Fix any errors or bugs in the program
9. Repeat Steps 6 through 8 over and over again until the program really works.

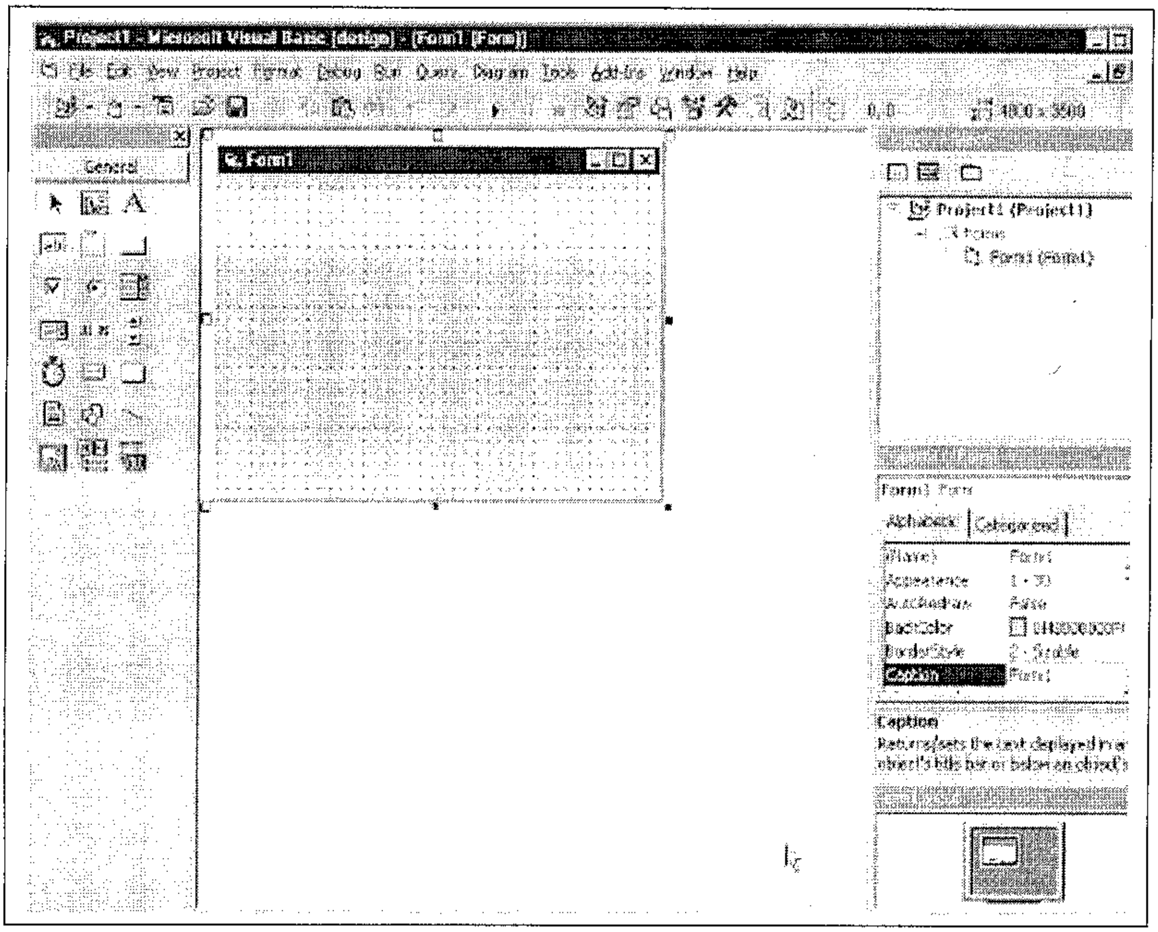

Figure 3-1: *The Visual Basic Integrated Development Environment presents a unified programming work area.*

Beside of using Visual Basic software, this project also uses Microsoft Excel worksheet program in order to draw the performance graphs and see the operating envelope of the compressor.

3.3 TESTING AND RESULT ANALYSIS

After the calculation part of the software is done, by using the real field data, the results are compared and analyzed in order to check the feasibility of the software itself. By doing this, the percentage of error produced by the software can be predicted either in the range that can be accepted or not for the betterment of the software. However, the most important thing here is to ensure that the software can be used in the real field environment.

CHAPTER 4

RESULTS AND DISCUSSION

Basically, the project concentrates more on the software application of Visual Basic 6 especially on the design of the user interface and the BASIC codes. The interfaces are as shown below:

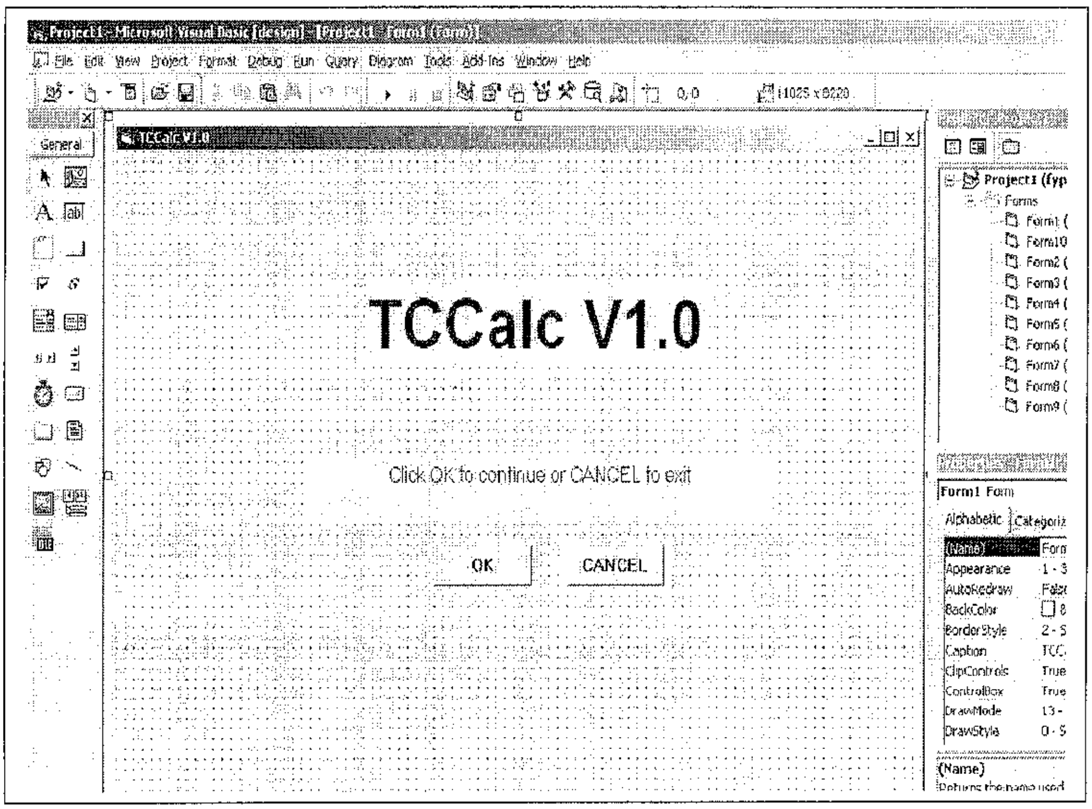

Figure 4-1: *TCCalc V1.0 Front Interface*

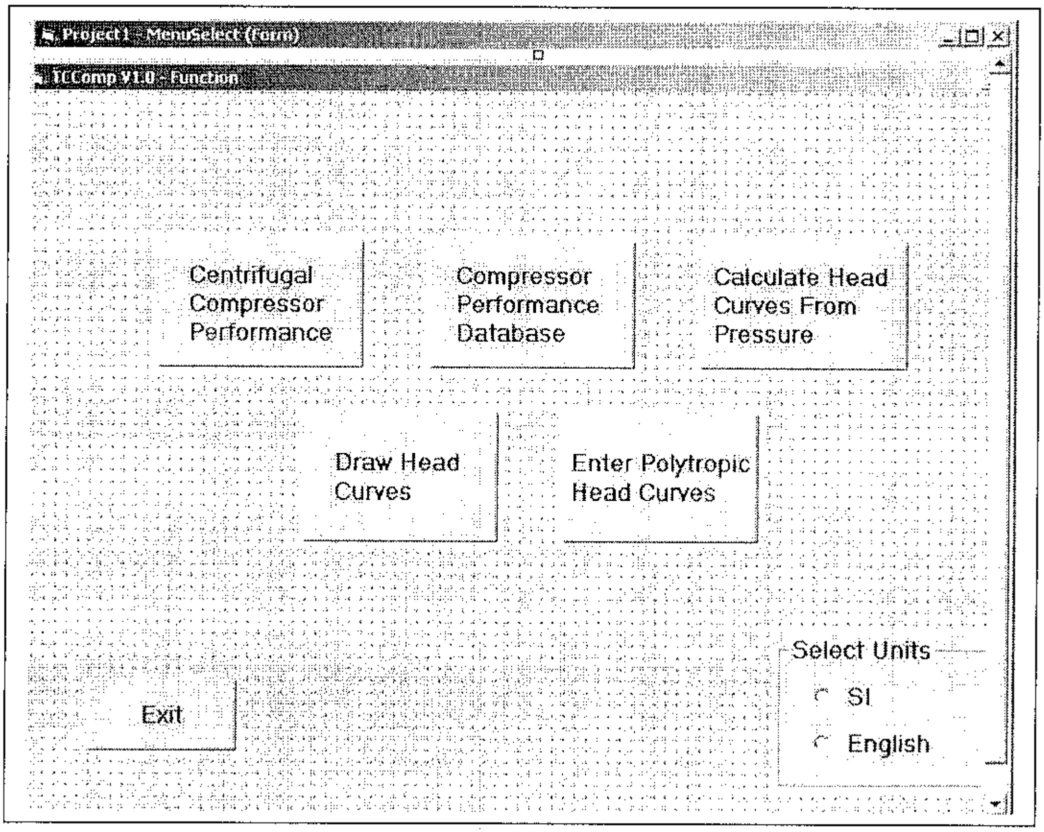

Figure 4-2: *TCCalc Menu Interface*

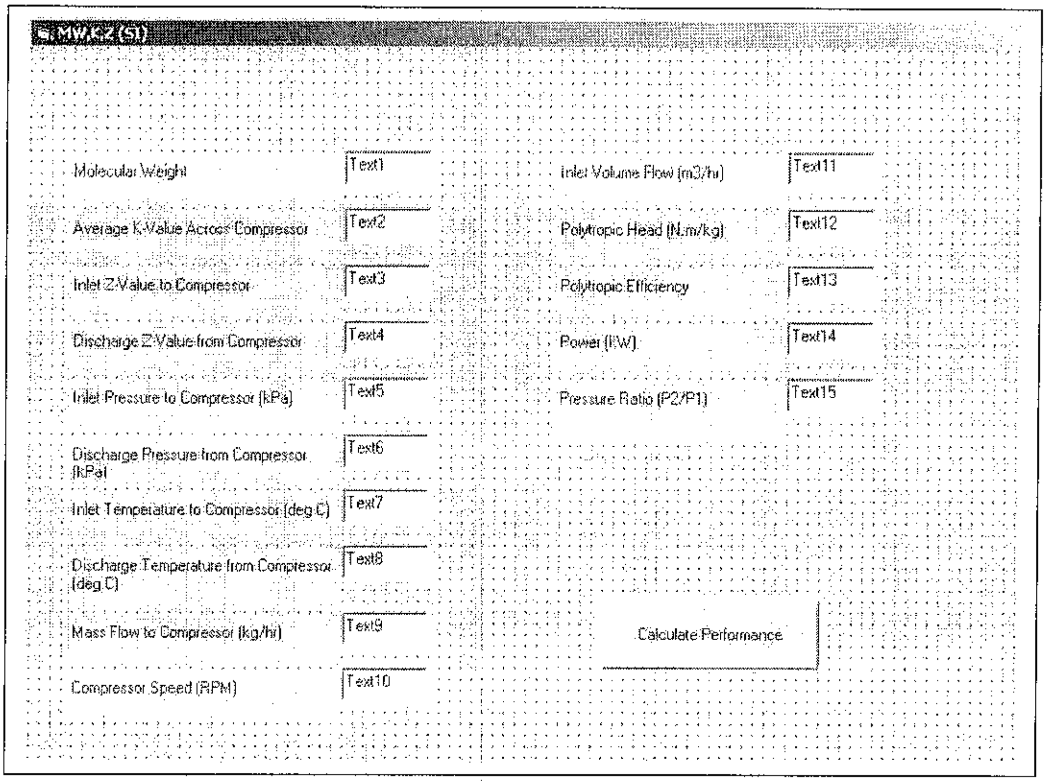

Figure 4-3: *One of TCCalc Input / Output Interfaces*

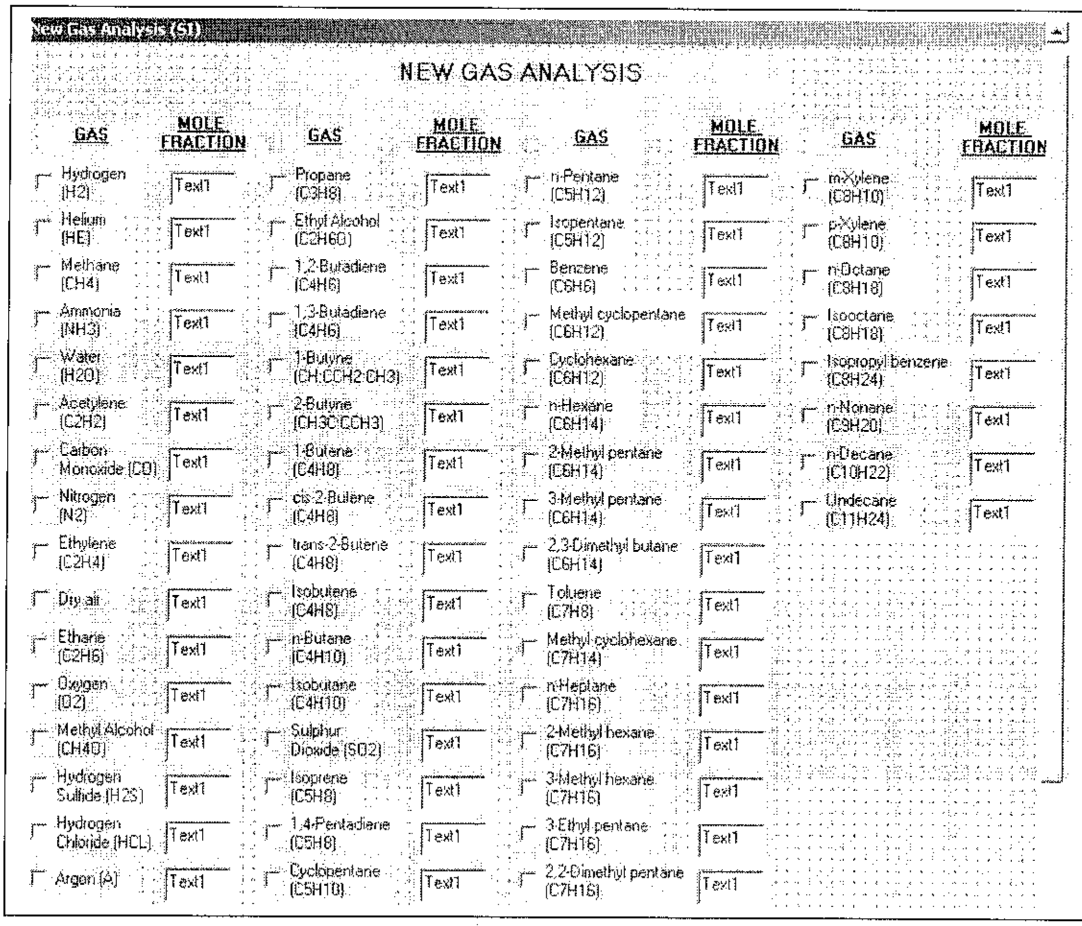

Figure 4-4: *TCCalc Gas Analysis*

4.1 RESULTS FROM THE PROGRAM

In order to check the feasibility of TCCalc program, two set of data from Pulai-A platform are used to compare with the calculation result of this program. Table 4-1 and Table 4-2 below show the centrifugal compressor performance field data.

4.1.1 Field Data

Table 4-1: *Pulai-A Compressor Field Performance Data (SI Unit)*

	LP Compressor	**HP Compressor**
Molecular Weight	23.5396	23.5396
Inlet Pressure (kPa)	1345.00	3852.31
Outlet Pressure (kPa)	3990.31	11721.00
Inlet Temperature (deg. C)	46.0	46.0
Outlet Temperature (deg. C)	149.1	165.3
K_1	1.258	1.258
K_2	1.214	1.208
Z_1	0.9711	0.9133
Z_2	0.9704	0.9441
Isentropic Head (J/kg)	134560	130695
Polytropic Head (J/kg)	139302	137110
Capacity (m^3/hr)	1812.6	548.2
Standard Flow (MMSCFD)	19.00	17.50
Power (HP)	1549	1593
Speed (RPM)	20509	20509
Pressure Ratio	2.967	3.043
Isentropic Efficiency	0.722	0.629
Polytropic Efficiency	0.748	0.660
Mass Flow (kg/hr)	22282.23	20523.20

Table 4-2: *Pulai-A Compressor Field Performance Data (Metric Unit)*

	LP Compressor	**HP Compressor**
Molecular Weight	23.5396	23.5396
Inlet Pressure (psia)	195.08	558.73
Outlet Pressure (psia)	578.75	1699.99
Inlet Temperature (deg. F)	114.8	114.8
Outlet Temperature (deg. F)	300.3	329.5
K_1	1.258	1.258
K_2	1.214	1.208
Z_1	0.9711	0.9133
Z_2	0.9704	0.9441
Isentropic Head (ft-lbf/lbm)	45019	43725
Polytropic Head (ft-lbf/lbm)	46605	45871
Capacity (ACFM)	1066.7	322.6
Standard Flow (MMSCFD)	19.00	17.50
Power (HP)	1549	1593
Speed (RPM)	20509	20509
Pressure Ratio	2.967	3.043
Isentropic Efficiency	0.722	0.629
Polytropic Efficiency	0.748	0.660
Mass Flow (lbm/min)	817.91	753.30

4.1.2 TCCalc Performance Result

Based on this real field data as reference, TCCalc program have computed the performance properties as shown in Figure 4-5 and Table 4-3 below. For the calculation and analysis purpose, only LP Compressor is used in the result discussion.

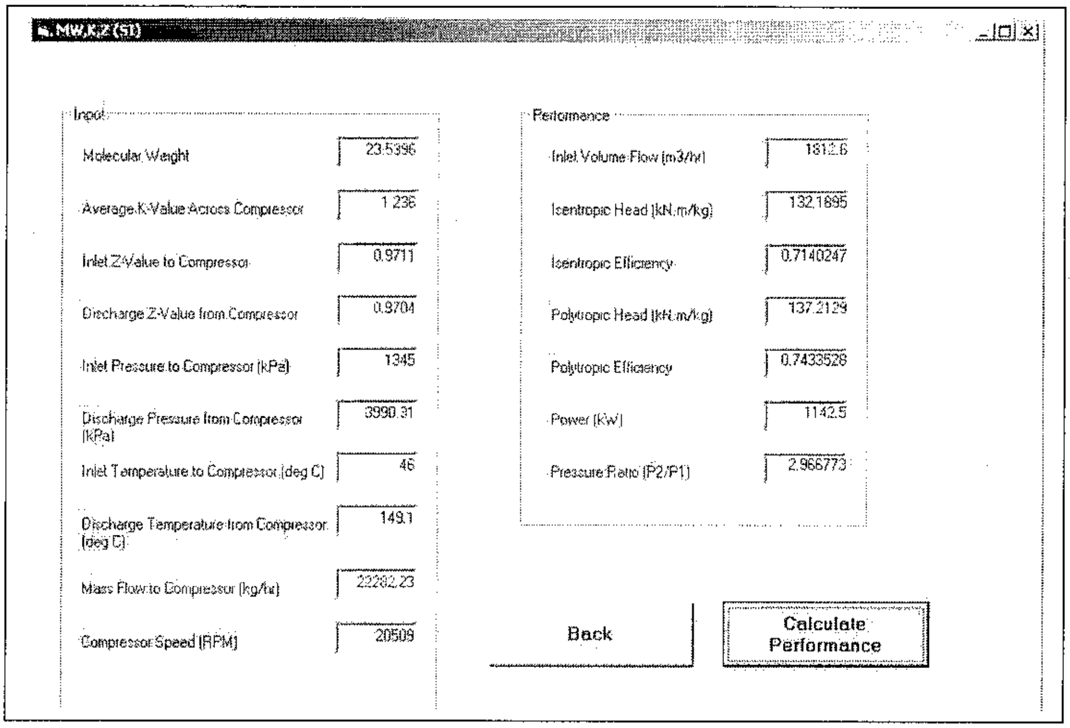

Figure 4-5(a): *Window's Interface for Pulai-A LP Compressor Performance (SI Unit)*

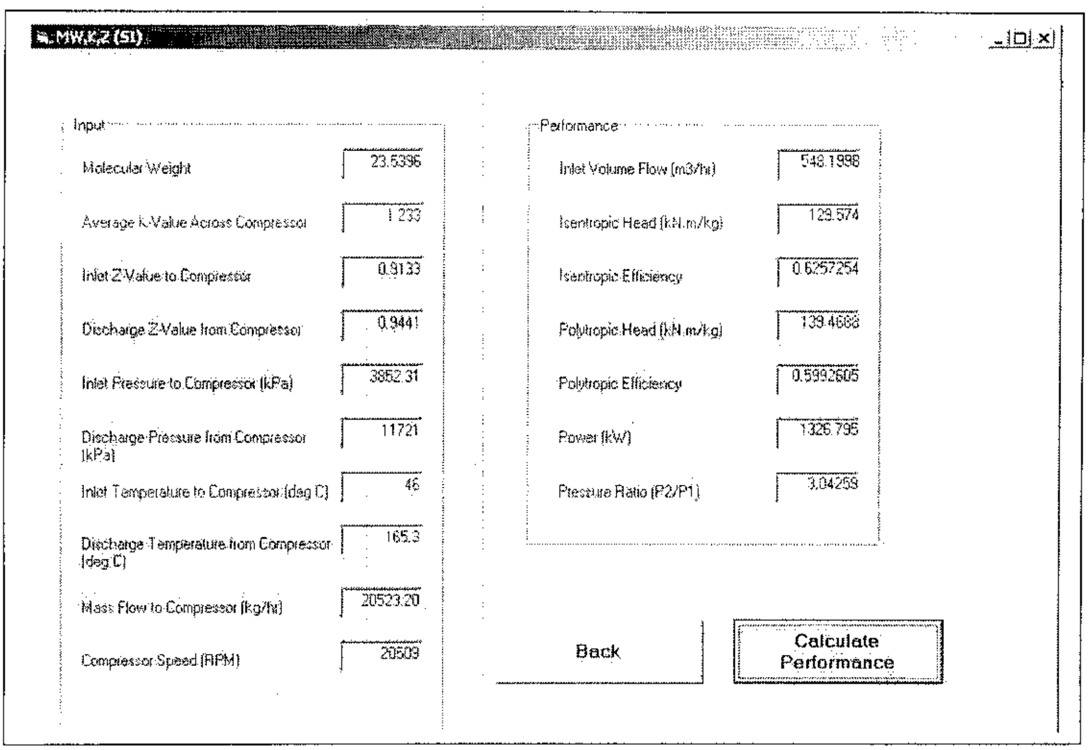

Figure 4-5(b): *Window's Interface for Pulai-A HP Compressor Performance (SI Unit)*

Table 4-3: *Performance Calculation Result for Both HP and LP Compressor (SI Unit)*

	LP Compressor	**HP Compressor**
Inlet Volume Flow (m^3/hr)	1812.6	548.19
Isentropic Head (kN.m/kg)	132.190	129.574
Isentropic Efficiency	0.714	0.626
Polytropic Head (kN.m/kg)	137.213	139.469
Polytropic Efficiency	0.743	0.599
Power (kW)	1142.5	1326.8
Pressure Ratio	2.967	3.043

4.1.3 Microsoft Excel Worksheet

After all the performance results are calculated using TCCalc program, the data is compared to the manufacturer's design performance curve using Microsoft Excel worksheet. By plotting the performance curves, user can estimate the operating envelope of that particular compressor either in an optimum area or under designated performance area. Hence, the calculated performance data becomes very significant here instead of just only calculated results. A sample of plotted performance curves is shown in Figure 4-6 below.

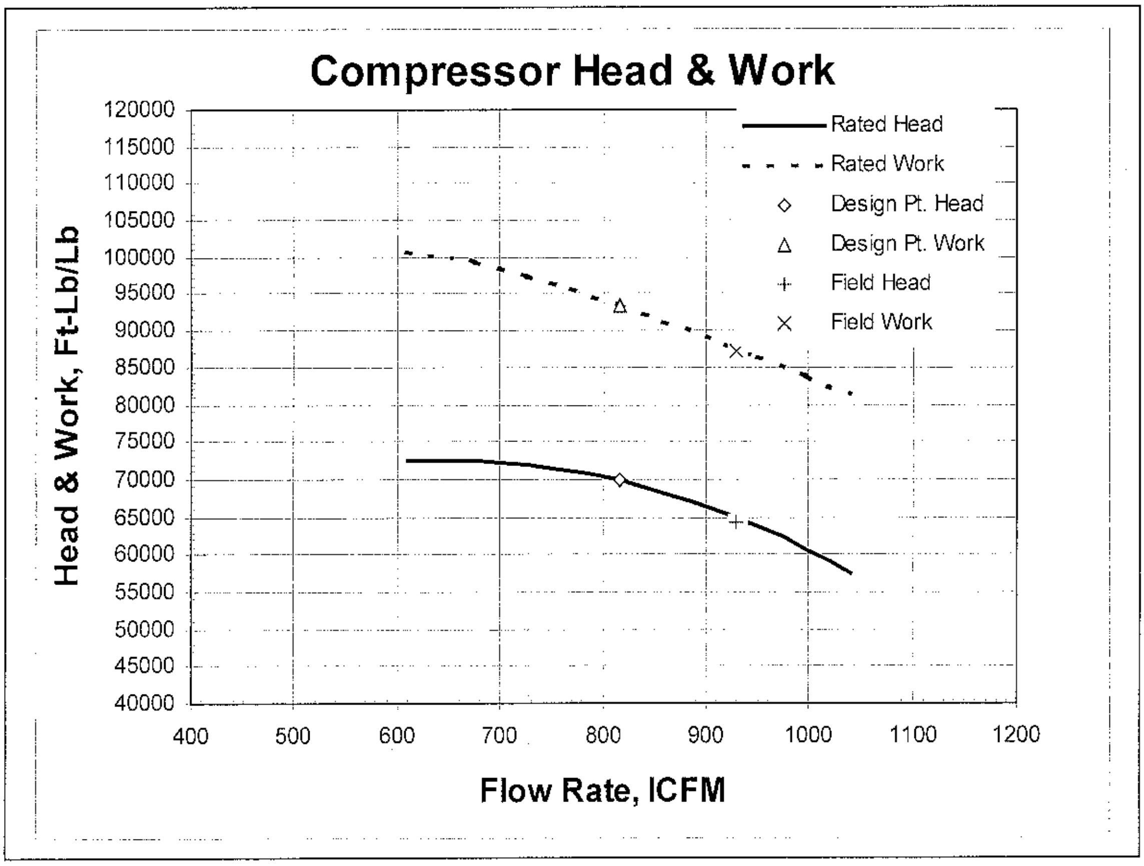

Figure 4-6: *Sample of Plotted Performance Curves Using Microsoft Excel Worksheet (Head and Work versus Flow Rate)*

4.2 DISCUSSION

Based on the field data from Pulai-A platform (LP Compressor), an error analysis of the TCCalc calculation results are analyzed. The sample calculation of errors is shown below:

$$\text{Percentage of Error (Isentropic Head)} = \frac{134{,}560 - 132{,}190}{134{,}560} \times 100 = 1.76\%$$

$$\text{Percentage of Error (Isentropic Efficiency)} = \frac{0.722 - 0.714}{0.722} \times 100 = 1.11\%$$

$$\text{Percentage of Error (Polytropic Head)} = \frac{139{,}302 - 137{,}213}{139{,}302} \times 100 = 1.50\%$$

$$\text{Percentage of Error (Polytropic Efficiency)} = \frac{0.748 - 0.743}{0.748} \times 100 = 0.67\%$$

$$\text{Percentage of Error (Power)} = \frac{1549 - (1142.5 \times 1.341)}{1549} \times 100 = 1.09\%$$

Table 4-4: *Percentage of Error for TCCalc Performance Calculation (LP Compressor)*

	Percentage of Error
Isentropic Head	1.76
Isentropic Efficiency	1.11
Polytropic Head	1.50
Polytropic Efficiency	0.67
Power	1.09

For this centrifugal compressor part, the error analysis is conducted by comparing the Pulai-A LP Compressor field data with TCCalc calculation results. By referring to the results (Table 4-4), the overall percentage of error for each of the LP Compressor performance properties are less than five percent and can be accepted. This error is due to the gas sample analysis from lab that is for the current one while the performance data is for year 1997. However, it does not give a major impact to the result of the calculation. By using the plotted performance curves in Excel worksheet, user can estimates the current operating area of particular compressor in order to predict the performance level of the compressor. If the compressor's performance point is under the rated line, some actions like corrective maintenance or overhaul should be taken to correct it and make it works on the desired operating envelope.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

The field data needed for the compressor performance calculation by the program are:

1. Gas analysis report from lab (Molecular weight and gas density is provided)
2. Inlet pressure, P_1
3. Inlet temperature, T_1
4. Discharge pressure, P_2
5. Discharge temperature, T_2
6. Compressor speed, N in RPM
7. Compressibility, Z
8. Adiabatic exponent, K
9. Mass Flow

Based on the error analysis, since the percentage of error is below 2%, it can be concluded that the program is feasible to use for compressor performance prediction with sufficient information of field data is supplied. Basically, from the results, a new performance curve has to be plotted in order to compare with the manufacturer-supplied curve. Then, the current operating envelop of the compressor can be determined either in an optimum condition or over the stable limit. In order to do this, a Microsoft Excel worksheet is prepared and attached with this program.

For the recommendation part, future works can be plan to combined the gas turbine and gas compressor performance calculation into a program. A database for future reference also can be programmed. The Excel worksheet also can be linked in the main program and the user does not need to reenter the calculated data. Further study and work on the calculation for flow analysis and compressibility factor without referring to the graph also an interesting part to improve this program.

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APPENDICES

Appendix 1-1	Project Milestone For The First Semester of The Project
Appendix 1-2	Project Milestone For The Second Semester of The Project
Appendix 2-1	Sample Calculation
Appendix 2-2	Sample of Gas Properties Calculation
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Appendix 4	Pulai-A Performance Data
Appendix 5	Pulai-A Performance Curves
Appendix 6	Bekok-A Compressors Performance Test Report
Appendix 7	Performance Evaluation of Centrifugal Compressors (By F.M. Odom, Solar Turbines)

Appendix 1-1

Project Milestone For The First Semester of The Project

APPENDIX 1-1

Project Milestone For The First Semester of The Project

No.	Detail/ Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	SW	EW
1	Selection of Project Topic																
	-Propose Topic																
	-Topic assigned to students																
2	Preliminary Research Work																
	-Introduction																
	-Objective																
	-List of references/literature																
	-Project planning																
3	Submission of Preliminary Report				15/8												
4	Project Work																
	-Reference/Literature																
	-Thermodynamic / compressor performance calculation																
5	Submission of Progress Report									22/9							
6	Project work continue																
	-Visual Basic familiarization																
	-Design the program																
7	Submission of Interim Report Final Draft													20/10			
8	Oral Presentation																
9	Submission of Interim Report																21/11

SW Study Week
EW Exam Week

Appendix 1-2

Project Milestone For The Second Semester of The Project

APPENDIX 1-2

Project Milestone For The Second Semester of The Project

No.	Detail/ Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	SW	EW
1	Project Work Continue																
	-Programming codes development																
2	Submission of Progress Report 1				●												
3	Project Work Continue																
	-Debug/ test the program																
4	Submission of Progress Report 2									●							
5	Project work continue																
	- Debug/ test and formalize the program																
6	Submission of Dissertation Final Draft													●			
7	Oral Presentation															●	
8	Submission of Project Dissertation																●

SW Study Week

EW Exam Week

Appendix 2-1

Sample Calculation

APPENDIX 2-1

SAMPLE CALCULATION:

1. Using the gas analysis,

   Mixture compositional : Molecular weight, M_{wt} = ________
   Density (kg/m^3) = ________

2. Field data:

   $P_{suction}$ = ________ psig → $P_{suction}$ = ________ psia
   $T_{suction}$ = ________ °F → $T_{suction}$ = ________ °R

   $P_{discharge}$ = ________ psig → $P_{discharge}$ = ________ psia
   $T_{discharge}$ = ________ °F → $T_{discharge}$ = ________ °R

3. Compositional properties:

Composition	% m_{wt}	P_{cr} psia	T_{cr} °R	% P_{cr} psia	% T_{cr} °R
Methane					
Ethane					
Propane					
Iso- butane					
N- butane					
Iso- pentane					
n- pentane					
Hexane plus					
Nitrogen					
Carbon dioxide					

4. Calculate reduce pressures, P_r and reduce temperature, T_r :

$$P_{r\ suction} = \frac{P_{suction}}{P_{cr}} = \_\_\_\_\_\_\_\_$$

$$P_{r\ discharge} = \frac{P_{discharge}}{P_{cr}} = \_\_\_\_\_\_\_\_$$

$$T_{r\ suction} = \frac{T_{suction}}{T_{cr}} = \_\_\_\_\_\_\_\_$$

$$T_{r\ discharge} = \frac{T_{discharge}}{T_{cr}} = \_\_\_\_\_\_\_\_$$

5. Estimate the compressibility factors, Z from curve

$Z_{suction}$ = ______ $Z_{discharge}$ = ______

6. Calculate constant pressure specific heat, C_p

Composition	**% m_{wt}**	**C_p (Btu/lbm-mol.°R)**	**% C_p (Btu/lbm-mol.°R)**
Methane			
Ethane			
Propane			
Iso- butane			
N- butane			
Iso- pentane			
n- pentane			
Hexane plus			
Nitrogen			
Carbon dioxide			

7. Calculate gas constant, R_o

$$R_o = \frac{R}{m_{wt}} = \_\_\_\_\_\_$$

8. Calculate isentropic exponent, k

$$k = \frac{c_p}{c_v}$$

English unit,

$$k = \frac{c_p}{c_p - 1.986} = \_\_\_\_\_\_$$

Metric unit,

$$k = \frac{c_p}{c_p - 8.314} = \_\_\_\_\_\_$$

9. Calculate isentropic efficiency, η_{isen}

$$\eta_{isen} = T_{suction}\left[\frac{\left(\frac{P_{discharge}}{P_{suction}}\right)^{\frac{k-1}{k}} - 1}{T_{discharge} - T_{suction}}\right]$$

$=$

10. Calculate isentropic head, H_{isen}

$$H_{isen} = R_o \times Z_{av} \times \frac{k}{k-1} \times T_{suction} \times \left[\left(\frac{P_{discharge}}{P_{suction}}\right)^{\frac{k-1}{k}} - 1\right]$$

$=$

11. Calculate polytropic efficiency, η_p

$$\eta_p = \frac{\ln\left[\left(\frac{P_{discharge}}{P_{suction}}\right)^{k-1/k}\right]}{} \div \ln\left[\left(\frac{T_{discharge}}{T_{suction}}\right)\left(\frac{Z_{discharge}}{Z_{suction}}\right)\right] = \_\_\_\_\_\_$$

12. Calculate polytropic exponent, n

$$\eta_p \times \frac{k}{k-1} = \frac{\text{n}}{\text{n - 1}}$$

$$n = \frac{\frac{\eta_p k}{k-1}}{\frac{\eta_p k}{k-1} - 1}$$

$n =$

13. Calculate polytropic head, H_{poly}

$$H_{poly} = \frac{Z_{suction} R_o T_{suction}}{\frac{n-1}{n}}\left[\left(\frac{P_{discharge}}{P_{suction}}\right)^{\frac{n-1}{n}} - 1\right]$$

14. Extrapolated actual flow using LP/HP Sectional curve (compressor speed and head),

$Q_{act} =$

15. Field measured flow, $Q_{act,measured} =$

16. Use the field measured flow to extrapolate head

17. Check whether there are **head** and **flow deficiency**

18. Calculate specific volume and mass flow

$$v = \frac{Z_{suction} \times R_o \times T_{suction}}{P_{suction} \times 144}$$

$$\dot{m} = \frac{Q_{act}}{v} =$$

19. Calculate gas horsepower

$$HP_{gas} = \frac{\dot{m} \times H_{poly}}{33000 \times \eta_p} =$$

20. Total train horsepower = LP train horsepower + HP train horsepower

21. Total indicated horsepower (get from the driver) = $\frac{\text{train horsepower}}{\text{engine efficiency}}$

Appendix 2-2

Sample of Gas Properties Calculation

APPENDIX 2-2

Sample of Gas Properties Calculation

For example from gas analysis:

Gas mixture	MOL %
Propane	89%
Butane	6%
Ethane	5%

Field data:

$P_1 = 20 psia$

$P_2 = 100 psia$

$N = 10650 RPM$

$Q = 5280 ICFM$

$T_1 = 40^0 F = 500^0 R$

$T_2 = 180.5^0 F = 640.5^0 R$

$$T_{average} = \frac{T_1 + T_2}{2} = \frac{40 + 180.5}{2} = 110^0 F$$

Gas mixture	(1) Mol% Each gas	(2) Mol Mass	(3) (1)x(2)	(4) Mass% [(3)/44.23] x100	(5) T_{cr} °R	(6) P_{cr} psi	(7) (1) x (5)	(8) (1) x (6)	(9) C_p Btu/mol-F	(11) (1) x (9)
Propane	89%	44.09	39.24	88.72%	666	617	592.7	549.1	16.58	14.76
Butane	6%	58.12	3.49	7.89%	766	551	46.0	33.1	22.53	1.35
Ethane	5%	30.07	1.50	3.39%	550	708	27.5	35.4	11.98	0.60
			44.23 Apparent Mol. Mass of Mixture				666.2 $T_{c(mix)}$	617.6 $P_{c(mix)}$		16.71 $C_{p(mix)}$

* (9) is obtained at inlet temperature. Average temperature should be used to minimize the error.

To find Z_1, first find P_{R1} and T_{R1}:

$$P_{R1} = P_1 / P_C , T_{R1} = T_1 / T_C$$

$$P_{R1} = \frac{20}{617.6} = 0.324, T_{R1} = \frac{40 + 460}{666.2} = 0.75$$

From Compressibility Chart, $Z_1 =$ 0.97

$$v_1 = ZRT/144P$$

$$v_1 = 0.97 \times \frac{1545}{44.23} \times \frac{(40+460)}{144 \times 20} = 5.88\,ft^3/lb$$

$$P_{R2} = P_2/P_C\,, T_{R2} = T_2/T_C$$

$$P_{R2} = \frac{100}{617.6} = 0.162, T_{R2} = \frac{640.5}{666.2} = 0.961$$

From Compressibility Chart, $Z_2 = 0.93$

$$v_2 = ZRT/144P$$

$$v_2 = 0.93 \times \frac{1545}{44.23} \times \frac{640.5}{144 \times 100} = 1.44\,ft^3/lb$$

Head

$$H_p = 72\left[\ln\left(\frac{P_2}{P_1}\right)\right](P_1 v_1 + P_2 v_2) = 72\left[\ln\left(\frac{100}{20}\right)\right](20 \times 5.88 + 100 \times 1.44) = 30300$$

Efficiency

$$\eta_p = \left(\frac{k-1}{k}\right) \div \left(\frac{n-1}{n}\right) \quad \text{where} \quad n = \frac{\ln(P_2/P_1)}{\ln(v_1/v_2)} = \frac{\ln(100/20)}{\ln(5.88/1.144)} = 1.144$$

$$\eta_p = \left(\frac{1.135-1}{1.135}\right) \div \left(\frac{1.144-1}{1.144}\right) = 0.95$$

This compressor cannot possibly have an efficiency of 95%. The k value should be recalculate using an average temperature.

For propane:

$C_p = 16.82 @ 50^0 F, 23.57 @ 300^0 F$ Interpolate for value @110^{0}F

$$C_p = 18.4 @ 110^0 F$$

The value for butane and ethane can be obtained in similar way:

For butane:

$$C_p = 24.81 @ 110^0 F$$

For ethane:

$$C_p = 13.14 @ 110^0 F$$

$$C_p(mix) = 0.89 \times 18.4 + 0.06 \times 24.81 + 0.05 \times 13.14 = 16.38 + 1.49 + 0.67 = 18.52$$

$$k(mix) = \frac{18.52}{18.52 - 1.99} = 1.12$$

Efficiency by using new value of k at average temperature:

$$\eta_p = \frac{(k-1)/k}{(n-1)/n} = \frac{(1.12-1)/1.12}{(1.144-1)/1.144} = 0.85$$

Comparing to the value calculated by the computer program (manufacturer), the reference value is 0.716. The efficiency cannot accurately be hand-calculated for this problem. Thus is common for high mole weight gases. The problem is due to the nonlinear relationship of the gas properties near the dew point. When looking at values far from the dew point, such as with air or nitrogen, the values are near linear and perfect gas laws are accurate.

Since efficiency cannot accurately be established, then also work and power cannot be established. We can however work backward from the driver to establish the gas power.

Appendix 3

PETRONAS Carigali Pulai-A Gas Analysis Test Report

SHORE GAS TERMINAL LABORATORY
PENINSULAR MALAYSIA OPERATIONS
RONAS CARIGALI SDN BHD
05, JALAN KUANTAN / TERENGGANU
0 KERTEH, TERENGGANU

09-8271943 FAX : 09-8271145

PETRONAS CARIGALI

TEST REPORT

'ORT NO . : **OGT LAB/11/2002/ 54**

'E : **14.11.2002**

IPLE ID	:	SR/059/2002	~~DATE / TIME RECEIVED~~	:	~~29/10~~ @ 1300 Hrs
IPLE LOCATION	:	PULAI A	STREAM NAME	:	SUCTION COMP. (LP)
IPLE DESCRIPTION	:	NATURAL GAS	STREAM PRESSURE	:	1400 kPa
.INDER SER. NO.	:	J 003	STREAM TEMP.	:	38 °C
IPLING DATE	:	19.10.2002	OPENING PRESSURE	:	220 psig
TOMER REF.	:	MIV 44670	ANALYSIS DATE	:	14.11.2002

COMPONENT	MOL %
C 1 - METHANE	71.3965
C 2 - ETHANE	4.5731
C 3 - PROPANE	1.5996
IC 4 - ISO-BUTANE	0.4659
NC 4 - N-BUTANE	0.4686
IC 5 - ISO-PENTANE	0.3648
NC 5 - N-PENTANE	0.2485
C 6+- HEXANE PLUS	0.5076
N 2 - NITROGEN	2.5937
CO 2 - CARBON DIOXIDE	17.7817
TOTAL	100.0000
MOLECULAR WEIGHT	23.5396
DENSITY (KG/M3)	0.9936

Reported by :

Huslan Sulong
AMIC (3996/99)

Appendix 4

Pulai-A Performance Data

TANDEM GAS COMPRESSOR PROGRAM P435 ---
:VISION NO. 7.0 RUN ON 11:15:54 26-MAR-97 ---

?REFIX	DIA	STAGES						
107	7.50	1ET	3DT	2DT	2DT	1DT	1DT	2CE
107	7.50	1CT	1CT	2BT	1BT	1BT	1BT	1BE

?AA	1345.00	3852.31	
?AA	3990.31	11721.00	
N, JOULES/KGM	134560.	130695.	
Y, JOULES/KGM	139302.	137110.	
PERCENT	9.510	4.328	
ITY, M3/HR	1812.6	548.2	
LOW, MMSCFD	19.00	17.50	
HP	1549.	1593.	3142. (TOTAL)
, RPM	20509.	20509.	
URE RATIO	2.967	3.043	
EG C	46.0	46.0	
EG C	149.1	165.3	
:IENCY, ISEN	0.722	0.629	
:IENCY, POLY	0.748	0.660	
: MARGIN	0.220	0.227	
:FIC GRAVITY	0.7985	0.7990	
	1.258	1.258	
	1.214	1.208	
	728.0	724.8	
	409.8	408.4	
	0.9711	0.9133	
	0.9704	0.9441	
50 DEG F	1.279	1.278	
300 DEG F	1.214	1.214	

PSIA	195.08	558.73	
PSIA	578.75	1699.99 ←	
:EN, FT-LBF/LBM	45019.	43725.	
)LY, FT-LBF/LBM	46605.	45871.	
;, PERCENT	9.510	4.328	
ACITY, ACFM	1066.7	322.6	
FLOW, MMSCFD	19.00	17.50 ←	
:R, HP	1549.	1593.	3142. (TOTAL)
:D, RPM	20509.	20509.	
SSURE RATIO	2.967	3.043	
DEG F	110.8	114.8	
ICIENCY, ISEN	0.722	0.629	
ICIENCY, POLY	0.748	0.660	
GE MARGIN	0.220	0.227	
IRC- T1 DEG F	117.8	125.6	
" STD FLOW	19.27	18.32	
)IAL CLRNC	0.0035	0.0035	

R TURBINES INCORPORATED
NE PERFORMANCE CODE REV. 2.81
)MER: PETRONAS CARIGALI / PULAI-A
:D: KL7-010

DATE RUN: 26-MAR-97
RUN BY: BLATTNER, TJ

CENTAUR 40-T4700
CS/MD
HI-AMBIENT
GAS
TCD-2 REV. 2.1
ES-1872

DATA FOR MINIMUM PERFORMANCE

l Type		CUSTOM [illegible]				
[illegible] Ratio		1.3750				
vation	Feet	69				
et Loss	in. H2O	3.0				
aust Loss	in. H2O	3.0				
essory on GP Shaft	Hp	31.0				
ine Inlet Temp.	Deg. F	71.6	80.6	86.0	95.0	
ative Humidity	%	95.0	95.0	95.0	95.0	
vation Loss	Hp	11	11	11	10	
et Loss	Hp	63	62	61	60	
aust Loss	Hp	30	30	29	29	
-Optimum NPT Loss	Hp	9	8	8	7	
box Loss	Hp	146	146	146	146	
box Efficiency		0.9644	0.9635	0.9628	0.9615	
ven Equipment Speed	RPM	20509	20509	20509	20509	
mum Equipment Speed	RPM	21530	21490	21467	21411	
Generator Speed	RPM	15000	15000	15000	14999	
ified Load	Hp	FULL	FULL	FULL	FULL	
Output Power	Hp	3947	3846	3777	3647	
Flow	MMBtu/hr	40.26	39.58	39.18	38.43	
Rate	Btu/Hp-hr	10100	[illegible]	[illegible]	[illegible]	
t Air Flow	lbm/hr	142648	139447	137354	133806	
ne Exhaust Flow	lbm/hr	145170	141927	139809	136214	
	psi(g)	129.7	127.2	125.4	122.2	
nlet Temp. (T5)	Deg. F	1149	1159	1165	1173	
ensated PTIT	Deg. F	1188	1198	1204	1212	
ust Temperature	Deg. F	847	860	869	881	

AS COMPOSITION (VOLUME PERCENT)
tu/SCF) = 1379.6 SG = 1.1452 W.I. @60F = 1289.1

0.0000	CH4 = 0.0000	C2H4 = 73.3599	C2H6 = 0.0000	
5.3700	C3H8 = 0.0000	C4 = 2.0900	C5 = 1.2400	
0.5800	C7 = 0.5600	C8 = 0.0000	CO = 0.0000	
15.7300	H2 = 0.0000	H2O = 0.6000	H2S = 0.0000	
0.4700	O2 = 0.0000	SO2 = 0.0001	He = 0.0000	

Appendix 5

Pulai-A Performance Curves

SOLAR TURBINES A CATERPILLAR COMPANY	CONTROL NO. SD-[illegible] Sh.1
	DATE 28 March 1997
	ENGINEER TJ Blattner/gm

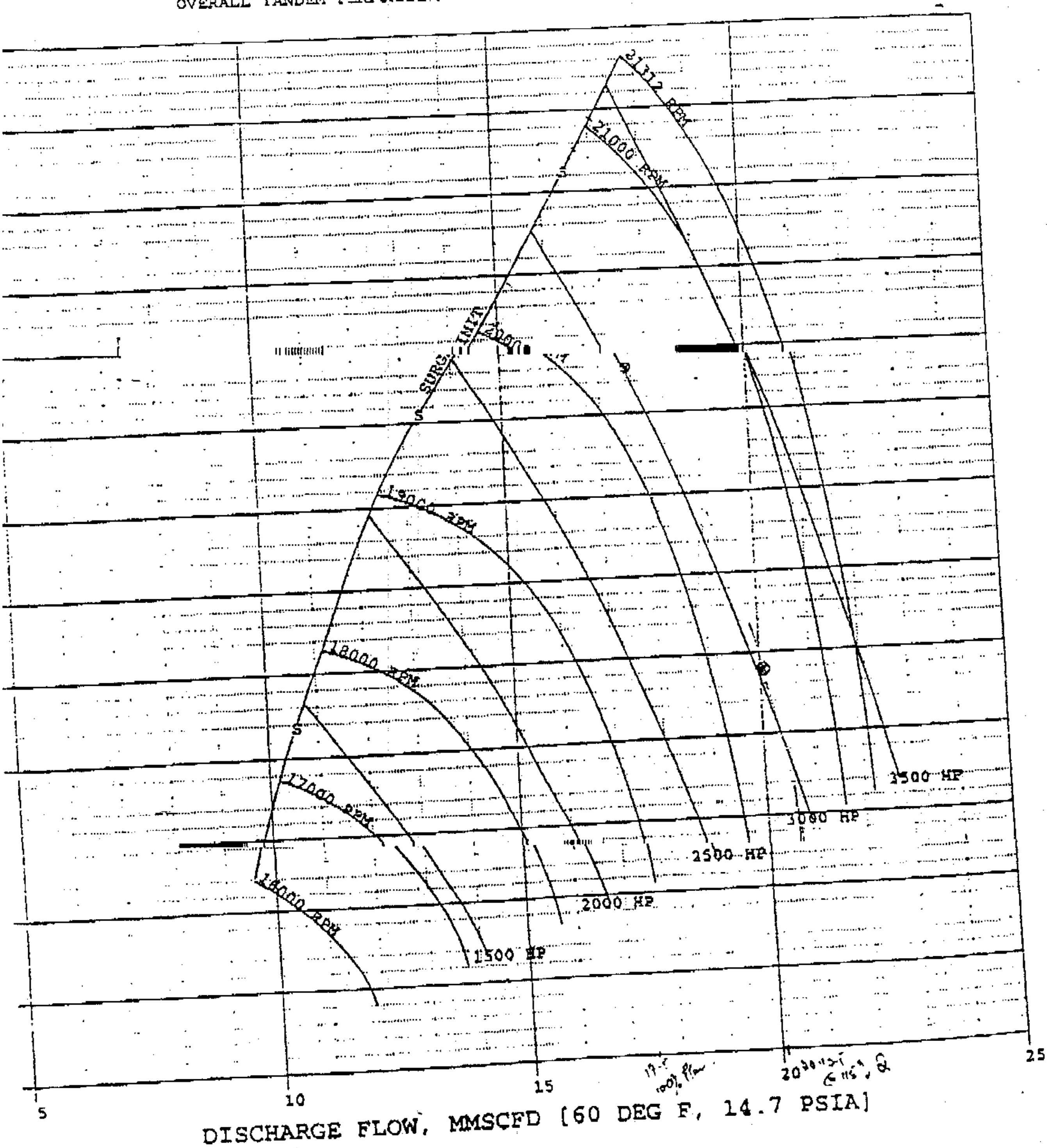
PETRONAS/CARIGALI
PULAI-A

CENTAUR 40-4700/HIGH AMBIENT
GBR = 1.375

KL7-010
OVERALL TANDEM PERFORMANCE
LP UNIT [C1607GRA]
(107) 1ET 3DT 2DT 2DT 1DT 1DT 2CE
.RATIO OF SPECIFIC HEATS 1.258
.SPECIFIC GRAVITY .7985
.SUCTION TEMPERATURE 46.0 DEG C
.SUCTION PRESSURE 1345 KPAA
.SQ1 = SUCTION FLOW, MMSCFD
(60 DEG F, 14.7 PSIA)

HP UNIT [C1607GRA]
(107) 1CT 1CT 2BT 1BT 1BT 1BT 1BE
.RATIO OF SPECIFIC HEATS 1.258
.SPECIFIC GRAVITY .7990
.INTERSTAGE TEMPERATURE 46.0 DEG C
.INTERSTAGE PRESSURE LOSS 138.0 KPAA
.SQ2 = SQ1 X .9210526
FOR INTERSTAGE FLOW LOSS

OVERALL TANDEM PERFORMANCE IS ESTIMATED ONLY



SOLAR TURBINES
A CATERPILLAR COMPANY

SD-23323 SH.2
DATE
29 Mar 1997
ENGINEER
TJ Blattner/gm

PETRONAS/CARIGALI
PULAI-A

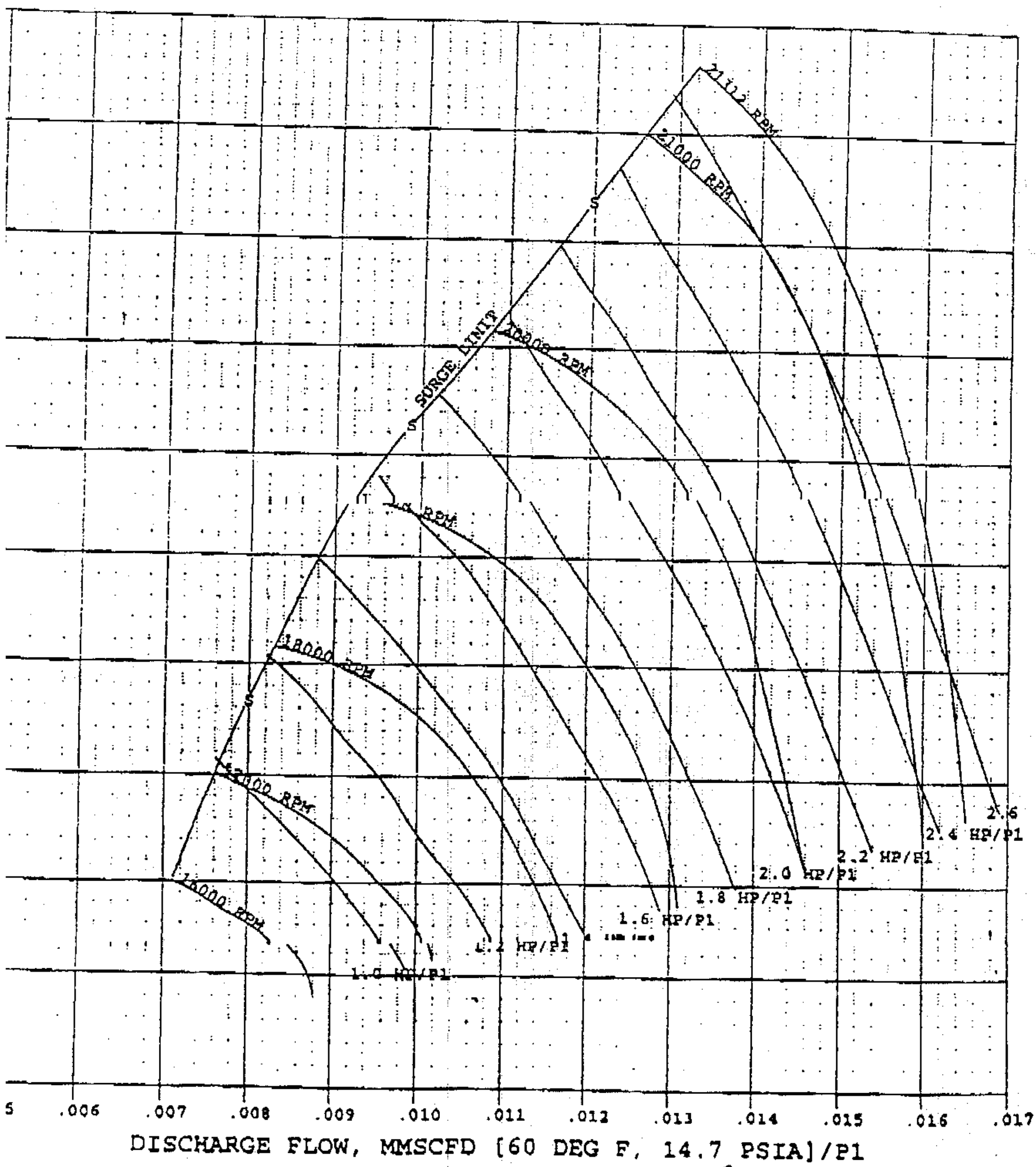
CENTAUR 40-4700/HIGH AMBIENT
GBR = 1.375

KL7-010

OVERALL TANDEM PERFORMANCE
LP UNIT (C1607GRA)
[107] 1ET 3DT 2DT 2DT 1DT 1DT 2CE
.RATIO OF SPECIFIC HEATS 1.258
.SPECIFIC GRAVITY .7985
.SUCTION TEMPERATURE 46.0 DEG C
.SUCTION PRESSURE 1345 KPAA
.SQ1 = SUCTION FLOW, MMSCFD

HP UNIT (C1607GRA)
[107] 1CT 1CT 2BT 1BT 1BT 1BT 1BE
.RATIO OF SPECIFIC HEATS 1.258
.SPECIFIC GRAVITY .7990
.INTERSTAGE TEMPERATURE 46.0 DEG C
.INTERSTAGE PRESSURE LOSS 138.0 KPAA

OVERALL TANDEM PERFORMANCE IS ESTIMATED ONLY



AL COMPRESSOR PERFORMANCE

ET 3DT 2DT 2DT-1DT 1DT 2CE

Solar Turbines
A Caterpillar Company

SD-23829 SHT. 1

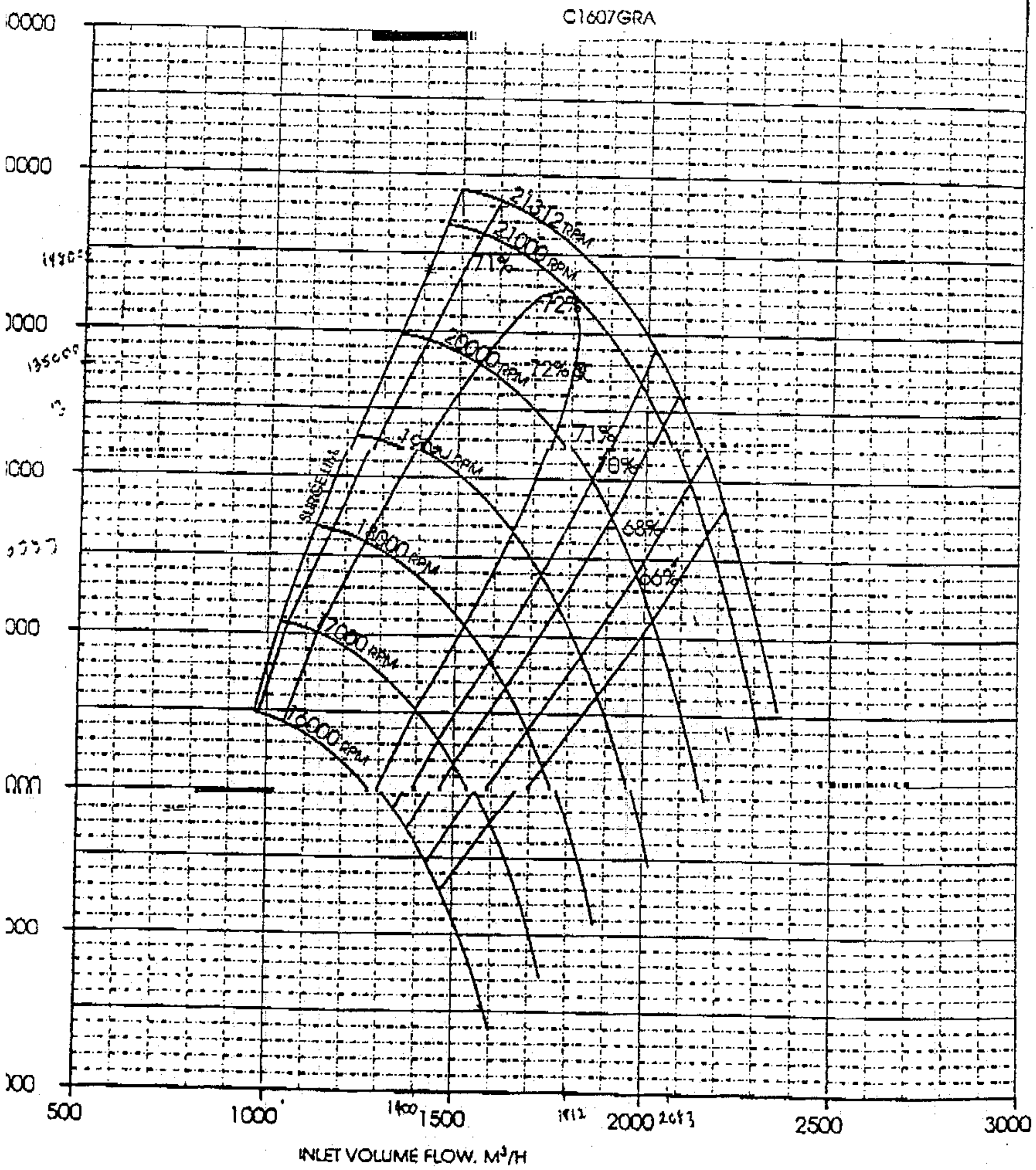
28 MAR 1997

TJ BLATTNER/GM

OF SPECIFIC HEATS 1.259
C GRAVITY 0.799
N TEMPERATURE 46.0°C
N PRESSURE 1345.03 KPA ABS.

PETRONAS CARIGALI
PULAI-A
LP UNIT OF 2 BODY TANDEM

KL7-010
CENTAUR 40-4700/H.A. GBR = 1.375
C1607GRA



. COMPRESSOR PERFORMANCE

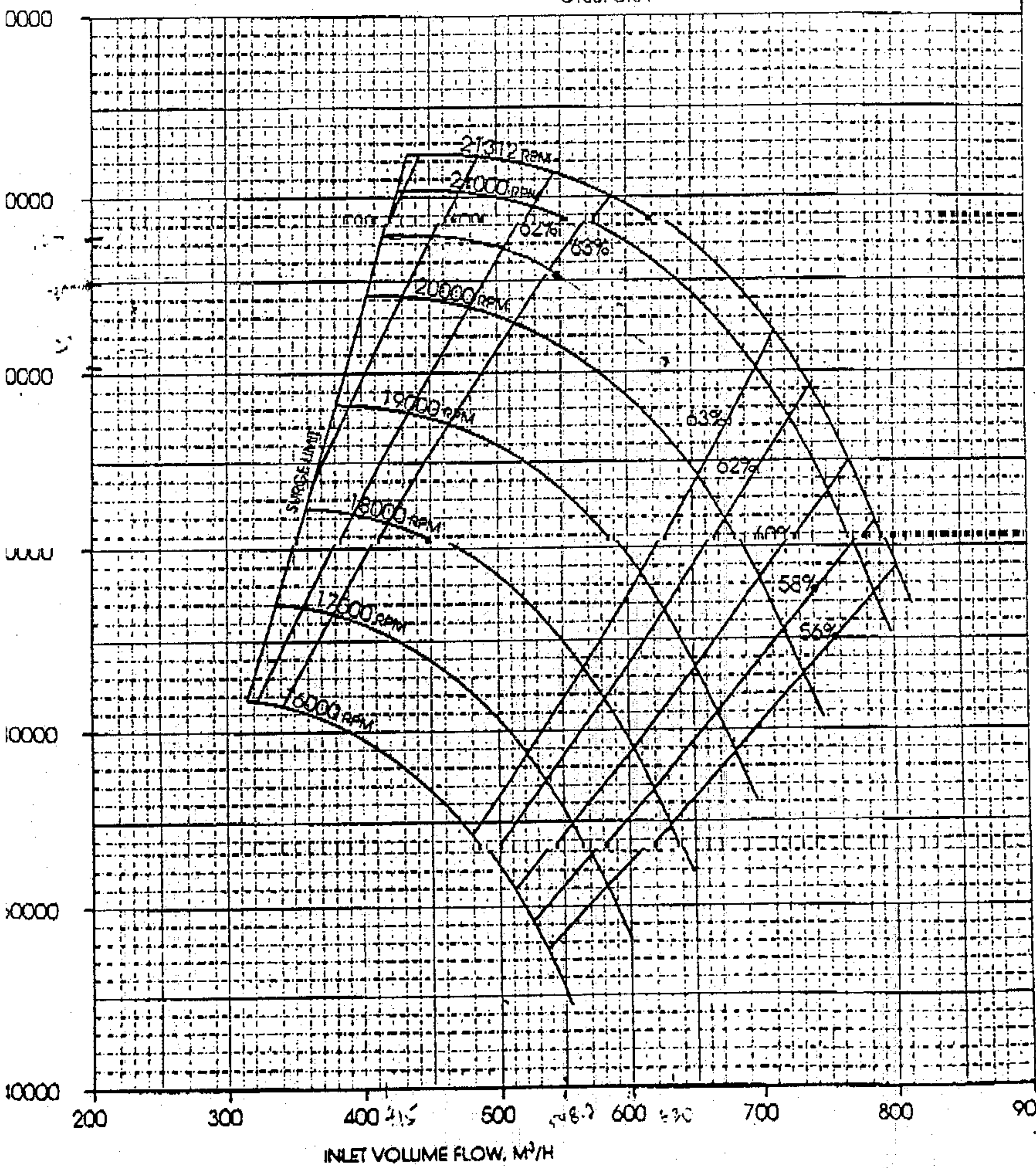
Solar Turbines
A Caterpillar Company

CONTROL NO	SD-23830 SHT 1
DATE	
ENGINEER	TJ BLATTNER/GM

PETRONAS CARIGALI
PULAI-A
HP UNIT OF 2 BODY TANDEM

KL7-010
CENTAUR 40-4700/H.A. GBR = 1.375
C1607GRA

)F SPECIFIC HEATS 1.258
: GRAVITY 0.799
N TEMPERATURE 46.0°C
N PRESSURE 3852.31 KPA ABS.



Appendix 6

Bekok-A Compressors Performance Test Report

M E M O R A N D U M

		Date :	March 25, 1992
To:	Ephraim Kouju N. Suryamurthy	Ref :	
		Code :	620 (TTL)
From:	Pang Kee Keng	Subj :	Bekok A C700A Compressors Performance Test

Machinery Group recent performance test on Bekok A C700A gas injection train indicates a possible fouling in the HP compressor. Thus, we recommend the rotor be inspected and replaced if necessary. Also, we recommend taking this opportunity to change-out the bearing capsules on the HP suction and discharge ends as the compressor vibration levels are approching the alarm limit. We propose that the change-out to be done in May to coincide with BEA platform shutdown for vessel inspection.

Compressor performance test

Attachments 1 and 2 show the test points plotted on the expected performance curves. As shown in attachment 1, the LP compressor performance is still within acceptable range. However, the HP compressor deficiency is about 10% and it increases with higher flowrates which indicates probable fouling in the compressor. The drop in isentropic head also means that the amount of gas the compressor can handle at a given speed will be reduced.

Attachments 3 and 4 depict the performance test results of the LP and HP compressor respectively. This results were used to plot the performance points in Attachments 1 and 2.

Gas sample analysis

Attachments 5 to 8 represent gas samples data taken during the performance test. Evaluation by TCOT on the gas samples show that the gas compositions have not changed significantly since 1989. Nevertheless, we would like BEA platform to update the gas sample record for their own perusal.

Recommendation

We recommend that the HP compressor rotor be inspected and replaced if neccessary together with the change-out of the HP bearing capsules. This change-out should be done in May to coincide with BEA platform shutdown for vessel inspection.

We would also like to express our appreciation to the platform personnel for the assistance provided in carrying out this performance test. Please contact Too Taik Luen at extn: 4375 should you require further clarification.

OTTL411M:nz
Attachments

cc: MS/RFL/BEA Plat. Supvr./File Bea-B2
circ: DRM/DEL/KCT/PKK/MIC/TTL

ATTACHMENT 1

C160 "2D-2D-3C-3C-2C-2C-2C-1C"

EPMI BEKOK "A", TRAIN "A"

LOW-PRESSURE UNIT

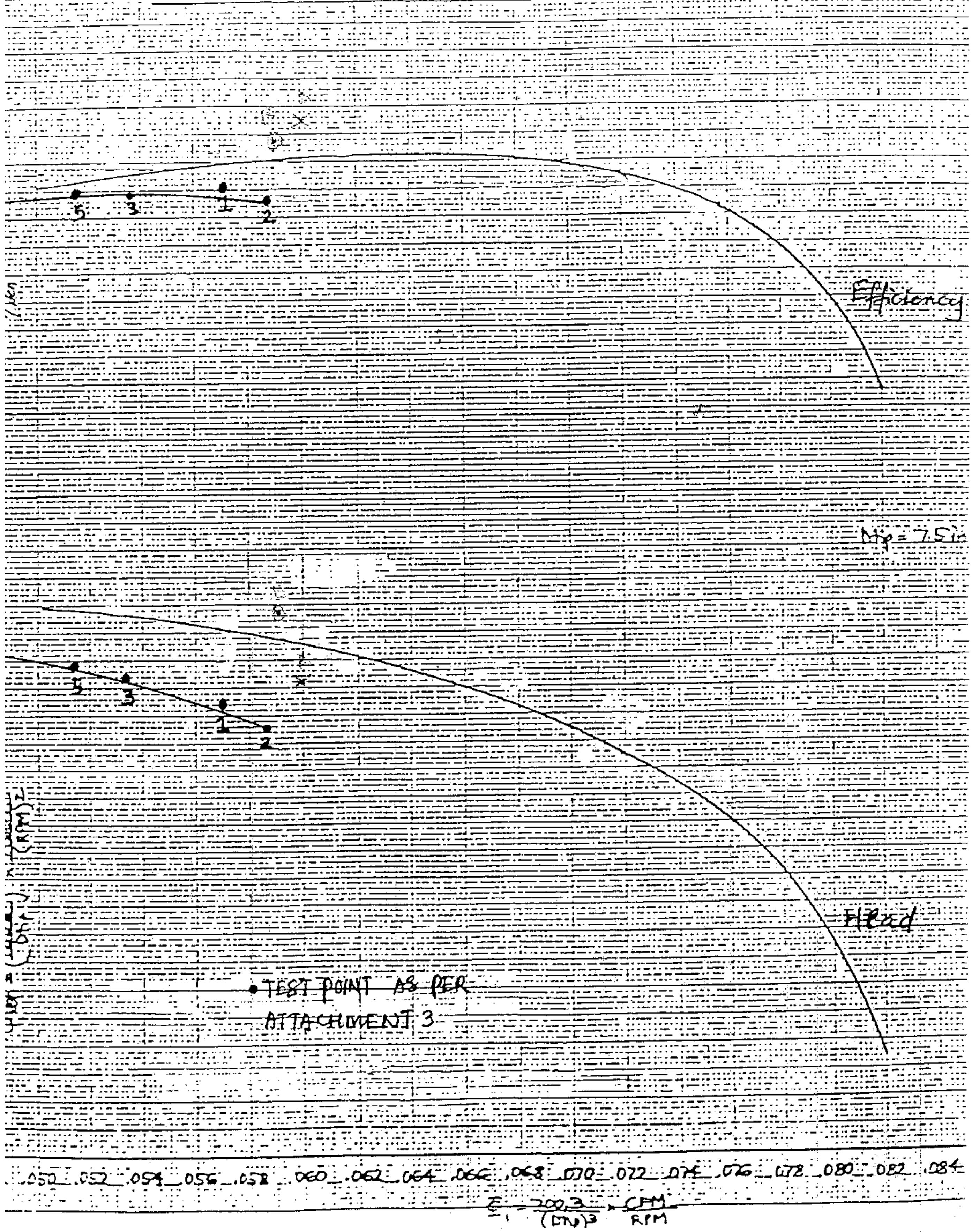
FIELD TEST RESULTS OF FEBRUARY 1992

SOLAR TURBINES INCORPORATED

CONTROL NO. CS-26462

DATE 8-19-87

ENGINEER Leon Sapira



ATTACHMENT 2

SOLAR TURBINES INCORPORATED

CONTROL NO. CS-26463

DATE 9-9-87

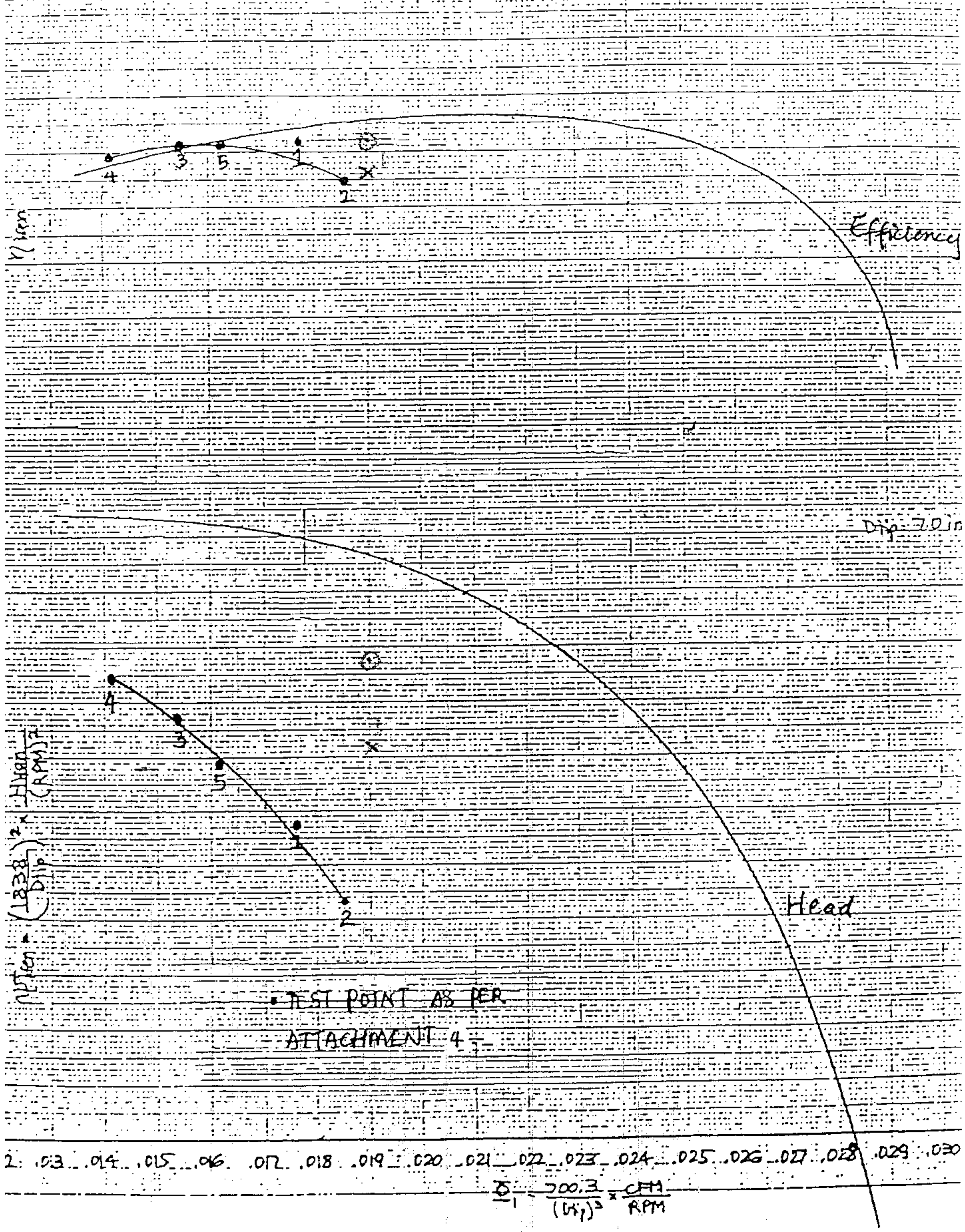
ENGINEER Leon Sapiro

C165 * 2B-3A-3A-2A-1A *

EPMI BEKOK "A" TRAIN "A"

HIGH-PRESSURE UNIT

FIELD TEST RESULTS OF FEBRUARY 1992



ATTACHMENT 3

TRAIN A LP COMPRESSOR TEST DATA AND RESULTS

	TEST POINT 1	TEST POINT 2	TEST POINT 3	TEST POINT 4	TEST POINT 5
COMPRESSOR SPEED (RPM)	20175	20225	20325	19650	19580
GAS SG	0.76	0.76	0.76	0.76	0.76
SUCTION PRESSURE (PSIG)	294.7	296	292	291.3	292.3
SUCTION TEMP (DEG. F)	85.5	84.1	84.1	83.9	84.1
DISCHARGE PRESSURE (PSIG)	1010.4	987.2	1058.8	1029.3	987.7
DISCHARGE TEMP (DEG. F)	279.9	275.5	289.1	286.6	276.5
ACTUAL INLET VOL (ACFM)	693.1	710.5	651.2	561.7	606.4
ISENTROPIC HEAD (FT-LBF/LBM)	48270	46890	50620	49450	47490
ISENTROPIC EFFICIENCY	67.6	66.8	66.9	66.03	67.3
GAS POWER (HP)	1854	1884	1834	1564	1596
ψ ISEN	7.12	6.88	7.36	7.69	7.44
Φ_1	0.0570	0.0583	0.0532	0.0475	0.0514

NOTE:

1 $\psi_{ISEN} = (1838/Dtip)^2 \times (Hisen/(RPM)^2)$

2 $\Phi_1 = (700.3/(Dtip)^3) \times (CFM/RPM)$

WHERE Dtip = 7.5 in

ATTACHMENT 4

TRAIN A HP COMPRESSOR TEST DATA AND RESULTS

	TEST POINT 1	TEST POINT 2	TEST POINT 3	TEST POINT 4	TEST POINT 5
COMPRESSOR SPEED (RPM)	20175	20225	20325	19650	19580
GAS SG	0.76	0.76	0.76	0.76	0.76
SUCTION PRESSURE (PSIG)	1000.0	986.7	1050.0	1022.5	985.0
SUCTION TEMP (DEG. F)	95.1	94.8	94.4	93.0	93.3
DISCHARGE PRESSURE (PSIG)	2073.3	1950.0	2356.7	2242.5	2040.0
DISCHARGE TEMP (DEG. F)	217.0	212.7	228.6	225.2	215.2
ACTUAL INLET VOL (ACFM)	174.3	182.8	153.0	135.5	154.3
ISENTROPIC HEAD (FT-LBF/LBM)	24460	22710	27190	26280	24310
ISENTROPIC EFFICIENCY	60.8	57.4	61.1	59.7	60.5
GAS POWER (HP)	1005.0	1022.0	1041.0	887.2	880.0
ψ ISEN	4.14	3.83	4.54	4.69	4.37
Φ_1	0.0176	0.0185	0.0154	0.0141	0.0161

NOTE:

1 ψ ISEN = $(1838/\text{Dtip})^2 \times (\text{Hisen}/(\text{RPM})^2)$

2 Φ_1 = $(700.3/(\text{Dtip})^3) \times (\text{CFM}/\text{RPM})$

WHERE Dtip = 7.0 in

ATTACHMENT 5

- ESSO PRODUCTION MALAYSIA INC. -

TERENGGANU CRUDE OIL TERMINAL

KERTEH

GAS ANALYSIS BY H.P. 5880A G.C.

SAMPLED BY	: BEKOK A		
SAMPLE	: HP DISCHARGE GAS		
LOCATION	: 700A BEKOK A	FLOWRATE :	N.A. M3/HR
DATE	: 21-02-92	TEMP. :	133.0 DEG.C
TIME (hrs)	: NA	PRESSURE :	13790.0 kPA
DATE TESTED	: 27-02-92		

- REFERENCE GAS -

COMPONENTS		CHART RESPONSE	MOL. %	GAS FACTORS
METHANE	C1	119448.000	29.640	0.0002481
ETHANE	C2	111280.000	20.170	0.0001813
PROPANE	C3	124144.000	23.060	0.0001858
ISO-BUTANE	I-C5	59357.800	8.000	0.0001348
N-BUTANE	N-C4	62299.300	7.980	0.0001281
ISO-PENTANE	I-C5	34044.400	4.010	0.0001178
N-PENTANE	N-C5	26282.700	3.000	0.0001141
* HEXANE +	C6+	0.000	0.000	
CARBON DIOXIDE	CO2	18406.800	3.990	0.0002168
NITROGEN	N2	2251.940	0.150	0.0000666
TOTAL			100.000	

- SAMPLE GAS -

COMPONENTS		CHART RESPONSE	UNNORM. MOL %	MOL %	MOL. WT FACTORS	MOL. WT.
METHANE	C1	365461.000	90.686	75.758	0.1604	12.1516
ETHANE	C2	78558.500	14.239	11.895	0.3007	3.5769
PROPANE	C3	38390.100	7.131	5.957	0.4409	2.6265
ISO-BUTANE	I-C5	11393.500	1.536	1.283	0.5812	0.7456
N-BUTANE	N-C4	11665.500	1.494	1.248	0.5812	0.7255
ISO-PENTANE	I-C5	4612.090	0.543	0.454	0.7215	0.3274
N-PENTANE	N-C5	2468.400	0.282	0.235	0.7215	0.1698
* HEXANE +	C6+	8706.960	0.794	0.663	0.9200	0.6102
CARBON DIOXIDE	CO2	13015.200	2.821	2.357	0.4401	1.0373
NITROGEN	N2	2687.060	0.179	0.150	0.2802	0.0419
TOTAL			119.705	100.000		22.013

* CORRECTED C6+ AREA = MEASURED C6+ AREA X 72/92

* MOL% C6+ = (MOL% OF I-C5 + MOL% OF N-C5) X $\frac{\text{(CORRECTED C6+ AREA)}}{\text{(I-C5 + N-C5 AREA)}}$

ATTACHMENT 6

SAMPLE : HP DISCHARGE GAS
LOCATION : 700A BEKOK A
DATE : 21-02-92

- SAMPLE GAS BTU ANALYSIS -

COMPONENTS		MOL %	GROSS BTU/CU.FT. FACTORS	GROSS BTU/CU.FT.	NET BTU/CU.FT. FACTORS	NET BTU/CU.FT.
METHANE	C1	75.758	10.100	765.154	9.090	688.638
ETHANE	C2	11.895	17.690	210.425	16.180	192.463
PROPANE	C3	5.957	25.170	149.942	23.160	137.968
ISO-BUTANE	I-C5	1.283	32.530	41.729	30.010	38.497
N-BUTANE	N-C4	1.248	32.620	40.719	30.100	37.573
ISO-PENTANE	I-C5	0.454	40.000	18.153	36.980	16.782
N-PENTANE	N-C5	0.235	40.090	9.436	37.070	8.725
* HEXANE +	C6+	0.663	55.030	36.500	51.000	33.827
CARBON DIOXIDE	CO2	2.357	0.000	0.000	0.000	0.000
NITROGEN	N2	0.150	0.000	0.000	0.000	0.000
TOTAL		100.000		1272.057		1154.473

** BTU/CU.FT. OF N-HEPTANE HAS BEEN ASCRIBED TO THE HEXANE PLUS FACTORS.

MOLECULAR WEIGHT	=	22.0126	
STREAM GAS DENSITY	=	90.9020	KG / M3
GAS DENSITY	=	0.9328	KG / M3
SPECIFIC GRAVITY	=	0.7600	
COMPRESSIBILITY	=	0.9962	
GROSS CALORIFIC VALUE	=	1272.0569	BTU / CU.FT.
GROSS HEATING VALUE	=	47.3956	MJ / M3

PETRONAS REPRESENTATIVE

EPMI-TCOT
SHIFT SUPERVISOR

LABORATORY TECHNICIAN
EPMI-TCOT

lj/fm120292

ATTACHMENT 7

- ESSO PRODUCTION MALAYSIA INC. -

TERENGGANU CRUDE OIL TERMINAL

KERTEH

GAS ANALYSIS BY H.P. 5880A G.C.

SAMPLED BY	: BEKOK A				
SAMPLE	: LP DISCHARGE GAS				
LOCATION	: 700A BEKOK A	FLOWRATE	:	N.A.	M3/HR
DATE	: 21-02-92	TEMP.	:	136.0	DEG.C
TIME (hrs)	: NA	PRESSURE	:	7000.0	kPA
DATE TESTED	: 27-02-92				

- REFERENCE GAS -

COMPONENTS		CHART RESPONSE	MOL. %	GAS FACTORS
METHANE	C1	119448.000	29.640	0.0002481
ETHANE	C2	111280.000	20.170	0.0001813
PROPANE	C3	124144.000	23.060	0.0001858
ISO-BUTANE	I-C4	59357.800	8.000	0.0001348
N-BUTANE	N-C4	62299.300	7.980	0.0001281
ISO-PENTANE	I-C5	34044.400	4.010	0.0001178
N-PENTANE	N-C5	26282.700	3.000	0.0001141
* HEXANE +	C6+	0.000	0.000	
CARBON DIOXIDE	CO2	18406.800	3.990	0.0002168
NITROGEN	N2	2251.940	0.150	0.0000666
TOTAL			100.000	

- SAMPLE GAS -

COMPONENTS		CHART RESPONSE	UNNORM. MOL %	MOL %	MOL. WT FACTORS	MOL. WT.
METHANE	C1	329510.000	81.765	76.390	0.1604	12.2529
ETHANE	C2	68571.700	12.429	11.612	0.3007	3.4917
PROPANE	C3	35377.700	6.571	6.139	0.4409	2.7069
ISO-BUTANE	I-C4	11222.800	1.513	1.413	0.5812	0.8213
N-BUTANE	N-C4	10865.100	1.392	1.300	0.5812	0.7557
ISO-PENTANE	I-C5	3840.440	0.452	0.423	0.7215	0.3049
N-PENTANE	N-C5	2016.850	0.230	0.215	0.7215	0.1552
* HEXANE +	C6+	7684.680	0.701	0.655	0.9200	0.6024
CARBON DIOXIDE	CO2	8371.270	1.815	1.695	0.4401	0.7461
NITROGEN	N2	2536.040	0.169	0.158	0.2802	0.0442
TOTAL			107.037	100.000		21.881

* CORRECTED C6+ AREA = MEASURED C6+ AREA X 72/92

* MOL% C6+ = (MOL% OF I-C5 + MOL% OF N-C5) X $\frac{\text{(CORRECTED C6+ AREA)}}{\text{(I-C5 + N-C5 AREA)}}$

ATTACHMENT 8

SAMPLE : LP DISCHARGE GAS
LOCATION : 700A BEKOK A
DATE : 21-02-92

- SAMPLE GAS BTU ANALYSIS -

COMPONENTS		MOL %	GROSS BTU/CU.FT. FACTORS	GROSS BTU/CU.FT.	NET BTU/CU.FT. FACTORS	NET BTU/CU.FT.
METHANE	C1	76.390	10.100	771.536	9.090	694.383
ETHANE	C2	11.612	17.690	205.413	16.180	187.879
PROPANE	C3	6.139	25.170	154.530	23.160	142.190
ISO-BUTANE	I-C5	1.413	32.530	45.969	30.010	42.408
N-BUTANE	N-C4	1.300	32.620	42.414	30.100	39.137
ISO-PENTANE	I-C5	0.423	40.000	16.905	36.980	15.628
N-PENTANE	N-C5	0.215	40.090	8.622	37.070	7.973
* HEXANE +	C6+	0.655	55.030	36.032	51.000	33.393
CARBON DIOXIDE	CO2	1.695	0.000	0.000	0.000	0.000
NITROGEN	N2	0.158	0.000	0.000	0.000	0.000
TOTAL		100.000		1281.421		1162.991

** BTU/CU.FT. OF N-HEPTANE HAS BEEN
ASCRIBED TO THE HEXANE PLUS FACTORS.

MOLECULAR WEIGHT	=	21.8813
STREAM GAS DENSITY	=	45.8539 KG / M3
GAS DENSITY	=	0.9272 KG / M3
SPECIFIC GRAVITY	=	0.7555
COMPRESSIBILITY	=	0.9962
GROSS CALORIFIC VALUE	=	1281.4213 BTU / CU.FT.
GROSS HEATING VALUE	=	47.7445 MJ / M3

PETRONAS REPRESENTATIVE

EPMI-TCOT
SHIFT SUPERVISOR

LABORATORY TECHNICIAN
EPMI-TCOT

Appendix 7

Performance Evaluation of Centrifugal Compressors

(By F.M. Odom, Solar Turbines)

Performance Evaluation of Centrifugal Compressors

F. M. Odom
Manager, Performance Analysis

ITRODUCTION

ıis paper describes the fundamental principles centrifugal compressor performance, com-'essor performance curves, and methods for ɔtaining and analyzing performance data. ɜing these you can determine the operating ɔndition of your centrifugal compressor. For a ore complete description of field performance sting for contractual performance guarantee ɜmonstration, see Solar's Engineering Specifi-ıtion ES-1973.

There is a companion paper to this one, titled erformance Evaluation of Gas Turbine Engines", ɔlar Publication No. 89570.

The first step to understanding compressor ɪrformance is to understand the performance ırves.

ƆLAR CENTRIFUGAL COMPRESSOR ƎRFORMANCE CURVES

ɪr all single-body compressors, Solar Turbines ɔorporated produces three types of perfor-ɜnce curves:

- Head versus Capacity
- Dimensional
- Semi-Dimensional

For tandem compressors (more than one com-ɜssor body on the same shaft), Solar produces o types of composite performance curves ıich predict the performance of the tandem as t were a single compressor.

- Dimensional
- Semi-Dimensional

In addition, a head versus capacity curve is ɔduced for each individual compressor body of andem unit.

All Solar compressor performance curves are ɔduced in whatever language and choice of its is requested by the customer. Single-body rformance curves can be automatically plotted computer in English, French, Spanish or ɪrman.

All Solar compressor performance curves are computer **predictions** of compressor performance. The prediction is based upon computed data for each individual stage configuration, and combined data for all of the stages operating together.

For every compressor performance curve, the base operating conditions of suction temperature, gas composition, and pressure (either suction or discharge) must remain constant. These base operating conditions are assumed to remain constant when the curve is computed, so if the curve is used to predict performance for other than these base conditions, some inaccuracy may occur. Therefore, on every Solar compressor performance curve, the base operating conditions used for computation of the curve are clearly printed in the heading area. These base conditions are:

- Specific Gravity
- Suction Temperature
- Constant Pressure (Suction or Discharge)
- Ratio of Specific Heats
- Pseudo criticals are assumed constant but not printed.

Specific gravity is a function of the gas composition. Ratio of specific heats is a function of gas composition and temperature. Either the suction or discharge pressure must be assumed to remain constant.

The magnitude of the effect of deviation from the base operating conditions is demonstrated in the Appendix.

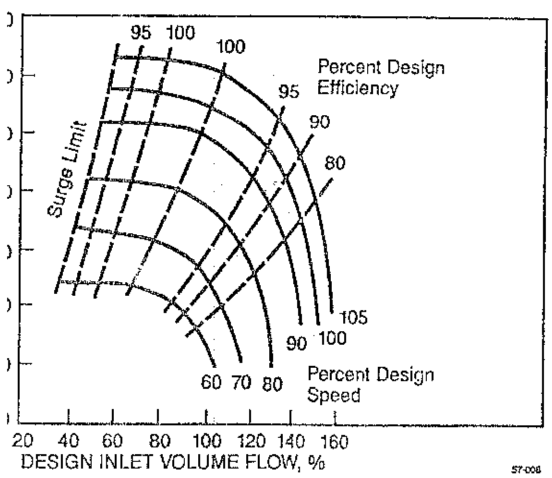
Operating a compressor at its peak efficiency requires, among other things, an understanding of its performance curve(s). There are many different formats for graphically showing the expected performance of centrifugal compressors. Understanding centrifugal compressor performance curves can be condensed into a single rule: The head versus capacity curve is the only curve necessary.

d versus Capacity Curve

head versus capacity curve (Figure 1) vs, on coordinates of isentropic head and al inlet volumetric flow rate, lines of constant ed, lines of constant adiabatic efficiency, and igle line showing the approximate location of surge limit. To plot a performance curve, the composition, suction temperature, and a sure (either suction or discharge) must be tant. The curve may not be accurate if it is l for other values than those assumed when plotted. **The head versus capacity curve is l most often because it is only slightly cted by even very large changes in the conditions** of gas composition, suction perature, and pressure. It is also used to k the condition of an operating compressor, omparing the actual efficiency and speed to efficiency and speed that the curve says the pressor ought to have.

esides speed, head and capacity are the two parameters that directly affect the per- ance of a centrifugal compressor. All other meters, such as pressure, temperature, cular weight, and standard volumetric flow only affect the performance indirectly. Chan- n any of these parameters do not significant- ange the shape of the head versus capacity e but simply change the location of the ating point on the curve.

l. "Head" is a term used to describe the unt of energy added to one unit of mass of as being compressed. It is the enthalpy rise suction to discharge. Enthalpy is a measure e energy contained in one unit of mass. Head function of the properties of the gas being pressed, the suction temperature, and the pressure ratio. Head can be described with a pressure-enthalpy diagram, as presented in Figure 2 for methane.



ure 1. Typical Head versus Capacity Curve

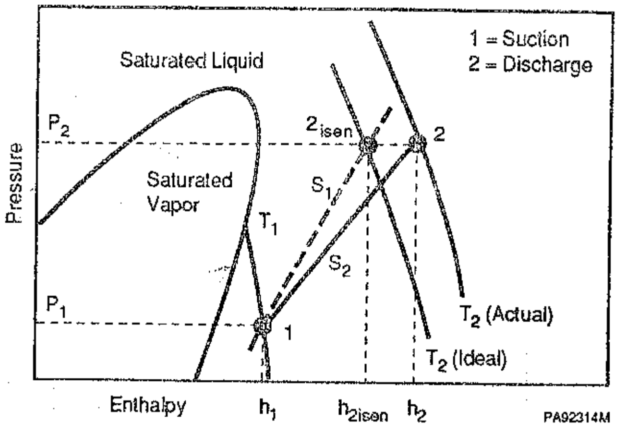


Figure 2. Isentropic Compression Process

The pressure-enthalpy diagram provides all of the thermodynamic information needed to calculate the head for the gas mixture to be compressed. Point 1 represents the suction condition of pressure and temperature. To increase the pressure, a perfectly efficient process must follow a line of constant entropy (internally reversible process), shown in Figure 2 as the dashed line from point 1 to point 2isen. The enthalpy at 2isen minus the enthalpy at 1 is defined as "isentropic head". Isentropic is defined as being measured along a constant entropy path. Notice that the discharge temperature at point 2isen is higher than the suction temperature at point 1. Thus, an ideal compression process increases the temperature of the gas.

However, because no compression process is perfectly efficient, it cannot follow a constant entropy path. It must follow a path of increasing entropy, shown in Figure 2 as the solid line from point 1 to point 2. The discharge temperature at point 2 is even higher than at point 2isen. The enthalpy at point 2 minus the enthalpy at point 1 is the actual enthalpy rise of a real compression process.

The isentropic head is a function only of the gas properties and the pressure ratio. The actual enthalpy rise is also a function of the compressor efficiency, defined as:

$$\text{Efficiency} = \frac{\text{Enthalpy(2isen)} - \text{Enthalpy(1)}}{\text{Enthalpy(2)} - \text{Enthalpy(1)}}$$

This efficiency is known as "isentropic efficiency" or "adiabatic efficiency". Adiabatic means with no heat transfer to or from the surroundings.

ıe heat transfer to the surroundings from most ɘntrifugal compressors is negligible, so adi- ɔatic efficiency is used as a synonym for entropic efficiency and is a suitable means of ɘtermining the efficiency and power consump- ɔn of a compressor.

ıpacity. Capacity is a term used to describe let volumetric flow rate. It is actually the velocity the gas entering the impeller that affects the ɜrformance of the compressor. However, be- ıuse the internal geometry of a compressor is ed, the velocity is directly proportional to the et volumetric flow rate, and flow rate is more ısily measured than velocity.

Figure 3 shows the velocity vectors at the inlet ıd outlet of a single impeller. The resultant vec- · C is the sum of the gas velocity vector W and ɘ impeller rotation vector U. The vector Cu is ɘ resultant gas tangential velocity vector. The locity (U) is perpendicular to the radius, while ɘ relative velocity (W) is tangential to the ıdes; thus, its direction depends on the blade gle. Assuming that the impeller rotates at a nstant speed, the impeller rotation vector u nains constant. Then, the only other thing ıich can affect the resultant vector is the gas locity vector W.

Euler's equation:

$$\text{Head} = U_2 \times Cu_2 - U_1 \times Cu_1 \qquad (1)$$

ɔws that the head rise is related to the velocity ıngles at the inlet and exit of the impeller.

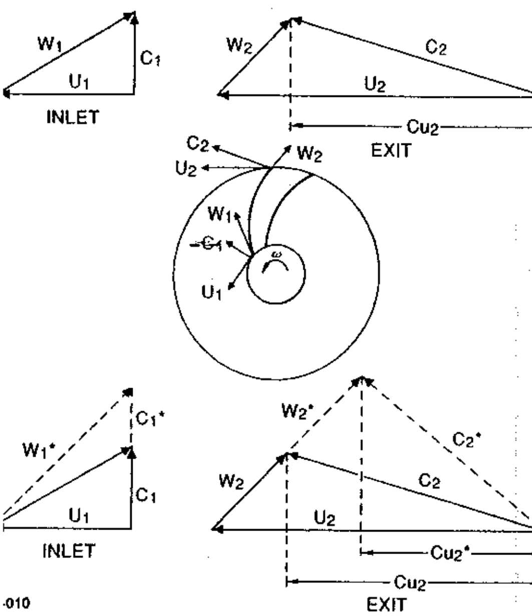


Figure 3. Impeller Velocity Triangles

For simplicity, assume that $Cu_1 = 0$. Thus, Head $= U_2 \times Cu_2$. Any increase in flow (gas velocity, W_2) reduces Cu_2, thus reducing head. This shows that the relative velocity (or flow into the impeller) directly affects the energy imparted to the flowing gas; thus, it directly affects the performance of the compressor.

Effect of Changing Gas Conditions on the Head versus Capacity Curve

Because the head versus capacity curve is the least affected by changes in gas composition and suction temperature, it is preferred over all other curves.

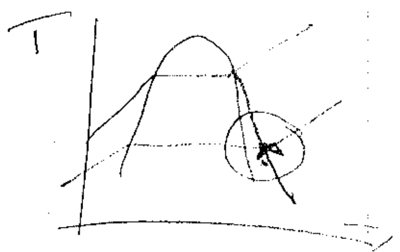
For applications at a relatively low Mach number, the head versus capacity curve is accurate, even if significant changes in gas composition or suction temperature occur. However, for applications that are at a relatively high Mach number (about 0.7 or higher), a small change in gas composition or suction temperature may make a noticeable change in the head versus capacity curve. For these high Mach number applications, even the head versus capacity curve may not be accurate enough for performance evaluation.

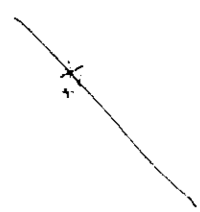
By definition, Mach number is the ratio between the gas velocity and the speed of sound in the gas at the same conditions of pressure and temperature. The compressor Mach number, also called "machine Mach number" or "reference Mach number", is defined by the ASME PTC-10 as:

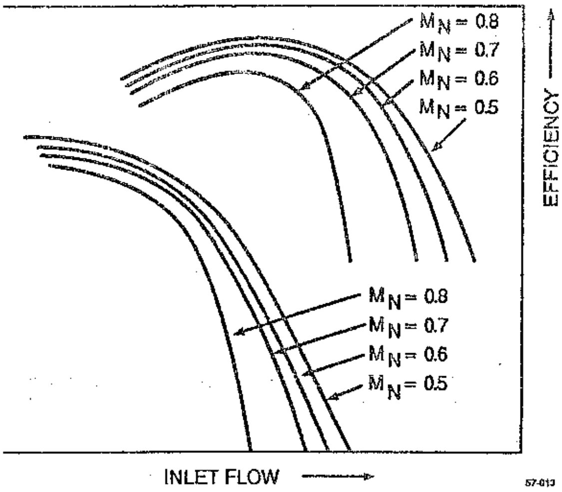
$$\text{Mach Number} = \frac{U_2}{\sqrt{g \times K1 \times R \times Z1 \times T1}} = \frac{0.0001054 \times \text{Diam} \times \text{rpm}}{\sqrt{(K1 \times Z1 \times T1)/SG}} \qquad (2)$$

The effect of higher Mach number is to move the surge limit to a higher flow and move the choke flow to a lower flow. Thus, higher Mach number reduces the flow range from surge to choke for a given speed. A higher Mach number also lowers efficiency slightly. The parameters which increase Mach number are higher speed, higher gas molecular weight, and lower suction temperature. Figure 4 shows the typical effect of Mach number on compressor stage performance.

However, this does not mean that changing Mach number will affect the shape of the head versus capacity curve. As shown in Figure 5, a







ıre 4. Typical Effect of Mach Number on Stage Performance

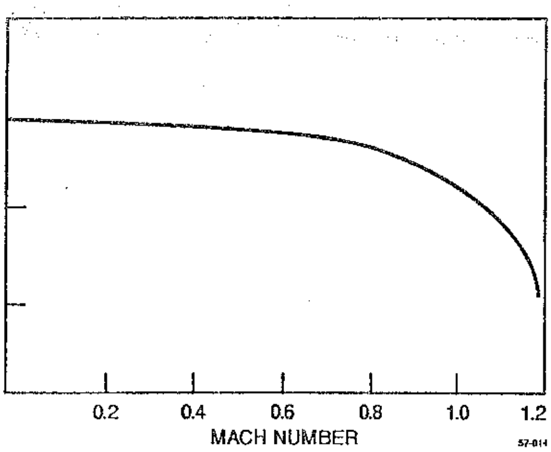


Figure 5. Mach Number Effect

ıthetical curve shape factor, as a function of ı number, remains relatively unchanged until ʌach number increases above 0.7.

ır most applications, the Mach number ıins relatively low and constant (below 0.7), ıe effect on the shape of the head versus ıcity curve is negligible. For relatively low ı numbers (less than 0.7), it takes a very ficant change in suction temperature or gas cular weight to make even a slight change e head versus capacity curve. As an exle, applications for commercial natural gas ɔrmal ground temperatures have relatively Mach numbers. In these applications, a ıge as large as 100°F (56°C) in suction temperature or 15% in gas molecular weight will have very little effect on the shape of the head versus capacity curve.

Examination of Eq. 2 for Mach number shows that pressure does not directly affect Mach number; thus, pressure does not directly affect the shape of the head versus capacity curve. Pressure does affect the compressibility factor (z) which, in turn, affects Mach number. Nevertheless, it takes such a significant change in pressure to affect the shape of the head versus capacity curve that the effects of pressure may be neglected.

Figures 6, 7, 8, and 9 show the effect of Mach number on the head versus capacity curve plotted for the same compressor at four different gas conditions. The four gas conditions cause four Mach numbers: 0.5, 0.6, 0.7, and 0.8. There is almost no change in the curve for a Mach number change from 0.5 to 0.6. Also, there is very little change in the curve for a Mach number change from 0.6 to 0.7. However, the same compressor curve changes noticeably when the Mach number increases from 0.7 to 0.8.

Therefore, for low Mach number applications, unless the application has extremely large changes in suction temperature or gas molecular weight, the head versus capacity curve can be considered valid for any operating condition. However, for applications which are already at a relatively high Mach number (about 0.7 or higher), changing gas conditions that make an additional increase in Mach number may make a noticeable change in the head versus capacity curve. For these high Mach number applications,

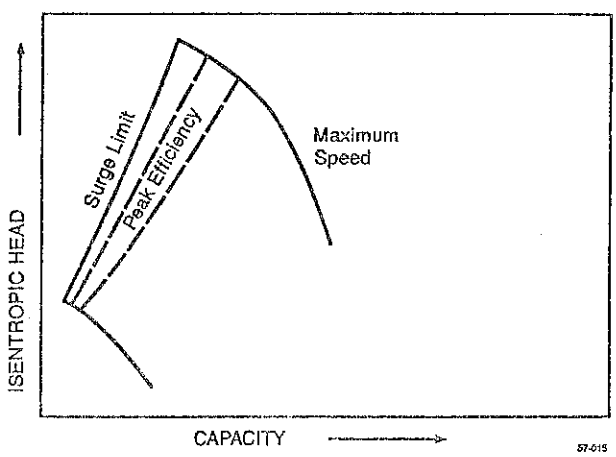


Figure 6. Head versus Capacity Curve at 0.5 Mach Number

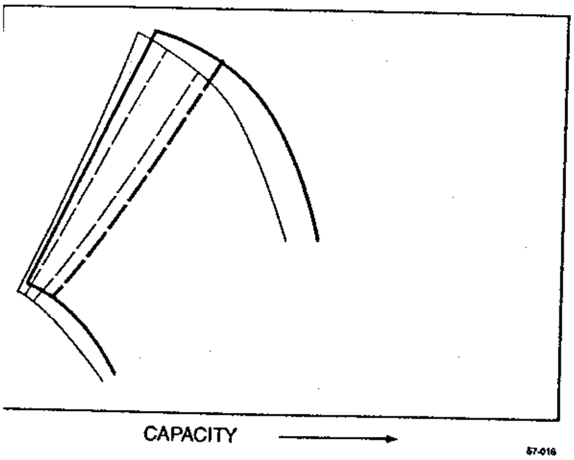


Figure 7. Head versus Capacity Curve at 0.6 Mach Number

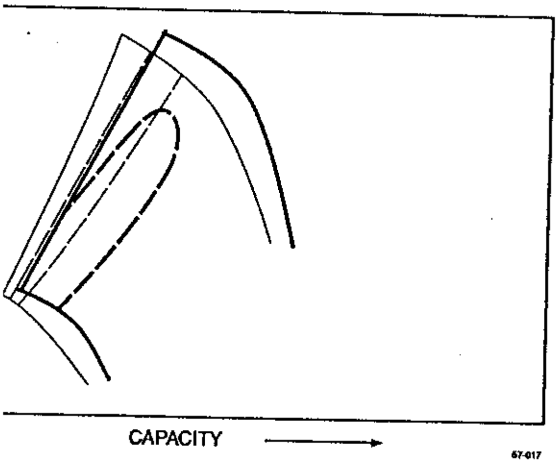


Figure 8. Head versus Capacity Curve at 0.7 Mach Number

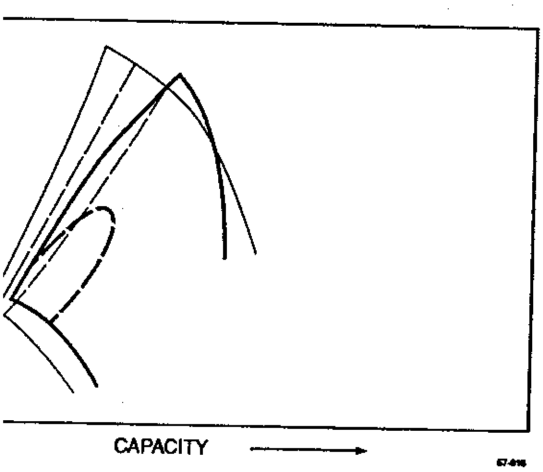


Figure 9. Head versus Capacity Curve at 0.8 Mach Number

even the head versus capacity curve may not be accurate enough for performance evaluation. Computer analysis is better suited for these applications. The computer accurately computes the gas properties and adjusts the compressor performance prediction for every operating condition, high Mach number or low.

Other Compressor Curve Formats

There are other curve formats which have specific, limited purposes, such as the dimensional, semi-dimensional, and composite tandem curves. To plot a performance curve, certain parameters must be held constant. These parameters are gas composition, suction temperature, and pressure (either suction or discharge). The compressor performance curve is plotted based on assumed values for these parameters. If these parameters change, the curve may no longer be valid.

Dimensional Curve. The purpose of the dimensional curve is merely to aid the potential purchaser in determining the operating range of the compressor. Plotted in units of pressure versus standard volumetric flow rate (Figure 10), this curve is solely for the purpose of bid evaluation.

If the curve is based upon a constant suction pressure (P1), the ordinate is a range of discharge pressure (P2). If the curve is based on a constant P2, then the ordinate is a range of P1. In order to have the ordinate in the customary ascending order, the dimensional curve appears upside-down when plotted for a constant P2.

The dimensional curve enables the user to read the power and speed required for a specified

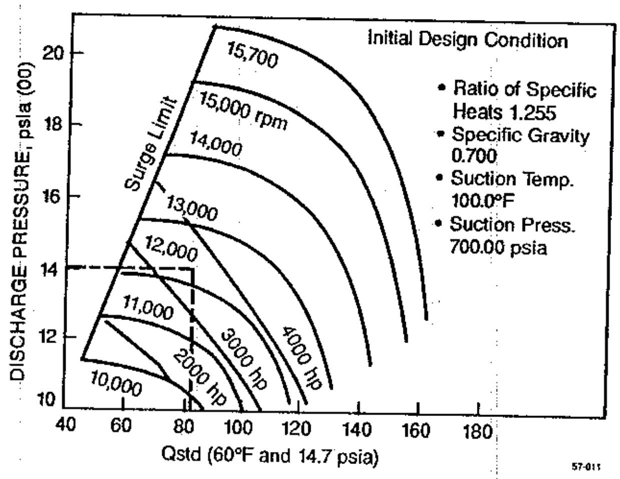


Figure 10. Dimensional Curve

ɪerating condition. Once the decision to pur-ıase has been made, the dimensional curve no longer of any value. **The shape of the mensional curve changes significantly for ıy gas composition, suction temperature, or essure other than those assumed as base ırameters.** This makes the dimensional curve ımpletely useless for any parameters other than ɔse printed on the curve.

ɜmi-Dimensional Curve. The semi-dimensional ırve is identical to the dimensional curve, except ɜt the abscissa and the ordinate scales and the es of constant power have been divided by the ıse pressure. This is useful for applications that ıve very stable, constant suction temperature ıd gas composition but with fluctuating suction ıd discharge pressures. **The semi-dimen-onal curve is accurate for any pressures, but ɛe the dimensional curve, is limited to only ɜt gas composition and suction temperature inted on the curve.** To use the semi-dimen-ɔnal curve simply multiply the values of pres-ıre, standard flow rate, and power by the actual ıse pressure.

Assume that the semi-dimensional curve in gure 11, based on a constant P1 of 500 psia, is ɜed for an operating condition that actually has P1 of 700 psia. Also, assume all other base ɪnditions remain the same. The desired P2 is ɪ00 psia and the power available is 3500 hp:

$$hp/P1 = 3500/700 = 5.0$$

$$P2/P1 = 1400/700 = 2.0$$

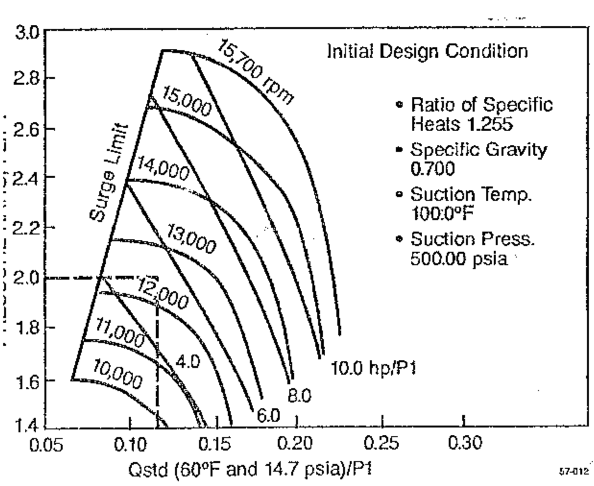


Figure 11. Semi-Dimensional Curve

From Figure 11 at P2/P1 = 2.0 and hp/P1 = 5.0, $Q_{std}/P1$ should be 0.115:

$$Q_{std}/P1 \times P1 = Q_{std}$$

$$0.115 \times 700 = 80.5$$

Thus, the standard volumetric flow (Q_{std}) to be expected is 80.5 mmscfd.

Composite Tandem Curve. Dimensional and semi-dimensional curves are also used to depict the performance of a multiple-body tandem compressor, as if the tandem compressor were a single-compressor body. This type of curve is known as a composite curve. The composite curve is only useful to assist the purchaser to see the operating range of the tandem compressor unit.

In addition to the assumptions which are made to plot a single-body curve, more assumptions must be made to plot a composite tandem curve. The interstage pressure drop, heat extracted via interstage cooling, flow shrinkage due to inter-stage condensation, and sidestream flow(s) must be assumed to remain constant. If the actual operating conditions do not agree exactly with these assumptions, then the composite tandem curve is of little value.

The only accurate way to evaluate the perfor-mance of a multiple-body tandem compressor is to evaluate each body individually, using the head versus capacity curve for that body. Unfor-tunately, using individual head versus capacity curves to determine the overall performance of a multiple-body tandem is a time consuming, trial-and-error calculation which is best suited for computers.

PC Computer Program

Until recently, the best tool available for you to accurately predict the performance of your com-pressor for varying operating conditions has been the head versus capacity curve.

However, if you have a high Mach number condition, even the head versus capacity curve may not be accurate enough.

Also, using individual head versus capacity curves to predict the overall performance of a multiple body tandem is a time consuming trial-and-error calculation. But advancements in the power of personal computers in the last few years have made it possible now to overcome this prob-lem of Mach number effect.

ıe same computer program which Solar uses 'edict compressor performance is now avail- to you. This program computes the gas erties and the performance prediction of compressor for any operating condition and ny Mach number, using a personal computer a math coprocessor.

'ailable for all units, this program is a more enient and accurate advancement over ıse of performance curves. It is the next- ration, performance prediction and evalua- ool.

e usefulness of composite overall tandem rmance curves is minimal because of all the nptions which must be made for interstage itions as well as suction conditions. Using dual-body, head versus capacity curves for ıll tandem-performance estimates is a time- ıming, trial-and-error calculation. Therefore, ıe with tandem compressors can significant- ıefit from this program.

ɜ head versus capacity curve does not ɟe significantly with changing gas condi- when the Mach number is low. At high Mach er, however, the head versus capacity changes with changing gas conditions, so sefulness for performance evaluation is ɟ. The PC program **corrects for changing number**, making it very valuable for those ːations at high Mach number (over 0.8).

all single-body compressors operating at ely low Mach number (less than ap- ıately 0.8), a head versus capacity curve is etely satisfactory for any performance ıtion. For those applications this program is ıvenience rather than additional perfor- ɜ evaluation capability; however, the con- ıces are significant:

ɔmpressibility factors and ratio of specific at is calculated internally, eliminating table ɔk-up or separate calculations.

nost any combination of 3 independent vari- les may be specified to obtain a checkpoint, ninating trial-and-error solutions.

ıgs of physical limits exceeded are printed, over maximum discharge pressure.

eckpoints can be printed for recordkeeping.

Checking the Condition of an Operating Compressor

To check the physical condition of an operating compressor, calculate the actual efficiency of the compressor using the adiabatic efficiency equation 8. Then, compare the actual efficiency and speed to the efficiency and speed that the head versus capacity curve for the compressor shows it should have. If they agree within measurement tolerances, about 6%, then the compressor is in good condition. If the actual efficiency is more than 6% lower than the curve says is should be, then investigate for reasons. Some causes of performance changes are contamination from foreign material such as paraffin, asphalt, sand and salt. Erosion and corrosion can cause performance deterioration. Worn or heat-damaged seals may cause excessive internal recirculation, thus lowering the efficiency.

Centrifugal compressors have a peak efficiency island. If operation is maintained within this island, the power required and, thus, the fuel consumed are minimized. Find out where the operating point is on the curve by calculating head and inlet volumetric flow with equations 3 and 4. Read the head versus capacity curve for the compressor to determine what the efficiency and speed should be at that operating condition.

If the compressor is not in the peak efficiency island, determine if something can be done to move toward the peak efficiency. For example, for multiple units in parallel, starting another unit will move the operating point of each compressor to a lower flow. Likewise, shutting down a unit will move the operating point of each compressor to a higher flow. Adding units at upstream or downstream pipeline stations will lower the head and, thus, move the operating point lower on the curve. Shutting down units at upstream or downstream stations will likewise move the operating point higher on the curve.

Prolonged operation in the lower efficiency area of the curve can cost a lot of fuel. If the operating conditions are expected to stay in the lower efficiency area, restaging the compressor may be advantageous. Restaging is simply changing the impellers and/or stators (and, thereby, the curve), so that it is most efficient where it is going to operate most of the time.

To maximize efficiency, keep the compressor in the peak efficiency area of the head versus capacity curve.

ESTING CENTRIFUGAL COMPRESSORS

uring the service life of a gas turbine-driven, entrifugal compressor set, it may be desirable evaluate the thermodynamic performance of e centrifugal gas compressor.

Normally, the object of a field performance test the determination of:

- Adiabatic Head (Head) versus actual inlet volume flow (Q_{act}) characteristic of the centrifugal compressor for the complete operating range, or design point.
- Adiabatic Efficiency (EFFY) versus inlet volume flow (Q_{act}) of the centrifugal compressor for the complete operating range, or design point.

Then, the measured efficiency and speed of e compressor are compared to the predicted iciency and speed from the performance rve. Some difference between measured and edicted values is normal. However, a trend of reasing difference over time is an indication at maintenance may be necessary.

ST CONDITIONS

. The test should be performed when pipeline conditions allow operation near the compressor design conditions.

. Steady state conditions should be established before data is taken. A minimum fifteen minute interval between test settings is suggested.

. In case serious inconsistencies in data arise, either during the test or during the data analysis, the test should be rejected in whole or in part, and the necessary repetitions should be made to comply with the test objective.

Any deviations in the test procedure from the guidelines presented in this report should be noted on the data sheet.

STRUMENTATION AND DATA QUIREMENTS

perly calibrated and selected instrumentation he primary requirement for obtaining satisfac- y field test data. A recommended list is furhed in Table 1.

f provisions are made during the construction ase of the compressor station to allow the tallation of the necessary instrumentation, ch as temperature wells and pressure taps, test can be conducted with a minimum of interruption to the operation. Valves should be installed ahead of pressure gauges, to allow changeout during engine-compressor operation.

The Field Test Check List will help in the preparation of the test.

The following compressor data should be taken:

- Inlet and discharge gas pressure
- Inlet and discharge gas temperature
- Gas flow
- Compressor speed
- Gas anslysis (mole percent of constituents
- Atmospheric conditions

Pressure Measurements

1. Compressor inlet and discharge pressure measurements should be made with calibrated pressure gauges or dead-weight gauges. Pressure gauges should be selected so that the minimum scale division is no more than 1% of the actual pressure reading. Dead-weight gauges are the preferred instruments and are normally capable of measuring pressure to within 0.5% of the actual pressure reading.
2. The measurement should be made as close as possible to the compressor flanges. If the point of measurement is located more than 10 feet from the compressor flange, the pressure measurement should be corrected to flange conditions by consideration of the calculated pressure drop. Package gauges are not acceptable due to sensing locations.
3. Pressure measurements should be made, if possible, at two locations at both the compressor inlet and discharge (in the same plane approximately 90 degrees apart).

Temperature Measurements

1. Compressor inlet and discharge gas temperatures should be measured with ASME type mercury-in-glass thermometers (or equivalent measuring devices) with a maximum of 0.2F graduations.
2. The thermowells should be filled with oil or mercury.
3. If the point of measurement is located more than 10 feet from the compressor flange, the temperature measurements, in particular on the discharge side, should be corrected to flange conditions by consideration of the calculated heat transfer.

Table 1. Recommended List of Calibrated and Selected Instrumentation

Data	Instrument	Range	Precision	Number (per Package)
Barometric Pressure	Barometer		0.01" Hg	1
Ambient Temperature	Hg thermometer	0-150°F	0.5°F	1
Suction Pressure	Dead weight/calibrated gauge		1 psi	1-2 (per compressor)
Discharge Pressure	Dead weight/calibrated gauge		1 psi	1-2 (per compressor)
Suction Temperature	Hg thermometer	0-150°F	0.2°F	1-2 per compressor)
Discharge Temperature	Hg thermometer	20-420°F	0.2°F	1-2 (per compressor)
Flow Meter Static Pressure	Dead weight/calibrated gauge		0.1 psi	1
Flow Meter Diff.	Manometer/Recorder	0-100"H_2O	0.1" H_2O	1
Flow Meter Temperature	Hg thermometer	0-150°F	0.2°F	1
Gas Comp. (Power Turbine x GB Ratio) Speed (counts)	Digital counter			1
Gas Sample Bottles				1

. Temperature measurements should be made, if possible, at two locations at both the compressor inlet and discharge (in the same plane approximately 90 degrees apart).

. Spare thermometers are highly recommended.

is Flow Measurement

. Gas flow measurement should be made by orifice meter run or equivalent flow measurement device. The size of the device should conform to recommendations made by the American Gas Association (Gas Measurement Committee Report #3) for the specific pipe size and flow range. Flow measurements can be made at either the compressor inlet or discharge side.

The static pressure at the flow measurement device should be measured with a calibrated pressure gauge or dead-weight gauge of the same accuracy as the compressor inlet and discharge pressure gauges.

Gas temperature at the flow measurement device should be measured with a mercury-in-glass thermometer (or equivalent) with a maximum of 0.5F graduations. (Thermowell to be filled with oil or mercury).

The differential pressure across the flow measurement device should be measured with a water manometer (or equivalent).

Compressor Speed

1. The compressor speed should be measured with an electronic digital counter connected to the power turbine magnetic pick-up.
2. Gearbox ratio, if any, should be recorded.

Gas Analysis

Samples of the gas being compressed should be obtained during each day of testing. The samples should be properly identified, including the location where the sample was obtained, date and time of day. The valves on the sample bottle should be closed tightly to prevent any leakage. The sample should be analyzed by a qualified laboratory so that the percent volume of each constituent in the gas can be accurately established.

Atmospheric Conditions

1. Barometric pressure should be recorded for each test setting with an aneroid or mercury barometer.
2. If a mercury barometer is used, the temperature at the barometer should be recorded to make any necessary temperature corrections to the barometric reading.
3. If a barometer is not available, record site elevation.

IELD TEST CHECK LIST

ıspection and Preparation

ıe following checks and calibration should be ərformed prior to test with items checked off as ɔmpleted:

		Completed
Pressure gauges used for suction, discharge and orifice checked against a standard orifice	(Suction)	______
	(Discharge)	______
	(Orifiice)	______
Thermometers or thermocouples used for suction, discharge and orifice checked against each other in the temperature range that will exist during the test	(Suction)	______
	(Discharge)	______
	(Orifice)	______
		______
Thermometers checked for proper immersion depth and stem insulation		______
Compressor flow and fuel orifices cleaned, inspected and checked for size		______
If flow recorder used for orifice pressure differential, check readout against U-tube manometer		______
Specific gravity of any manometer fluids other than water or mercury checked		______
Obtain or make schematic of station showing critical dimensions, diameters, tap offs, scrubber locations, measurement and orifice locations		______
Check all possible sources of flow leakages which could affect compressor flow determination (bypass lines, vent lines, spurs, fuel tap-offs, etc.)		______

ɔmpressor Flow Orifice Data

ifice Location =_____ (suction or discharge)

fice Size =_____

ɔe I.D. =_____

ɔe Pressure Taps =_____ (flange or pipe)

ıssure Tap Location =_____ (upstream or downstream)

ΞST PROCEDURE

ȷure 12 is a typical gas compressor head vs. lumetric flow performance curve as normally ovided by the manufacturer. A curve of this ɔe can be used for most variations in gas con- :ions of pressure, temperature, and gravity. For ɔper analysis, the actual test data should be ‹en at constant compressor speed and varying əssure ratio. This can normally be done by ınipulation of the compressor suction and dis- arge valves. It is suggested that at the most, five evenly spaced test settings be selected at or near the compressor design speed to provide a full flow range curve, or optionally a design/ operating point be run.

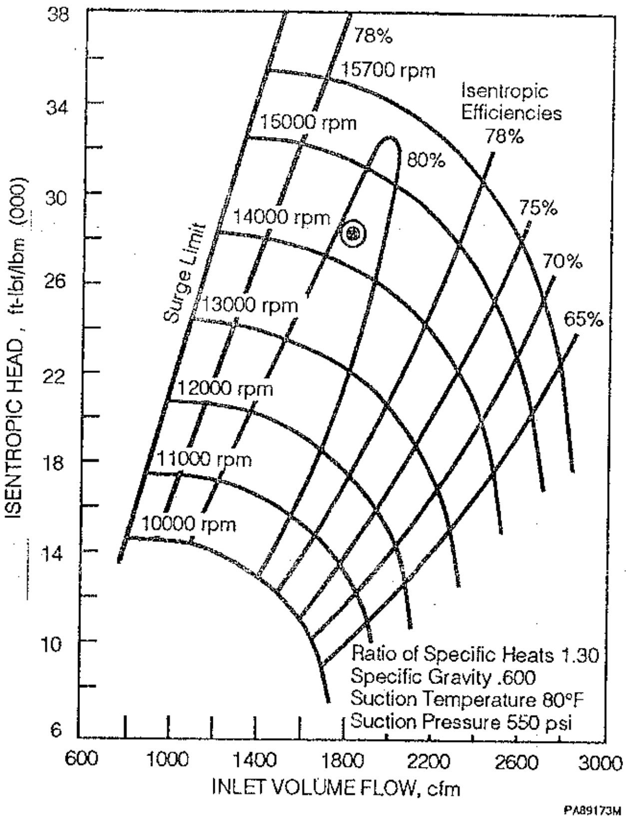


Figure 12. C3063RGA Compressor Performance

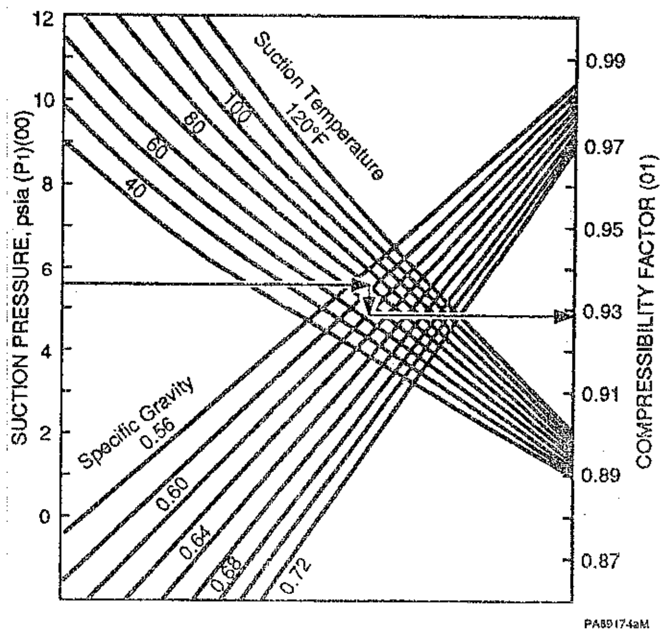


Figure 13. Natural Gas Compressibility Factor

ATA ANALYSIS

ıe test data should be converted into param-
ɜrs as shown on Figure 12 by use of the follow-
ȷ equations: (It should be noted that these
uations are approximate.)

ɜntropic Head

$$\text{Head} = C1\,\frac{(T1 + C2)Zave}{(K\text{-}1/K)SG}\left[\left(\frac{P2}{P1}\right)^{\frac{K-1}{K}} - 1\right] \quad (3)$$

tual Inlet Flow

$$Q_{act} = C3\,Q_{std}\,Z1(T1+C2)/P1 \quad (4)$$

wer Required

$$PWR = C4\ \text{Head}\ Q_{std}\ SG/(EFFY \times EMCH)$$

$$EFFY = \text{Adiabatic Efficiency (\%)} \quad (5)$$

$$EMCH = \text{Mechanical Efficiency* (Decimal)}$$

charge Temperature

$$T2 = T1 + \frac{(T1 + C2)}{EFFY/100}\left[\left(\frac{P2}{P1}\right)^{\frac{K-1}{K}} - 1\right] \quad (6)$$

ssure Ratio (P2/P1)

$$\frac{P2}{P1} = \left[\frac{C5\ PWR\ (EMCH \times EFFY)(K-1)}{Q_{std}(T1+C2)Z_{ave}\ K} + 1\right]^{\frac{K}{K-1}} \quad (7)$$

Isentropic Efficiency (%)

$$EFFY = \frac{(T1 + C2)}{(T2 - T1)}\left[\left(\frac{P2}{P1}\right)^{\frac{K-1}{K}} - 1\right] \times 100 \quad (8)$$

NOMENCLATURE

Compressor Data Symbol	Nomenclature
EFFY	Isentropic compressor efficiency (%)
EMCH	Mechanical Efficiency* (Decimal)
Head	Isentropic head developed
K	Ratio of specific heats
N_c	Gas compressor speed
P_b	Site barometric pressure
P1	Gas compressor inlet static pressure
P2	Gas compressor discharge static pressure
P_f	Differential pressure at compressor flow device
P_f	Static pressure at compressor flow device
PWR	Power required by compressor, or available from driver
q	Work factor
Q_{act}	Volumetric flow at inlet conditions
Q_{std}	Standard volumetric flow
SG	Gas specific gravity
T1	Gas compressor inlet temperature
T2	Gas compressor discharge temperature
T_f	Gas temperature at compressor flow device
Z	Compressibility factor

The K value of the gas should be determined for the average or inlet gas temperature in the compressor. After the test data has been reduced in terms of Adiabatic Head, inlet volumetric flow and compressor efficiency, a **preliminary** comparison can be made with the compressor manufacturer's curve. The power absorbed by the compressor, which should be equivalent to the gas turbine output power, is derived from equation (5). A sample problem follows to show how a preliminary comparison could be made.

UNITS	AMERICAN	S.I.	METRIC
C1	53.350	287.04	29.269
C2	459.67	273.15	273.15
C3	19.631	97.5×10^{-6}	61.764×10^{-6}
C4	0.16057	34.0×10^{-6}	351.71×10^{-6}
C5	0.11674	102.45	97.1
Head	$ft\text{-}lb_f/lb_m$	J/kg_m	$m\text{-}kg_f/kg_m$
PWR	hp	kW	kW
P	psia absolute	kPa absolute	bara absolute
Q_{act}	cfm	m^3/sec	m^3/min
Q_{std}	mmscfd (60°F/14.7 psia)	sm^3/hr (15°C/760mm)	Nm^3/hr (0°C/760mm)
T	°F	°C	C

*Mechanical Efficiency is approximately 98 percent

Z can be approximated with Figure 13.

The final comparison of compressor performance data should be reduced using a computer program utilizing an equation of state to calculate isentropic head and efficiency. The equation of state calculates total enthalpy values at suction and discharge conditions and compares the results with isentropic values expected. This PC computer program is available from Solar's Systems Analysis Department.

EXAMPLE

This sample gas compressor performance evaluation is for a C3063RGA compressor (Figure 12).

- Suction Pressure = 550 psia
- Discharge Pressure = 1000 psia
- Suction Temperature = 80°F
- Discharge Temperature = 180°F
- Q_{std} = 100 mmscfd
- SG = 0.600
- K = 1.300
- Gas Compressor Speed = 14,500 rpm

1. From Figure 13, determine Z.

   P1 = 550 psia

   T1 = 80°F + 460 = 540°R

   SG = 0.60

   Therefore, Z = 0.929

2. Calculate adiabatic (or isentropic) head from equation (3).

$$\text{Head} = \frac{(T1+C2)(Z)}{(SG)} \; \frac{(C1)(K)}{(K-1)} \left[\left(\frac{P2}{P1} \right)^{\frac{K-1}{K}} - 1 \right]$$

$$= \frac{(540)(0.929)}{(0.60)} \; \frac{(53.3)(1.3)}{(1.3-1)} \left[\left(\frac{1000}{550} \right)^{\frac{1.3-1}{1.3}} - 1 \right]$$

$$= 28{,}567 \text{ ft-lb}_f/\text{lb}_m$$

3. Calculate inlet flow (Q_{act}) from equation (4)

$$Q_{act} = \frac{C3\,(Q_{std})(T1+C2)(Z)}{P1}$$

$$= \frac{19.63\,(100)(540)(0.929)}{550}$$

$$= 1790 \text{ cfm}$$

4. Plot Head and Q_{act} values on Figure 12 and record efficiency and rpm.

   EFFY = 80%

   rpm = 14,300

5. Calculate compressor test efficiency from package data and equation (8).

$$\text{EFFY} = \frac{T1 \left[\left(\frac{P2}{P1} \right)^{\frac{K-1}{K}} - 1 \right]}{T2-T1} \text{ X } 100$$

$$= \frac{540 \left[\left(\frac{1000}{550} \right)^{\frac{1.3-1}{1.3}} - 1 \right]}{640-540} \text{ X } 100$$

$$= 80\%$$

6. Calculate horsepower required from equation (5).

$$\text{PWR} = \frac{C4 \text{ X (Head)}(Q_{std})(SG)}{\text{EFFY X EMCH}}$$

$$= \frac{0.16057\,(28567)(100)(0.60)}{80(0.98)}$$

$$= 3517 \text{ hp}$$

As can be seen in this idealized case, predicted efficiency matches test results, and the measured compressor speed of 14,500 rpm is slightly higher than the predicted speed of 14,300. This discrepancy is not significant. But, if the difference between actual speed and predicted speed increases, compressor contamination may be increasing.

TEST REPORTING

A minimum of five points plus mild surge are obtained at the design speed. The first data point is obtained near choke and the sequence is continued with points of increasing pressure ratio while maintaining constant speed and stabilized flow.

In order to proceed from one test point to the next, the flow is decreased in increments by throttling the discharge valve. Once a given setting is reached in the compressor, the discharge emperature is monitored until it has been deternined that the temperature has stabilized, before proceeding to record data. The minimum time between settings is usually more than 20 minutes.

To determine mild surge, the flow is reduced intil detecting the first sign of instability, which ould consist of manometer oscillation or **surge ıoise**. Since the detection of needle oscillation r noise can be uncertain, a more precise proceure can be applied; it consists of monitoring the scilloscopes displaying the compressor rotor rbits, at suction and discharge ends, for orbit rowth, and monitoring the rotor frequency pectrum in a real-time analyzer for increases in ıe amplitude of vibration. In some cases, when esting at very low pressure, any of the methods ist described may fail to detect the beginning of stability; in that case, the procedure consists of lotting the isentropic head coefficient versus the let flow coefficient for several data points in the ea of expected surge, to determine the point here the head reaches its peak, which is then efined as the **mild surge point**.

Once the mild surge point is determined, the ow is increased to the minimum necessary to abilize the flow, and data is recorded. Then the w is again reduced to determine the mild surge oint.

During the compressor aerodynamic test, in ldition to the entire design speed line, the mild rge points of at least two more speeds are etermined to define the compressor **surge line limit of stability**.

When determining the beginning of instability ild surge point), the compressor should not be aintained in that condition more than the miniım time required to record the parameters that termine its flow location.

Test results are usually presented in the dimennal form of inlet volume flow versus isentropic ad. However, because it is unlikely that the t points can all be taken at exactly the same speed, the test data analysis relies heavily on the non-dimensional map of isentropic head coefficient, efficiency and work factor versus inlet flow coefficient.

NON-DIMENSIONAL CHARACTERISTICS

Inlet Flow Coefficient

The inlet volume flow is conventionally non-dimensionalized by referring it to a fictitious flow, corresponding to the impeller tip velocity passing through the projected frontal area of the impeller. Thus, the **inlet flow coefficient** is defined as:

$$\Phi 1 = \frac{Q1}{(\pi \times D2^2/4) \times U2} \qquad (9)$$

$$\Phi 1 = \frac{700.3}{(D2)^3} \times \frac{Q1}{N} \qquad (10)$$

Isentropic Head Coefficient

The isentropic head is non-dimensionalized by referring it to a hypothetical dynamic head, corresponding to the impeller tip velocity. Thus, the **isentropic head coefficient** is defined as:

$$\psi_{isen} = \frac{H_{isen}}{U2^2/2g} \left(\frac{1838.3}{D2}\right)^2 \times \frac{H_{isen}}{N^2} \qquad (11)$$

$$\psi_{isen} = 2g \times J \times Cp \times T1 \times \left[(P2/P1)^{\frac{k-1}{k}} - 1 \right] / U2^2 \qquad (12)$$

From this relationship, and known gas properties such as specific heat at constant pressure (Cp), specific heat ratio (k), and gas temperature at compressor inlet (T1), it is possible to determine the pressure ratio (P2/P1) or the discharge pressure (P2), if the suction pressure (P1) is also known. Obviously, the rotational speed (N) and the impeller tip diameter (D2) are required to calculate the tip speed (U2).

Many technical publications use a definition of head coefficient based on a hypothetical dynamic head ($U2^2/g$), thus producing a value of ψ which is half of the one obtained by formulas 11 and 12.

Isentropic Efficiency

The efficiency is also a function of the inlet flow coefficient. As the flow coefficient is decreased, impeller positive incidence increases and eventually positive blade stalling occurs, accompanied a decrease in blading efficiency. Conversely, the flow coefficient is increased, negative lling or choke is eventually reached, also :ompanied by decreased blade efficiency.

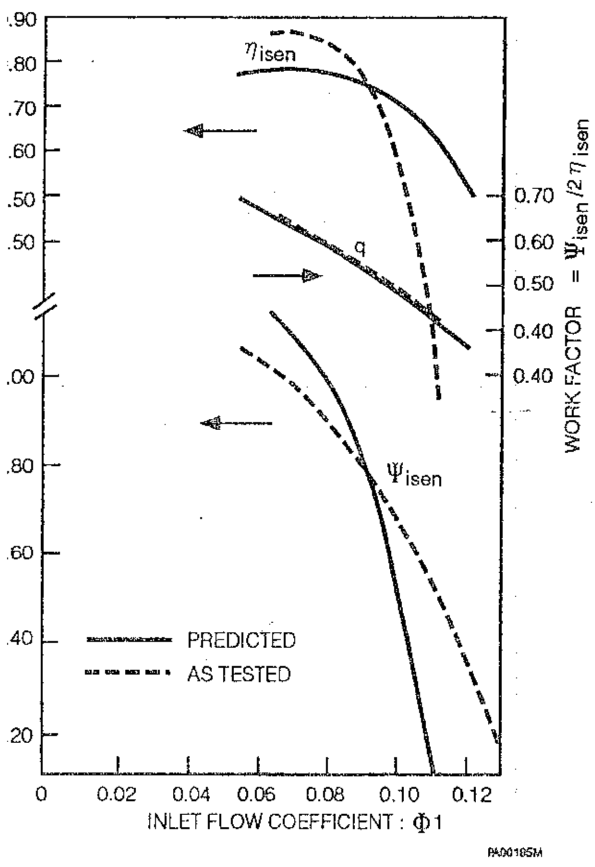
Thus, a unique relationship exists between ciency and flow coefficient, as well as be- ›en the head coefficient and flow coefficient for jiven stage. This relationship is shown by iensionless performance maps of ψ_{isen} and ›n versus $\Phi 1$, as shown in Figure 14.

The previous statement of unique relationship jlects Mach number and Reynolds number :cts; therefore, it is valid only when assuming ›ration within a certain range of Mach number 1 Reynolds number, where the differences in ir effects are negligible.

rk Factor or Actual Head Coefficient

› work factor (q) is a non-dimensional param- r which relates the isentropic head coefficient he isentropic efficiency.

The work factor is obtained by referring the ual head to twice the hypothetical dynamic ıd corresponding to the impeller tip velocity.



gure 14. Flow Range, Head and Efficiency Test Result Example

Thus, the work factor or **actual head coefficient** is defined as:

$$q = \frac{H_{act}}{U2^2/g} \tag{13}$$

$$\therefore q = \frac{9495}{D2}^2 \times \frac{Z_{av}}{SG} \times \frac{k}{k-1} \times \frac{(T2-T1)}{N^2} \tag{14}$$

Also, since:

$$q = \frac{H_{isen}/\eta_{isen}}{U2^2/g} \tag{15}$$

$$\therefore q = \frac{\psi_{isen}}{2\,\eta_{isen}} \tag{16}$$

The work factor versus inlet flow coefficient characteristic is essentially a straight line, except near surge and stroke. This peculiarity makes it a useful tool to determine the average isentropic efficiency versus inlet flow coefficient characteristic based on scatter data obtained during testing. The isentropic head coefficient versus inlet flow coefficient characteristic is a curve of decreasing slope towards surge which is not difficult to average from test data. The accurate averaging of the efficiency versus flow curve is much more difficult to achieve due to its positive and negative slopes, as its maximum value is located somewhere between surge and choke. Therefore, a standard procedure consists of averaging the isentropic head and work factor versus inlet coefficient lines, and then calculating from them the average efficiency line.

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APPENDIX

lead versus Capacity Curves for Single-Body Compressors

'he performance curve shows, on coordinates of ıead and inlet flow, lines of constant speed rpm), lines of constant adiabatic efficiency (%), nd a single line showing the surge limit. In merican units, Head is expressed in ft-lb_f/lb_m nd inlet flow is expressed in cubic feet per ninute (cfm). This type of curve is most often sed to depict compressor performance beause it is only slightly affected by changes in the ase conditions of pressure, temperature, and as composition. Head and inlet flow can easily e converted to any desired units with these quations.

A sample head versus flow curve has been lotted for a theoretical compression requireient. Then the same curve has been replotted ır each of the following changes:

- Increase SG from 0.7 to 0.8
- Decrease T1 from 100 to 50
- Decrease SG from 0.7 to 0.6
- Increase P1 from 500 to 700
- Increase T1 from 100 to 150
- Decrease P1 from 500 to 425

Examples of each are in Figures 15 to 22. It can be seen that these relatively significant changes in base conditions cause only moderate changes in the head vs. flow curve. Note that changes in SG and T1 produce only moderate changes in the curve, and changes in P1 produce almost no change at all. Use the transparency of initial conditions supplied with this book to overlay the changed conditions, observe how little the curve changes.

Also plotted on these example curves is a theoretical surge control line. This surge control line is plotted on each curve without adjusting the surge control calibration. This demonstrates that moderate changes in the base conditions have very little, if any, effect on the protection given by the surge control system.

The last sample plot is based on the same compressor and same base conditions, but is based on a constant discharge (P2) pressure rather than on a constant suction pressure. It can be seen that virtually no change occurs.

The design operating point is shown on each example performance curve with ◉ . Notice that changes in the base conditions do not affect the curve very much, but these changes do affect where the operating point falls on the curve.

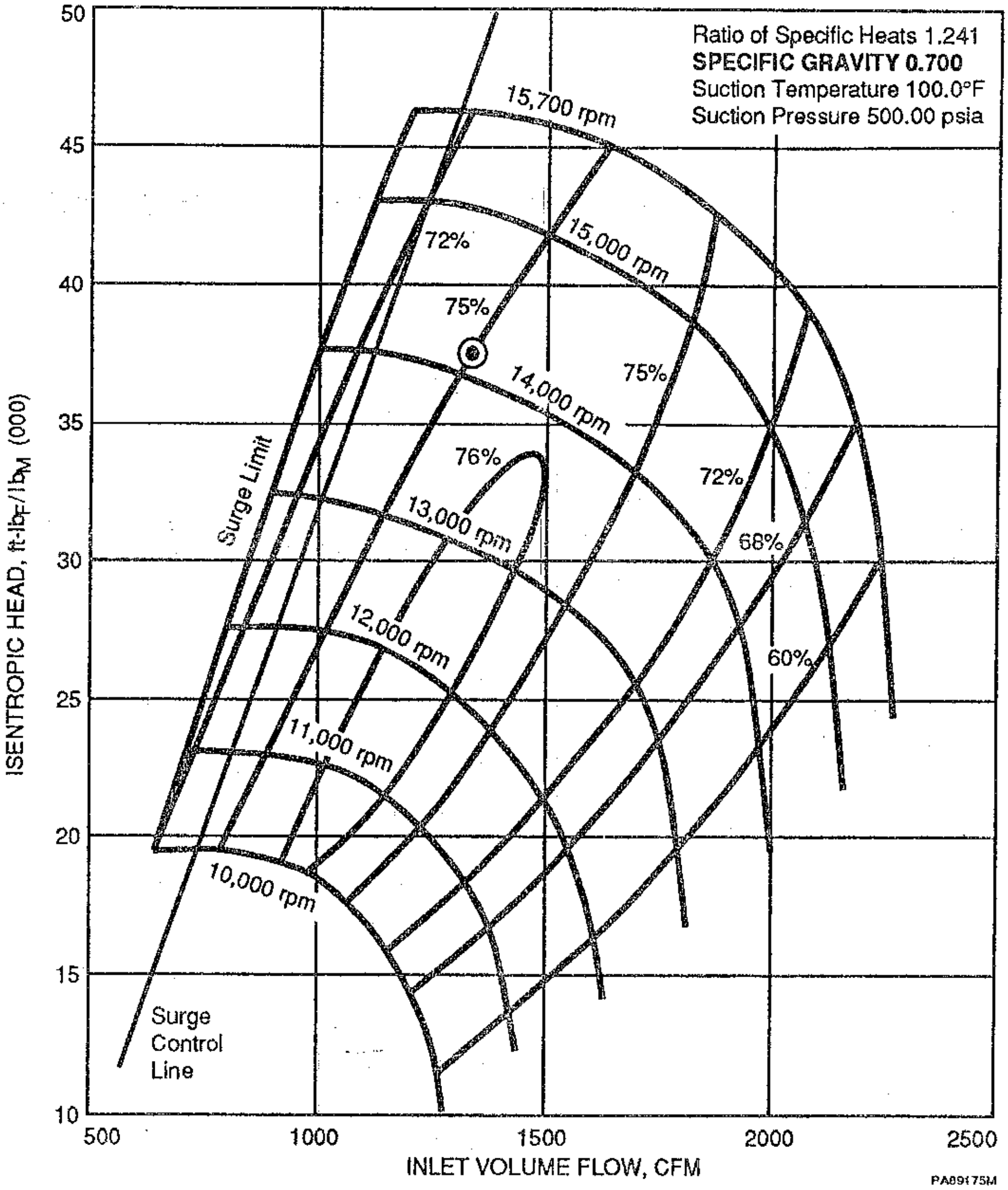

Figure 15. Initial Design Condition

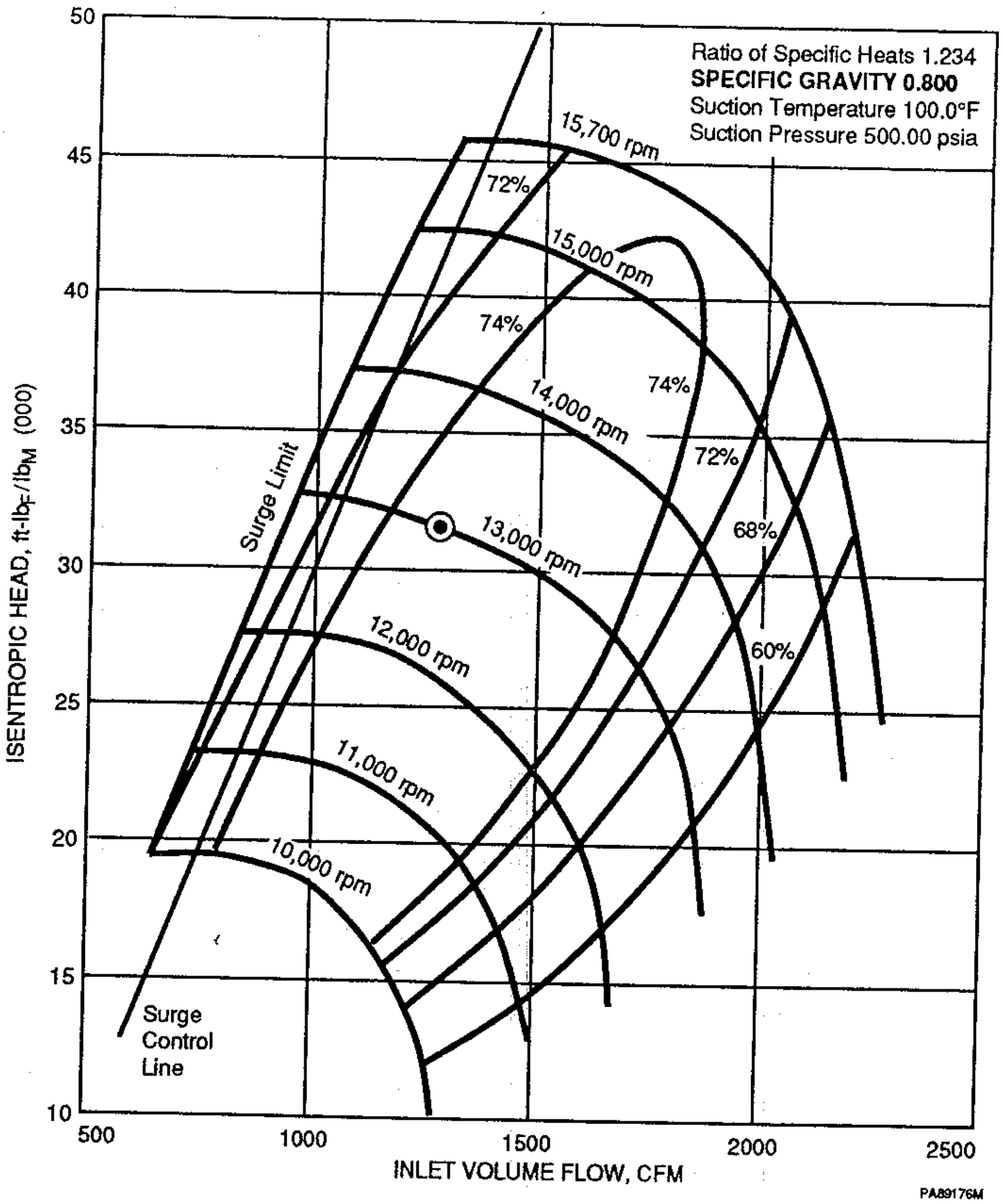

Figure 16. Higher Specific Gravity (SG was 0.7)

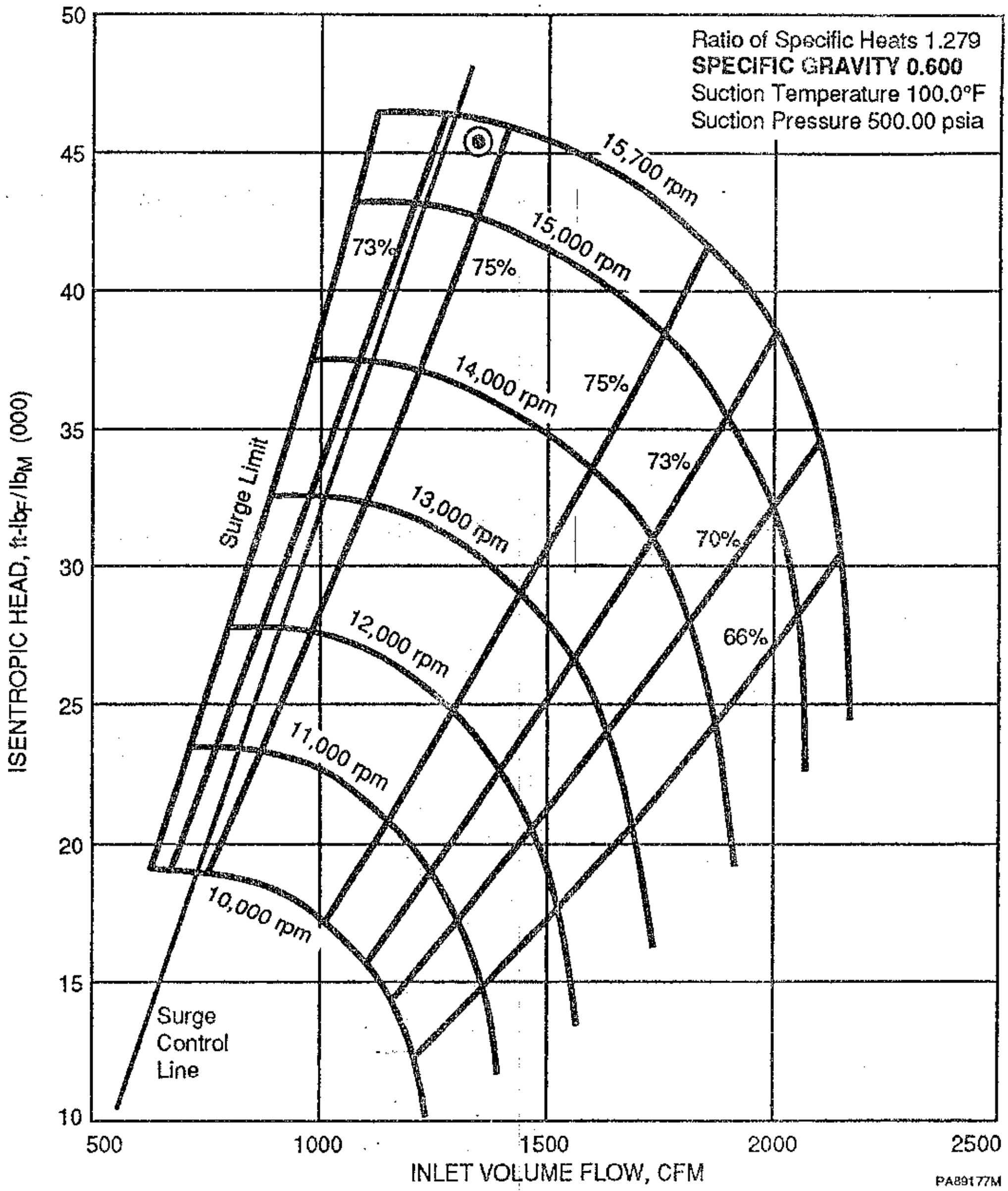

Figure 17. Lower Specific Gravity (SG was 0.7)

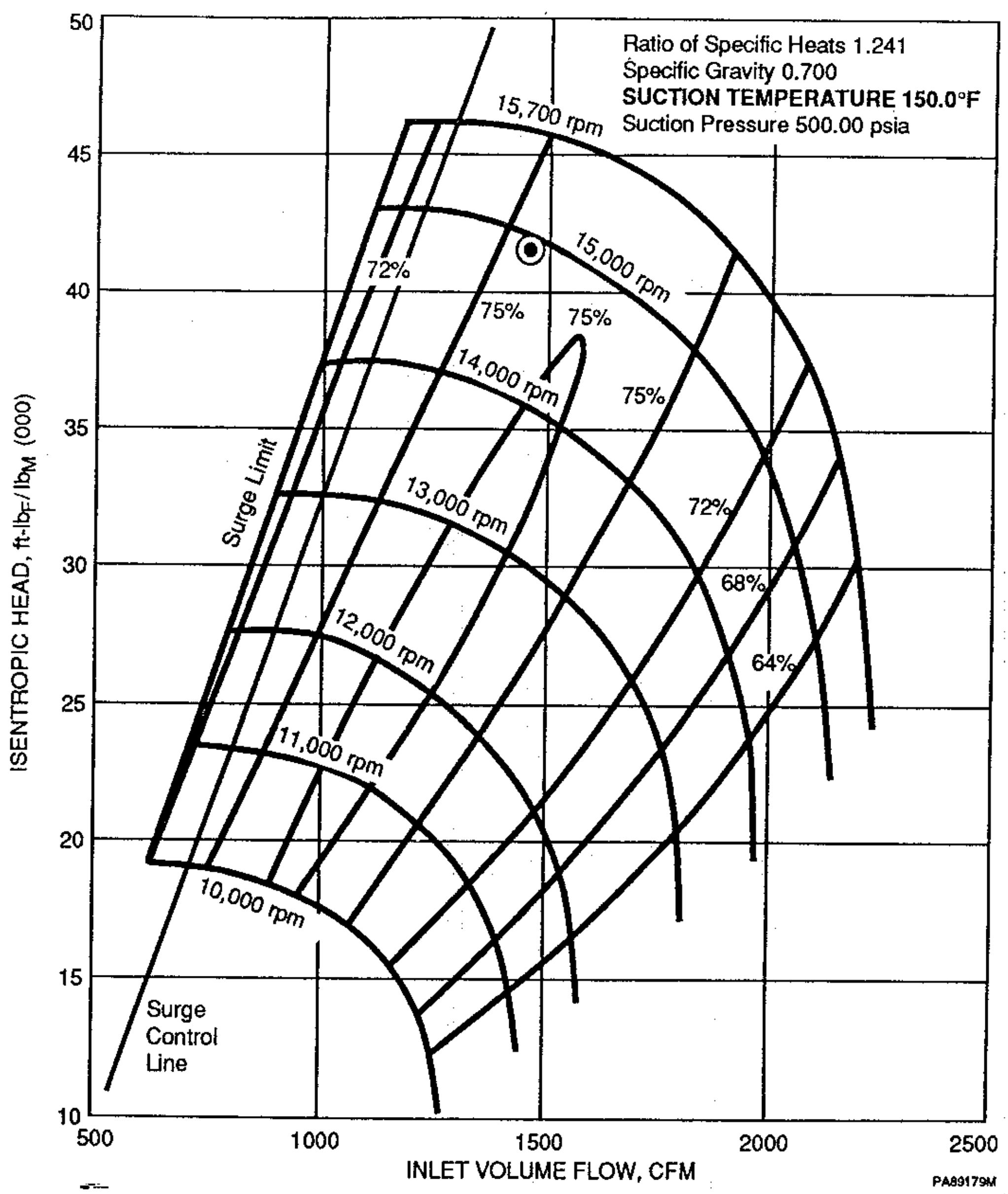

Figure 18. Higher Suction Temperature (T_1 was 100°F)

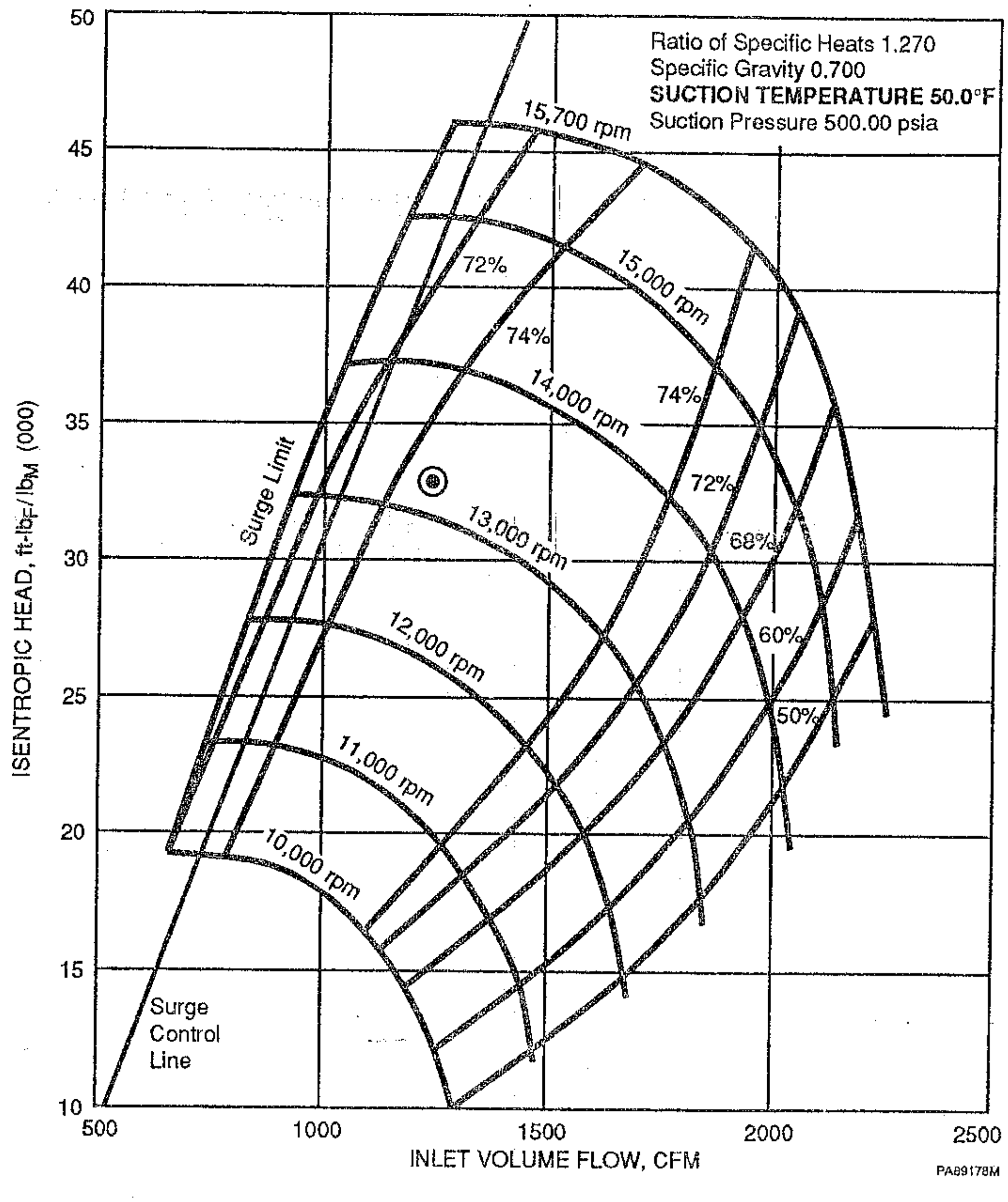

Figure 19. Lower Suction Temperature (T_1 was 100°F)

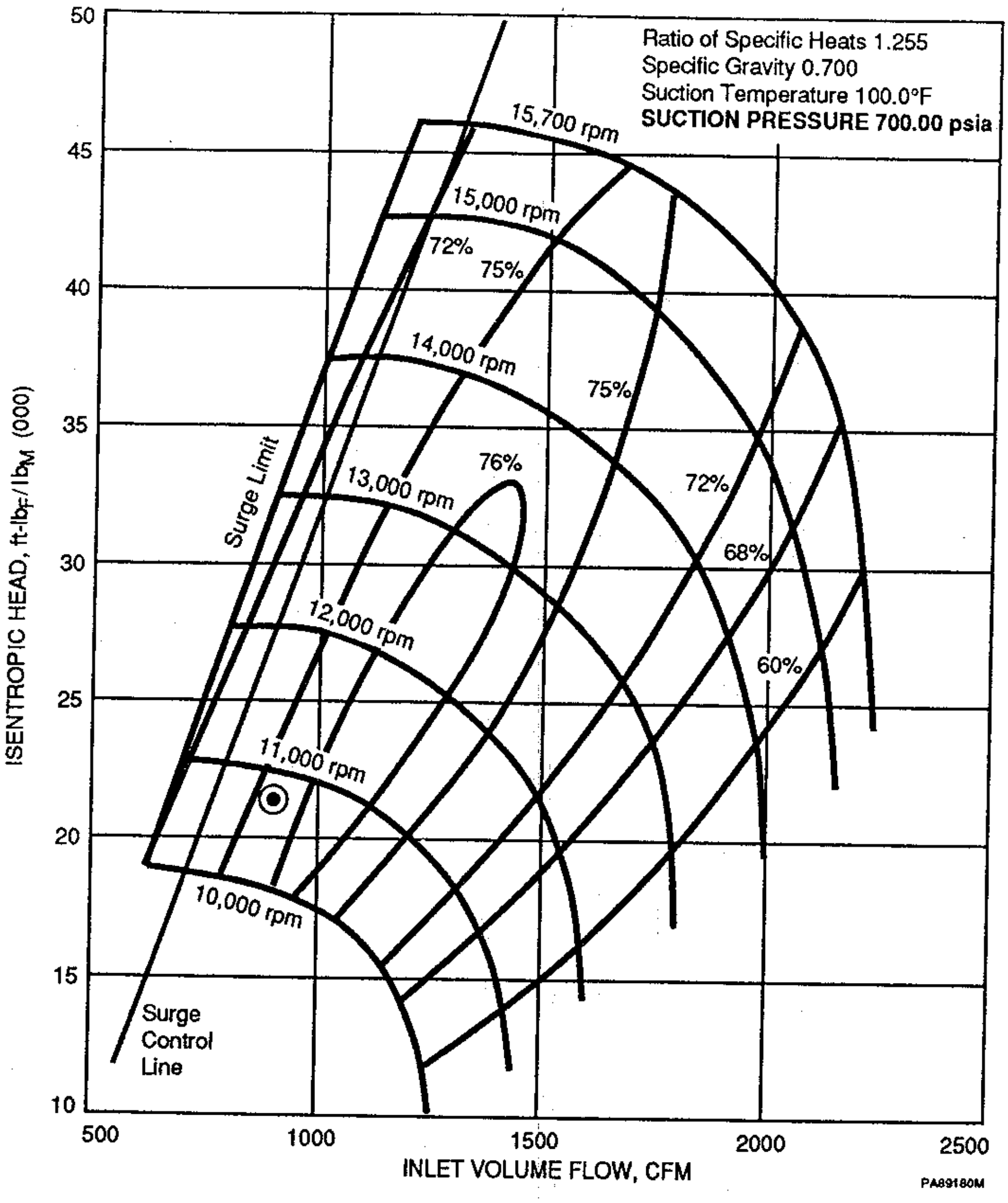

Figure 20. Higher Suction Pressure (P_1 was 500 psia)

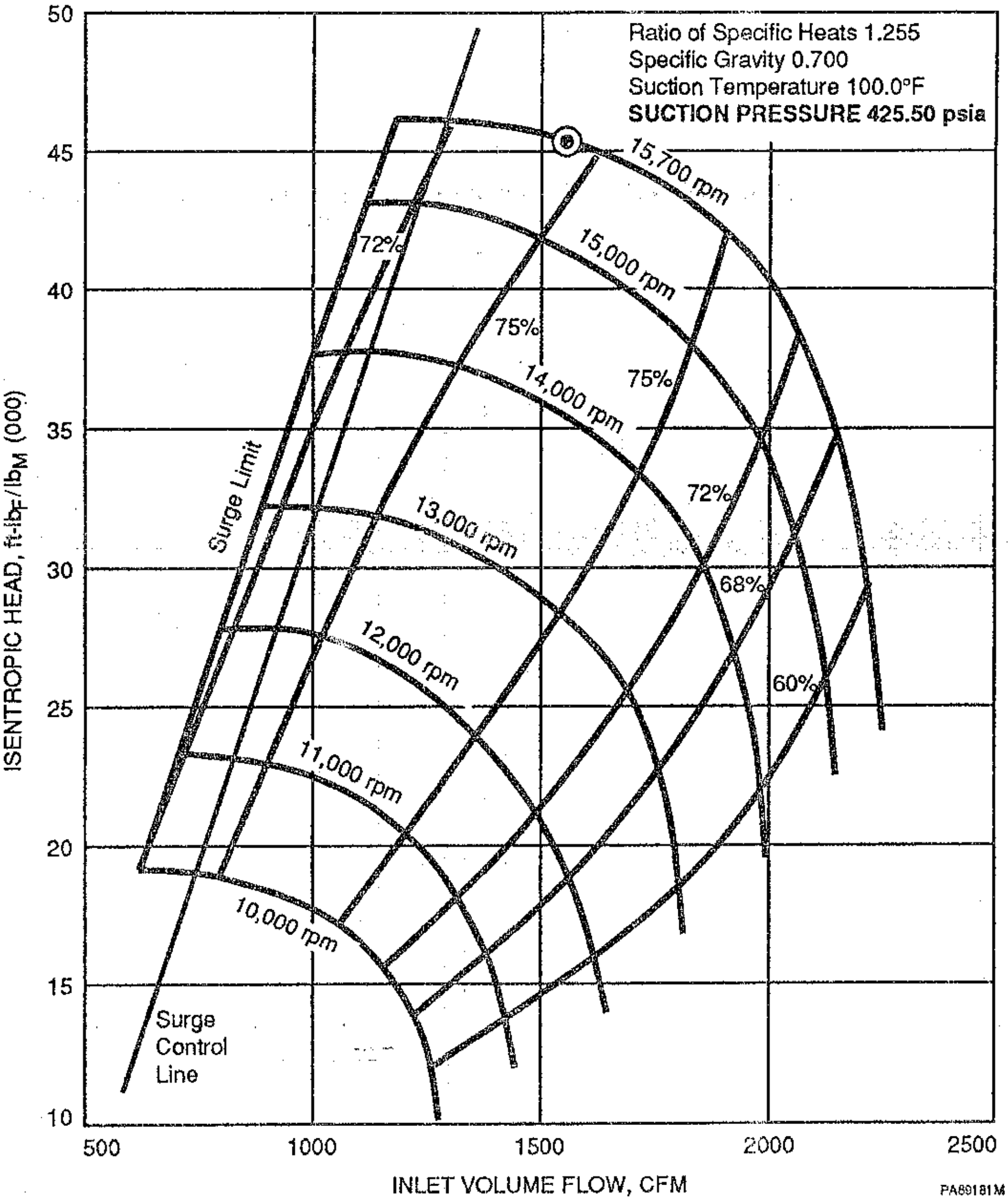

Figure 21. Lower Suction Pressure (P_1 was 500 psia)

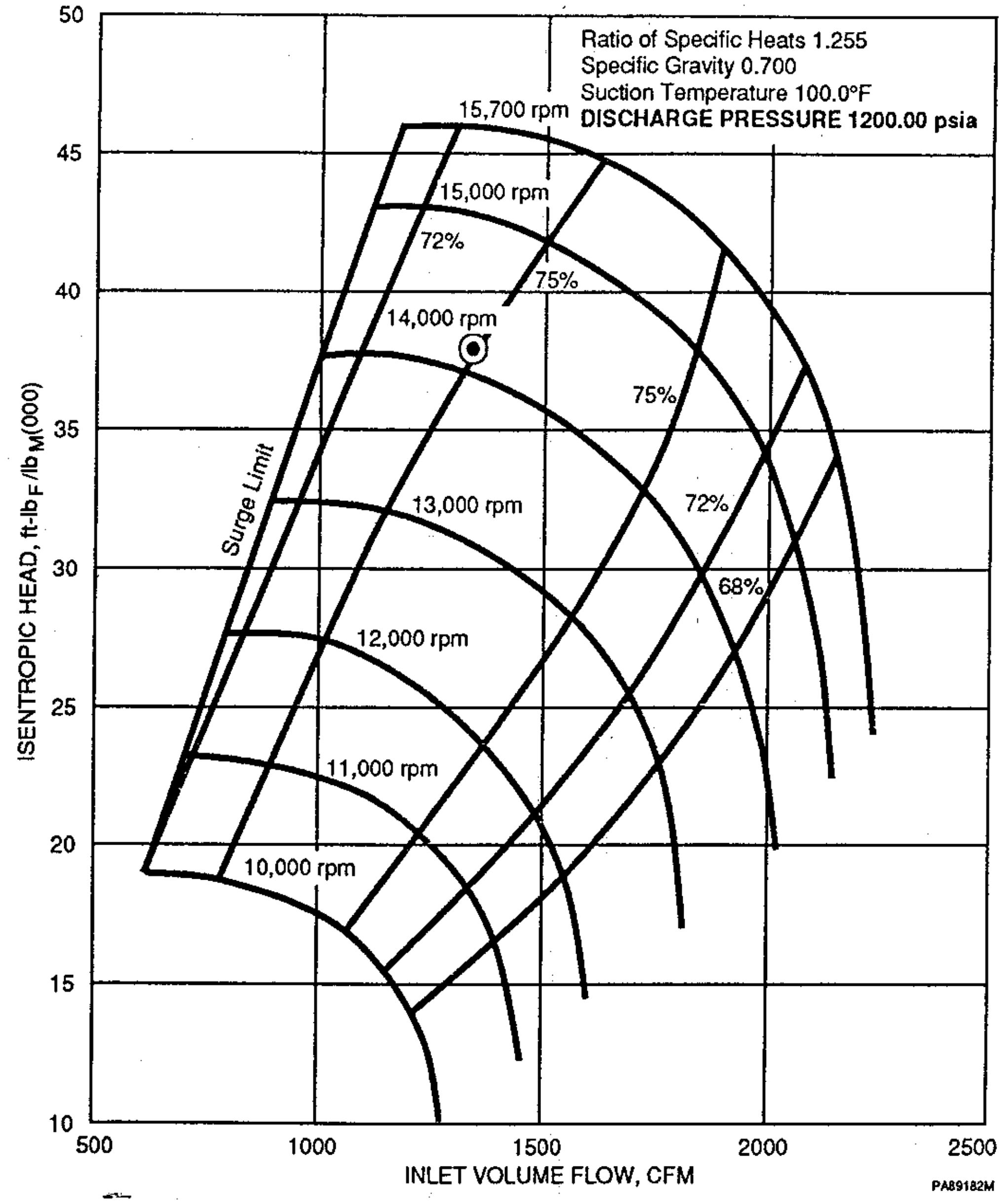

Figure 22. Initial Design Condition at Constant Discharge Pressure