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Surge Testing of Natural Gas Pipeline Centrifugal Compressors

by

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Abstract

Operating a gas pipeline centrifugal compressor within the unstable region of operation can lead to a destructive phenomenon known as surge, which can result in mechanical damage to the compressor. The users of such compressors rely on surge control systems to prevent operation in the unstable region.

The only way to properly setup a surge control system is to locate the true boundary of the unstable region through a surge test. Because a surge test involves bringing the operation of the compressor near the unstable region, it places both the compressor, and the associated production, at risk.

The reported work outlines the development and subsequent successful implementation of a surge testing procedure that was developed by the author to allow the surge testing of pipeline compressors with a manageable degree of risk.

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NOMENCLATURE

A	Area (m^2)
C (scalar quantity)	Absolute Velocity (m/s)
C (bold, vector quantity)	Absolute Velocity (m/s)
C_p	Constant Pressure Specific Heat (kJ/kg·K)
d	Diameter (m)
E_{ff}	Efficiency (dimensionless)
h	Specific Enthalpy (kJ/kg)
H_d	Head or Specific Enthalpy Increase (kJ/kg)
$K_{1,2,3,etc.}$	Constants
k	Ratio of Specific Heats
L	Internal Loss
M	Mach Number, Mass (kg), Molecular Mass (kg/kmol)
$\dot{m}$	Mass Flow Rate (kg/s)
N	Rotational Speed (rpm)
p	Pressure (N/m^2)
Q	Total Heat Transfer (J), Volumetric Flow Rate (m^3/s)
R	Universal Gas Constant (kJ/kmol·K)
r	Radius (m)
T	Torque (N·m), Absolute Temperature (K)
U (scalar quantity)	Tangential Velocity (m/s), Total Internal Energy (J)
U (bold, vector quantity)	Tangential Velocity (m/s)
W	Total Work (J)
$\dot{W}$	Power (J/s)
W (bold, vector quantity)	Relative Velocity (m/s)
Z	Compressibility, Number of Impeller Vanes

Greek Symbols

ß	Angle of Relative Velocity Vector From Axial or Radial Direction (radians)
Σ	The Summation of
μ	Angle of Absolute Velocity Vector From Axial or Radial Direction (radians), Absolute Viscosity ($\frac{N.s}{m^2}$)
ø	Angle (radians)
Δ	Differential
∂	Density (kg/m^3), Partial Derivative
Ω	Angular Velocity (rad/s)

Subscripts

0	Conditions in Suction Header Upstream of Reducing Flange
1	Conditions at Guide Nozzle Inlet
2	Conditions at Impeller Eye
3	Conditions at Impeller Periphery
4	Conditions Downstream of Diffuser
5	Conditions at Volute Outlet
6	Conditions Downstream of Discharge Nozzle
7	Conditions in Discharge Header
a	Axial
abs	Absolute
ad	Adiabatic
Ave	Average
c	Compressor, Counter-Whirl (IGV)
D	Design Conditions
d	Discharge, Diffusion

ELB	Elbow Meter
EYE	The Eye of the Impeller
f	Skin Friction
H	Off Design, High
i	Incidence
isen	Isentropic
L	Off Design, Low
NG	Natural Gas
o	Total (Stagnation) Conditions Note: This subscript will always follow a numerical subscript, wherever applicable.
p	Pre-Whirl (IGV)
r	Radial
s	Static Conditions, Suction Note: This subscript will always follow a numerical subscript, wherever applicable.
t	Tangential
ø	Means "At the Angle ø"

Superscripts

•	Time Rate of Change
–	Average
'	Indicates Conditions Resulting from an Isentropic Process

Abbreviations

CORR	Corrected
c/w	Complete With
DE	Drive End
D/T	Down Time
FP	Flow Pattern
IGV(s)	Inlet Guide Vane(s)
I/O	Input/Output

I/P	Current to Pneumatic
l.e.	Leading Edge(s)
LHS	Left Hand Side
mA	Milliampere
MCRS	Maximum Continuous Running Speed
MOS	Minimum Operating Speed
MW	Megawatt
NDE	Non-Drive End
OP	Operating Point
PLC	Programmable Logic Computer
RHS	Right Hand Side
RTI	Return to Idle
RVO	Recycle Valve Open Line
SBL	Surge Backup Line
SCL	Surge Control Line
SG	Specific Gravity
SLL	Surge Limit Line
t.e.	Trailing Edge(s)

CHAPTER ONE: INTRODUCTION

It is necessary for the user of centrifugal natural gas pipeline compressors to confirm, through field testing, the true performance of each aero assembly in the fleet (see Appendix A for a description of an aero assembly). The fleet consists of aero assemblies that are in service and in storage. The user will already have been supplied, by the manufacturer, with some indication of expected performance for each aero assembly. This expected performance is often the result of a shop test done at the manufacturer's facility.

The true performance of an aero assembly must be confirmed through a field test, for it is impossible for a user to make intelligent decisions regarding the existing or future capabilities of a pipeline without knowing the performance to expected of each aero assembly. This knowledge will strongly influence decisions regarding the need to purchase new aero assemblies, which currently cost in the range of \$0.6 million to \$0.7 million Canadian.

Part of knowing the true performance of an aero assembly is knowing the true position of the surge limit line (SLL). The SLL defines the boundary between the stable and unstable operating regions of a centrifugal compressor. Operation in the unstable region will lead to surging of the compressor, which can result in lost production and costly mechanical damage.

To prevent operation in the unstable region, surge control systems are installed. The surge control system prevents surge by opening a recycle valve to bring the operating point (OP) back into the stable region of operation. Running a compressor with the recycle valve open is inefficient, and often leads to a unit shutdown with associated lost production.

If the SLL is positioned too conservatively, inefficient operation and compressor shutdowns will result. If it is placed not conservatively enough, lost production and costly mechanical damage will result. The determination of the true position of the SLL is a step that must be taken to allow the efficient operation of pipeline compressors.

1.1 Natural Gas Pipelines in Canada: A Brief History

The year 1853 saw the first commercial use of natural gas in Canada at Trois Rivieres, Quebec. This gas was supplied via fifteen miles of cast iron pipe: Canada's first natural gas pipeline. By 1950, the consumption of natural gas in Canada was still insignificant relative to other forms of energy, with only the province of Alberta consuming notable amounts. In 1955, natural gas surpassed wood as a fuel and energy source, and by 1960 it supplied more energy than hydro-electric power. By 1964 natural gas equalled coal in the production of energy, and by 1969 natural gas was being used to produce 20% of Canada's energy consumption, second only to crude oil (Kilbourn, 1970).

Today, natural gas is used to produce approximately 27% of Canada's energy consumption. It is transported from the gas fields of Alberta, Saskatchewan, and British Columbia to markets all across Canada through a massive and ever expanding network of natural gas pipelines. Today's pipelines are no longer cast iron: they are constructed from the best available steel, coated with modern epoxy coatings, and operated at pressures up to 9928 kPag.

1.2 Pipeline Compressors

When natural gas flows through a pipeline, friction with the inner walls of the pipe causes some of the energy in the gas to be converted from pressure to thermal energy. The gas loses thermal energy through the transfer of heat from the gas, through the walls of the pipe, to the surrounding ground. This loss of energy results in a gradual decrease in pipeline pressure. Since pressure is the driving force causing the flow of fluid in any piping system, it is necessary to add energy to the gas along the pipeline to keep the gas flowing. This addition of energy is achieved by the installation of compression packages.

The most popular pipeline compression package being installed in Canada today is a gas turbine driven centrifugal pipeline compressor in the 28 Megawatt (MW) power range (see Appendix B for a description of the compression package). Single stage pipeline centrifugal compressors typically produce overall, absolute, stagnation pressure ratios in the range of 1.2 to 1.3, while multi-stage units produce ratios as high as 1.6 to 1.7.

1.3 An Introduction to Surge

Centrifugal compressors fall under the category of dynamic type compressors (as opposed to positive displacement) and are known to be sensitive to operating conditions in that conditions associated with high head, low flow, or both, can result in a destructive phenomenon known as surge (Shepherd, 1956; Shepherd, 1960; Ferguson, 1963; Kovàts, 1964; Dixon, 1978; Balje, 1981; Cohen et al., 1987; Sayers, 1990).

1.3.1 The Cause of Surge

There exists a region associated with high head and low flow that is known to be unstable for this type of compressor (Shepherd, 1956; Shepherd, 1960; Ferguson, 1963; Kovàts, 1964; Dixon, 1978; Balje, 1981; Cohen et al., 1987; Sayers, 1990). Operation within this region is possible, but not recommended. An unexpected increase in system resistance can momentarily cause a drop in compressor flow, along with a drop in compressor output pressure. This combination of occurrences can result in a sudden reversal of flow through the compressor.

1.3.2 The Danger of Surge

The mechanical damage resulting from one or more surge events can be extensive. Immediately preceding the surge itself there will be a rotating stall, which will cause the rotor to vibrate at amplitudes three to four times normal levels. This vibration can damage the labyrinth seals around the impeller eye and damage the dry gas seal(s). The sudden flow reversal will cause rapid rotor thrust load reversals, possibly damaging the thrust bearings. There will be flow pulsation and high frequency vibration (also due to the reverse flow) which could further damage the labyrinth and dry seals. Loud, explosive noises are often heard.

Surge has even been known to cause the aero assembly (depending on compressor design) to rotate inside the compressor case. The cost of the damage in parts alone can be several hundred thousand dollars. The cost of lost production can be even higher, depending on how long the unit is down for repair.

1.3.3 How Surge is Avoided

To prevent the occurrence of surge, control systems are put in place to monitor the flow through the compressor and the pressure rise across it. The OP is determined by the surge controller based on two pressure differential signals, one indicating flow and the other indicating head, and is compared to the setpoints representing the SLL. When the OP gets within a specified distance from the SLL, the surge control system will begin to open the recycle valve to reduce the pressure rise and increase the flow.

This method of prevention works well as long as the true position of the SLL is known, allowing the surge control system to be set up correctly.

1.4 Objectives

The objectives of the work described herein are as follows:

- To confirm, through field testing, the effect of changes in variable geometry on centrifugal compressor performance and hence the need for surge testing following geometry changes.

- To develop an acceptable procedure for the surge testing of centrifugal pipeline compressors, including the method of bringing the OP near the unstable operating region.[1]

- To determine the most reliable indication that the OP is near the unstable operating region, and to determine the theoretical explanation behind the indication, i.e. the physical phenomenon causing it.

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[1] Although surge testing consultants are available to undertake surge testing for a significant fee, their methods are usually proprietary and are often viewed as being incomplete by the operators of pipeline compressors.

CHAPTER TWO: THEORY OF CENTRIFUGAL COMPRESSORS

This thesis assumes a sound understanding of the theory behind the operation of centrifugal compressors. Some topics will be reviewed and discussed, but only those that are considered to be directly related to the content of the thesis. Some of the excellent references listed can be consulted as required (Shepherd, 1956; Ferguson, 1963; Kovàts, 1964; Dixon, 1978; Cohen et al., 1987; Sayers, 1990).

2.1 The Presentation of Compressor Performance

The presentation of compressor performance is usually accomplished by one of two common methods: the generalized compressor performance curves, or the compressor characteristic map.

The users of natural gas pipeline compressors often use plots of efficiency, isentropic head, and actual head versus inlet flow at the rated (or reference) speed only. This manner of presentation is referred to as generalized compressor performance curves (see Figure 2.1).

The presentation of compressor performance across the entire operating range of the machine, under a full range of operating conditions, is referred to as a compressor characteristic map (see Figure 2.2). The compressor characteristic map displays the rotational speeds required for the compressor to produce specific head and flow rate combinations, while also indicating the efficiency to be expected for each combination.

Since the work outlined in this research is closely tied to compressor performance, this Section will review briefly some of the theoretical concepts affecting compressor performance.

2.1.1 Euler Head versus Flow

The Euler head represents the theoretical, ideal, specific enthalpy addition to the flow due to the increase in angular momentum the flow receives as it passes through the impeller. It is expressed as a work increment, or addition, to the flow in units of kJ/kg.

The Euler head is based on the representation of the flow of gas leaving the impeller by a single relative velocity vector. This single relative velocity vector is assumed to leave the impeller at the precise angle of the trailing edge of the impeller vane. The prediction of the magnitude of the vector is based on the assumption of a uniform velocity distribution in the impeller channel.

The relationship between the theoretical specific enthalpy addition (the Euler head) and the mass flow of fluid being handled is determined by the shape of the vanes of the impeller. With back-swept impeller vanes, which is the geometry of choice for pipeline compressors, there exists a linear relationship of negative slope between Euler head and flow (see Figure 2.3).

The thinking behind the use of back-swept impeller vanes in pipeline compressors warrants a brief discussion.

From the point of view of impeller stress, radial vanes are the most advantageous because they introduce no complex bending stresses (Shepherd, 1956). With a centrifugal compressor impeller, it is either impeller stress, or shock losses, that limits rotational speed. A centrifugal compressor handling natural gas is unlikely to experience shock losses because of the relatively low molecular mass of the natural gas. This leaves stress as the most likely factor to limit the rotational speed of such a compressor, which would seem to indicate the radial vane as the preferred geometry.

It must, however, be remembered that the primary objective of a pipeline compressor is to produce a large static pressure rise in the working fluid. This implies a relatively high degree of reaction, which is defined as "... the ratio of the energy transfer... resulting in a change of static pressure in the rotor to the total energy transfer in the rotor." (Shepherd, 1956). The back-swept vane produces an acceptably high transfer of energy along with a relatively high reaction ratio (Shepherd, 1956), and avoids placing unreasonable demands on the diffusing capabilities of the fixed components downstream of the impeller by keeping the absolute discharge velocity relatively low. It is these design considerations that leads to the exclusive use of back-swept vanes in pipeline compressors.

Note also that a negative slope of the head versus flow relationship is, in general, good for the prevention of surge because it is advantageous to have an increase in delivery pressure associated with a decrease in flow (this will be discussed further in Section 3.1).

2.1.2 Actual Head versus Flow

Because of physical realities occurring within the impeller, the Euler head is unattainable. A phenomenon called fluid deviation (Shepherd, 1956) causes the single relative velocity vector mentioned above to be at a slight negative angle relative to the angle of the trailing edge (t.e.) of the impeller vane. This phenomenon is also referred to as slip. The velocity distribution in the impeller channel is in fact not uniform either.

These two factors reduce the Euler head to what is referred to as the actual head. The actual head curve is also linear, and of slightly less negative slope than the Euler head curve (see Figure 2.3). The actual head can be thought of as the specific work done on the fluid. In terms of power, it can be thought of as the thermodynamic power input to the fluid. This power is referred to as the absorbed power because it represents the amount of shaft power the fluid is able to absorb. The actual head consists of the useful total pressure rise, as well as the unwanted thermal energy, or temperature rise, occurring simultaneously in the fluid. This unwanted thermal energy is the result of internal losses.

2.1.3 Isentropic Head versus Flow

The isentropic head is a measure of the useful specific enthalpy addition required if the fluid were to undergo a reversible, adiabatic (isentropic) compression process from the actual suction header pressure (p_{0o}) to the actual discharge header pressure (p_{7o}). The temperature rise involved would be only that associated with such a process, thus there would be no unwanted thermal energy. The isentropic head is related to the actual head in the following manner: it is the specific work that would be required by an isentropic compression process to achieve the same pressure rise as the actual process.

Internal losses are the reason for the difference between actual head and isentropic head in relation to compressor performance. They are caused by skin friction, turbulence, incidence, drag, and shock. Internal losses represent energy that is available to the gas, but not usefully utilized. Because the energy is available, it contributes to the actual head. The difference between the actual head and the isentropic head is the internal losses. If one were to total up the internal losses and subtract them from the actual head curve, the result would be the isentropic head curve (see Figure 2.3)

Of the internal losses mentioned above, incidence and drag losses are most relevant to the upcoming discussion. Incidence refers to the angle of attack of the approaching fluid relative to an immersed body similar in shape to an airfoil. Examples of such immersed bodies are the inlet guide vanes (IGVs), the vanes of an impeller, and the diffuser vanes (if they exist). Drag is also considered in terms of the similarity between the above mentioned vanes and an airfoil. Friction and pressure losses both contribute to the total drag on an airfoil.

The losses due to angles of incidence at the leading edges of the impeller vanes are identified as L_i on Figure 2.3. Incidence results in flow separation (and resulting turbulence) as well as pressure drag on the airfoil-like impeller vanes.

2.1.4 Generalized Compressor Performance Curves

The main advantage of displaying information at the rated speed only is it gives a quick and clear comparison of performance over time for a given compressor. Note that the uppermost curve on the generalized display (Figure 2.1) is a relationship between isentropic efficiency and flow. Isentropic efficiency is given by,

$$E_{ff,Isen} = \frac{h_{\mathit{7o}} - h_{\mathit{0o}}}{h_{7o} - h_{0o}} \qquad (2.1)$$

where,

$E_{ff,Isen}$	= isentropic efficiency (dimensionless)
h_{7o}	= isentropic stagnation enthalpy, discharge header (kJ/kg)
h_{0o}	= isentropic stagnation enthalpy, suction header (kJ/kg)
h_{7o}	= actual stagnation enthalpy, discharge header (kJ/kg)

If the performance of a compressor is being questioned, a site performance test is carried out (see Appendix C). It is to the tester's advantage to take all test points as close to the compressor rated speed as possible. Any test points taken more than 10% from rated speed should be used with caution, and any test points taken more than15% from rated speed should be disregarded. All test points are corrected back to the rated speed via the familiar fan laws,

$$Q_{CORR} = Q_{ACT} \cdot \frac{N_{RATED}}{N_{ACT}} \qquad (2.11)$$

$$H_{d,CORR} = H_{d,ACT} \cdot \left(\frac{N_{RATED}}{N_{ACT}}\right)^2 \qquad (2.12)$$

where,

Q_{CORR}	= corrected volumetric flow rate (m³/s)
Q_{ACT}	= actual volumetric flow rate (m³/s)
N_{RATED}	= rated rotor rotational speed (rpm)

N_{ACT} = actual rotor rotational speed (rpm)
$H_{d,CORR}$ = corrected isentropic enthalpy increase (kJ/kg)
$H_{d,ACT}$ = actual isentropic enthalpy increase (kJ/kg)

The reason for not using the fan laws for correcting points more than 15% from the reference speed is that this technique neglects changes in viscosity and hence Reynolds number, thereby leaving doubt as to the validity of the results (Shepherd, 1956).

After the test points are corrected to the rated speed, they are plotted against the results of previous tests on the compressor. If the actual head versus flow rate relationship has changed, it could mean a problem with the IGVs or the impeller itself. If the actual head plots the same as before but the isentropic head and hence isentropic efficiency has changed, it could mean a worn or damaged labyrinth eye seal, the impeller is not properly positioned axially, or a problem with the diffuser or volute.

In addition to helping pinpoint some of the above mentioned problems, the generalized curves are good for showing the difference in compressor performance before and after a significant change in the configuration of the aero assembly, such as a repositioning of the IGVs, or the removal of, or installation of, diffuser vanes. This topic will be discussed in more detail in Section 2.2.

2.1.5 The Compressor Characteristic Map

The performance curves discussed thus far have served to indicate compressor performance at a single rotor speed. The compressor characteristic map indicates the relationship between overall pressure ratio (or isentropic head) and mass flow rate (or actual inlet volumetric flow rate) for a range of rotor speeds, with isentropic efficiency curves superimposed on the isentropic head curves (Figure 2.2).

The compressor characteristic map is a common way for the manufacturers of gas pipeline compressors to display to potential end users the results of shop tests on their compressors. They will often plot overall compressor pressure ratio (stagnation basis) versus mass flow rate as shown in Figure 2.2. For a plot such as this, the test gas would be specified, as well as the test inlet conditions.

Natural gas pipelines tend to display the performance of their compressors in a slightly different way. As shown in Figure 2.4, they plot the isentropic specific work input to the natural gas (or head) versus the actual inlet volumetric flow rate. The conversion from mass flow rate to actual inlet volumetric flow is trivial. The conversion from pressure ratio to isentropic specific work input is not so trivial. The isentropic efficiency, mentioned previously in Subsection 2.1.4, is given by,

$$E_{ff,Isen} = \frac{h_{7'o} - h_{0o}}{h_{7o} - h_{0o}} \tag{2.1}$$

or,

$$E_{ff,Isen} = \frac{C_{pAve} \cdot (T_{7'o} - T_{0o})}{C_{pAve} \cdot (T_{7o} - T_{0o})} \tag{2.2}$$

where,

C_{pAve} = average constant pressure specific heat (kJ/kg·K)
$T_{7'o}$ = isentropic stagnation temperature, discharge header (K)
T_{0o} = stagnation temperature, suction header (K)
T_{7o} = stagnation temperature, discharge header (K)

which is valid for any adiabatic flow of a perfect gas, whether reversible or not (Shapiro, 1953). The isentropic head is given by

$$H_{d,Isen} = h_{7'o} - h_{0o} \tag{2.3}$$

or,

$$H_{d,Isen} = C_{p,Ave} \cdot (T_{7o} - T_{0o}) \tag{2.4}$$

where,

$H_{d,Isen}$ = isentropic specific enthalpy increase (kJ/kg)

The following equation:

$$\frac{T_{7o}}{T_{0o}} = \left(\frac{p_{7o}}{p_{0o}}\right)^{\frac{(k-1)}{k}} \tag{2.5}$$

where,

p_{7o} = stagnation pressure, discharge header (N/m²)
p_{0o} = stagnation pressure, suction header (N/m²)
k = ratio of specific heats (dimensionless)

is valid for an isentropic process involving a perfect gas (Van Wylen & Sonntag, 1985). Rearranging Equation 2.5:

$$T_{7o} = T_{0o} \cdot \left(\frac{p_{7o}}{p_{0o}}\right)^{\frac{(k-1)}{k}} \tag{2.6}$$

substituting Equation 2.6 into 2.4:

$$H_{d,Isen} = C_{pAve} \cdot \left[T_{0o} \cdot \left(\frac{p_{7o}}{p_{0o}}\right)^{\frac{(k-1)}{k}} - T_{0o}\right] \tag{2.7}$$

The following equation:

$$C_{p,Ave} = \frac{k}{(k-1)} \cdot R \tag{2.8}$$

where,

R = gas constant, natural gas (kJ/kg·K)

is valid for a substance that obeys the perfect gas law (Shapiro, 1953). Substituting Equation 2.8 into 2.7:

$$H_{d,isen} = T_{00} \cdot \frac{k}{(k-1)} \cdot R \cdot \left[\left(\frac{p_{70}}{p_{00}}\right)^{\frac{(k-1)}{k}} - 1\right] \tag{2.9}$$

In an attempt to handle deviations of the working fluid from ideal gas behavior, the compressibility factor is included in Equation 2.9, resulting in,

$$H_{d,isen} = T_{00} \cdot Z_{ave} \cdot \frac{k}{(k-1)} \cdot R \cdot \left[\left(\frac{p_{70}}{p_{00}}\right)^{\frac{(k-1)}{k}} - 1\right] \tag{2.10}$$

with Z_{ave} being the average compressibility factor based on suction and discharge conditions. Note that Equation 2.10 allows the calculation of the isentropic specific work input to the gas by knowing the pressure ratio, the suction temperature, and the properties of the natural gas.

2.1.6 The Operating Envelope

On a compressor characteristic map, the useful operating range, or operating envelope, is bounded on the left by the surge control line, on the right by the capacity limit, on the top by the maximum continuous speed, and on the bottom by the minimum operating speed (see Figure 2.2). Each boundary will be discussed briefly.

Surge Control Line (SCL)

The SCL defines the line at which the surge controller will begin to take action (i.e., open the recycle valve) to keep the operating point out of the unstable region. It is usually located at 110% of the flow at the SLL. Operation to the left of the SCL is only possible if the surge control system is not able to perform its function.

Capacity Limit

The capacity limit defines the point when either the absolute or the relative fluid velocity has reached the local sonic velocity at some location in the compressor. The most likely locations for this are the inducing section of the impeller and the vaned diffuser throat. This condition is referred to as choking.

In general, the condition is characterized by a rapid dropping off of the isentropic head versus flow characteristic curve. In fact, the slope of the characteristic will become vertical, indicating the choked condition and indicating infinite internal losses due to the shock wave located inside the compressor. Under these circumstances, there may be negligible useful pressure rise across the compressor.

Maximum Continuous Running Speed (MCRS)

The maximum allowable rotational speed of an impeller will be dictated either by the fear of shock losses, or unacceptably high stresses in the impeller caused by centrifugal forces. It is also possible, in the case of a gas turbine driven pipeline compressor, that stresses in the power turbine rotor could limit the rotational speed of the compressor since the two machines are mechanically coupled together. It is unlikely that shock losses would be a factor in a properly designed compressor handling natural gas under normal operating conditions (i.e. not a choked flow situation). In fact, a properly designed turbo compression package should be able to run with the gas generator at full power under normal operating conditions before the MCRS of the compressor is reached (i.e., one would never strive to limit the package power based on power turbine/compressor speed).

Nonetheless, the MCRS exists as a set point in the unit controls, and it is "...the speed at least equal to 105 percent of the highest speed required by any of the specified operating conditions" (API Standard 617, 1988). In other words, the unit will be capable of meeting all performance expectations at 100 percent speed, and it is allowed to run up to 105 percent speed at which point the stresses in the impeller will still be below the allowable limits and at which point internal shock losses will not be expected under normal operation conditions.

Minimum Operating Speed (MOS)

The MOS is defined as "...the lowest speed at which the manufacturer's design will permit continuous operation." (API Standard 617, 1988). It is usually dictated by the existence of a known first critical speed of the compressor rotor, and it will be a setpoint in the unit control set at 118% of the rotational speed at which the known critical speed exists (API Standard 617, 1988). In fact, the MOS is typically the setpoint at which the programmable logic computer (PLC) sends a command to the surge control system telling it to fully open the recycle valve thereby taking the unit off line (see Chapter Four).

2.2 The Effects of Variable Geometry

A discussion on variable geometry is included because variations in geometry can significantly alter the performance of a given aero assembly, and in fact, a variation in geometry constitutes one of the conditions under which a surge test is conducted.

The term variable geometry is not intended to indicate that the geometry of the aero assembly can be changed during operation. There are gas pipeline compressors driven by constant speed electric motors that have true variable IGVs because the guide vanes are used to load and unload the compressor. Typically, the geometry that is variable in a gas turbine driven pipeline compressor is limited to the IGVs and the diffuser vanes. Any changes in geometry will require that the compressor be shut down and at least partially disassembled.

IGVs are variable in that they can be rotated on a radial axis perpendicular to the rotor axis (Figure 2.5). Diffuser vanes are not typically variable, but they can be removed or replaced.

2.2.1 Inlet Guide Vanes

Internal losses due to angles of incidence at the leading edges of the impeller vanes was introduced in Subsection 2.1.3. A theoretical understanding of these losses is best gained through the use of velocity triangles. Figure 2.6(a) shows graphically the effect that varying flow rate has on the angle at which the gas approaches the leading edges (l.e.) of the impeller vanes. For the purposes of this discussion, the tangential velocity of the impeller l.e. ($\mathbf{U_2}$) is assumed to be constant, and the incoming gas is assumed to have only an axial velocity component (i.e. $\mathbf{C_{2a}} = \mathbf{C_2}$).

Note that an increase in mass flow rate results in a decrease in the angle β_2, which results in a negative angle of incidence, while a decrease in mass flow rate results in an increase in the angle β_2, which results in a positive angle of incidence. Lower than design flow, with the associated positive incidence angle, is said to begin affecting compressor performance at relatively small incidence angles, whereas relatively large negative angles of incidence due to higher than design flow can be experienced before significant losses are incurred (Shepherd, 1956; Ferguson, 1963). In Figure 2.6(a), peak impeller efficiency is represented by the vector triangle resulting in zero incidence.

IGVs can be used to alter the performance of a centrifugal compressor in that they can be used to impart a swirling motion to the flow as it moves toward the impeller eye, i.e. they give the incoming flow a tangential velocity component. Theoretically, this should change the energy transfer in the impeller and change the location of peak compressor efficiency relative to the mass flow rate through the compressor.

Figure 2.6(b) shows the theoretical effect of IGVs on the velocity vector triangles at the leading edges of the impeller vanes with $\mathbf{U_2}$ held constant. The vector triangles indicate that in order to maintain the correct angle (β_2) for the relative velocity vector ($\mathbf{W_2}$), the flow rate would have to increase with counter-whirl IGVs and decrease for pre-whirl IGVs relative to the flow required for axial IGVs.

This would seem to indicate that the point of peak efficiency would occur, relative to axial IGVs, at a higher flow for counter-whirl IGVs and at a lower flow for pre-whirl IGVs.

We have implied that the transfer of energy occurring in the impeller is also affected by IGV position. To explain this, a form of the Euler turbine equation, named for the Swiss mathematician Leonhard Euler (1707-1783), is introduced (Fox and McDonald, 1985).

$$\frac{\dot{W}_{Euler}}{\dot{m}} = U_3 \cdot C_{3t} - U_2 \cdot C_{2t} \qquad (2.13)$$

where,

$\dot{W}_{Euler}$ = power (J/s)
$\dot{m}$ = mass flow rate (kg/s)
U_3 = tangential impeller velocity at periphery (m/s)
C_{3t} = absolute tangential fluid velocity, impeller periphery (m/s)
U_2 = tangential impeller velocity at eye (m/s)
C_{2t} = absolute tangential fluid velocity, impeller eye (m/s)

The Euler turbine equation is a mathematical expression for the increase in angular momentum the fluid receives from the impeller. The result is the specific work input to the fluid in kJ/kg, and it represents the Euler head previously discussed in Subsection 2.1.1. Note on Figure 2.6(b) that with an axial IGV position, the tangential component of the incoming fluid (C_{2t}) is equal to zero, so the second term on the RHS drops out. With pre-whirl IGVs, C_{2t} takes on a positive value, and the specific work input, or head, is reduced. With counter-whirl IGVs, C_{2t} takes on a negative value, and the specific work input is increased.

Results from field performance tests undertaken by the author (Figures 2.7, 2.8, and 2.9) show the results of field performance testing done on identical axial-entry pipeline compressors with 36 inch flanges rated at 26 MW. For all three tests identical diffuser vanes were in place, but there were three different IGV positions involved: axial, 20° pre-whirl, and 20° counter-whirl respectively.

The conclusions drawn from the results of these field tests are as follows:

- The flow rate at which peak efficiency is attained is the same in all three cases (approximately 15.5 m^3/s).
- The peak efficiency for pre-whirl IGVs (85.8%) is higher than the peak efficiency of both axial (84.4%) and counter-whirl (83.8%) IGVs.
- The isentropic specific work input is lowest for pre-whirl IGVs (39.0 kJ/kg), with axial IGVs producing a higher value (45.0 kJ/kg), and the counter-whirl IGVs producing the highest value (47.0 kJ/kg).

Thus the theoretical expectations previously outlined with respect to Figure 2.6 are confirmed by field testing with respect to specific work input, but not with respect to the flow rate at which the point of peak efficiency occurred.

Although the field test data do not cover the entire operating range of the compressor, it does indicate in all three cases that the point of maximum isentropic efficiency has been located. Surge testing was not attempted as a part of these particular performance tests, so test data at relatively lower flow rates was limited by the positioning of the manufacturer's SLL. The fact that manufacturer's SLLs are often placed too conservatively will be proven in Section 6.1.

Another point worth noting is that although the point of peak efficiency appears to be unchanged by the IGV positions tested here, the compressor characteristic at flows lower than maximum efficiency, based on extrapolations of test data, appears to vary with IGV position. This implies that the point of instability could also vary with IGV position, which supports the need for surge testing following any changes to IGV position.

2.2.2 Vaned and Vaneless Diffusers

Aero assemblies come equipped with vaned or vaneless diffusers, and some of them can be installed either way. With respect to diffuser design, there is basically a choice between relatively good performance at all operating conditions, or relatively high peak performance at design conditions with relatively low performance at off-design conditions. The former design objective is attained with a vaneless diffuser, the latter by designing the diffuser with stationary vanes.

Vaneless Diffusers

The flow through the vaneless portion of a diffuser follows a path known as a logarithmic spiral as the gas undergoes diffusion governed by the laws of continuity and conservation of angular momentum (see Figure 2.10; Shepherd, 1956; Dixon, 1978; Sayers, 1990). A vaneless diffuser, although providing acceptable levels of peak efficiency with relatively low losses in performance for off-design conditions, does require a physically large relative diameter in order to be successful. This is so because it would be reasonable to assume that the gas would travel up to 180° around the vaneless diffuser passage after leaving the impeller tip (Dixon, 1978). This can result in relatively high frictional losses.

The angular distance the gas moves through as it travels radially through the diffuser is based on the angle of the absolute flow velocity vector at the impeller periphery, μ_3. The smaller μ_3, the less the angular distance of the gas, i.e. it takes a more "direct" radial route through the diffuser. The path the gas takes though a vaneless diffuser is intimately related to static pressure recovery as well as frictional loss. A detailed discussion of vaneless diffuser design is beyond the scope of this thesis, however, let it suffice to say that at some optimum angle μ_3 the vaneless diffuser will perform the best. Performance will drop off for other angles of μ_3, but not dramatically.

Vaned Diffusers

If diffuser vanes are used, they will be located downstream of the impeller periphery by the appropriate diametral clearance (.10 to .20 d_3). Once inside the diffuser channel, the fluid will follow a linear path before spilling into the volute, thus allowing the gas to follow a much shorter path as it undergoes the diffusion process (see Figure 2.10). This approach serves to reduce the conventional skin friction losses, but it also introduces the internal losses associated with angles of incidence previously mentioned in Subsection 2.1.3.

A vaned diffuser plays a role in determining the point of peak compressor efficiency because the leading edges of the diffuser vanes will be angled such that they will receive the gas from the impeller without incidence at design conditions. As soon as off-design conditions are encountered, incidence will occur with resulting turbulence and drag. Extreme off-design performance can result in a wake at the trailing edges of the diffuser vanes, resulting in turbulent volute flow.

It is possible that the diffuser vanes will be designed such that they will enhance the peak efficiency of the compressor without changing the location at which that peak efficiency occurs, as indicated by the following quotation from Balje: "At the design point, the inducer and diffuser angles are assumed to be matched; that is, both elements operate without incidence" (Balje, 1981). This, however, need not be the case. Diffuser vanes can be designed to try to hold the overall peak efficiency of a compressor at a location other than the location it would have without the vanes. Proof of this fact will be discussed later.

Compared to performance without diffuser vanes, the installation of properly designed diffuser vanes should have the following effect on the performance of a compressor:

- The location of peak isentropic efficiency may or may not change.
- The peak isentropic efficiency will be higher than before.

- The isentropic efficiency versus flow relationship will have a more pronounced peak than the flatter curve of a vaneless diffuser.
- The actual head versus inlet flow relationship will not change.
- The isentropic head will be higher at peak efficiency than before, and will follow the same trend as the isentropic efficiency (which it must with the actual head being unchanged).

The design feature of the diffuser vane that determines the flow rate it will favour for peak efficiency is the angle of its leading edge. Note that the leading edge of the diffuser vane will form the same angle μ_3 with a radial line perpendicular to the rotor axis as will the absolute velocity vector as it leaves the periphery of the impeller[2].

The vector triangles on Figure 2.10 indicate that with a given diffuser l.e. angle and a given impeller t.e. tangential velocity ($\mathbf{U_3}$), an increase in flow would increase the magnitude of the relative velocity vector $\mathbf{W_3}$ and cause a positive angle of incidence, while a decrease in flow would decrease the magnitude of $\mathbf{W_3}$ and cause a negative angle of incidence. Consequently, we would expect performance losses associated with incidence angles to be more severe for flow rates above design rather than below design as is the case at the impeller l.e. This expectation is based on the relationship between incidence angles and performance losses discussed in Subsection 2.2.1.

Before leaving the discussion of the effects of diffusers, it is worth referring to Figure 2.11, which shows the results of field performance tests done on a gas pipeline compressor with and without diffuser vanes. The compressor is of an axial-entry design with 36 inch flanges rated at 26 MW. The test with the vaneless diffuser (20 February 1995) was done by the author. The test with the vaned diffuser (15 March 1994) was done by a colleague of the author.

The following items are noteworthy regarding these performance test results:

---

[2]If the effects of friction and compressibility are accounted for, the angle at the diffuser vane would actually be slightly larger than at the impeller vane as shown in Figure 2.10.

- As expected, the actual head versus flow relationship is not changed with the removal of the diffuser vanes.
- With the removal of the diffuser vanes, peak efficiency is reduced from over 88% to approximately 86%, and the efficiency versus flow rate relationship takes on a broader, flatter appearance.
- With the diffuser vanes removed, peak efficiency occurs at a higher flow (20 m^3/s) than with the diffuser vanes in place (18 m^3/s). This is an example of the diffuser vanes holding peak efficiency at a lower flow than it would otherwise occur at. Obviously, the vaneless diffuser performed better with a smaller angle μ_3, leading to a more direct radial path through the diffuser.
- With the removal of the diffuser vanes, the isentropic head drops at peak efficiency in a manner similar to the efficiency curve as expected.

Thus it is again clear, as it is with IGV position, that a surge test is required to determine the point of instability following the installation of, or removal of, diffuser vanes.

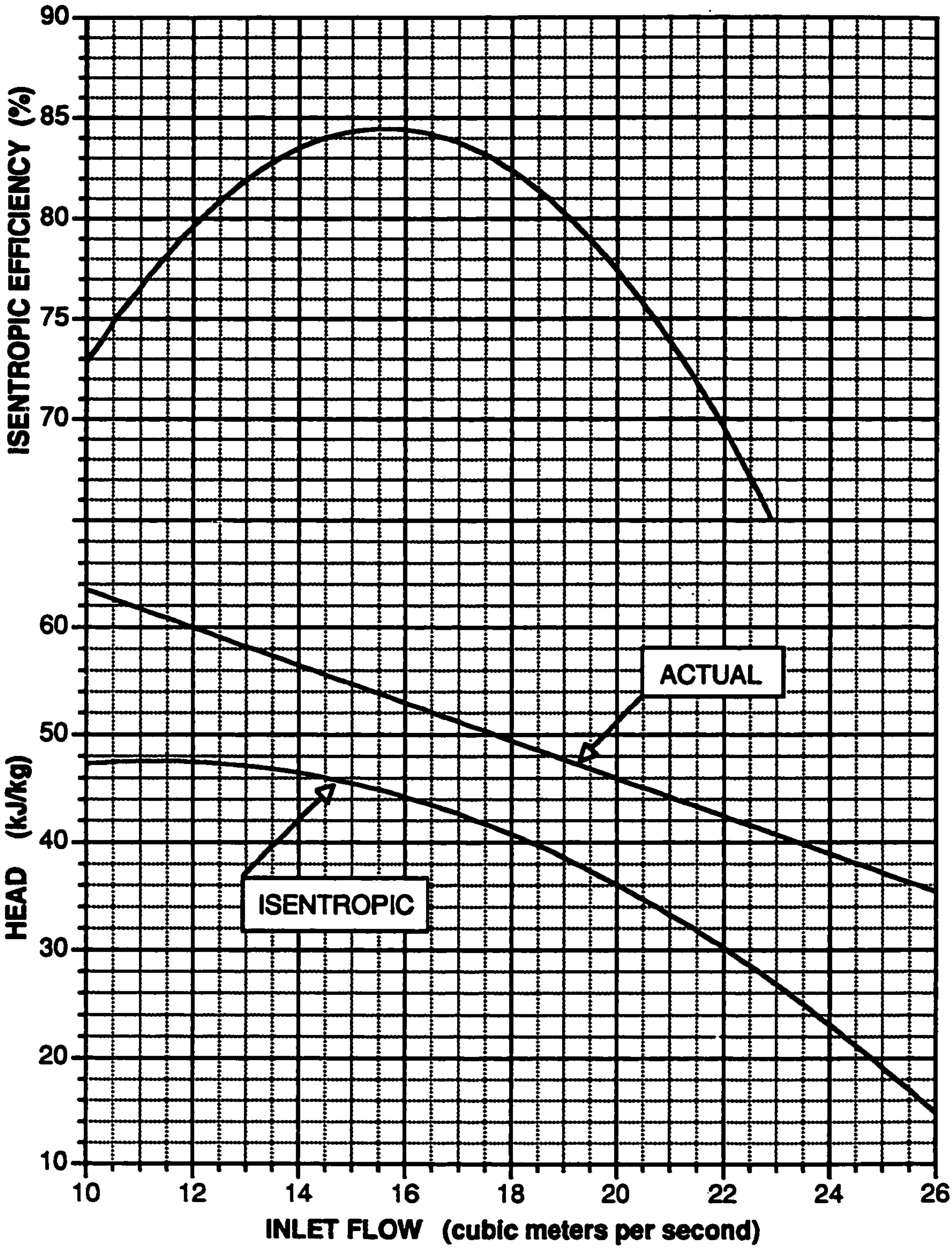

Figure 2.1 Typical Generalized Performance Curves

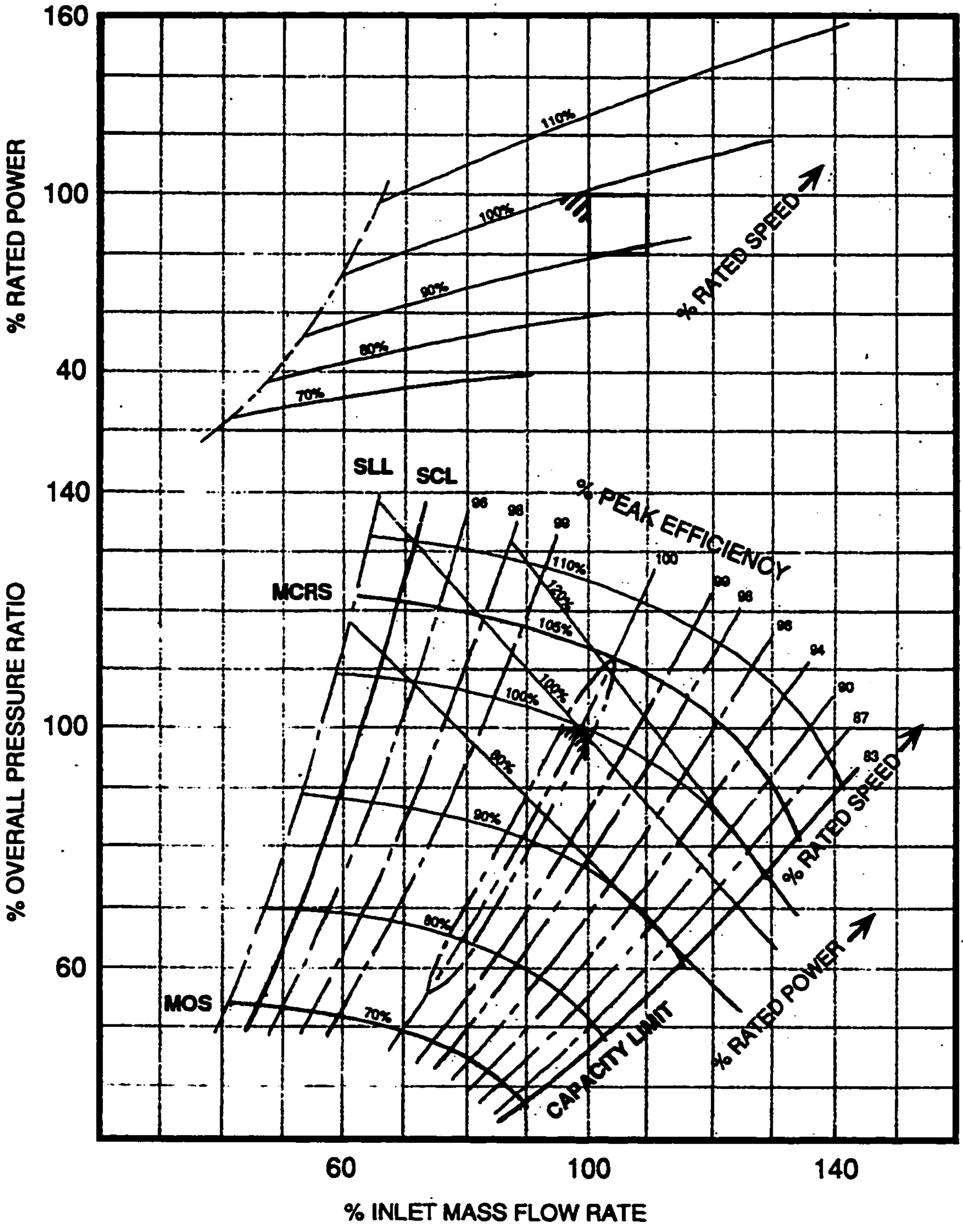

Figure 2.2 Compressor Characteristic Map: Pressure Ratio versus Mass Flow Rate

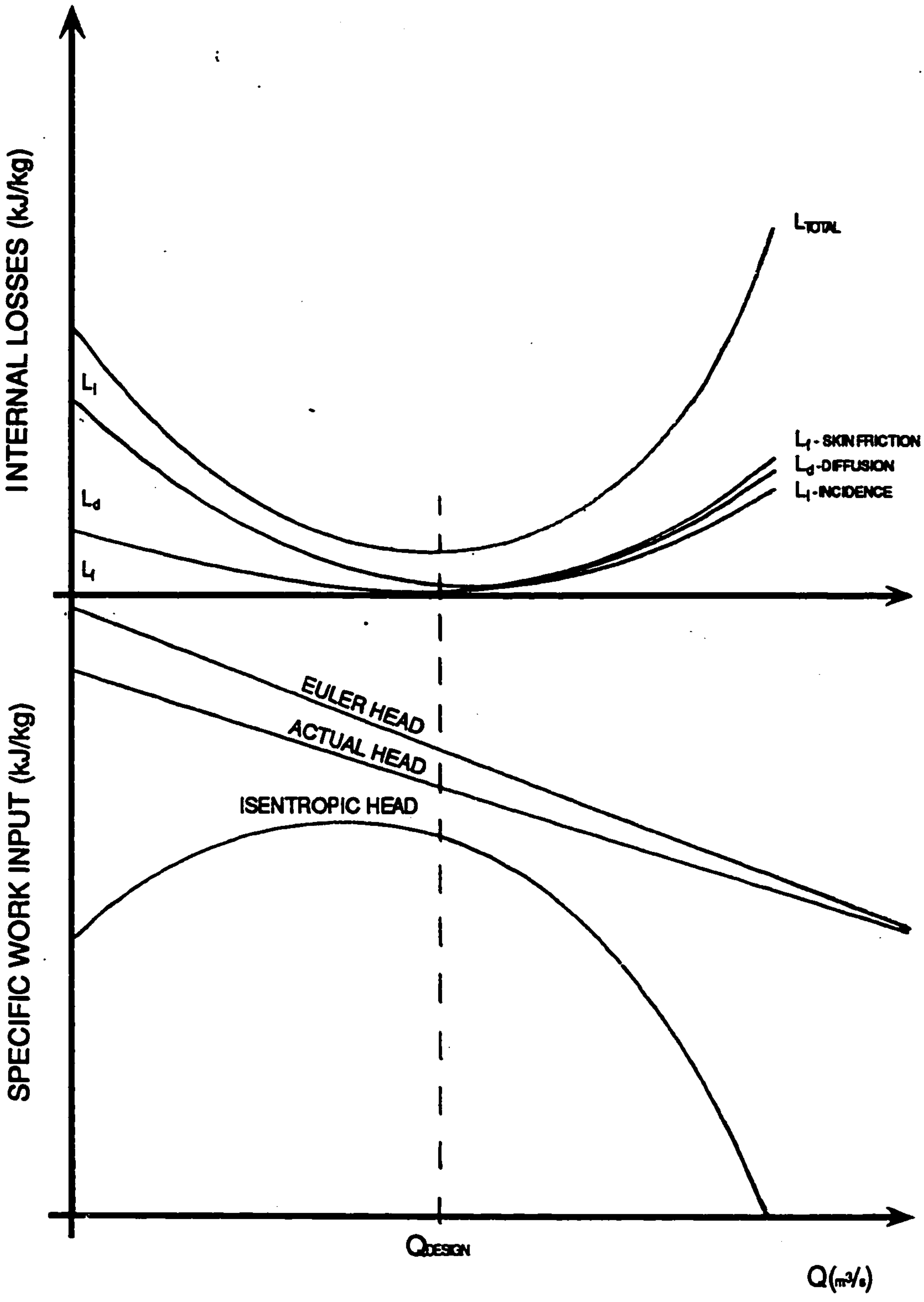

Figure 2.3 Slip and Internal Losses

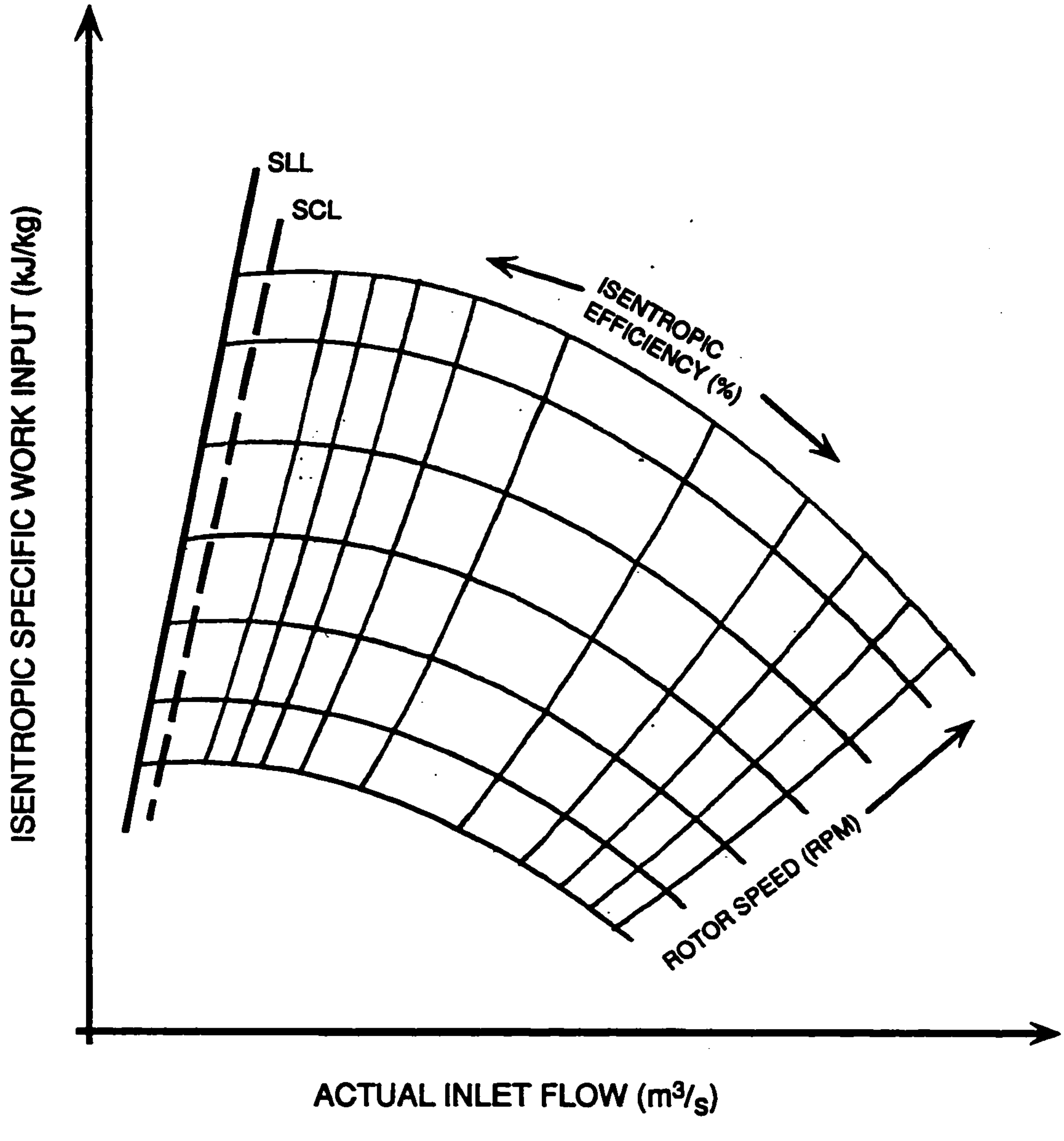

Figure 2.4 Compressor Characteristic Map: Isentropic Head versus Inlet Flow Rate

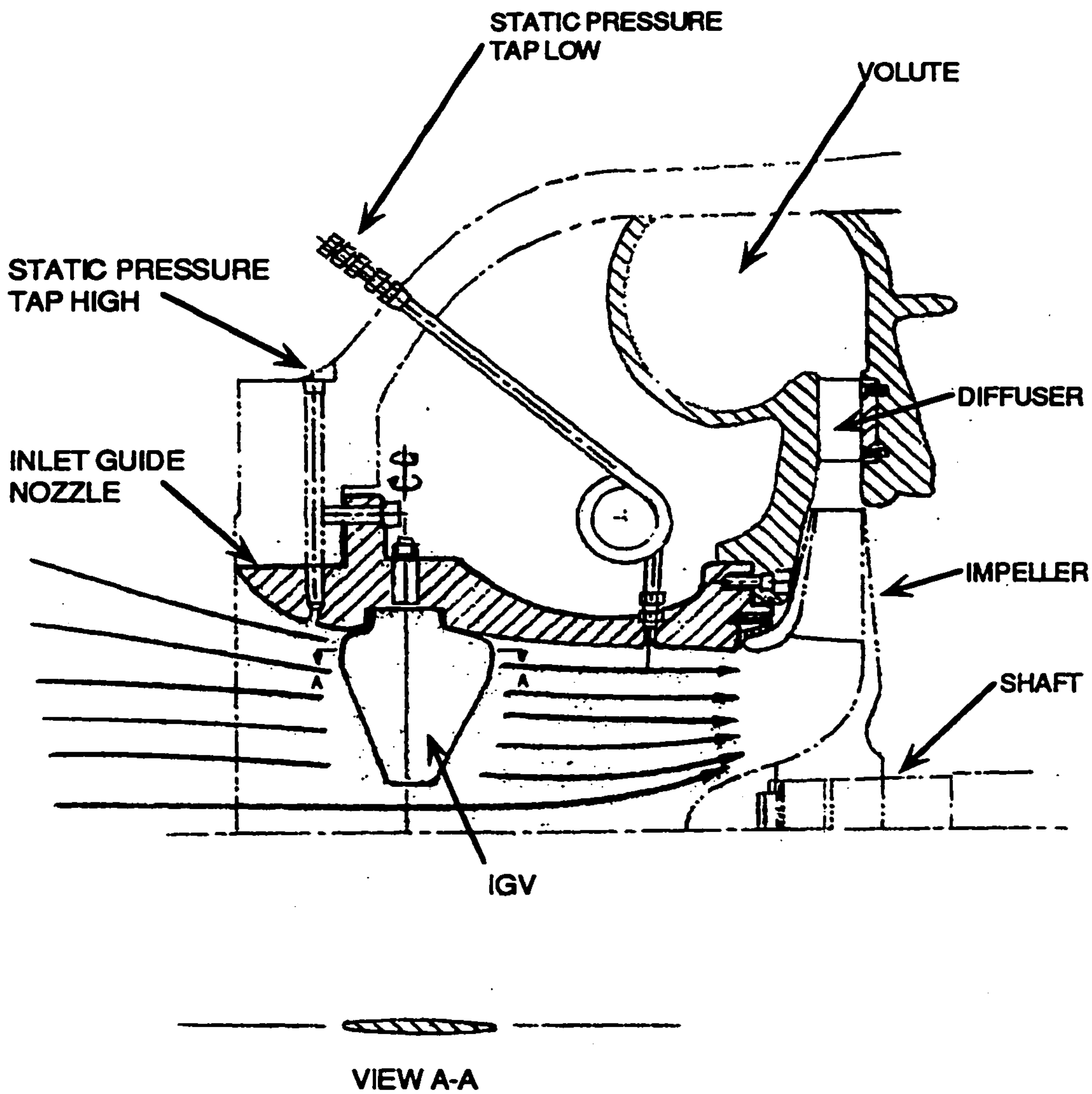

Figure 2.5 The Aero Assembly Including IGVs

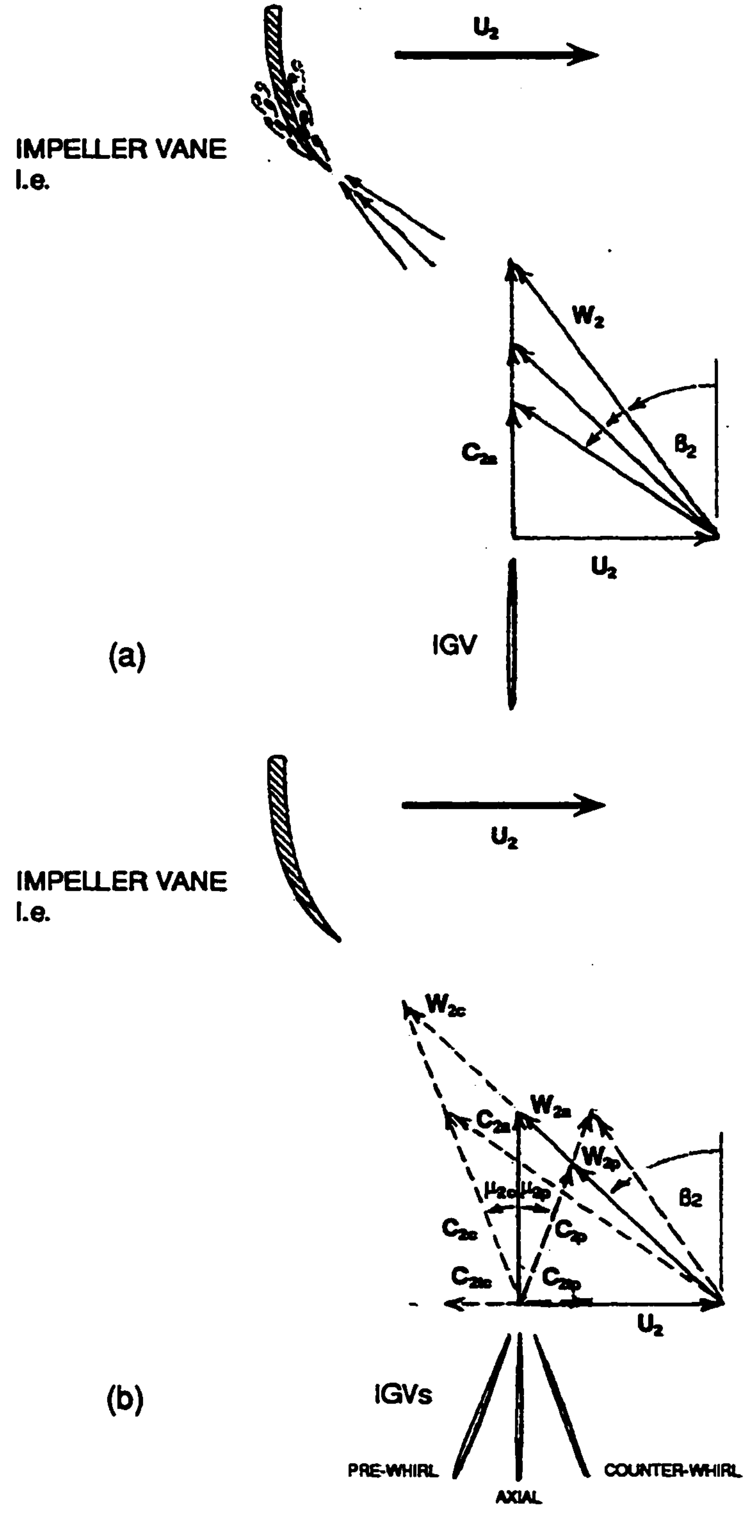

Figure 2.6 (a) Effect of Varying Mass Flow Rate on Inlet Velocity Triangles,
(b) Effect of IGV Position on Inlet Velocity Triangles

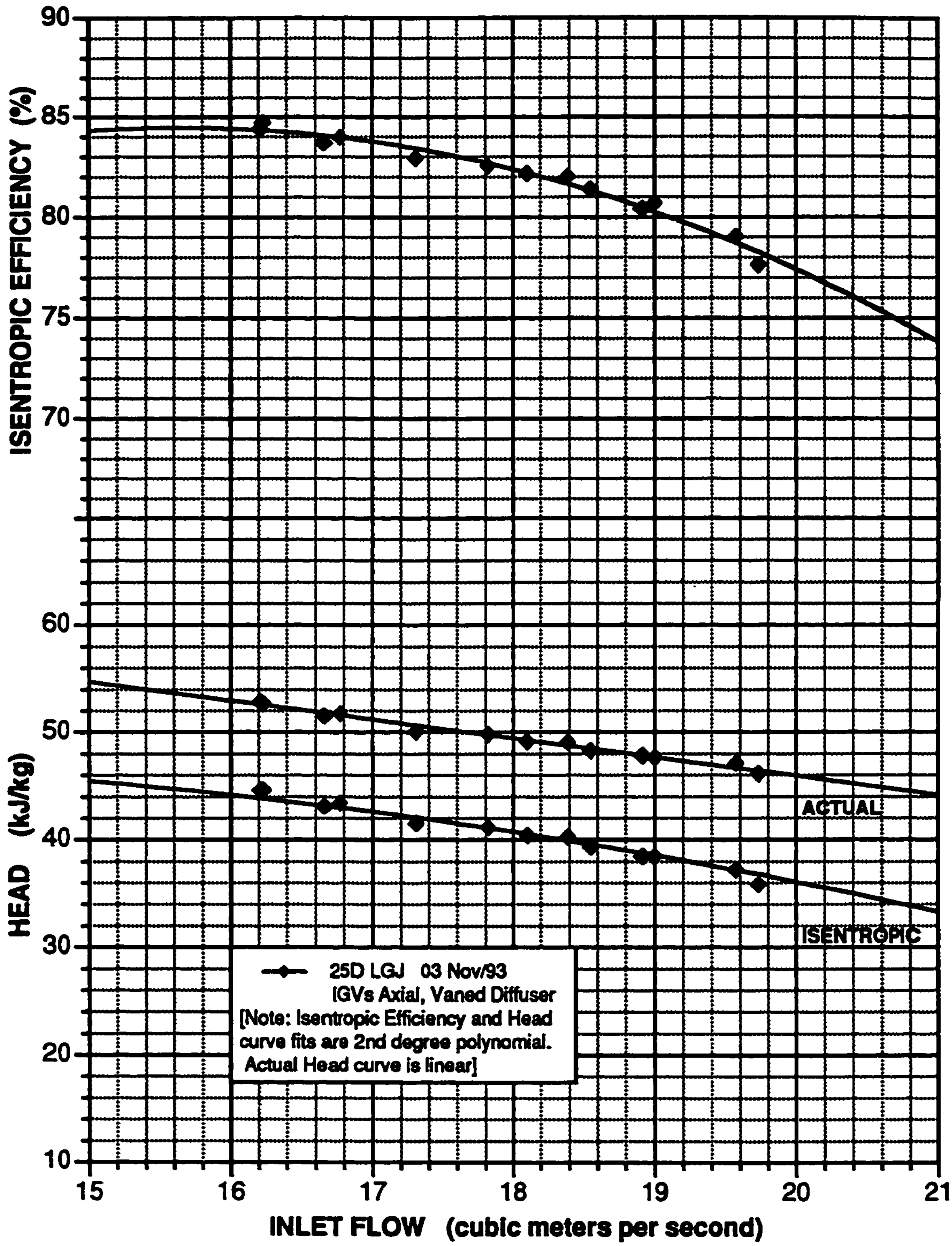

Figure 2.7 Field Test Results: Axial IGVs, Vaned Diffuser (4800 rpm)

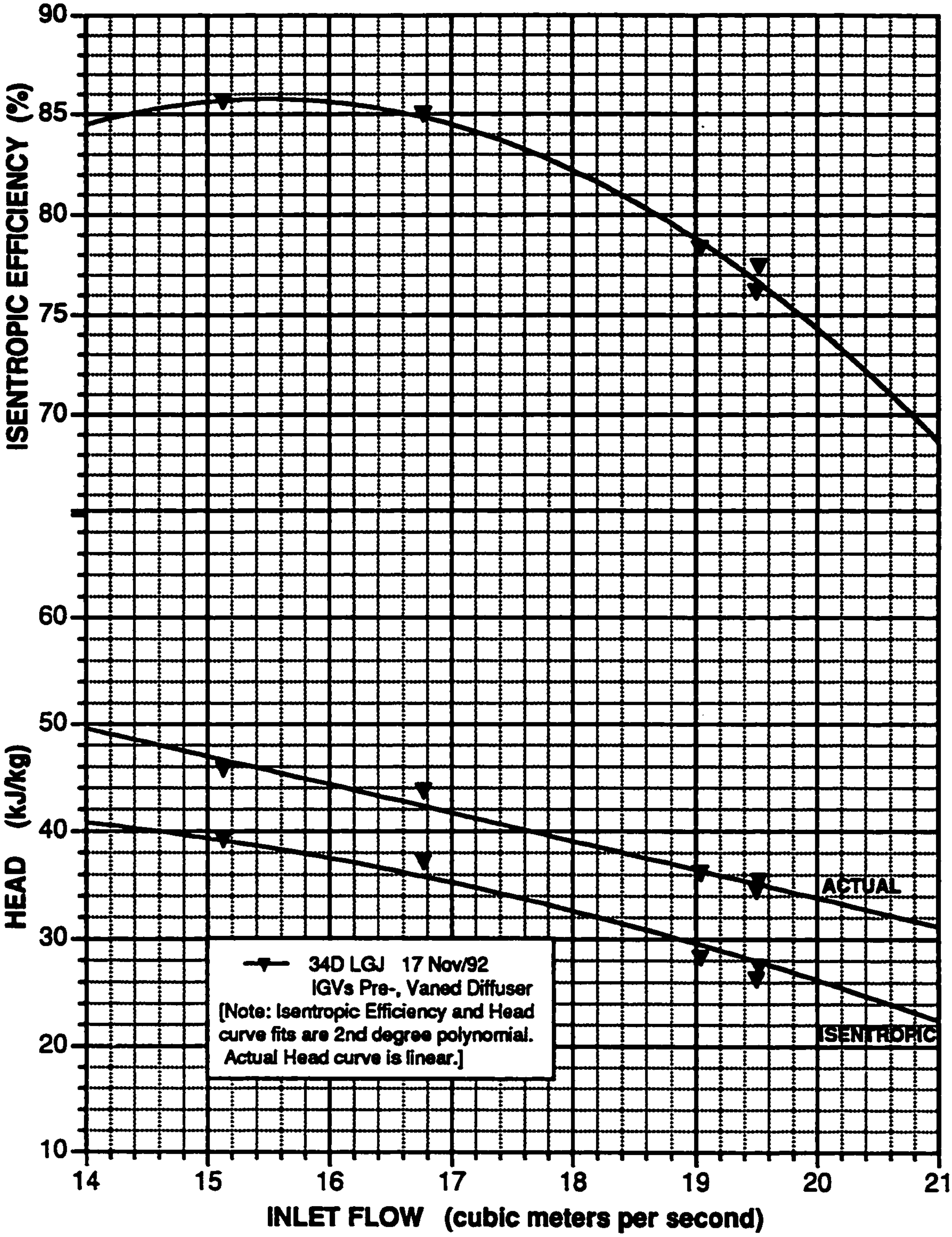

Figure 2.8 Field Test Results: Pre-Whirl IGVs, Vaned Diffuser (4800 rpm)

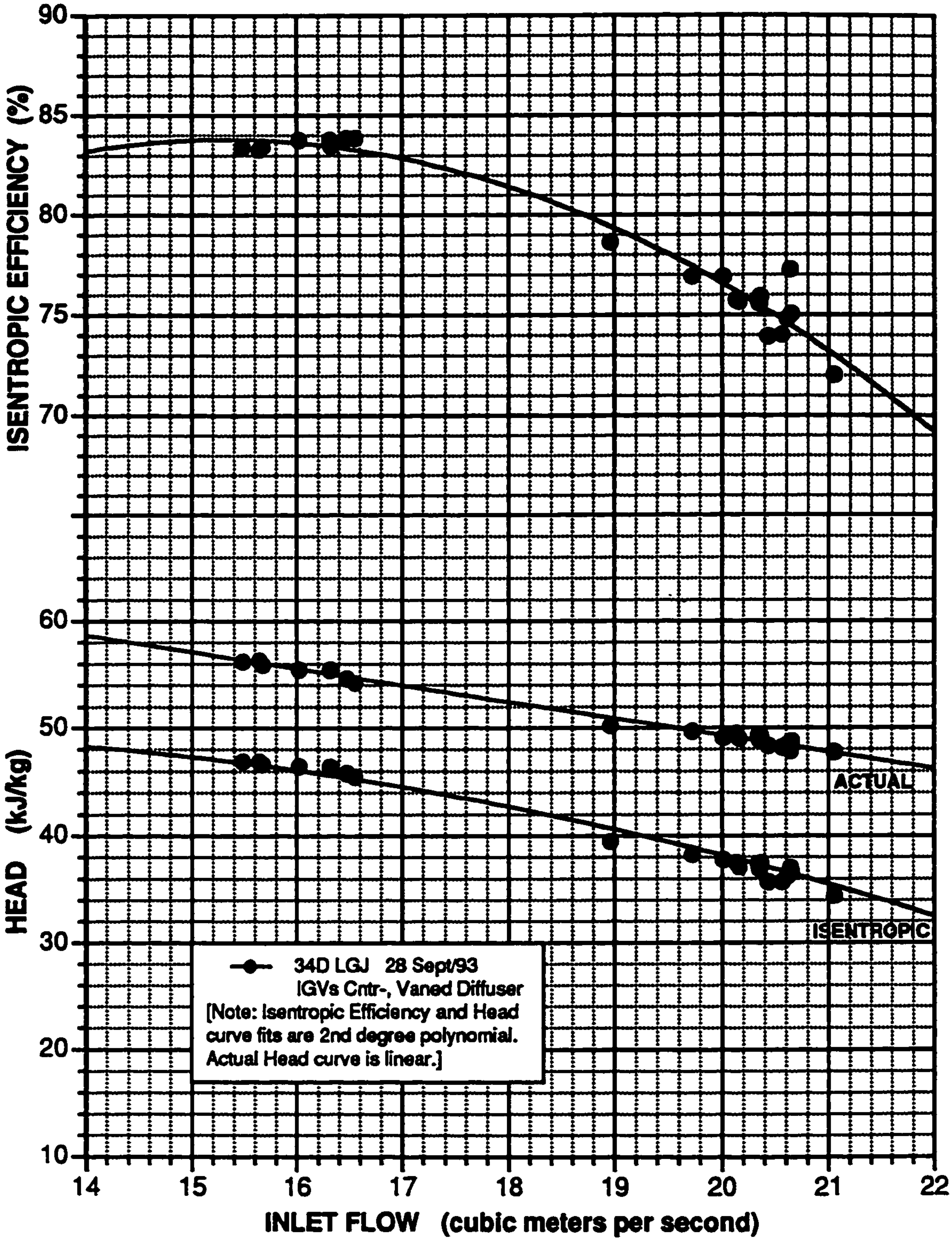

Figure 2.9 Field Test Results: Counter-Whirl IGVs, Vaned Diffuser (4800 rpm)

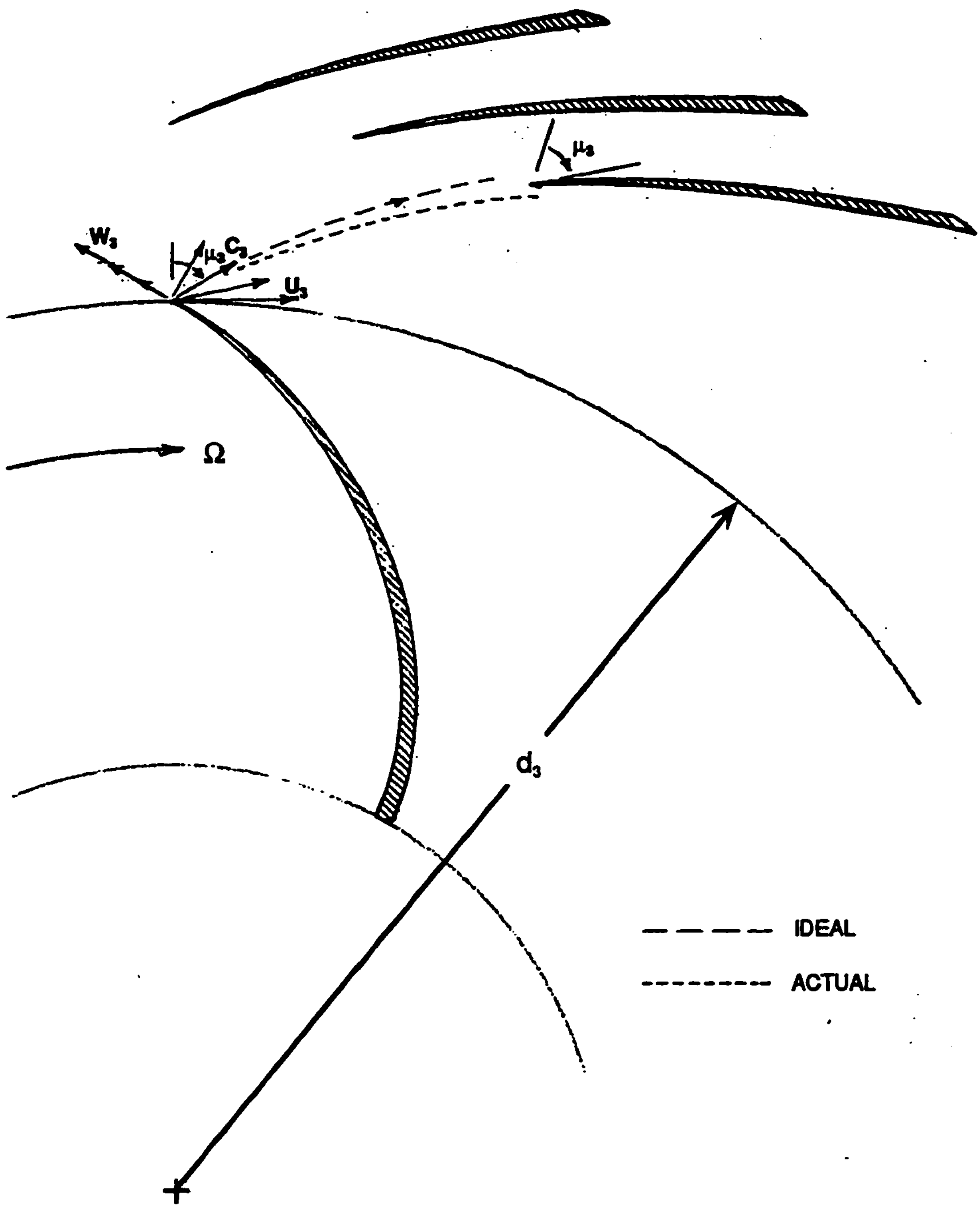

Figure 2.10 The Vaned Diffuser

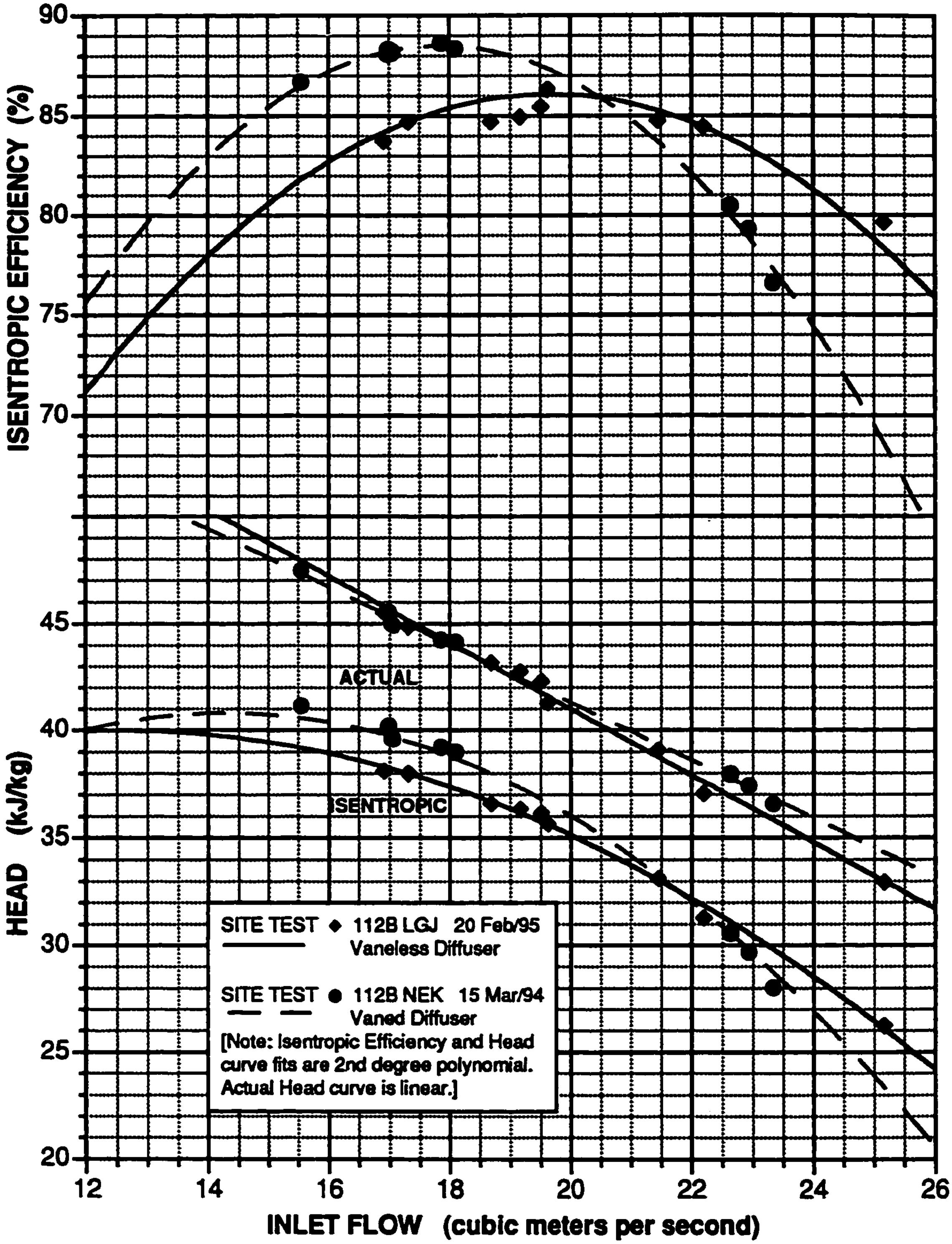

Figure 2.11 Field Test Results: Vaneless versus Vaned Diffuser (4950 rpm)

CHAPTER THREE: SURGING IN CENTRIFUGAL COMPRESSORS

The phenomenon of surge seems to be intimately linked to the phenomenon of rotating stall in that rotating stall will always precede a surge cycle. Rotating stall, on the other hand, can be present without being accompanied by a surge event (Sayers, 1990).

3.1 The Unstable Operating Condition

The point of maximum isentropic head on a compressor characteristic defines the dividing line between stable and unstable operation (see Figure 3.1). Note that the point of maximum head is well to the left of the point of maximum efficiency, meaning that the losses associated with flow rates below design, particularly positive angles of incidence at the leading edges of the impeller vanes, are just beginning to have an effect on performance.

The region to the right of the point of maximum head is considered stable for the following reason: consider the operating point A on Figure 3.1. It is located at the intersection of the compressor characteristic at the rated speed, some system resistance curve i, the compressor is operating at 100% of rated power, and at maximum efficiency. If the resistance of the system changed rather suddenly to curve ii, the compressor would have to produce the head corresponding to point B in order to maintain the same flow as before. The rate of power produced by the gas turbine driving the compressor will not change suddenly. In fact, under normal operating conditions, it will be held constant by a setpoint within the gas generator's fuel control system. The power is therefore unlikely to change at all. The rotational speed of the rotor will also not change suddenly due to the effects of inertia. The end result is that the operating point will travel initially toward Point C until the excess power increases rotor speed to Point D, where stability will be regained following the transient.

Consider now the operating point E. The compressor is operating at the rated speed on a system resistance curve iii at 90% of rated power. Again, the resistance of the system is suddenly increased, this time to curve iv. The compressor would now have to produce the head corresponding to point F in order to maintain the same flow rate, but immediately following the increase in system resistance, for the same reasons as stated above, the operating point moves down a steep portion of the rated speed curve to point G, or possibly even lower. If the compressor has time to react to the upset, the existing power level will eventually increase the rotor speed to point H, where operation might settle out (Ferguson, 1963).

The difference in response to an upset involving increased system resistance is clear. In the stable region, more head is called for, and more head is available, even at constant rotor speed. In the unstable region, more head is called for, but more head is not available until the unit increases rotor speed. Immediately following the upset, the head produced by the compressor drops, with dire consequences that will be discussed shortly.

3.2 Rotating Stall

Rotating stall is a phenomenon that will occur either at the leading edges of the impeller vanes, or at the leading edges of the diffuser vanes (if they exist). Rotating stall is initiated by a non-uniformity in the flow, typically an excessive angle of incidence (Sayers, 1990). This being the case, it is clear that one would watch for rotating stall in the impeller vanes at flows below design (unstable region), and in the diffuser vanes at flows above design (stable region), based on the conditions under which the respective vanes are known to develop positive angles of incidence.

The mechanism of rotating stall related to the leading edges (l.e.) of the impeller vanes is displayed graphically in Figure 3.2. A rate of flow below design causes a positive angle of incidence at the l.e. of the impeller vanes (streamline A). Because of the non-uniformity of either the individual blade passages or the entering flow, one passage stalls before the others (Ferguson, 1963), or several blades may stall in separate patches (Dixon, 1978). The stalled channel disturbs the flow to the channels adjacent to it. The channel ahead of it (relative to direction of rotation) is forced to accept flow at a more negative angle of incidence (streamline C), thus it recovers. The channel behind is forced to accept flow at a an even greater positive angle of incidence, causing it to stall (streamline B). However, the very act of causing the passage behind to stall also initiates the recovery of the passage originally stalled. Thus the disturbance moves against the direction of impeller rotation or, conversely, in the direction of blade lift (Ferguson, 1963), at a frequency related to shaft speed (Sayers, 1990), disturbing flow in the passages as it goes.

Given that the stall is carried with the impeller, but travels against the direction of rotation, the frequency of this particular phenomenon could be expected to occur at a frequency that is some fraction of rotor rotational frequency. Rotating stall in the diffuser vanes would be fundamentally the same, but, as pointed out, would be expected to occur at flows above design. Another difference is that in the diffuser vanes, the stall is moving in vanes that are not rotating and it is moving in the direction of impeller rotation. It is therefore difficult to predict the frequency at which this phenomenon would become apparent. It could show up at a frequency slightly less than the vane passing frequency, or at whatever frequency it traveled round the diffuser ring, i.e. much less than rotor frequency.

As inferred above, rotating stall has been known to occur in both the stable and unstable regions of compressor performance (Balje, 1981; Cohen et al., 1987; Sayers, 1990). It is believed to be an important cause of instability and poor performance (Cohen et al., 1987) and is also believed to lead to surging if it occurs in the unstable operating range (Cohen et al., 1987; Sayers, 1990). Rotating stall has been observed in compressors with both vaned and vaneless diffusers, and it is believed that a reduction in the number of vanes in a vaned diffuser reduces the intensity of stall (Ferguson, 1963). Although there is a rotating disturbance in the flow, the integrated flow through the compressor is steady in time (Ferguson, 1963). The condition is self-sustaining, and if allowed to persist, can cause blade failure if the natural frequency of the blade is excited (Dixon, 1978).

3.3 Surge

The dividing line between stable and unstable operation is generally believed to be the boundary of what is known as the surge zone in centrifugal compressor performance. The operation of a compressor to the left of this dividing line is not considered conservative (Shepherd, 1960). If one were to join the points of maximum head on a compressor characteristic with a smooth curve, this curve would be known as the SLL (see Figure 2.2). If a compressor is forced to operate in the vicinity of the SLL, rotating stall can begin to take place as described in Section 3.2. If the system resistance is increased further, there can be an increase in the number of stalled passages, leading finally to compete stall of the rotor (Ferguson, 1963). This forces the operating point down the positively sloped portion of the characteristic (i.e. in the unstable region) as described in Section 3.1.

Once the discharge pressure of the compressor has been reduced by the conditions described above, the compressor is in the undesirable situation of having the pressure in the discharge header instantaneously higher than discharge pressure at the outlet flange. This can result in a rapid reversal of flow through the compressor along with the associated rotor thrust-load reversals. After the pressure in the discharge header has been relieved somewhat by the reversal of flow, it is believed that the event can be repeated if the system resistance has not been reduced from the level that initiated the whole process.

The events described above are known as surge events, and it is often implied that they take place very rapidly, i.e. in the order of one second or less. Most authors are, however, hesitant to attach a precise time span to the event. The surging of a compressor is usually associated with a general increase in noise level which is thought to be indicative of a pulsation of the flow and of mechanical vibration. The higher vibrational "...frequencies are known to be due to rotating stall and are of the same order as the rotational speed of the impeller." (Dixon, 1978). At the high pressures typical of gas pipeline operation (6894 kPag), "...the effect may be almost explosive, with continued operation likely to lead to physical damage due to impact loads and high-frequency vibration." (Shepherd, 1960).

The flow pulsation and high frequency vibration mentioned above may well be the result of the fluid flowing radially inward through the diffuser vanes (if they exist), back down the logarithmic spiral (Figure 2.10) and approaching the trailing edges of the backward sloped impeller vanes at an almost perpendicular angle. Some authors suggest that the rotor actually reverses rotational direction during a surge cycle (Balje, 1981). This may be reasonable for a machine with a rotor (including the driver's rotor) of small mass moment of inertia. It seems unlikely in the case of a pipeline compressor. The mass moment of inertia of the compressor rotor alone is significant, never mind the rotor of the power turbine. If there did happen to be enough reverse torque on the impeller to reverse rotational direction of the whole train, the coupling would most likely be sheared.

As discussed in Section 1.3.2, the damages that can result from one or more surge events can be extensive: surge should be avoided at all cost. The prevention of surge is the topic of Chapter Four.

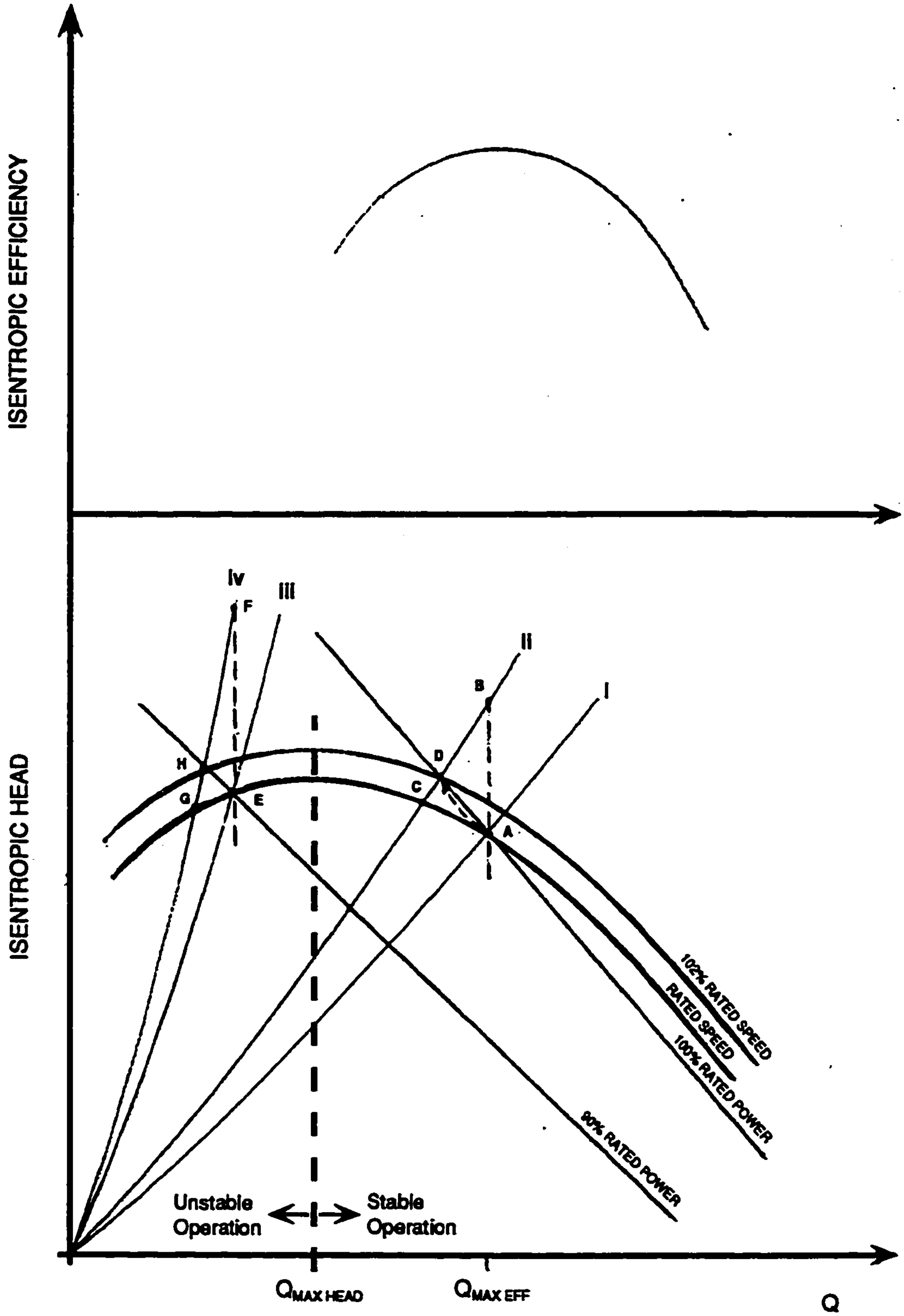

Figure 3.1 The Unstable Operating Range

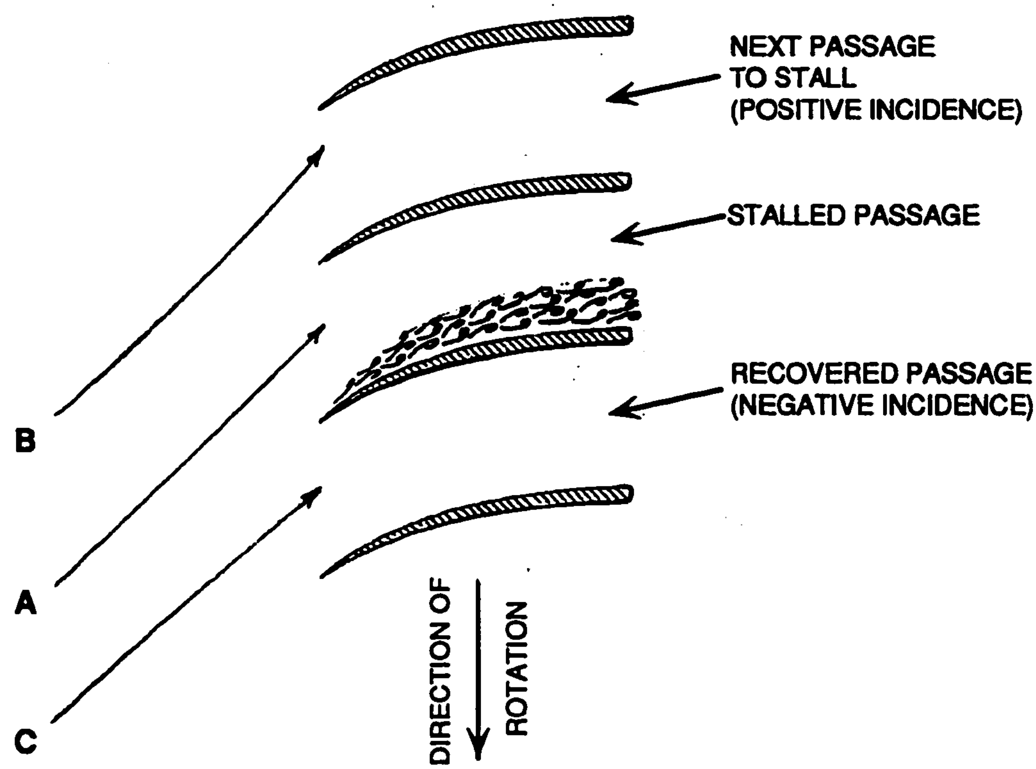

Figure 3.2 Rotating Stall at Impeller Vane Leading Edges

CHAPTER FOUR: SURGE CONTROL

The purpose of a surge control system is to prevent the possibility of a compressor operating in the unstable region as defined in Section 3.1. The system accomplishes this by monitoring the flow through, and the pressure rise across, the compressor (i.e. the operating point) and comparing the current operating point to known setpoints (the SCL). The accurate determination of the setpoints is the objective of the surge test.

The surge controller influences the operating point (OP) of the compressor via a recycle valve. By opening the recycle valve, the surge controller reduces the pressure rise across the compressor, and increases the flow through it, thereby moving the OP away from the unstable region.

The surge control system consists of recycle line piping, a recycle valve c/w associated controls, head and flow differential pressure transmitters, the surge controller itself, and a current-to-pneumatic (I/P) transducer. The basic layout of a typical surge control system is shown in Figure 4.1.

4.1 The Recycle System

The piping layout for a typical recycle system consists of the recycle line piping and the recycle valve as shown in Figure 4.1. This system is referred to as a "hot" recycle system because no effort is made to cool the recycled gas before it re-enters the unit suction header. At the present time hot recycle systems are, almost without exception, the type found on gas pipeline compressors. A "cold" recycle system involves the use of heat exchangers to cool the discharge gas. The cold systems offer definite advantages in that the compressor, recycle valve, and piping are not subjected to the relatively high gas temperatures that are inevitable with hot systems. However, the heat exchangers, along with the associated valves and piping, represent a significant initial outlay of capital. Cold recycle systems are therefore only found on gas compressors where gas coolers have been installed to cool the discharge gas for pipeline coating or capacity reasons.

4.2 Pressure and Flow Measurements

In order to keep the compressor operating in the stable flow region, the surge controller must constantly monitor the OP and compare it to the SCL. For the purposes of surge control, two differential pressure signals are used to indicate head and flow. The difference in static pressure between entry to the compressor and the eye of the impeller (see Figure 2.5, and Figure 4.1) is taken to be indicative of the mass flow rate through the compressor (see Appendix D). The difference in static pressure between the discharge header and the suction header (see Figure 4.1) is taken to be indicative of the head produced by the compressor (see Appendix D).

4.3 Instrumentation

Differential pressure transmitters are used to read the two differential pressure signals referred to in Section 4.2. Transmitters such as this take the pneumatic pressure differential as input, and output an electrical current between 4 and 20 milliamps (mA) based on the calibration of the device. The required maximum input range must be known when the transmitters are selected. The operating point M (Figure 4.2a), at the intersection of the SLL and the maximum power curve, is used to determine the required maximum transmitter range (see Appendix D). There is a linear relationship between the pneumatic input and the electrical current output within the working range of the transmitter. The pneumatic inputs come from the compressor and the headers as shown in Figure 4.1. The transmitters themselves are located in the plant near the compressor. The 4 to 20 mA output of the transmitters is hard-wired to the surge controller in the control room.

The output of the surge controller is a 4 to 20 mA signal to the I/P transducer, which is mounted on the recycle valve as part of the recycle valve control. Similar to the differential pressure transmitters, there is a linear relationship between 4 to 20 mA input signal, and the 3 to 15 psig pneumatic output signal to the recycle valve positioner. 20 mA to the I/P means 15 psig to the valve positioner calling for the recycle valve to close. 4 mA to the I/P means 3 psig to the valve positioner calling for the recycle valve to open, i.e. fail-safe. The recycle valve positioner and other devices associated with the recycle valve control are not shown in detail in Figure 4.1 so as not to clutter the diagram. Fail-safe solenoid valves are shown in the energized state. Should they be de-energized, the recycle valve would immediately go 100% open.

4.4 The Surge Controller

Modern digital surge control systems will exist either as a stand-alone, black box type of dedicated control, or within a programmable logic computer (PLC) along with other unit control systems.

If a digital surge control system exists as a dedicated system, it will receive the analog inputs from the head and flow transmitters directly, and it will send the analog output directly to the I/P transducer at the recycle valve. It will receive discrete inputs from the unit PLC to indicate the required operating state: normal, shutdown, or purge. It will also send discrete outputs to the PLC to make it aware of events such as "controller output failure", "analog input out of range", or "operating point in the unstable region".

If a digital surge control system exists within the unit PLC, it will communicate with the instrumentation in the plant via input/output (I/O) blocks. On the input side of the surge control system, the I/O blocks will receive the 4 to 20 mA analog signals from the head and flow transmitters, drop the current across a resistor to produce 0 to 5V, digitize the voltage, and send the signal on to the PLC as a bit count.

On the output side, the PLC will send a bit count to the I/O block, the I/O block will perform the reverse of the above process, and send a 4 to 20 mA analog signal to the I/P transducer at the recycle valve. With the PLC based system, the discrete communication mentioned in relation to the stand-alone system is redundant: the PLC looks after all alarms and operating states.

From a functionality point of view, the two different forms of digital surge control are identical. In addition to this, dedicated systems from different manufacturers are in use, and software packages for PLC based systems from different software suppliers are in use. For that reason, Figure 4.1 simply shows a box identified as a surge controller communicating with the transmitters and the I/P transducer. In the case of a dedicated system, the surge controller is intended to include the black box along with the portion of the PLC that is working with it. In the case of a PLC based system, the surge controller is intended to include the PLC and the associated I/O blocks.

4.4.1 The Domain of the Surge Controller

As discussed in the previous section, the surge controller monitors the OP of the compressor based on the analog inputs from the head and flow transmitters. Figure 4.2(b) shows the domain of the surge controller, and Figure 4.2(a) shows the corresponding control lines on the compressor characteristic. The OP shown on Figure 4.2(b) is at approximately 60% of full head transmitter span and 30% of full flow transmitter span.

Point M on Figure 4.2(a) is used for the set up of the surge controller. By the methods described in Appendix D, the expected analog inputs corresponding to point M are determined, and they are used to define the SLL in the domain of the surge controller as shown in Figure 4.2(b). In this case, it was expected that at point M, under typical operating conditions, the chosen head transmitter would be at 70% of span, and the chosen flow transmitter would be at 25% of span. A straight line drawn through these coordinates and the origin forms the SLL. In the domain of the surge controller, this represents the SLL shown on the compressor characteristic in Figure 4.2(a).

On the compressor characteristic, the SCL is typically placed at approximately 110% of the flow at the SLL for any given head. Because the actual inlet flow is proportional to the square root of the differential pressure signal from the flow meter in the compressor, in the domain of the surge controller the SCL is located at $(1.10)^2$, or at 121% of the flow transmitter reading on the SLL for any given value of head transmitter output. The SCL is therefore shown on Figure 4.2(b) at $(1.10)^2 \times 25\% = 30\%$ of flow transmitter span and 70% of head transmitter span. The SCL represents the setpoints used by the surge controller to determine when action (output) is required.

The recycle valve open (RVO) line is typically located midway between the SLL and the SCL at 105% of SLL flow, and the surge backup line (SBL) is typically located immediately behind the SLL at 97% or 98% of SLL flow. The control lines are shown on the surge controller domain in Figure 4.2(b). The relative positioning of the lines presented here is only typical. The protection can be more conservative, but it is unlikely that it would ever be less conservative. Since all of the control lines are referenced, either directly or indirectly, to the SLL, if the SLL is moved, all of the other lines move with it. This feature proves to be convenient during a surge test, which will be discussed in Chapter Five. The positioning of the control lines described above is accomplished by the inputting of configuration constants to the surge controller. The positioning of the control lines must be manipulated during a surge test.

4.4.2 Surge Controller Response

The surge controller will constantly monitor the position of the OP relative to the SCL. As long as the OP is to the right of the SCL (and the recycle valve is closed), no action is taken by the surge controller, and the output would remain at 20 mA.

PI Response (Closed-Loop)

If the OP were to move toward the SCL by either an increase in head or a decrease in flow (or both), and if the OP were to make contact with the SCL, proportional plus integral (PI) action would be taken by the surge controller. The aggressiveness of the PI response is determined by two factors: the constants input by the user, and the error. The error is defined in different ways by different systems, but it is always indicative of the distance between the OP and the SCL.

The surge controller, when initiating PI response to protect the compressor, should be capable of maintaining the OP on (or near) the SCL by controlling recycle valve position. When a surge controller operates in this manner, it is said to be properly tuned. Tuning is accomplished by adjusting the constants mentioned above until the PI response of the controller is aggressive enough to protect the compressor under most operating conditions, but not so aggressive as to cause erratic recycle valve action. This type of response is referred to as closed-loop response because the effect of changes in surge controller output is always being monitored based on the error.

Open-Loop Response

If the PI response can't keep the OP from continuing to move to the left (toward the unstable region), it will eventually contact the RVO line. The response of the controller when the RVO line is touched by the OP is called open-loop response because this response is not based on the error as the PI response is. Open-loop response involves the opening of the recycle valve in predetermined step sizes (typically 25% of full stroke) with a predetermined time interval between the step openings (typically 0.5 sec) until the OP is to the right of the SCL and moving away from the unstable region. The open-loop output then decays back down to zero at a rate that is determined by the configuration of the surge controller.

Under all but the most unusual circumstances, the combination of closed- and open-loop response outlined above is adequate to protect the compressor and keep the OP to the right of the SLL. There are times, however, when an upset is so severe, and it happens so quickly, that the OP does cross to the left over the SLL and the SBL. The crossing of the SLL initiates no controller action. The main purpose of the SLL is to be a reference for the other control lines. The crossing of the SBL will initiate a unit shutdown, or at the very least, the sending of a return to idle (RTI) command to the unit PLC.

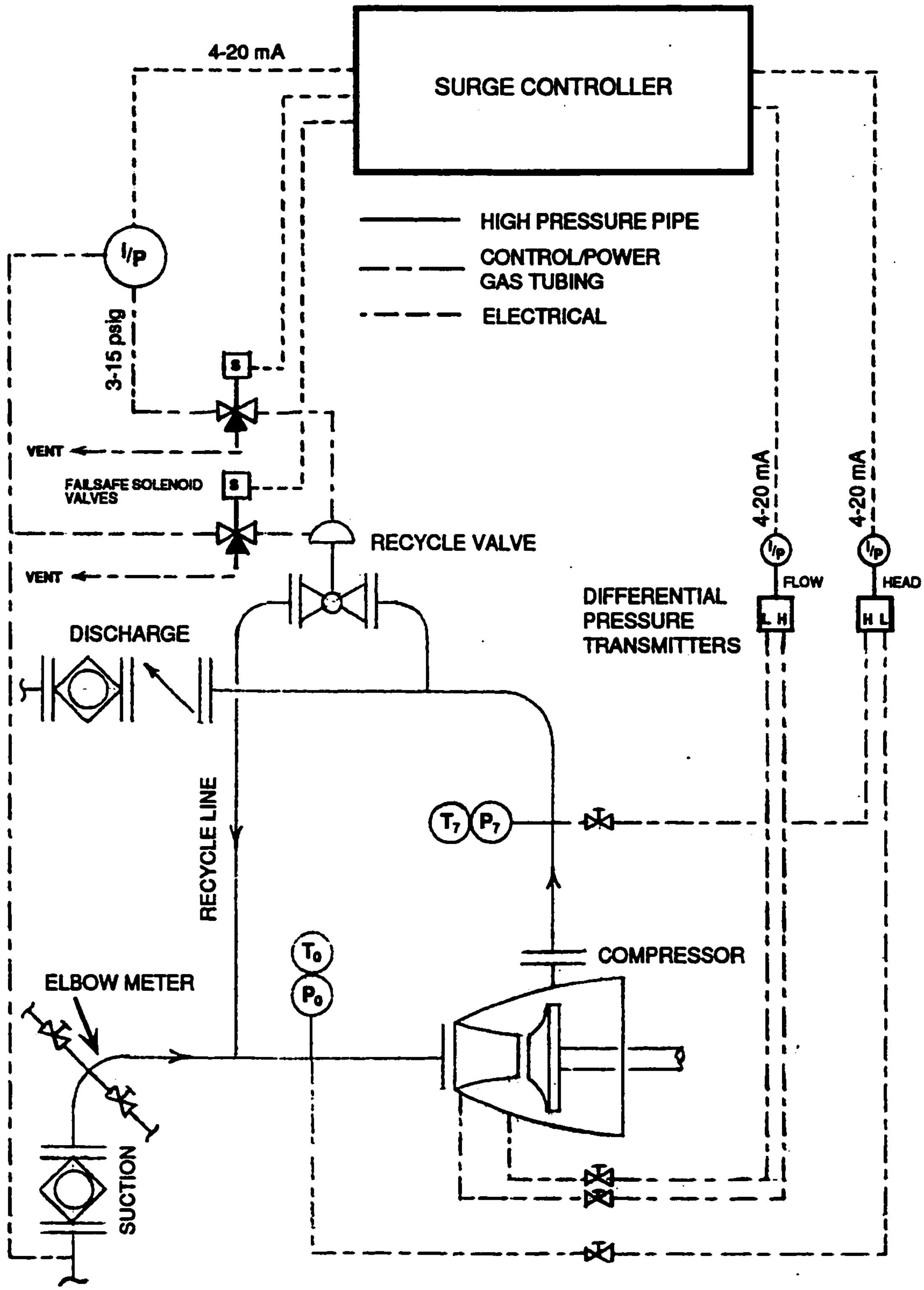

Figure 4.1 Surge Control System: Instrumentation and Piping Layout

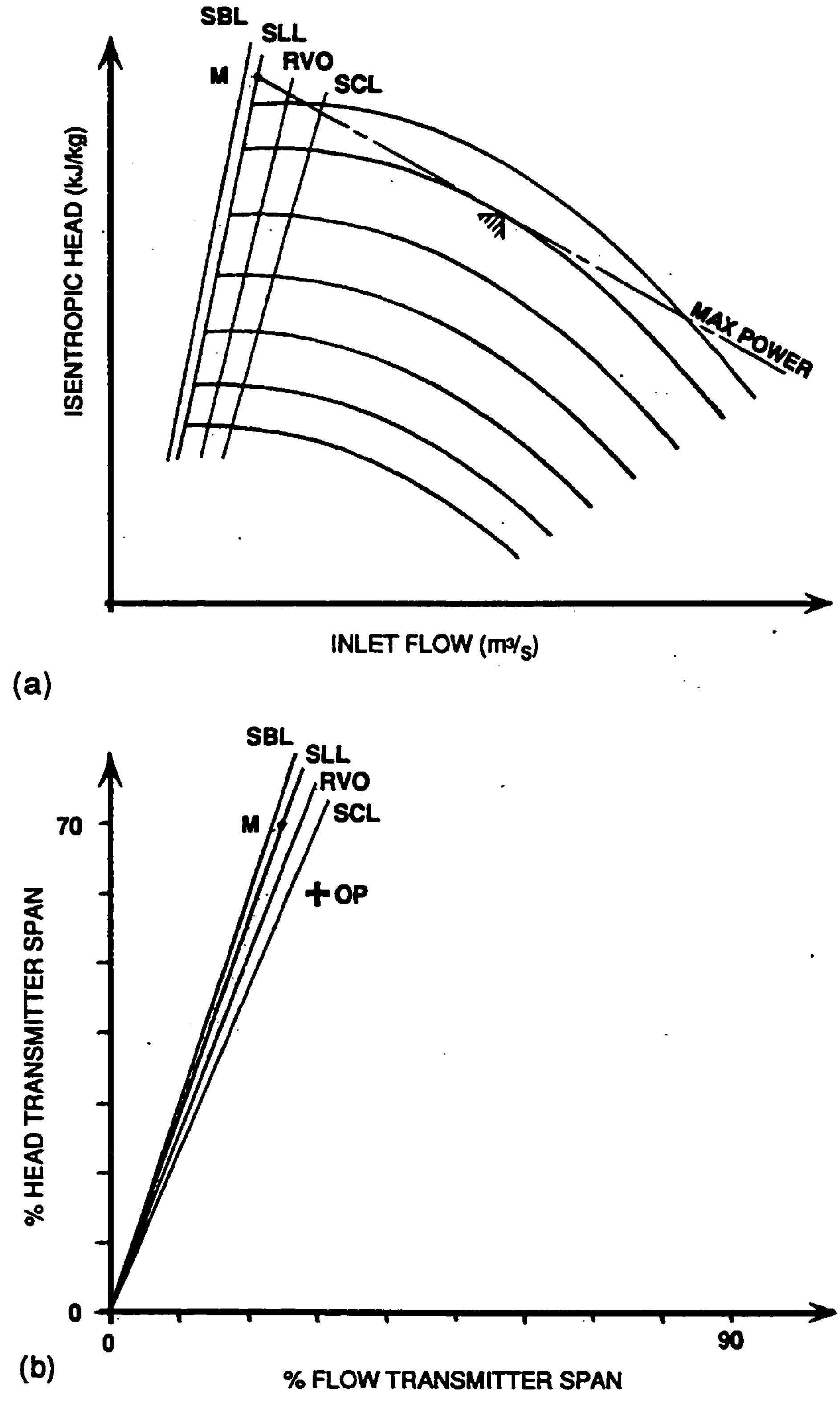

Figure 4.2 (a) Surge Control and The Compressor Characteristic (FP-301); (b) The Domain of The Surge Controller

CHAPTER FIVE: SURGE TESTING CENTRIFUGAL COMPRESSORS

This chapter describes in detail the procedure developed by the author for the surge testing of centrifugal pipeline compressors.

The development of the procedure began with the gathering of all available knowledge from colleagues of the author. This formed the foundation of the new procedure. Equipment manufacturers and the vendors of surge control systems were consulted, with some providing advice. Surge testing consultants were contracted to undertake surge tests for a fee while being observed by the author. Their methods, although not considered complete, were used for the framework of what eventually became the pre-test activities and the test procedure itself. The author then applied the final modifications, which brought both the pre-test activities and the test procedure up to the required standards, i.e. the standards considered acceptable by the owner of the compressors to be tested.

5.1 Situations Requiring a Surge Test

Aero assemblies are discussed in Appendix A. Sometimes several aero assemblies are manufactured under the same design. Such a family of aero assemblies are considered to be identical from a performance point of view, so a surge test on any one of the aero assemblies in the family will determine the performance of all. Any time an aero assembly representing an unfamiliar design, be it a new aero assembly or one that has undergone geometry changes, is added to the fleet, a complete performance and surge test is required (see Appendix C). A surge test is not something that is undertaken unless it is deemed to be absolutely necessary. The reasons for this prudence will be discussed in the next section.

5.2 The Objectives of a Surge Test

The primary objective of the surge test is to determine the true position of the SLL as discussed in Chapter Three. Knowing the true position of the SLL allows the surge control system to be properly setup as discussed in Chapter Four, and provides advantages that will be discussed in Chapter Six. An unavoidable fact in any surge test is that the compression unit, and the productivity of that unit, is being placed at risk.

Natural gas pipeline companies typically have a group of people who are dedicated to determining the true performance of each aero assembly on the fleet (Performance Analysis) and a group of people who are dedicated to the operation, maintenance, and productivity of the compressors (Field Operations). The former group must obtain the information they seek through the surge testing of compressors for the reasons outlined in Chapter One. The latter group are primarily concerned with keeping the compressor running and keeping it out of situations in which it could be damaged.

The individual who undertakes the performance and surge test (the tester) typically belongs to the Performance Analysis Group mentioned above, but he must work with members of the Field Operations Group to accomplish the test. The tester is in charge of the test, and it will be carried out under his direction. The completion of a successful surge test requires good judgment on the part of all involved personnel. It is essential for the tester to have the confidence, support, and cooperation of Field Operations.

To call the test to be subsequently described a surge test is really a misnomer: the idea is to bring the compressor to the point of instability, i.e. rotating stall, and no further. It should never be the intention of the tester to actually surge the compressor. Sometimes consultants, or the vendors of surge control systems, will propose that actually surging a compressor during a surge test is quite acceptable. This practice is seldom advocated by the owners of pipeline compressors.

No matter how carefully a surge test is conducted, there is always a strong possibility that the unit will be shut down and maybe even damaged. If the unit is shut down, it often takes up to several hours to get the compressor back on line, and by then the enthusiasm of Field Operations will have been seriously diminished. If the test results in damage to the compressor, the confidence of Field Operations in the tester will be lost and further surge testing at that particular site will be out of the question, regardless of the potential gains.

So, although the primary theoretical objective of the test is related to gaining knowledge of compressor performance, no less important are the practical objectives related to carrying the test off without shutting down or damaging the compressor.

5.3 Pre-Test Activities

Since the surge control system must function correctly under test conditions, it is necessary to check the operation of the entire system before the test begins. This portion of the test procedure is referred to as the pre-test activities. The pre-test activities outlined below usually take approximately four hours of downtime (D/T).

- The tubes feeding the pneumatic signal to the head and flow transmitters are blown out. If liquid is present in the tubes, it can cause erroneous transmitter readings.

- The calibration of the head and flow transmitters is checked, and they are re-calibrated as required. The as-found and the as-left calibration is recorded. The digital readout on the surge controller representing analog inputs is checked at zero, midpoint, and maximum span for both transmitters. This will ensure that no hard-wire problems exist between the transmitters and the surge controller.

- Both transmitters are leak tested, and all leaks are eliminated. A leak can cause erroneous transmitter readings, especially if the leak is located close to the transmitter.

- The surge controller is placed in manual mode and the fail-safe solenoids are energized. Recycle valve response is checked by varying surge controller output and comparing to recycle valve position. The stroking speed of the valve is checked in both directions: opening stroke should take less than two seconds, closing stroke less than ten seconds. Dead zones at either end of valve stroke must be checked and minimized as required. Dead zones should not exceed two to three percent of controller output, especially coming off the fully closed stops. Recycle valve action must be checked for hysteresis and dealt with as required.

- Recycle valve control is checked by connecting an electric current generator to the I/P transducer out in the plant in place of the controller output signal. Step changes are initiated in recycle valve position, and any overshoot is noted. Overshoot should be less than 25% of step magnitude. Oscillations in recycle valve position must be checked and eliminated as required. The settling time must also be checked. Settling time should be less than 2 seconds.

Although the activities outlined above require D/T, if they are carried out in a thorough manner they guarantee that the surge control system will function predictably, not only during the surge test, but for normal operation following the test. There are often unexpected problems discovered during this portion of the undertaking, even at locations staffed with competent and conscientious personnel. It is always in the best interests of all concerned to complete the pre-test activities.

5.4 The Surge Test Procedure

After the pre-test activities have been completed and the compressor is back on line, a spectrum analyzer[3] is connected to the appropriate output jacks on the front panel of the vibration monitoring equipment in the control room. Proximity probes measuring radial rotor displacement in the X and Y directions are usually located at the drive end (DE) and non-drive end (NDE) of the

[3] The spectrum analyzer is a device that is capable of displaying vibrational activity in the frequency domain.

compressor (two at each location). An additional two proximity probes are usually available to measure axial rotor displacement.

If the spectrum analyzer being used is a single channel model, a proximity probe monitoring radial rotor displacement is chosen as close to the impeller as possible (usually NDE). If a second channel is available, it is connected to one of the probes monitoring rotor axial displacement. If additional channels are available, a proximity probe measuring radial shaft displacement at the location further from the impeller can be monitored (usually DE). The sub-synchronous vibrational activity representative of normal operation is either noted, or a printout is obtained (see Figure 5.1). The amplitude of the 1X running speed peak is noted, along with any other prominent peaks at sub-synchronous frequencies.

The unit is placed in local manual control to maintain constant compressor speed. The surge test begins with the manipulation of the control lines in the domain of the surge controller. The positions of these lines are determined by the configuration constants input into the surge controller. The first step is to move the SBL a safe distance to the left of the SLL, approximately 90% of SLL flow should be adequate. The reason for this is to avoid unnecessary unit shutdowns during the surge test.

The current deviation displayed on the surge controller read-out should be noted. The deviation is an indication of the distance between the OP and the SCL. The SLL is moved out to the right until the deviation is zero. Recall that all control lines are positioned directly or indirectly relative to the SLL, so moving this line moves all lines. Note that it is tempting to simply begin reducing the flow while leaving the surge protection as found. This method is dangerous, since it assumes that the SLL is accurately placed, and also that the surge control setup was done properly. If these assumptions are incorrect, the unit could be surged before the surge protection is even reached. A safer method is to move the surge protection out to the OP, and then reduce the flow to the unit in a manner that maximizes the degree of flow control while also maintaining maximum unit protection.

The surge controller is switched to the manual mode, and the recycle valve is opened to 40-50% of valve capacity[4]. The error will be increased as the OP moves to the right in the domain of the surge controller. The OP flow, or total flow, which is the flow through the compressor, is now made up of the flow coming to the unit from the yard manifolding (net flow), and the flow being recycled through the recycle system (recycle flow). It is prudent to always leave the unused capacity of the recycle valve at no less than 50% so that the total flow to the unit can be significantly increased in little time if required. This approach also prevents the compressor discharge gas from becoming too hot. The objective of this method is to allow the total flow to be controlled by the recycle valve rather than by a much larger, slower, and unpredictable valve (or some other, possibly even more difficult to control, method).

It is now necessary to bring the OP to the left in the domain of the surge controller by reducing the net flow to the compressor. The means of accomplishing this will vary with the situation, but some common methods are listed below:

- Station power can be increased, if it is available, in parallel with the unit being tested. This action will tend to pull flow from the unit being tested and increase the head across the station.

- Station power can be decreased, if it is available, in series with the unit being tested. This action will increase the head across the unit being tested and reduce the flow through the station.

- The access the unit has, on the suction side, to the main lines can be cut off by manipulating crossover, side, and tie-over valves.

- The flow to the unit can be throttled by partial closure of the unit suction valve.

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[4]Note that this may or may not correspond to 40-50% of valve stroke, depending on the characteristic of the recycle valve.

Once the deviation has again been reduced to zero, it is an opportune time to check the tuning of the surge controller by switching the controller back to the automatic mode. The controller should be able to maintain the OP on the SCL in a steady fashion with smooth recycle valve action. Any erratic action should be dealt with at this point by adjusting the constants input into the PI portion of the controller response.

The controller is switched back to manual mode, and the control lines are further manipulated by moving the SCL to the left in the surge controller domain. The RVO line will follow the SCL, maintaining a constant relative position. As the SCL is moved left, the recycle valve is slowly closed to keep the OP in contact with the SCL, and thus keep a zero deviation. From this point on, the readout of the spectrum analyzer must be closely watched because the OP flow is being made less than what it was for the as-found normal operating condition, i.e. the surge tester is entering uncharted waters.

The surge control line is further moved to the left until it corresponds with the surge limit line. The recycle valve open line is positioned at approximately 97% of surge limit line flow. The optimum position for the recycle valve open line relative to the surge control/surge limit lines during the test can be determined by watching the fluctuation in the flow signal. The objective is to have the recycle valve open line close enough behind the surge control/surge limit lines to immediately throw the recycle valve open if the operating point unexpectedly moves to the left of the surge control/surge limit lines, but not to have it so close that it is activated by the low end spikes of the flow transmitter. The recycle valve open line will serve as a safety backup in case the surge tester loses control of the test. If the operating point makes contact with the recycle valve open line during the test, the controller will be switched back to automatic mode and the recycle valve will be immediately opened to 100%. If the surge backup line, positioned at 90% of surge limit line flow, is contacted by the operating point, the unit will be shut down and the fail-safe solenoids will be de-energized. Figure 5.2 shows the position of the control lines reflecting the changes outlined above. The control lines will remain in these relative positions for the remainder of the surge test.

The test is continued by alternating between moving the control lines back to the left in unison, and reducing the output to the recycle valve to keep the OP near the control lines. While this is being done, one must watch the spectrum analyzer for signs of impending compressor surge.

If the recycle valve reaches the fully closed position, it must be re-opened to 40-50% of capacity to further reduce the net flow to the unit until the OP again makes contact with SCL/SLL, i.e. the deviation is again zero. The control lines are moved further left, thus reducing the recycle flow and hence total flow to the unit. The process is repeated until the spectrum analyzer indicates that an unstable region of operation has been reached. This will mark the termination of the test.

The position of the SCL/SLL at the termination of the test represents the true position of the SLL in the domain of the surge controller for the aero assembly being tested. By the methods outlined in Appendix D, the SLL can be transferred to the dimensional compressor characteristic as the confirmed SLL. Following the test, the control lines are returned to the relative positions they assumed prior to the test for normal operation.

A schematic representation of the sequence of operations included in the surge test procedure has been presented in Figure 5.4 as a flowchart.

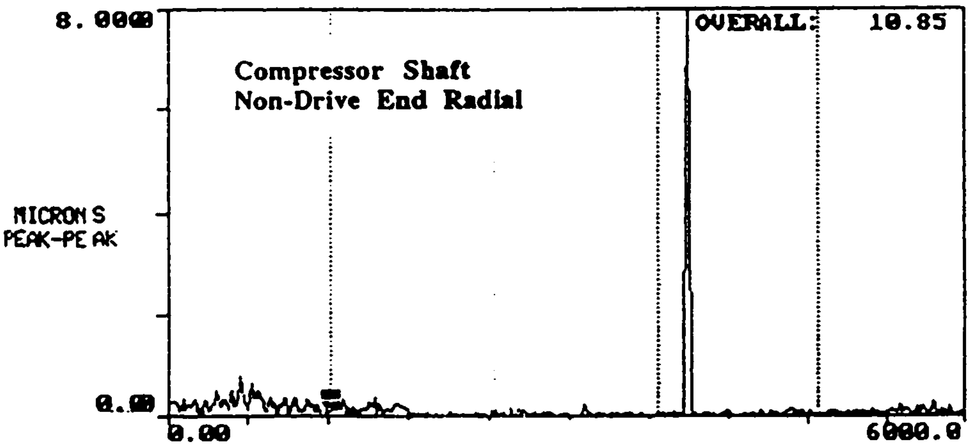
5.5 The Detection of Imminent Surge

An example of a spectrum analyzer display for a compressor under normal stable operation is shown in Figure 5.1. There is always a spike at 1X running speed (the synchronous frequency), but there is usually relatively minor vibrational activity at the sub-synchronous frequencies for normal operation as shown.

The operation of a compressor in the unstable region is known to create unusual vibrational activity at 1X running speed and lower frequencies, i.e. at sub-synchronous frequencies. The spike at 1X running speed will often reach amplitudes 3 to 5 times that for normal operation. At sub-synchronous frequencies the vibrational activity increases from negligible amplitudes for normal operation to amplitudes in the order of 50% to 75% or even 100% of that at the 1X frequencies. In addition to this, the vibrational activity will not be steady with time, but will come and go, almost like waves on the ocean, except in a less predictable manner. It is often the case that a predominant vibrational spike will appear at approximately 0.7X running speed under these circumstances.

When significant vibrational activity relative to that for normal operation as described above is observed, it means that the compressor is operating in the unstable region and is probably beginning to experience rotating stall. If the compressor is pushed much further into the unstable operating range, it is likely that the compressor will experience a surge as discussed in Chapter Three. Since it is never the intention to actually surge the compressor, the surge test is terminated at this point, and the setpoints existing in the surge controller for the SCL/SLL are considered to be indicative of the correct position for the SLL in the surge controller's domain for the aero assembly being tested.

Figures 5.1 and 5.3 are printouts from a spectrum analyzer taken during a surge test that was carried out by a colleague of the author. Figure 5.1 represents normal operation, and Figure 5.3 represents operation believed to be in the unstable region. Both Figures are generally consistent with the experience of the author, although it must be understood that no two compressors react in exactly the same manner to operation in the unstable region.



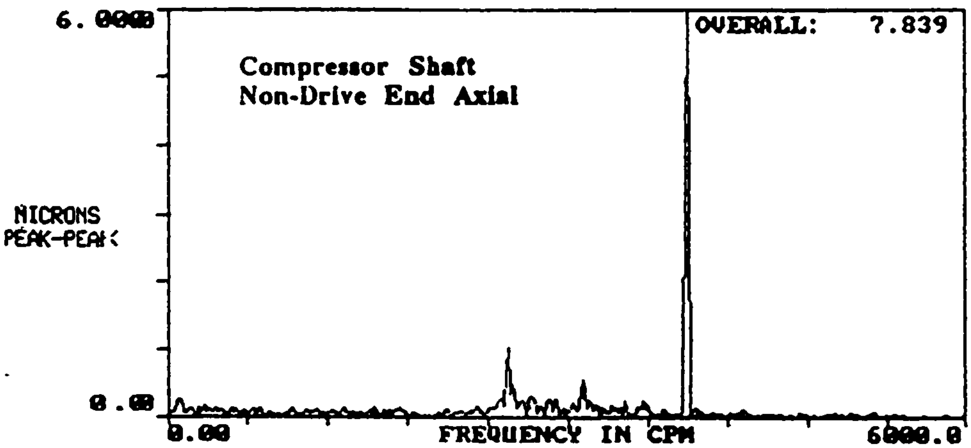


Figure 5.1 Spectrum Analyzer Displays: Normal Operation

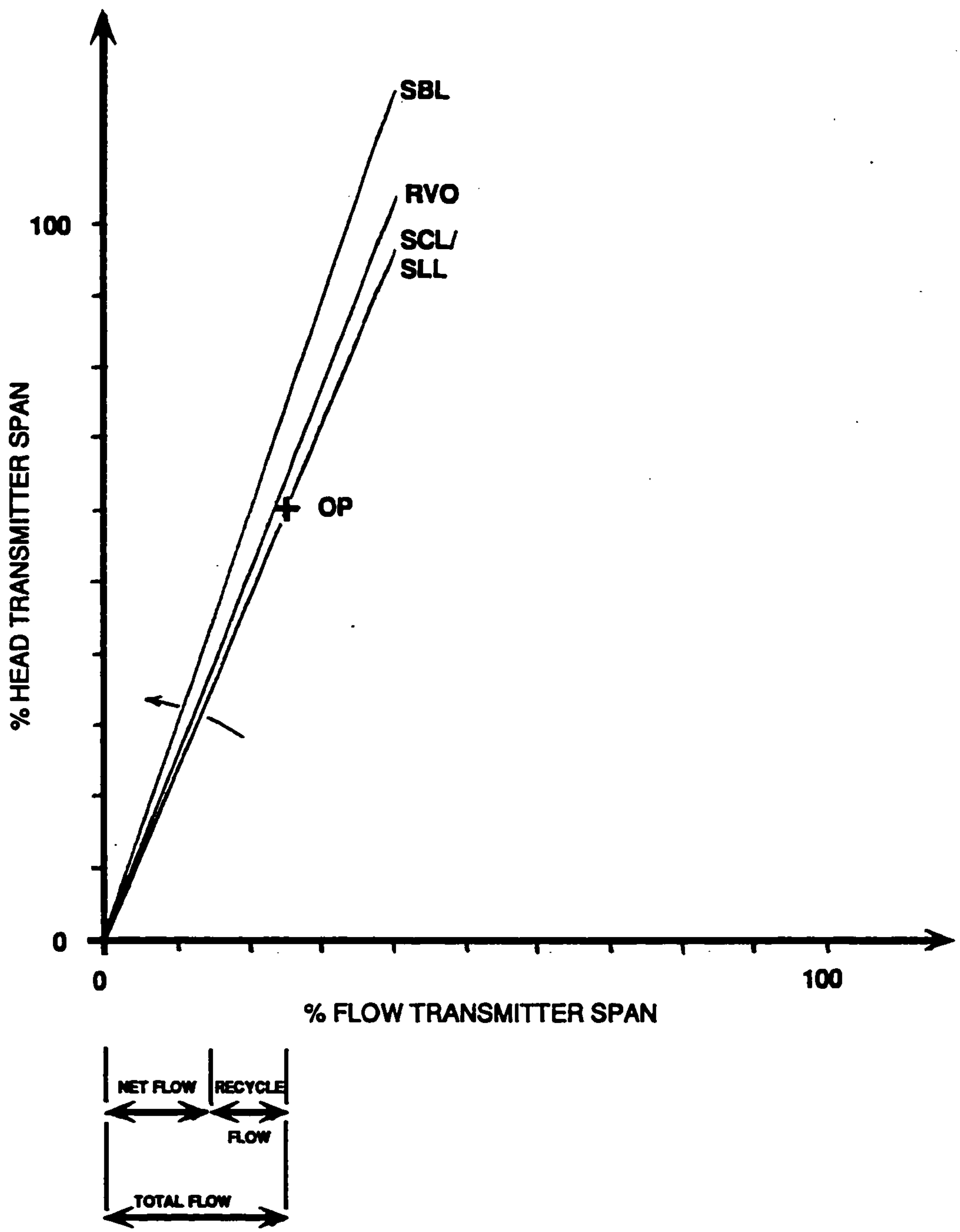


Figure 5.2 Control Lines Positioned for Surge Test

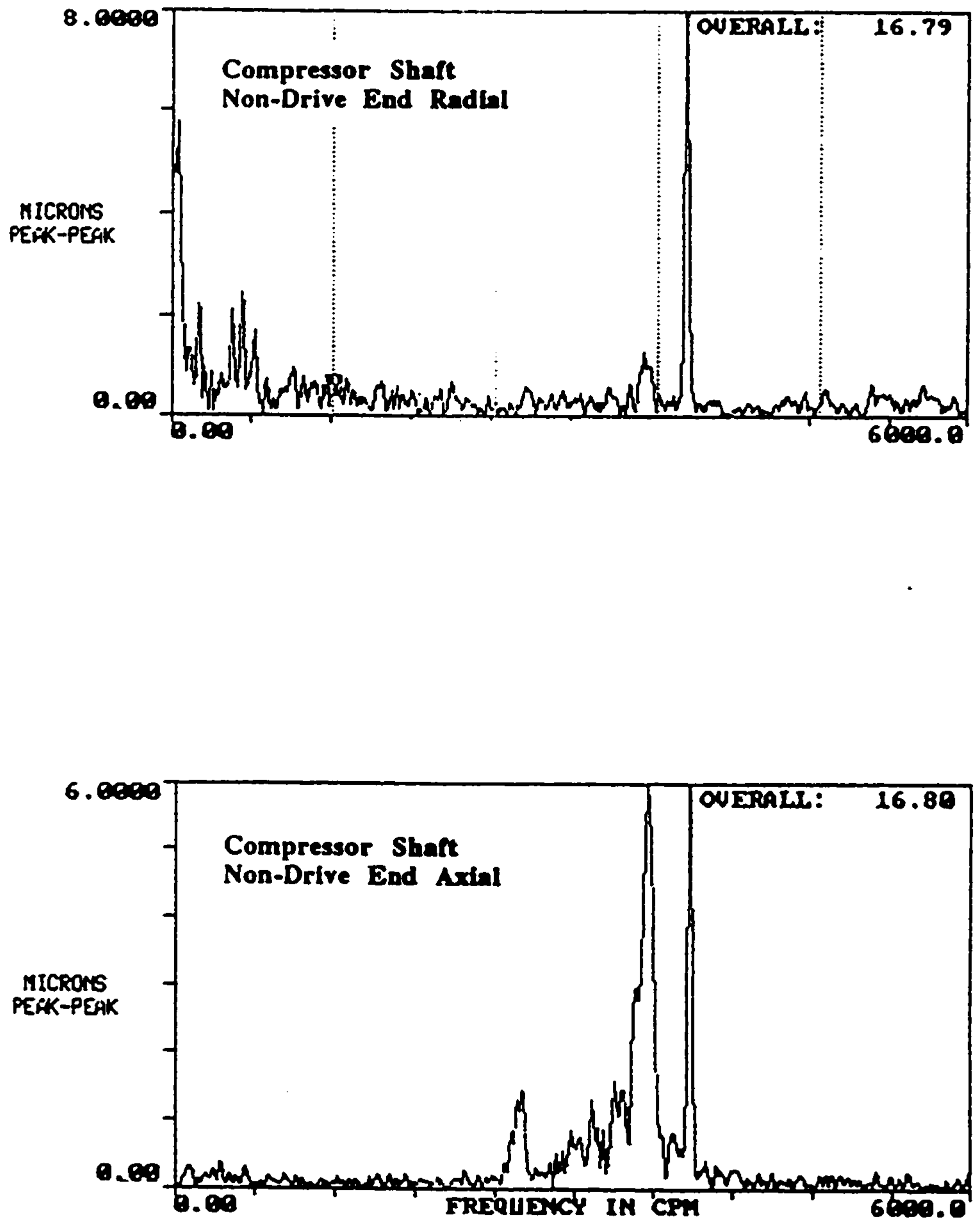


Figure 5.3 Spectrum Analyzer Displays: Unstable Operation

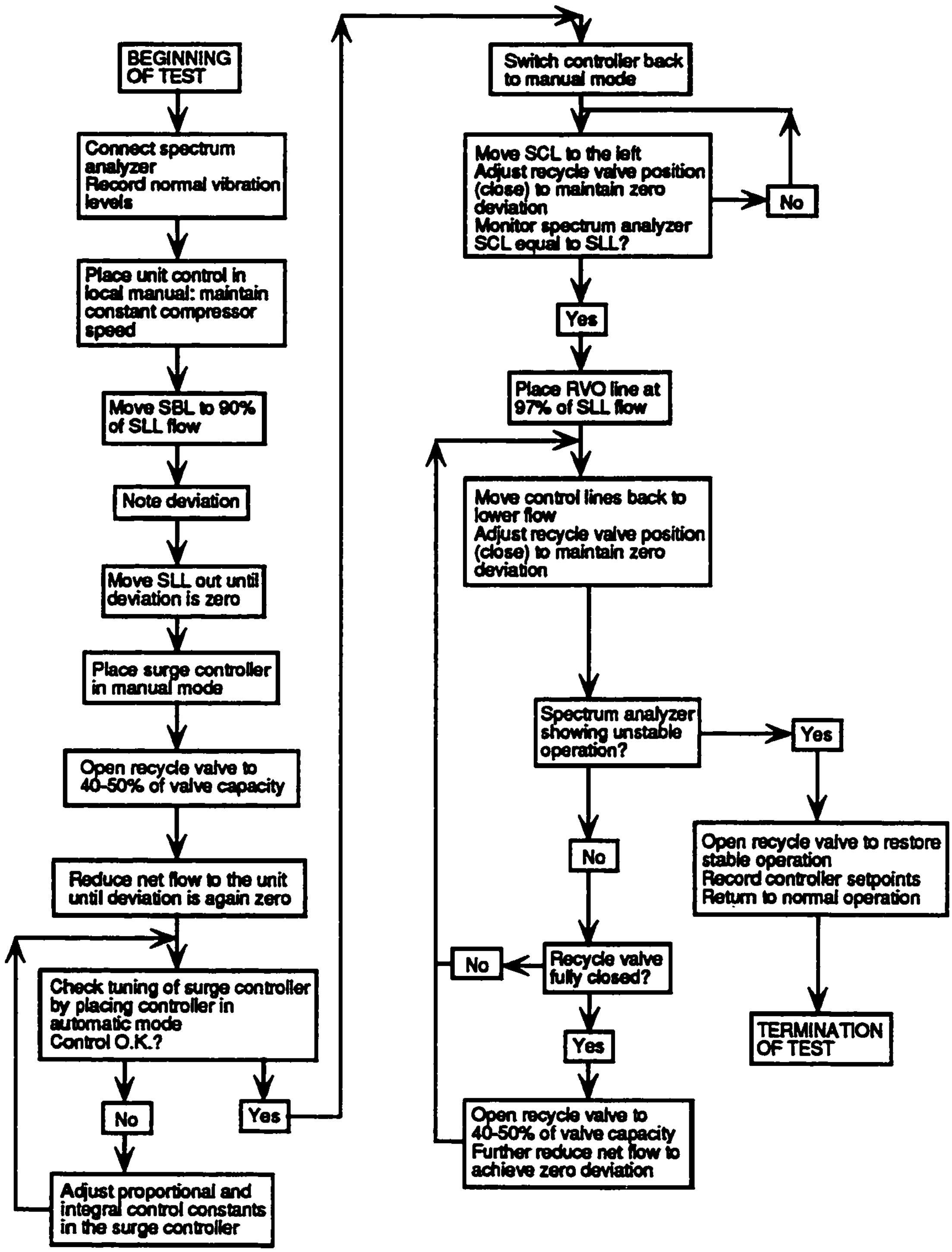


Figure 5.4 The Surge Test Procedure

CHAPTER SIX: THE RESULTS OF SURGE TESTING

A successful surge test will establish the true boundary of the unstable region. This information will allow the avoidance of future operating problems and will promote operator confidence. The true position of the SLL, relative to the previously predicted position, will result in one of three conclusions: the predicted SLL was placed too conservatively, the predicted SLL was accurately placed, the predicted SLL was not conservative enough. These three situations are each discussed in turn below.

6.1 Predicted SLL Positioned Overly Conservative

Under the previous surge control settings, the recycle valve would have been opened prematurely under operating conditions involving relatively low flow. The premature opening of the recycle valve leads to, at the very least, extremely inefficient operation of the gas compressor plant because valuable energy is lost through the dumping of discharge gas directly back to the suction header.

In the case of a hot recycle system (as discussed in Chapter Four), other serious problems are encountered while recycling. The temperature of the gas flowing through the compressor and the recycle valve can become relatively high. This can result it the unit shutting down on high temperature, and possibly being off line for several hours while the compressor cools down. If the high temperature protection system malfunctions for some reason, significant damage can be incurred by the compressor and the recycle valve if hot gas is recycled for a prolonged period of time.

The unnecessary opening of recycle valves is therefore something to be avoided. Although it is frequently difficult to quantify the economic advantages of discovering some extra operating range on a pipeline compressor, it is clear that a more desirable situation has been attained.

Figures 6.1, 6.2, and 6.3 show the results of six surge tests undertaken by the author, using the newly developed test procedure, where extra operating range was gained as a result of the test. The long solid line bounding the crosshatched area on the right represents the position of the SLL as predicted by the manufacturer. The heavier solid line bounding the crosshatched area on the left represents the true position of the SLL as confirmed through a surge test. The crosshatched area represents the additional operating range gained through the surge test.

6.2 Predicted SLL Position Accurate

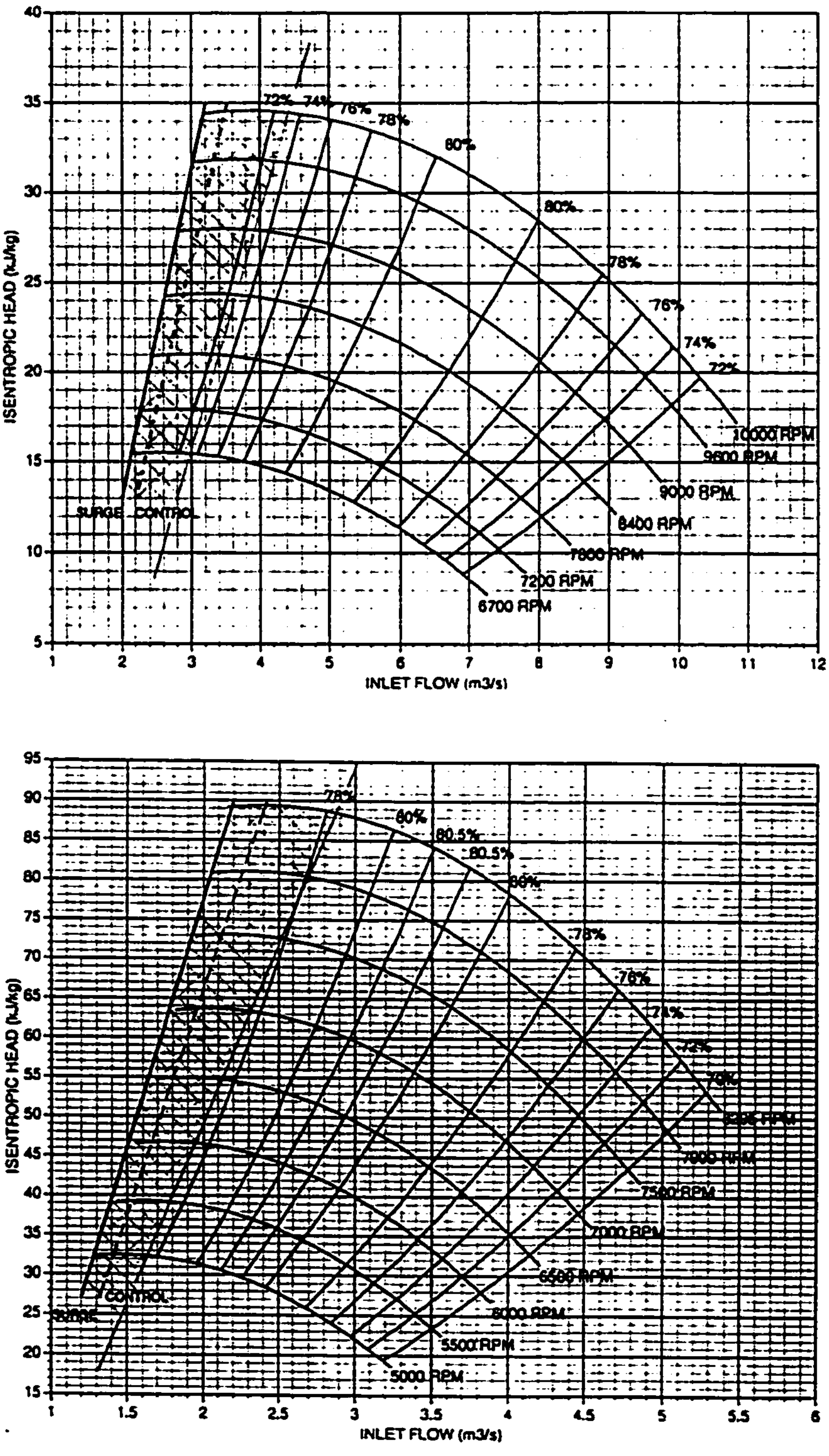
In this situation it can be said that the operators of the compressor can now rest assured that the surge control system was, and is, setup appropriately, and the unit is being adequately protected while also being allowed to operate as efficiently as possible.

6.3 Predicted SLL Position Not Conservative Enough

A situation like this indicates that the setup of the surge control system prior to the surge test would not have been able to protect the compressor from entering an area of unstable operation. This is why the surge protection is moved out to the OP at the beginning of a surge test as discussed in Section 5.4. If this practice is not followed, a compressor can be surged before the surge protection is even reached.

A surge test with this result should cause the owner of the compressor to feel fortunate it wasn't damaged before the test could be undertaken. Experience has shown, however, that this situation is relatively unlikely. Manufacturers are usually extremely careful to recommend overly conservative positions for SLLs rather that the alternative, i.e. they are more interested in avoiding liability than in saving the owner money.

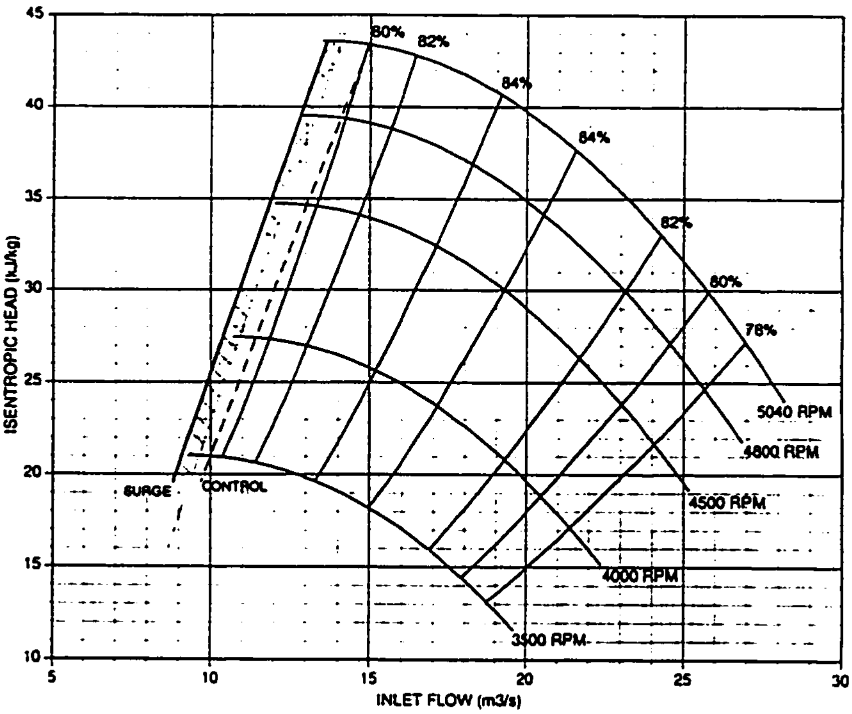
In any event, although some operating range might seem to have been lost through the test, the lost operating range resided in the unstable region, and is best avoided anyway. Following the test, the compressor will be allowed to operate only in the safe, stable region.



(a)

(b)

Figure 6.1 Results of Surge Testing: (a) January 1994, FP-301; (b) March 1994, FP-295



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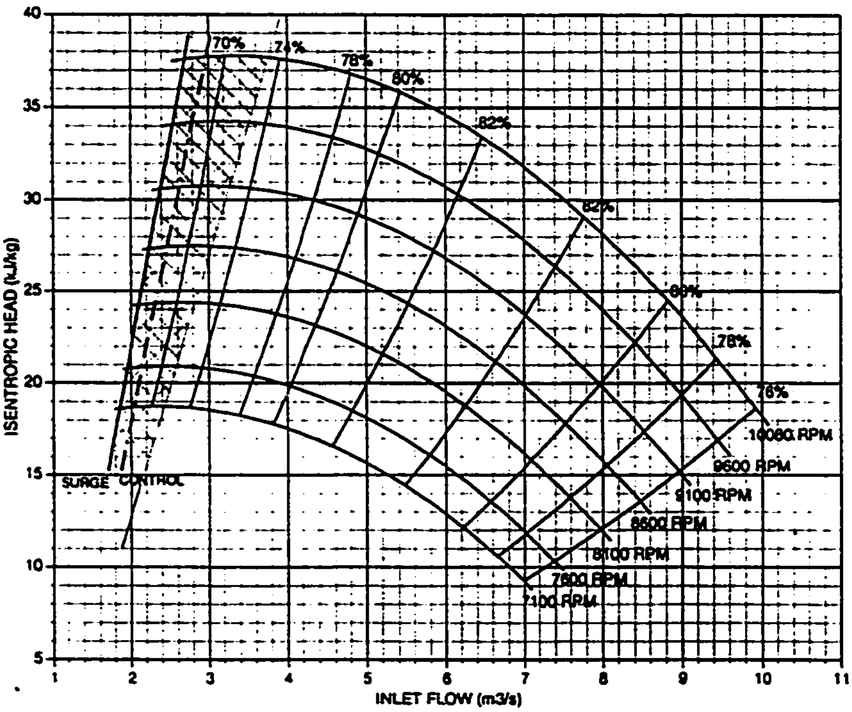
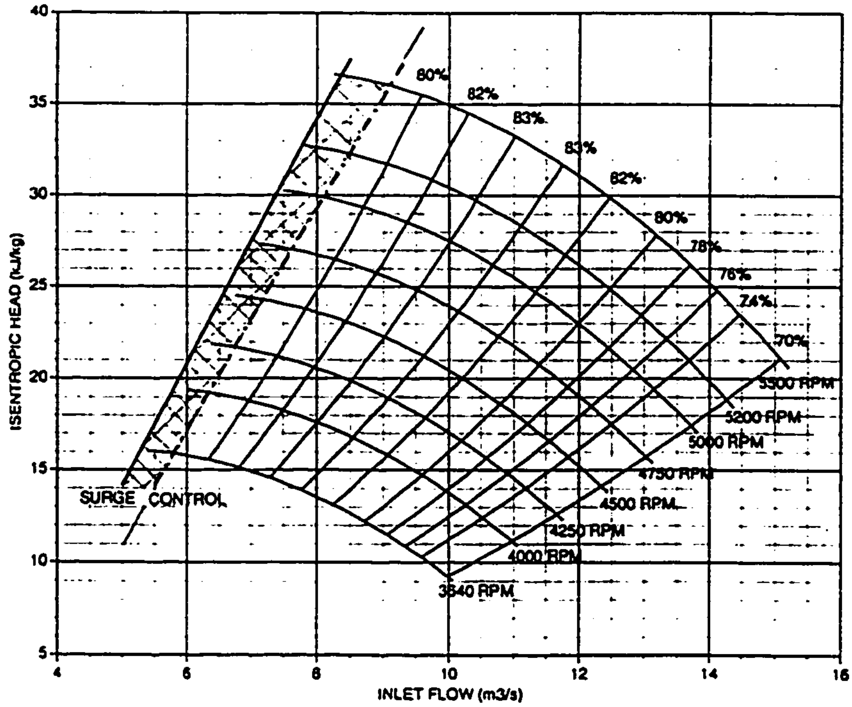
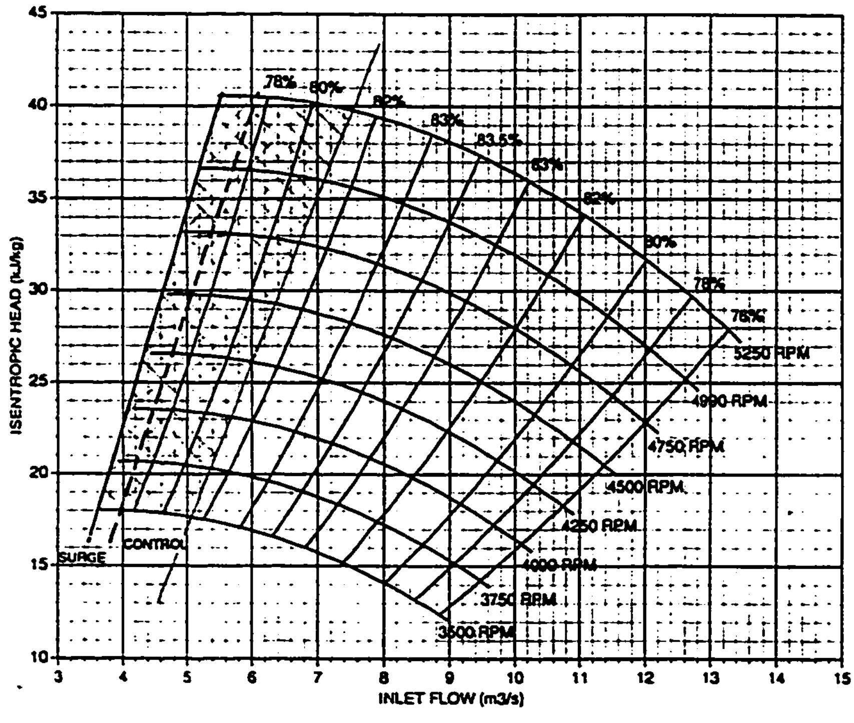


(b)

Figure 6.2 Results of Surge Testing: (a) June 1994, FP-326; (b) July 1994, FP-300



(a)



(b)

Figure 6.3 Results of Surge Testing: (a) August 1994, FP-267; (b) March 1995, FP-361

CHAPTER SEVEN: CONCLUSIONS AND SUGGESTIONS FOR FURTHER WORK

7.1 Conclusions

The conclusions of this research are as follows:

- The performance of a centrifugal compressor, and hence the shape of its characteristic curve, is influenced by geometry changes involving IGVs and diffuser vanes. This fact confirms the need for surge testing following all such changes in geometry.

- This work has resulted in the development of an acceptable procedure for the surge testing of centrifugal gas pipeline compressors. It has been proven that with this procedure surge testing can be undertaken effectively, without causing damage, while the compressor is on line.

- An increase in vibrational amplitudes at approximately 0.7X rotor rotational speed will always precede a surge cycle. This increase in vibrational activity is caused by rotating stall at the leading edges of the impeller vanes, and is a reliable indication that a surge event is imminent.

7.2 Suggestions for Further Work

Further work in the area of surge testing could include the trending of the more prominent parameters involved, including suction pressure, discharge pressure, compressor flow rate, speed, and coupling power. The results of such work, along with the equipment required to trend the parameters during a surge test, could assist a surge tester in detecting the approach of unstable operation, thus increasing his chances of carrying off the test without damage to the machine.

Further work in the area of surge control could include the fine tuning of the PI response. Adjusting the constants in the controller for more effective response could be based on the trending of the error in response to forced controller setpoint changes.

Further work in the area of performance testing could involve testing on similar compressors with various IGV positions to attempt to resolve the discrepancy between theoretical expectations and actual field test results as noted in Subsection 2.2.1.

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APPENDIX A

THE AERO ASSEMBLY

A centrifugal gas pipeline compressor is a relatively large piece of rotating equipment. When a gas compressor plant is being installed, the compressor will be delivered to the site as a complete unit. The weight of the compressor will be in the order of 31000 kg. The compressor will be placed on steel sole plates which are cast into the reinforced concrete foundation on which the compression package will sit. The compressor will be aligned cold to the power turbine such that there will be acceptable alignment between the two pieces of equipment at normal operating temperatures. The compressor case is made of thick cast steel. Incorporated into the compressor case are the compressor feet and the inlet and outlet flanges. The compressor feet are held by anchor bolts to the sole plates below, and the flanges are bolted to the suction and discharge headers. The compressor case is thus a permanent fixture in the gas compressor plant and will, almost without exception, reside in the plant until the plant is retired.

There are several removable components contained within the compressor case, including the inlet guide nozzle, the impeller, the diffuser, and the volute (see Figure 2.5). These components are instrumental in the transfer of energy that takes place within the compressor. These components are referred to as the aero assembly. The discharge nozzle is typically incorporated into the compressor case, and is therefore not considered to be part of the aero assembly. The compressor shaft may or may not remain with an aero assembly.

The design and manufacture of the components making up the aero assembly will determine the expected performance of the compressor. In fact, the aero assembly is designed and manufactured to achieve a desired performance. The components making up the aero assembly are designed and matched to work together as a unit, and are therefore usually not interchangeable with those of another aero assembly. An aero assembly will always be given its own identification number, and each component in the aero assembly will bear that number. The components of the aero assembly are always shipped and stored together when not in use.

The performance of any given gas compressor can be changed (within reason) by changing the aero assembly[5]. The performance of each aero assembly will be recorded on generalized performance curves and on a compressor characteristic map as discussed in Chapter Two. The operators of a natural gas pipeline (transmission) system will own a fleet of aero assemblies, with some in service, some in storage, and some undergoing repairs.

[5]Overhung compressors (compressors with both journal bearings on the same side of the impeller) are typically limited to only one impeller, so changing the aero assembly will result in relatively subtle differences in performance. Beam style compressors (compressors with a journal bearing on each side of the impellers) are capable of accommodating more than one impeller, and are therefore easier to modify to meet a broader range of performance requirements.

APPENDIX B

DESCRIPTION OF A TYPICAL MODERN GAS PIPELINE COMPRESSION PACKAGE

The purpose of this Appendix is to help clarify some of the terminology used throughout the report.

A modern day natural gas pipeline compression package consists of a driver, and the driven equipment. Drivers are required in many industries and in many applications because they provide shaft power. The drivers that are being considered here are called gas turbines. The gas turbine driver consists of a gas generator and a power turbine. The gas generators will either be a land/marine model of an aircraft engine (hence the term aircraft derivative), or a light duty industrial equivalent. The exhaust gas power produced by the gas generator is converted to shaft power by a power turbine. There is no mechanical coupling between the gas generator and the power turbine, only an aerodynamic coupling exists between these two components.

The driven equipment will vary with industry and application. Electric generators, pumps, and compressors are common. The driven equipment that is the topic of discussion in this report is the gas pipeline centrifugal compressor. The compressor will be mechanically coupled to the gas turbine driver (specifically the power turbine output shaft). The compressor is tied-in to the natural gas pipeline and the power transmitted from the gas turbine to the compressor is used to compress the gas in the pipeline.

APPENDIX C

THE FIELD PERFORMANCE TEST

The measurement of compressor performance in the field is done for two reasons: to verify the claims of a manufacturer in the case of a new aero assembly; to investigate suspected internal damage to the compressor.

The primary objective of a field performance test is to determine the relationship between both isentropic head and isentropic efficiency versus compressor flow rate. It is necessary to measure the inlet (station 0) and outlet (station 7) flow pressure and temperature. Figure 4.1 shows the approximate locations, relative to the compressor, where the measurements are taken. The procedure is complicated somewhat by the relatively high process gas pressure (up to 6894 kPag) and the hazardous environment created by the presence of the natural gas.

Up to this point it has been implied (Equations 2.1 through 2.10) that compressor performance is discussed in terms of total, or stagnation, conditions. It is not, however, stagnation conditions that are measured in the field. No attempt is made to measure stagnation pressure. The pressure measurement is taken through a 3mm diameter hole in the pipe wall, resulting in a static pressure measurement. The dynamic component of stagnation pressure is, however, insignificant with mean process gas velocity being approximately 10 to 30 m/s in the suction and discharge headers where the measurements are taken.

The measurement of temperature is accomplished by using a high pressure thermowell with a temperature measuring probe inside. The accuracy of this method is enhanced by the following details of manufacture, construction, and application:

- Both the probe and the thermowell are made of stainless steel, which has a relatively low coefficient of thermal conductivity.

- The cross-sectional area of the probe and even the thermowell is relatively small.
- A film of oil is used between the probe and the thermowell bore to reduce contact resistance.
- The outer wall of the pipe is covered with a thick layer of fibreglass insulation cladded in aluminum, which must virtually eliminate temperature gradients in the gas flow.
- The temperature measuring element within the probe is located right at the tip.
- The gas flow is turbulent.

The temperature measured in this manner would be neither total nor static, but would be a value somewhere in between. This is, however, a moot point since the insignificance of the dynamic energy component has already been pointed out.

With the actual inlet and outlet conditions being known, the actual head can be determined. By assuming constant entropy from inlet to outlet, along with the known outlet pressure, the isentropic head can be determined. Once the actual and isentropic head is known, the isentropic efficiency can be calculated.

Since the performance of a compressor is always plotted with rate of flow as the abscissa, the flow rate must also be measured. The most common method is an elbow meter. A differential pressure gauge is used to measure the difference in static pressure between the outside and the inside of the bend. Centrifugal effects distort the velocity profile as the fluid passes through the elbow, and also cause the static pressure to be higher at the outside of the bend. The pressure differential is proportional to the square of the rate of flow through the elbow, and if the flow pressure and temperature is known, then the mass flow rate can be determined (see Appendix D for a discussion on elbow meters). The elbows are usually located close enough to the flanges of the compressor to allow the use of inlet or outlet compressor flow conditions at the elbow.

Either a suction or discharge elbow can be used as long as the correct corresponding flow conditions are also used. The ideal case is to have an elbow meter on both sides of the compressor, allowing the tester to do a mass flow balance across the compressor to convince himself of the accuracy of his measurements.

The performance tester will gain enough information to determine the impeller power and the isentropic impeller power. In most cases this is all of the information that is available, or for that matter, required. There is, however, one more piece of information that is typically available on modern pipeline compression packages.

Since the late 1980s, it has become common to have a torque meter installed between the power turbine and compressor coupling hubs. The output of this device is essentially the coupling power. This added piece of information provides confirmation of the measurements taken during a pipeline compressor performance test, and also allows a quick and convenient indication of power turbine output power if it is desired to evaluate the performance of the gas turbine without taking pipeline compressor measurements.

APPENDIX D

THE COMPRESSOR CHARACTERISTIC - SURGE CONTROLLER DOMAIN LIAISON

This Appendix deals with the general approach that must be used in to allow the transformation of information from the compressor characteristic (Figure 4.2a) to the domain of the surge controller (Figure 4.2b), and vice versa for the purposes of surge controller setup or the revision of compressor characteristics based on surge testing.

Surge Controller Setup

The initial setup of a surge controller intended to protect a new aero assembly is always based on performance information supplied by the vendor. The information is typically in the form of a compressor characteristic map as shown in Figure 2.2. The first step to be taken by the owner of the new aero assembly would likely be to convert the information supplied by the vendor into the more familiar plot of isentropic head versus inlet flow (Figure 2.4), although this step is not essential from the point of view of surge controller setup. In any event, the task is to arrive at expected differential pressure signals for head and flow corresponding to a performance point on the SLL representing maximum absorbed compressor power (point M on Figure 4.2a). Differential pressure signals are all that will be available to the surge controller to allow it to monitor the performance of the compressor, as discussed in Chapter Four. Once the expected signals are known, head and flow transmitters are selected to handle the entire operating range of the compressor. The expected head and flow signals, in percent of transmitter span, are input into the surge controller. A straight line drawn through this point and the origin will define the SLL in the domain of the surge controller, as shown in Figure 4.2b.

Revision of Compressor Characteristics

In coming at the problem the other way, a surge test on a pipeline compressor will result in differential pressure signals corresponding to the performance point at which unstable operation was encountered. Based on this information, it is necessary to define a new SLL on the plot of isentropic head versus inlet flow (Figure 2.4).

Following is an outline of the mathematical relationship between the differential pressure signals and the corresponding performance on the compressor characteristic for both head and flow. These relationships provide the links between the compressor characteristic (Figure 4.2a) and the domain of the surge controller (Figure 4.2b). In both cases, one must know the properties (i.e. Z_{ave}, R, k, M_{NG}), and also either know, or be prepared to predict, the pressure and temperature of the gas being handled.

Head

The isentropic head from the compressor characteristic is related to the differential pressure rise across the compressor by rearranging Equation 2.10 as shown below.

$$\Delta p_c = p_{0o} \cdot \left\{ \left[\frac{H_{dISEN} \cdot 10^3}{Z_{ave} \cdot R \cdot T_{0o}} \cdot \frac{k-1}{k} + 1 \right]^{\left(\frac{k}{k-1}\right)} - 1 \right\} \qquad \text{D1}$$

Equation D1 allows one to move one way or the other between the pressure rise across the compressor, Δp_c, and the isentropic head, H_{dISEN}.

Flow

The actual inlet flow from the compressor characteristic is related to the differential pressure signal produced by the suction elbow meter by the following relationship established for a elbow type centrifugal flow meter (ASME Research Report on Fluid Meters, 1971).

$$Q = K_{ELB} \cdot \sqrt{\frac{\Delta p_{ELB} \cdot Z \cdot R \cdot T_{0s}}{p_{0s} \cdot M_{NG}}} \qquad \text{D2}$$

where,

K_{ELB} = elbow flow constant (m^2)
Δp_{ELB} = differential pressure measured at the elbow meter (N/m^2)
T_{0s} = fluid temperature at the suction elbow (K)
p_{0s} = static pressure of fluid at the suction elbow (N/m^2)
M_{NG} = molecular mass, natural gas (kg/kmol)

The above equation is based on the following assumptions:

- The elbow is in a horizontal plane.
- A uniform distribution of velocity exists over the cross section of the elbow.
- There is no loss of either pressure or velocity from inlet face to outlet face.
- There is no effect of fluid viscosity.
- A uniform distribution of pressure exists over the inside and outside walls of the elbow.
- There is a steady state of flow with respect to time.

K_{ELB} is a flow constant that incorporates the flow coefficient of the elbow meter as well as the geometric details of the elbow, including the elbow radius, the elbow diameter, and the cross-sectional flow area. Chapter Four specifies that it is the eye meter within the compressor that is used for the surge control flow signal. There are several reasons for this:

- If the elbow meter is located outside the recycle loop, it will not measure the total flow through the compressor.
- The eye meter produces a stronger differential pressure signal.
- The eye meter is physically closer to the compressor.

The flow constant for the elbow meter is relatively easy to determine because of the geometric simplicity of the device. An eye flow meter can be geometrically simple in an axial-entry compressor, but this is not the case with a side-entry compressor where the eye flow meter can consist of a reducing flange, the case inlet nozzle, a collector, and finally the inlet guide nozzle. The accepted method for determining the flow coefficient of the eye meter is to record several differential pressure readings simultaneously from the suction elbow meter and the eye meter for various compressor flow rates. When the two sets of readings are plotted against each other, they should form a straight line running through the origin. Since a surge test is almost always done in conjunction with a complete performance test, several readings from the flow meters will be available. The flow coefficient for the eye meter can then be determined by using the slope of this linear relationship as follows:

$$K_{EYE} = K_{ELB} \cdot \sqrt{\frac{\Delta p_{ELB}}{\Delta p_{EYE}}} \qquad \text{D3}$$

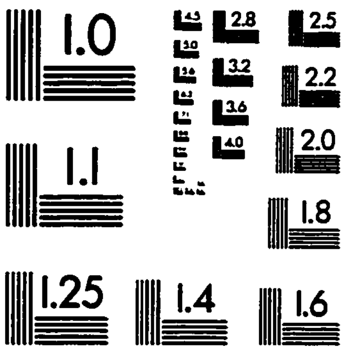
where,

K_{EYE} = eye flow constant (m^2)
Δp_{EYE} = differential pressure measured at the eye meter (N/m^2)

The substitution of Equation D3 into Equation D2 allows one to move one way or the other between the differential pressure signal produced by the eye flow meter, Δp_{EYE}, and the actual volumetric flow rate through the compressor, Q.

Since a uniform velocity profile is one of the fundamental assumptions on which the development of the theoretical equations for differential flow meters is based, it is acknowledged that the eye flow meter would not be a relatively accurate method to be used for the measurement of compressor flow rate. It is not, however, intended for this purpose: the elbow meter is. The intended use of the eye meter is to provide an indication of flow rate relative to the SCL, and for this purpose the accuracy of the method is acceptable.

TEST TARGET (QA-3)



150mm

6″

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