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Integrating Dynamic Machinery Performance with Component Condition to Optimize Reliability

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By

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Objective: To instruct how Condition Monitoring of Dynamic Machinery (Pumps and Compressors) can be optimized when integrated with trending performance.

This Tutorial will consist of practically discussing the parameters needed to accurately trend performance and all major components of Dynamic Machinery. Topics included will be the following:

- Effects of the Process on Dynamic Pumps and Compressors
- Performance Monitoring
- Component Condition Monitoring
    - Rotor
    - Journal Bearings
    - Thrust Bearings
    - Seals
    - Auxiliary Systems

The course material presented will approximately total 70 minutes, leaving 20 minutes for question during and after the presentation.

THE EFFECT OF THE PROCESS ON MACHINERY RELIABILITY & COMPONENT CONDITION MONITORING

- **Centrifugal Pumps And Drivers**
- **Component Condition Monitoring**
  - **Principles**
  - **Component Monitoring Guidelines**
    - **Pump & Compressor Rotors**
    - **Bearings**
    - **Pump & Compressor Seals**
    - **Auxiliary Systems**

INTRODUCTION

The effect of the process on machinery reliability is often neglected as a root cause of machinery failure. It is a fact that process condition changes can cause damage and/or failure to every major machinery component. For this discussion, the most common type of Driven Equipment — Pumps will be used.

There are two (2) major classifications of pumps, positive displacement and kinetic, centrifugal types being the most common. A positive displacement pump is shown in Figure 1. A centrifugal pump is shown in Figure 2.

POSITIVE DISPLACEMENT PLUNGER PUMP

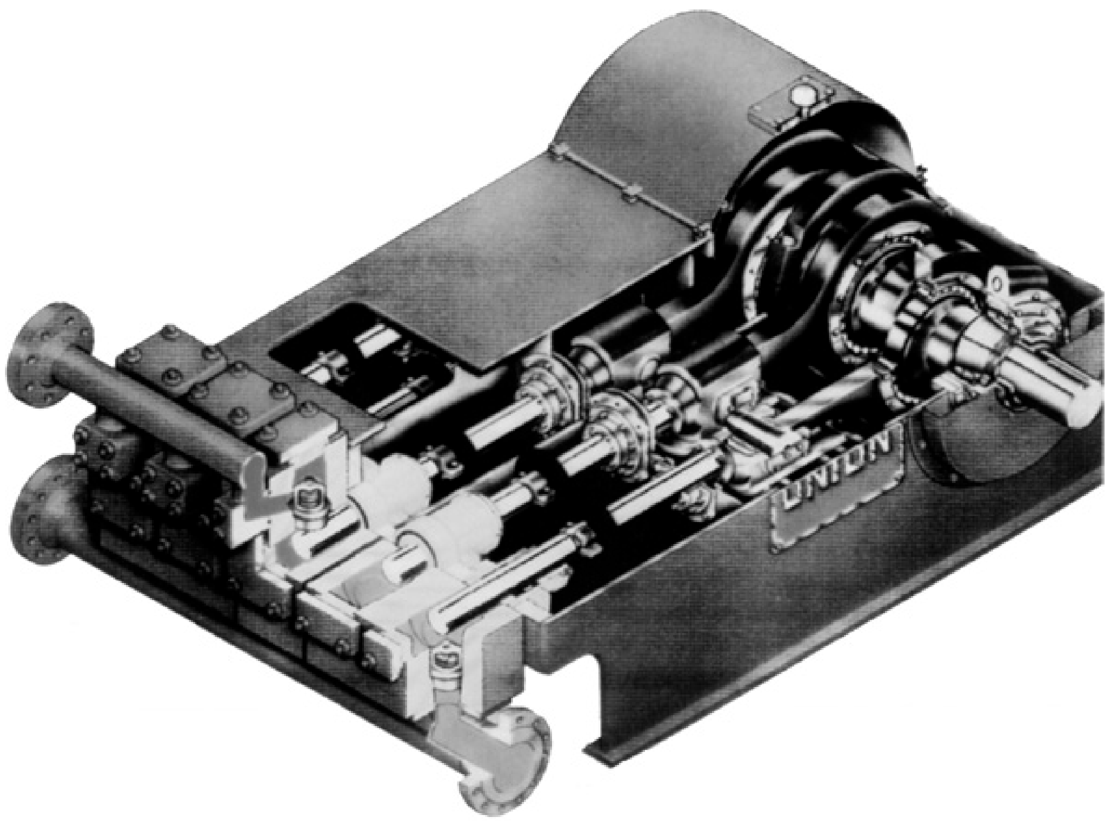


FIGURE 1

CENTRIFUGAL PUMP

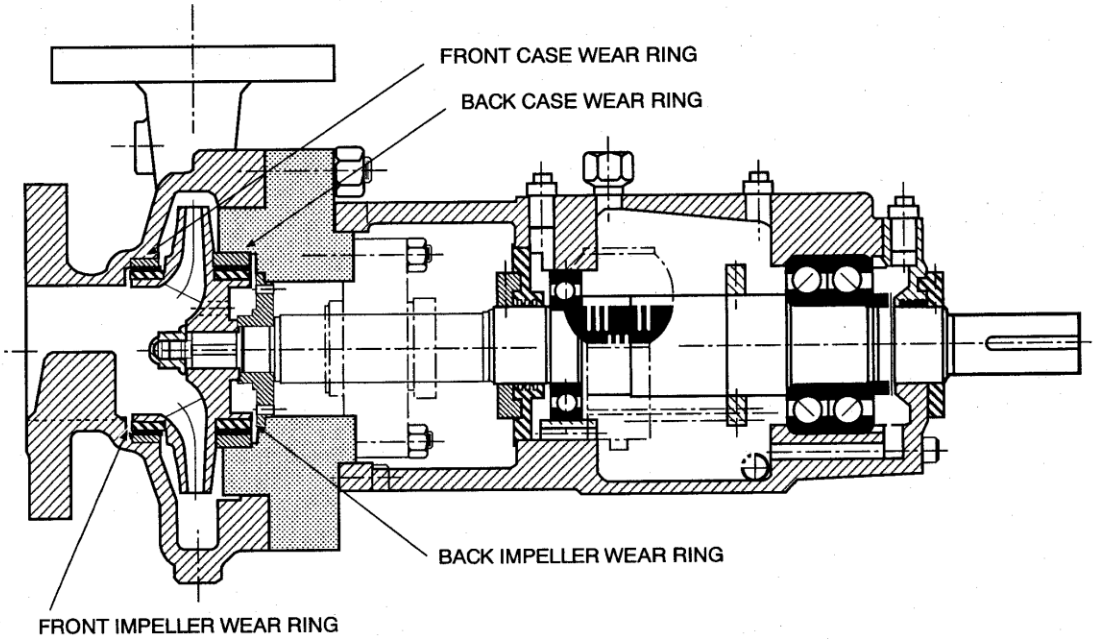


FIGURE 2

It is most important to remember that all driven equipment (pumps, compressors, fans, etc.) react to the process system requirements. They do only what the process requires. This fact is noted in Figure 3 for pumps.

PUMP PERFORMANCE

- Pumps produce the pressure required by the process
- The flow rate for the required pressure is dependent on the pump's characteristics

FIGURE 3

Centrifugal (Kinetic) Pumps and Their Drivers

Centrifugal pumps increase the pressure of the liquid by using rotating blades to increase the velocity of a liquid and then reduce the velocity of the liquid in the volute. Refer again to Figure 2.

A good analogy to this procedure is a football (soccer) game. When the ball (liquid molecule) is kicked, the leg (vane) increases its velocity. When the goal tender (volute), hopefully, catches the ball, its velocity is significantly reduced and the pressure in the ball (molecule) is increased. If an instant replay "freeze shot" picture is taken of the ball at this instant, the volume of the ball is reduced and the pressure is increased.

The characteristics of any centrifugal pump then are significantly different from positive displacement pumps and are noted in Figure 4.

CENTRIFUGAL PUMP CHARACERISTICS

- Variable flow
- Fixed Differential Pressure produced <u>for a specific flow*</u>
- Does not require a pressure limiting device
- Flow varies with differential pressure (P_1-P_2) and/or specific gravity

* assuring specific gravity is constant

FIGURE 4

Refer again to Figure 3 and note that all pumps react to the process requirements.

Based on the characteristics of centrifugal pumps noted in Figure 4, the flow rate of all types of centrifugal pumps is affected by the Process System. This fact is shown in Figure 5.

A CENTRIFUGAL PUMP IN A PROCESS SYSTEM

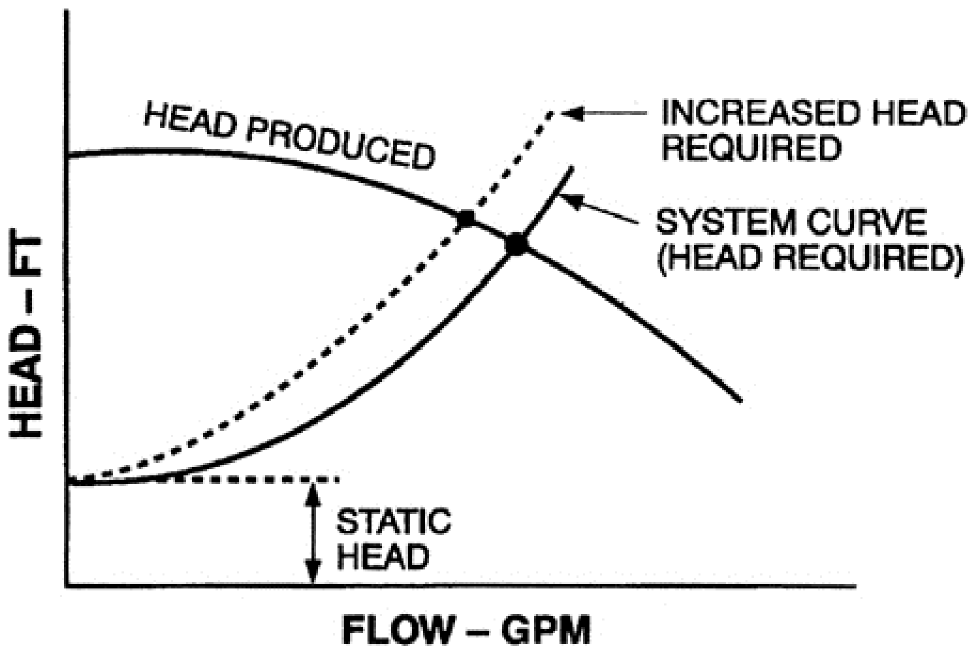


FIGURE 5

Therefore, the flow rate of any centrifugal pump is affected by the process system.

A typical process system with a centrifugal pump installed is shown in Figure 6.

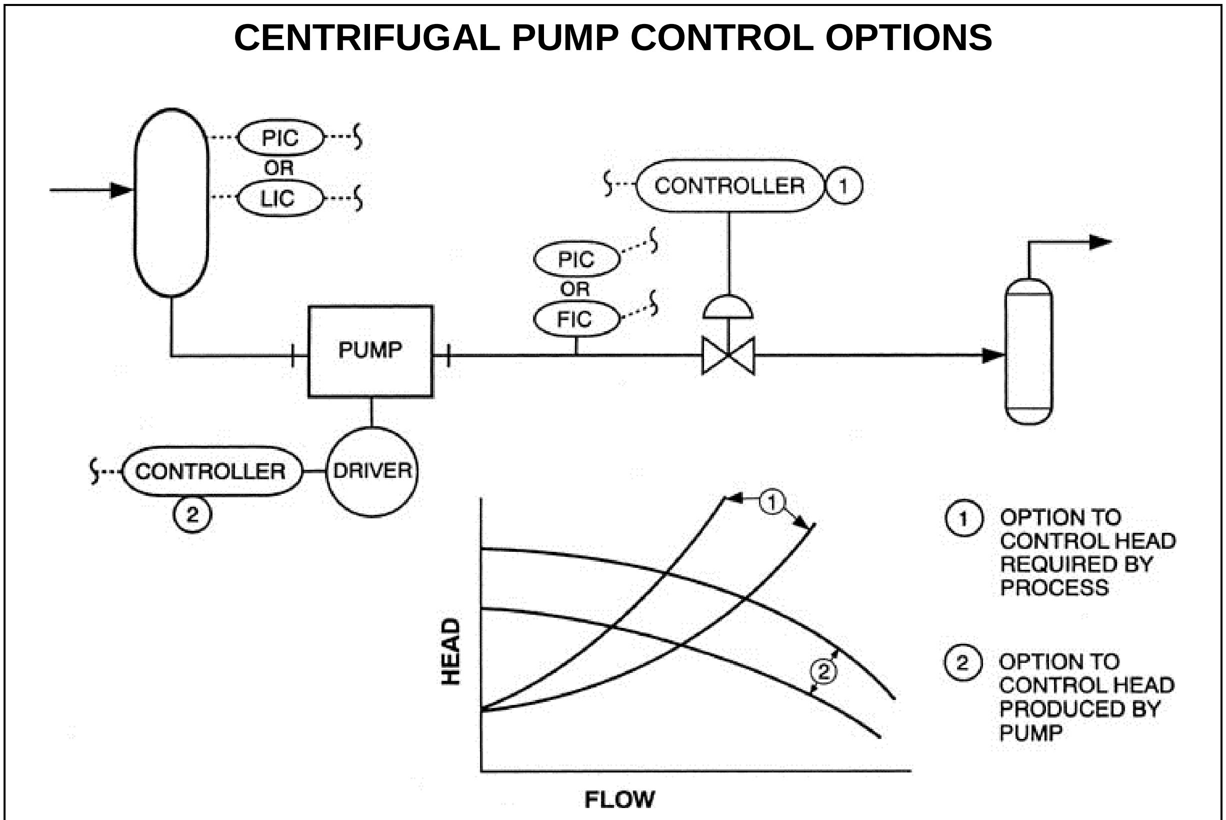


FIGURE 6

The differential pressure required (proportional to head) by any process system is the result of the pressure & liquid level in the suction and discharge vessel and the system resistance (pressure drop) in the suction and discharge piping.

Therefore, the differential pressure required by the process can be changed by adjusting a control valve in the discharge line. Any of the following process variables (P.V.) shown in Figure 6, can be controlled:

- Level
- Pressure
- Flow

As shown in Figure 5, changing the head required by the process (differential pressure divided by specific gravity), will change the flow rate of any centrifugal pump!

Refer to Figure 7 and it can be observed that all types of mechanical failures can occur based on <u>where the pump is operating based on the process requirements.</u>

CENTRIFUGAL PUMP COMPONENT DAMAGE AND CAUSES AS A FUNCTION OF OPERATING POINT

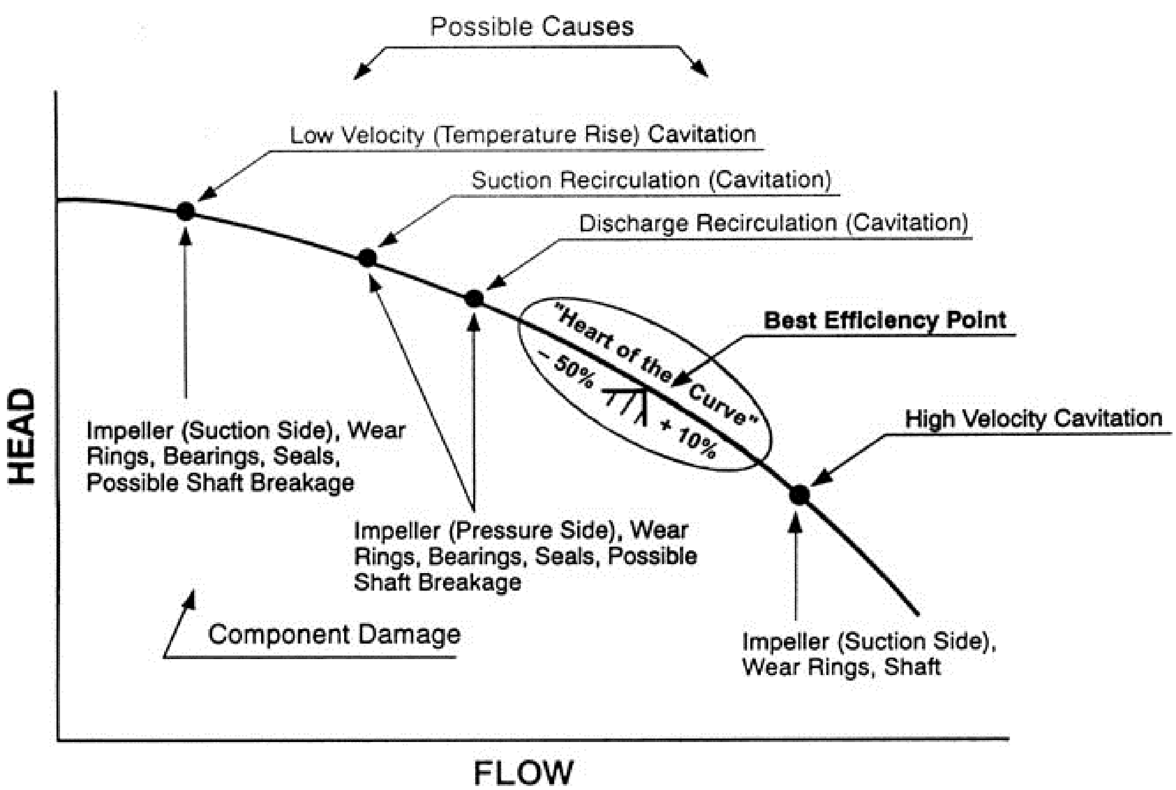


FIGURE 7

Since greater than 95% of the pumps used in this refinery are centrifugal, their operating flow will be affected by the process. Please refer to Figure 8 which shows centrifugal pump reliability and flow rate is affected by process system changes.

CENTRIFUGAL PUMP RELIABILITY

- Is affected by process system changes (system resistance and S.G.)
- It is not affected by the operators!
- Increased differential pressure (P_2 — P_1) means reduced flow rate
- Decreased differential pressure (P_2 — P_1) means increased flow rate

FIGURE 8

At this point it should be easy to see how we can condition monitor the centrifugal pump operating point. Refer to Figure 9.

CENTRIFUGAL PUMP PRACTICAL CONDITION MONITORING

- Monitor flow and check with reliability unit (RERU) for Significant changes
- Flow can also be monitored by:
    - Control valve position
    - Motor amps
    - Steam turbine valve position

FIGURE 9

Driver reliability (motors, steam turbine and diesel engines) can also be affected by the process when centrifugal driven equipment (pumps, compressor and fans) are used.

Refer to Figure 10 and observe a typical centrifugal pump curve.

A TYPICAL CENTRIFUGAL PUMP PERFORMANCE CURVE

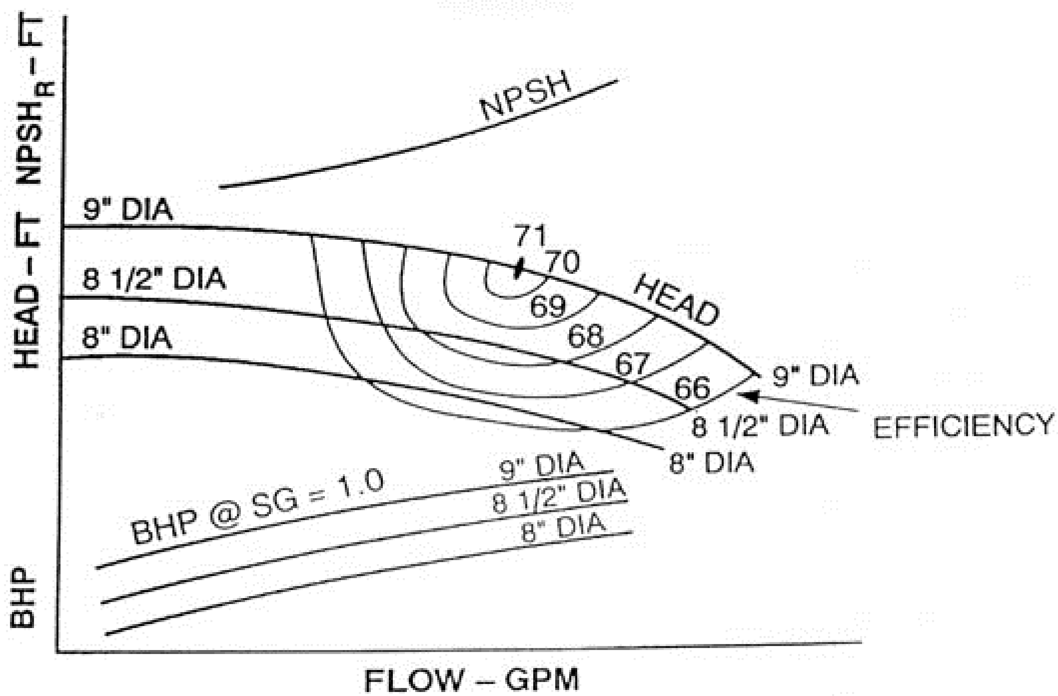


FIGURE 10

Since the flow rate will be determined by the process requirements, the power (BHP) required by the driver will also be affected. What would occur if an 8 $^{1}/_{2}$" diameter impeller were used and the head (differential pressure) required by the process was low? Answer: Since the pressure differential required is low, the flow rate will increase and for the 8 $^{1}/_{2}$" diameter impeller, the power required by the driver (BHP) will increase.

Therefore, a motor can trip out on overload, a steam turbine's speed can reduce or a diesel engine can trip on high engine temperature. These facts are shown in Figure 11.

EFFECT OF THE PROCESS ON DRIVERS

- Motors can trip on overload
- Steam turbines can reduce speed
- Diesel engines can trip on high engine temperature

FIGURE 11

Auxiliary System Reliability is also affected by process changes. Auxiliary systems support the equipment and their components by providing ... clean, cool fluid to the components at the correct differential pressure, temperature and flow rate.

Typical auxiliary systems are:

- Lube Oil Systems
- Seal Flush System
- Seal Steam Quench System
- Cooling Water System

The reliability of machinery components (bearings, seals, etc.) is directly related to the reliability of the auxiliary system. In many cases, the root cause of the component failure is found in the supporting auxiliary system.

As an example, changes in auxiliary system supply temperature, resulting from cooling water temperature or ambient air temperature changes, can be the root cause o component failure. Figure 12 presents these facts.

COMPONENT (BEARING AND SEAL) RELIABILITY

- Is directly related to auxiliary system reliability
- Auxiliary system reliability is affected by process condition changes.
- "Root causes" of component failure are often found in the auxiliary system.

FIGURE 12

As a result, the condition of all the auxiliary systems supporting a piece of equipment must be monitored. Please refer to Figure 13.

“ALWAYS "THINK SYSTEM"

- Monitor auxiliary system condition
- Inspect auxiliary system during component replacement

FIGURE 13

THE MAJOR MACHINERY COMPONENTS

Please refer again to Figure 7 which shows how process condition changes can cause damage and/or failure to any pump component.

Regardless of the type of machinery, the major component classifications are the same. The major machinery components and their systems are shown in Figure 14.

MAJOR MACHINERY COMPONENTS & SYSTEMS

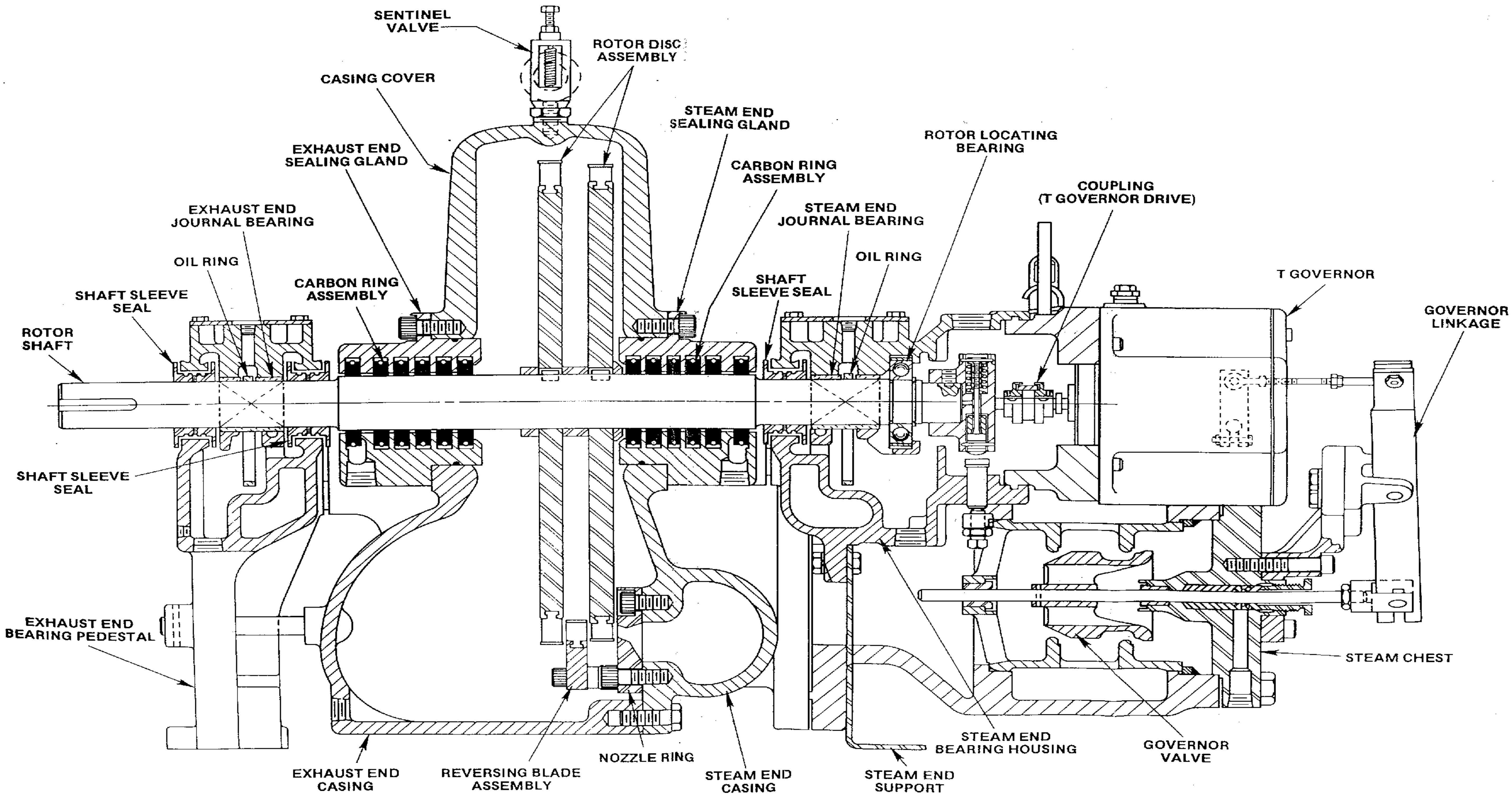
- Rotor
- Radial Bearing
- Thrust Bearing
- Seal
- Auxiliary Systems

FIGURE 14

Regardless of the type of machinery, if we monitor the condition of each major component and its associated system, we will know the condition of the machine!

Refer to Figure 15 and define the major components and their associated systems.

The concept of component condition monitoring will be the focus of this workshop. If we know what to monitor to determine the condition of each component, we can eliminate unscheduled shut downs and failures. The component condition monitoring procedure is shown in Figure 16.



TYPICAL GENERAL PURPOSE (PUMP DRIVE) TURBINE ASSEMBLY
BYRIH TURBINE

FIGURE 15

COMPONENT CONDITION MONITORING

- **Define** Each Major Component
- **List** Condition Monitoring Parameters
- **Obtain** Baseline Data
- **Trend** Data
- **Establish** Threshold Limits

FIGURE 16

During this workshop, the following specific examples of component condition monitoring will be presented:

Component	Site Visit
Rotor	1
Bearing & Lubrication Systems	2
Mechanical Seal & Associated Systems	3
Rotor & Bearing Systems	4

The function of each component, the parameters to monitor and their normal limits (where action should be taken) will be presented for each component example.

Next, a site visit will be conducted. Workshop groups will be assigned a specific component to condition monitor.

After each site visit, workshop groups will return to the classroom and determine the condition of the observed component and prepare an action plan if required.

The remainder of this module contains useful information concerning the major machinery components and auxiliary systems, parameters to be monitored along with their action limits. This information will be used during the workshop.

SPECIFIC MACHINERY COMPONENTS AND SYSTEM MONITORING PARAMETERS AND THEIR LIMITS.

The following pages contain information concerning what parameters should be monitored for each major machinery component to determine its condition. In addition, typical limits are noted for each component.

These limits represent the approximate point at which action should be planned for maintenance. They are not intended to define shutdown values.

THE ROTOR

Rotor condition defines the performance condition (energy & efficiency) of the machine. Figure 17 presents this value for a pump.

PUMP PERFORMANCE MONITORING

1. Take value at minimum flow (shut off discharge valve)

2. Measure:
    - P_1
    - P_2
    - Driver bhp
    - Specific gravity

3. Calculate:

    A. Head Produced $\dfrac{FT - LB_F}{LBM} = \dfrac{\Delta P \times 2.311}{S.G.}$

    B. Pump Efficiency (%) = $\dfrac{{}^{*}HD \times GPM \times S.G.}{3960 \times BHP}$

4. Compare to previous value:

    If > -10% perform maintenance

*** WHERE HD IS DEFINED IN ITEM A ABOVE.**

FIGURE 17

RADIAL BEARINGS

Figures 18 & 19 present the facts concerning anti-friction and hydrodynamic (sleeve) radial or journal bearing condition monitoring.

CONDITION MONITORING PARAMETERS AND THEIR ALARM LIMITS

JOURNAL BEARING (ANTI-FRICTION)

PARAMETER	LIMITS
1. BEARING HOUSING VIBRATION (PEAK)	.4 INCH/SEC (10 MM/SEC)
2. BEARING HOUSING TEMPERATURE	185°F (85°C)
3. LUBE OIL VISCOSITY	OFF SPEC 50%
4. LUBE OIL PARTICLE SIZE	
• NON METALLIC	25 MICRONS
• METALLIC	ANY MAGNETIC PARTICLE IN THE SUMP
5. LUBE OIL WATER CONTENT	BELOW 200 PPM

FIGURE 18

CONDITION MONITORING PARAMETERS AND THEIR ALARM LIMITS

JOURNAL BEARING (HYDRODYNAMIC)

PARAMETER	LIMITS
1. RADIAL VIBRATION (PEAK TO PEAK)	2.5 MILS (60 MICRONS)
2. BEARING PAD TEMPERATURE	220°F (108°C)
3. RADIAL SHAFT POSITION*	> 30° CHANGE AND/OR 30% POSITION CHANGE
4. LUBE OIL SUPPLY TEMPERATURE	140°F (60°C)
5. LUBE OIL DRAIN TEMPERATURE	190°F (90°C)
6. LUBE OIL VISCOSITY	OFF SPEC 50%
7. LUBE OIL PARTICLE SIZE	> 25 MICRONS
8. LUBE OIL WATER CONTENT	BELOW 200 PPM

* EXCEPT FOR GEARBOXES WHERE GREATER VALUES ARE NORMAL FROM UNLOADED TO LOADED

FIGURE 19

THRUST BEARINGS

Figures 20 & 21 show condition parameters and their limits for anti-friction & hydrodynamic thrust bearings.

CONDITION MONITORING PARAMETERS AND THEIR ALARM LIMITS

THRUST BEARING (ANTI-FRICTION)

PARAMETER	**LIMITS**
1. BEARING HOUSING VIBRATION (PEAK)	
• RADIAL	.4 IN/SEC (10 MM/SEC)
• AXIAL	.3 IN/SEC (1 MM/SEC)
2. BEARING HOUSING TEMPERATURE	185°F (85°C)
3. LUBE OIL VISCOSITY	OFF SPEC 50%
4. LUBE OIL PARTICLE SIZE	
• NON METALLIC	>25 MICRONS
• METALLIC	ANY MAGNETIC PARTICLES WITH
SUMP	
5. LUBE OIL WATER CONTENT	BELOW 200 PPM

FIGURE 20

CONDITION MONITORING PARAMETERS
AND THEIR ALARM LIMITS

THRUST BEARING (HYDRODYNAMIC)

PARAMETER	**LIMITS**
1. AXIAL DISPLACEMENT*	>15-20 MILS (0.4-0.5 MM)
2. THRUST PAD TEMPERATURE	220°F (105°C)
3. LUBE OIL SUPPLY TEMPERATURE	140°F (60°C)
4. LUBE OIL DRAIN TEMPERATURE	190°F (90°C)
5. LUBE OIL VISCOSITY	OFF SPEC 50%
6. LUBE OIL PARTICLE SIZE	> 25 MICRONS
7. LUBE OIL WATER CONTENT	BELOW 200 PPM

* AND THRUST PAD TEMPERATURES > 220° F (105°C)

FIGURE 21

SEALS

Figure 22 presents condition parameters and their limits for a pump liquid mechanical seal.

CONDITION MONITORING PARAMETERS
AND THEIR ALARM LIMITS

PUMP LIQUID MECHANICAL SEAL

PARAMETER	LIMITS
1. STUFFING BOX PRESSURE	< 25 PSIG (175 KPA) ** ABOVE SUCTION PRESSURE
2. STUFFING BOS TEMPERATURE	BELOW BOILING TEMPERATURE FOR PROCESS LIQUID
3. FLUSH LINE TEMPERATURE	+/- 20°F (10°C) FROM PUMP CASE TEMP
4. *PRIMARY SEAL VENT PRESSURE (BEFORE ORIFICE)	> 10 PSI (70 KPAG)

* ON TANDEM SEAL ARRANGEMENTS ONLY
** TYPICAL LIMIT – THERE ARE EXCEPTIONS (SUNDYNE PUMPS)

FIGURE 22

AUXILIARY SYSTEMS

Condition monitoring parameters and their alarm limits are defined in Figures 23 & 24 for lube and pump flush systems.

CONDITION MONITORING PARAMETERS AND THEIR ALARM LIMITS

LUBE OIL SYSTEMS

	PARAMETER	LIMITS
1.	OIL VISCOCITY	OFF SPEC 50%
2.	LUBE OIL WATER CONTENT	BELOW 200 PPM
3.	AUXILIARY OIL PUMP OPERATING OPERATING	YES/NO
4.	BYPASS VALVE POSITION (P.D. PUMPS)	CHANGE > 20%
5.	TEMPERATURE CONTROL VALVE POSITION	CLOSED, SUPPLY TEMPERATURE > 130°F (55°C)
6.	FILTER ∆P	> 25 psid (170 KPAG)
7.	LUBE OIL SUPPLY VALVE POSITION	CHANGE +/- 20%

FIGURE 23

CONDITION MONITORING PARAMETERS AND THEIR ALARM LIMITS

PUMP SEAL FLUSH
(SINGLE SEAL, FLUSH FROM DISCHARGE)

PARAMETER	LIMITS
1. FLUSH LINE TEMPERATURE	+/- 20^0 F (+/-10°C) OF PUMP CASE TEMPERATURE
2. SEAL CHAMBER PRESSURE	< 25 PSI (175 KPA) ABOVE SUCTION PRESSURE

FIGURE 24

THE CONCEPT OF FLUID HEAD

- **Definition**
  - **Head Required**
  - **Head Produced**
- **Paths Of Compression**
- **The Different Types Of Gas Head**
- **The Thermodynamic Path Of All Multistage Compressors – Polytropic Head**
- **Proper Dynamic Compressor Curve Formats**
- **The Universal Curve-Performance Parameters**
- **The Predicted, Test And Actual Curves**
- **Estimated Labyrinth Leakage**
- **Re-Rating Existing Compressors**

INTRODUCTION

Without a doubt one of the most confused principles of turbo-compressor design in my experience has been that of fluid head. I have found that understanding this concept can best be achieved by recognizing fluid head is the energy required to achieve specific process requirements.

Head is the energy in foot-pounds force required to compress and deliver one pound of a given fluid from one energy level to another. One of the confusing things about this concept is that the industry persists in defining head in feet. Head should be expressed in foot-pounds force per pound mass or British Thermal Units per pound. A British Thermal Unit per pound of fluid is equal to exactly 778 foot-pounds force per pound mass of that fluid.

Remember, when we deal with fluid head, a fluid can be either a liquid or a gas depending upon the conditions of the fluid at that time. Ethylene for instance, can be either a liquid or a gas depending on its pressure and temperature. If it is a liquid, an ethylene pump will be used and the energy required to increase the pressure of the liquid from P_1 to P_2 will be defined as head in foot-pound force per pound mass. Conversely, if the conditions render it a vapor, an ethylene compressor would be used to achieve the same purpose. We will see in this section that the amount of energy required to compress a liquid or a gas the same amount will be significantly higher in favor of the gas because the gas is at a much lower density than that of a liquid. Understanding a Mollier Diagram is an important aid to understanding the concept of fluid head. Every fluid can have a Mollier Diagram drawing which expresses energy on the X axis and pressure on the Y axis for various temperatures. Increasing the pressure of a vapor will result in increased energy required.

Having defined the concept of head as that of energy, one must now investigate the different types of ideal (reversible) gas heads, namely isothermal, isentropic or adiabatic and polytropic. All of these types of fluid head simply describe the path that the gas takes in being compressed. It must be remembered that any type of head can be used to describe a reversible compressor path as long as the Vendor uses the appropriate head and efficiency in his data reduction calculations. In this section we will show the assumptions for various kinds of heads and the relative difference in their values. In addition, the definitions of each type will be stated.

DEFINITION

The definition of head required by any fluid compression process is presented in Figure 1.

HEAD REQUIRED ***is the energy in foot pounds required to compress and deliver one pound of a given fluid from one energy level to another.***

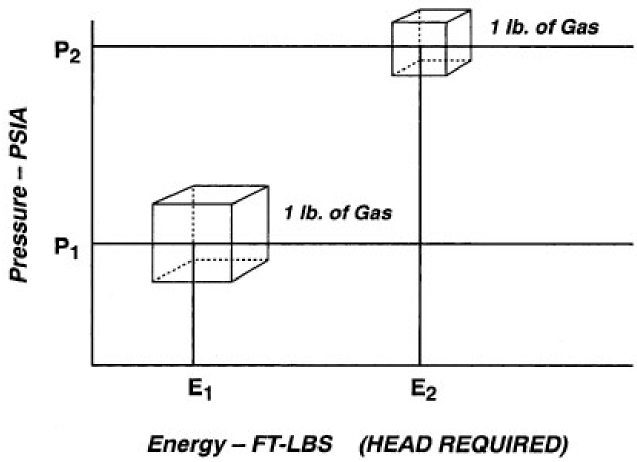


FIGURE 1

In any compression process, the amount of energy required to compress one pound of mass of a specific gas, at a given temperature from compressor suction flange (P_1) to the discharge flange (P_2) is defined as head required.

Figure 2 demonstrates how the density of a fluid significantly affects the amount of energy (head) required in a compression process.

FLUID HEAD

LIQUID

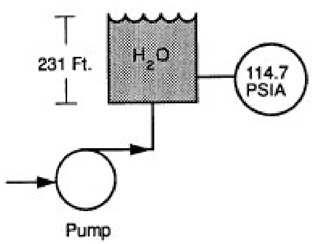
$$\underset{(Ft.)}{HEAD} = \frac{2.311 \times \Delta P}{S.G.}$$

Water

ΔP = 100 PSI

HEAD = 231 Ft.

P_1 = 14.7 PSIA , T_1 = 100°F



GAS

$$\underset{(Ft.)}{HEAD} = \left[\frac{1545}{M.W.}\right](T_1)\left(\frac{K}{K-1}\right)(Z)\left[\frac{P_2}{P_1}^{\left(\frac{K-1}{K}\right)} - 1\right]$$

Nitrogen

ΔP = 100 PSI

HEAD = 86,359 Ft.

P_1 = 14.7 PSIA , T_1 = 100°F

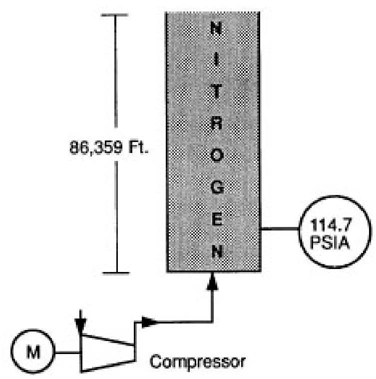


FIGURE 2

Water with a density of 62.4 lbs/ft^3 requires only 231 ft-lb force per lb mass to compress the liquid 100 PSI. Note also, the equation for head required by a liquid is independent of temperature. On the other hand, nitrogen with a density of 0.07 lbs/ft^3 requires approximately 350 times the energy! This is because in both the case of water and nitrogen, the process requires that the fluid be compressed 100 PSI. However, since the mass of nitrogen is only 0.1% of the mass of water, a much greater amount of energy is required to compress the gas.

Figure 3 shows the head produced characteristics of positive displacement and dynamic compressors.

COMPRESSOR HEAD PRODUCES CHARACTERISTICS

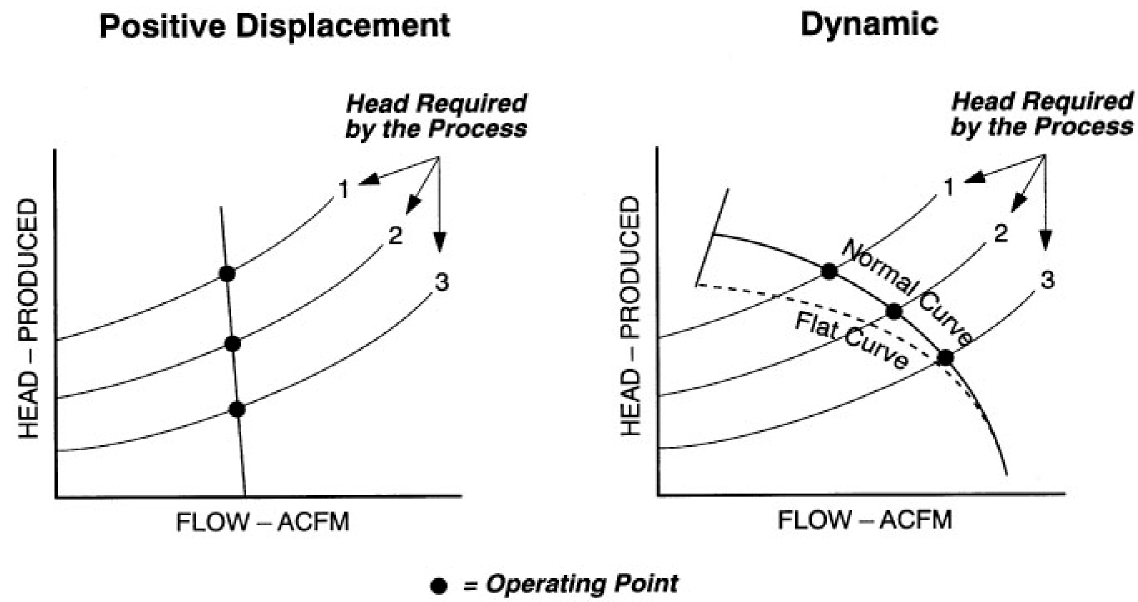


FIGURE 3

Note that regardless the amount of head required by the process, the flow rate of a positive displacement compressor is not affected. On the other hand, a dynamic compressor's flow rate is significantly affected by changes in the head required by the process. This is because the characteristic of any dynamic compressor is that it can only produce a greater amount of energy at a lower flow rate. The reason for this characteristic will be explained in a subsequent module. Therefore, any increase in the head required by a process will reduce the flow rate of a dynamic compressor. This is an extremely important fact because reduced flow rate in a dynamic compressor can lead to extreme, long term mechanical damage to the compressor unit. Also note in Figure 3 that the flatter the head produced curve a dynamic compressor possesses, the greater the effect of head required upon flow rate.

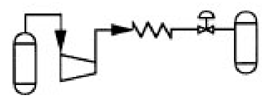
HEAD REQUIRED

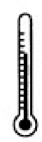
Thorough understanding of the concepts of head required by the process and head produced by the compressor is absolutely essential if dynamic compressor operation is to be understood.

It has been my experience that a lack of understanding exists in the area of dynamic compressor performance and often leads to a much greater emphasis upon the compressor's mechanical components (impellers, labyrinths, seals, bearings and shafts). In many cases, the root cause of dynamic compressor mechanical damage is that the head required by the process system exceeded the capability of the dynamic compressor.

Figure 4 presents the factors that determine the head (energy) required by any process.

HEAD (ENERGY) REQUIRED

- Head (Energy) Required is Determined by:
  - System Resistance (P_2 / P_1) 
  - Gas Composition (MW,Z,K) 

  - Inlet Temperature (T_1) 

- Head Required Increases If:
  - (P_2 / P_1) Increases
  - MW Decreases
  - (T_1) Increases
- Head Required Decreases If:
  - (P_2 / P_1) Decreases
  - MW Increases
  - (T_1) Decreases

FIGURE 4

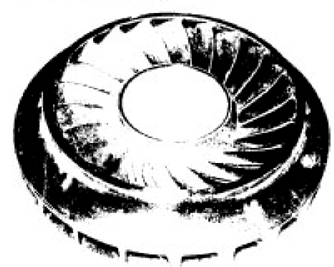
Note that the head required by the process is inversely proportional to the gas density. If the gas density decreases, the head required by the process will increase. Gas density will decrease if gas temperature increases, inlet pressure decreases or molecular weight decreases. If the head required by the process increases, the flow rate of any dynamic compressor will decrease as shown in Figure 3. If the gas density increases, the head required by the process will decrease and dynamic compressor flow rate will increase.

HEAD PRODUCED

In Figure 5, the factors that determine the head produced by a dynamic compressor are presented.

HEAD (ENERGY) PRODUCED

- Head (Energy) Produced is Determined by:
  - Compressor Impeller Design



- Head Produced by the Impeller,
  - Increases with Tip Speed
    - Impeller Diameter
    - Compressor Speed (RPM)
  - Increases with Decreasing Flow

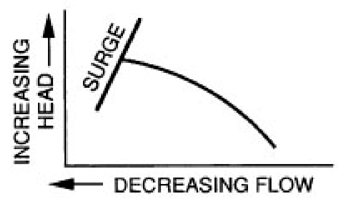


FIGURE 5

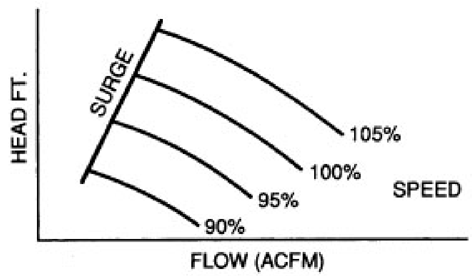
Simply stated, for a given impeller vane shape, head produced by a dynamic compressor is a function of impeller diameter and impeller speed. Once the impeller is designed, it will produce only one value of head for a given shaft speed and flow rate.

The only factor that will cause a lower value of head to be produced than stated by the compressor performance curve is if the compressor has experienced mechanical damage or if it is fouled.

Figure 6 shows how the need for dynamic compressor inspection can be determined.

THE TURBO-COMPRESSOR CURVE

- The Turbo-Compressor Head (Energy) vs. Flow Curve is *Fixed by Compressor Design*



- For a Given Speed and Flow, If an Operating (Test) Point *Does Not Fall on the Curve*,* the Unit Requires Maintenance

* *Within ± 10% (Accounting for Instrument Calibration Error)*

FIGURE 6

If for a given flow rate and shaft speed, the head produced falls below the value predicted by greater than 10%, the compressor should be inspected at the first opportunity. Having explained the concept of head required by the process, the method of calculating the head required by a process system needs to be discussed.

PATHS OF COMPRESSION

Figure 7 presents a typical Mollier Diagram plotted pressure vs. energy.

IDEAL PATHS OF COMPRESSION

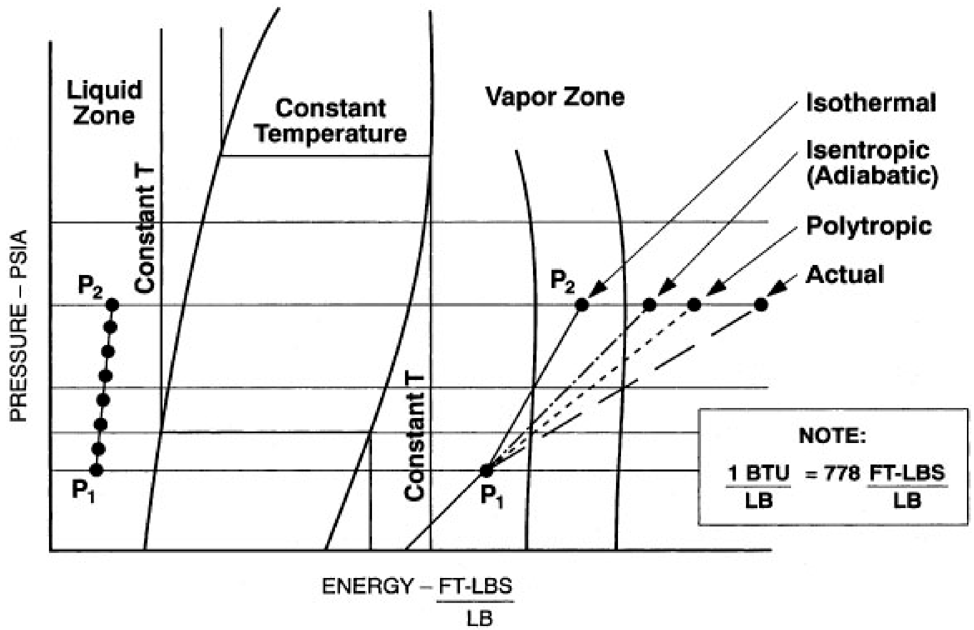


FIGURE 7

A Mollier Diagram can be drawn for any pure fluid or fluid mixture. Usually, Mollier Diagrams are prepared only for pure fluids since any change in fluid mixture will require a new Mollier Diagram to be prepared. The Mollier Diagram can be used to determine the head required by the process system for any liquid, saturated vapor or vapor compression process. Observe that for liquid compression, the amount of energy required to increase the pressure from P_1 to P_2 is *very* small. However, for compression of a vapor, the amount of energy required to compress from P_1 to P_2 is very large as previously explained. Refer back to Figure 3 of this module and study the equations that are used to determine the head required for a liquid and a vapor. In addition to many more parameters being required for calculation of head required for a vapor, certain ideal assumptions must be made.

THE DIFFERENT TYPES OF GAS HEAD

A vapor can be ideally compressed by any one of the following reversible thermodynamic paths:

- Isothermal - constant temperature
- Isentropic (adiabatic) - no heat loss
- Polytropic - temperature not constant and heat lost and most closely follows the actual path of compression since the impeller efficiency determines the amount of energy per unit of weight.   Polytropic Exponent – BTU's/LB or KJ/KG

$$\frac{n-1}{n} = \underbrace{\frac{k-1}{k}}_{\text{Polytropic Efficiency}}$$

ACTUAL HEAD

The actual path that any compressor follows in compressing a vapor from P_1 to P_2 is equal to the reversible path divided by the compressor's corresponding path efficiency. Therefore, we can write the following equation:

$$\text{Actual Head} = \frac{\text{Head Isothermal}}{\text{Eff'y. Isothermal}} = \frac{\text{Head Isentropic}}{\text{Eff'y Isentropic}} = \frac{\text{Head Polytropic}}{\text{Eff'y Polytropic}}$$

When evaluating compressor bids from different vendors quoting different types of reversible heads the above equation proves useful. If the head produced by one vendor divided by the corresponding efficiency is not equal to the corresponding values quoted by the competition   Better start asking why!

Figure 8 defines the different types of ideal heads and shows the relative difference between their values as compared to polytropic head.

IDEAL COMPRESSION HEADS – RELATIVE VALUES

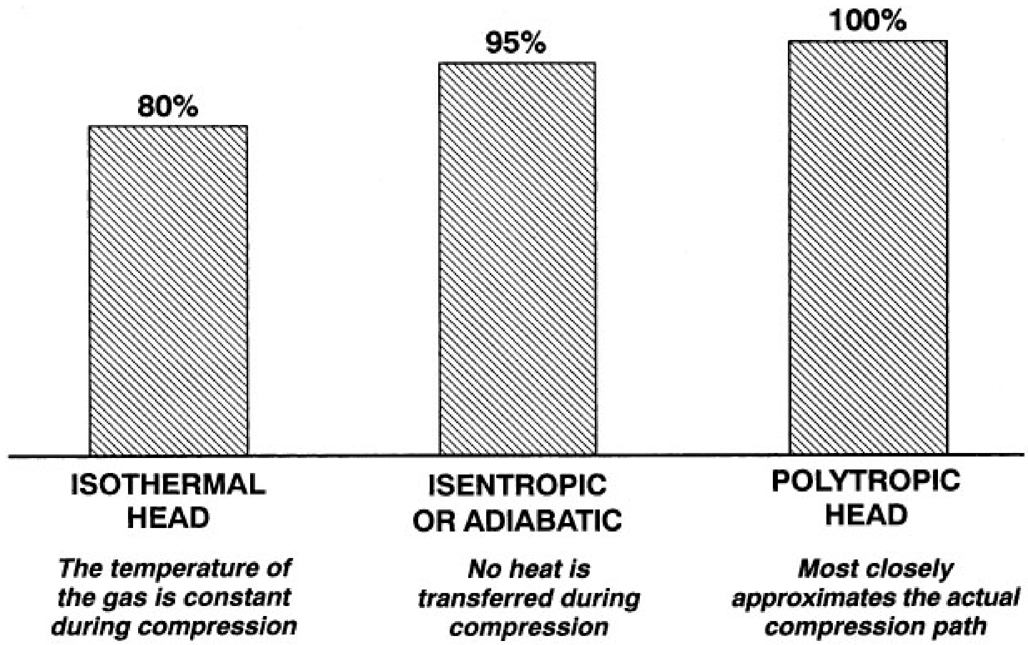


FIGURE 8

The definition of polytropic head is confusing and difficult to understand if not investigated further.

The ideal gas head equations are described in Figure 9.

THE THERMODYNAMIC MODEL OF ALL MULTISTAGE DYNAMIC COMPRESSORS

The energy produced by all multistage dynamic compressors is presented as a curve showing polytropic head on the Y axis and actual flow on the X axis. Also usually shown on the same curve is polytropic efficiency. Power vs. flow usually is shown on a separate curve. Polytropic head and efficiency are used since vendors can add head algebraically produced by each impeller in a multistage compressor since the head produced is a function of individual impeller ploytropic efficiency. Refer to Figure 9 for definitions of z, cp, cv, k and n (see actual polytropic performance curves).

At this point, the instructors will calculate:

- z, k and n using an actual gas analysis for one of your plant compressors
- Polytropic head and efficiency using:
    - o Benedict Webb Rubin (BWR) equation of state
    - o The ploytropic equations (excel spreadsheet)

Values will be plotted for given flows on one of your performance curves.

IDEAL GAS HEAD EQUATIONS

Isothermal	Isentropic (Adiabatic)	Polytropic

$$HD = \left(\frac{1546}{M.W}\right)(T_1)(Z_{AVG})\left[L_N\left(\frac{P_2}{P_1}\right)\right] \quad HD = \left(\frac{1545}{M.W}\right)(T_1)\left(\frac{K}{K-1}\right)(Z_{AVG})\left[\left(\frac{P_2}{P_1}\right)^{\left(\frac{K-1}{K}\right)}-1\right] \quad HD = \left(\frac{1545}{M.W}\right)(T_1)\left(\frac{n}{n-1}\right)(Z_{AVG})\left[\left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)}-1\right]$$

Where:

$\frac{1545}{\text{M.W}}$ = Gas constant "R"

P_2= Discharge Pressure – PSIA

T_1 = Inlet temp. – °R

P_1 = Inlet pressure – PSIA

°R = 460 + °F

K = Ratio of specific heats C_p/C_v

Z_{AVG} = Average compressibility

$$\frac{(Z_1+Z_2)}{2}$$

n = Polytropic exponent

$$\frac{(n-1)}{n} = \left(\frac{K-1}{K}\right)\left(\frac{1}{\eta_{POLY}}\right)$$

Ln = Log to base e

η_{POLY} =Polytropic efficiency

FIGURE 9

Note that the only difference between isentropic and polytropic head is the values;

$\frac{K-1}{K}$ and $\frac{n-1}{n}$

Also note that $\frac{n-1}{n} = \frac{\frac{K-1}{K}}{\eta \text{ poly}}$

Now if η poly - 100%, $\frac{n-1}{n} = \frac{K-1}{K}$ or

Polytropic Head = Isentropic Head

Therefore, I think of $\frac{n-1}{n}$ as a correction

Factor to $\frac{K-1}{K}$ that will most closely approximate the actual compressor path for a given compressor (refer back to Figure 7)

Remember, polytropic head is an ideal reversible compression path. Today, most Compressor vendors have adopted polytropic head as their standard.

DYNAMIC COMPRESSOR CURVES FORMAT

Finally, Figure 10 presents the different ways that compressor compression performance can be formatted.

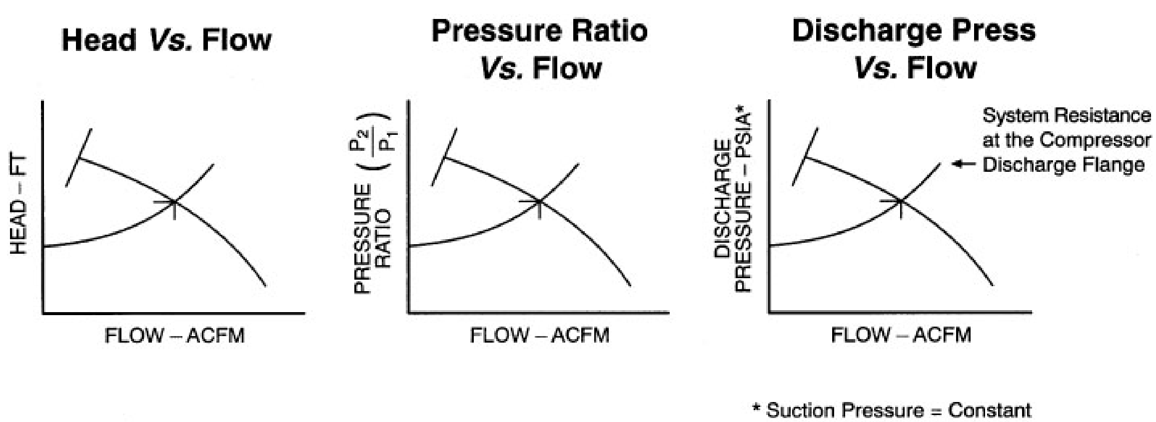


FIGURE 10

Head vs flow is always preferred because the head produced by a dynamic compressor is not significantly affected by gas density. However, compression ratio or discharge pressure are! That is, compression ratio and discharge pressure curves are invalid if the inlet gas temperature, inlet pressure or molecular weight changes! This may seem confusing at first, but refer back to Figure 5 which shows how head is produced by a dynamic compressor. It is a function only of impeller diameter and shaft speed. Gas density influences the head required by the process (refer to Figure 4).

THE UNIVERSAL CURVE PERFORMANCE PARAMETERS

Various parameters have been established within the compressor industry to classify and quantify the different types of compressors available. The first of these parameters is called *specific speed*, which is a term used to classify compressor impellers on the basis of their performance and proportions regardless of their actual size or the speed at which they operate. Since specific speed is a function of impeller proportions, it is constant for any series of impellers having the same proportions or for one particular impeller operating at any speed.

Specific speed can be defined as the speed in revolutions per minute at which an impeller would rotate if reduced proportionately in size so as to deliver one cubic foot of gas per minute against a total head of one foot. It is found from the equation in Figure 11.

SPECIFIC SPEED EQUATION

$$\text{Specific speed} = N_S = \frac{N\sqrt{Q}}{H^{3/4}}$$

Where N = operating speed, r/min
Q = design flow, icfm
H = design head, ft

FIGURE 11

It is not necessary to grasp the physical significance of specific speed to make use of it. Specific speed should be considered to be a type of characteristics of the impeller which specifies its general proportions and behavior.

Impeller and blade tests have confirmed that the higher the specific speed, the higher the attainable efficiency. See Figure 12.

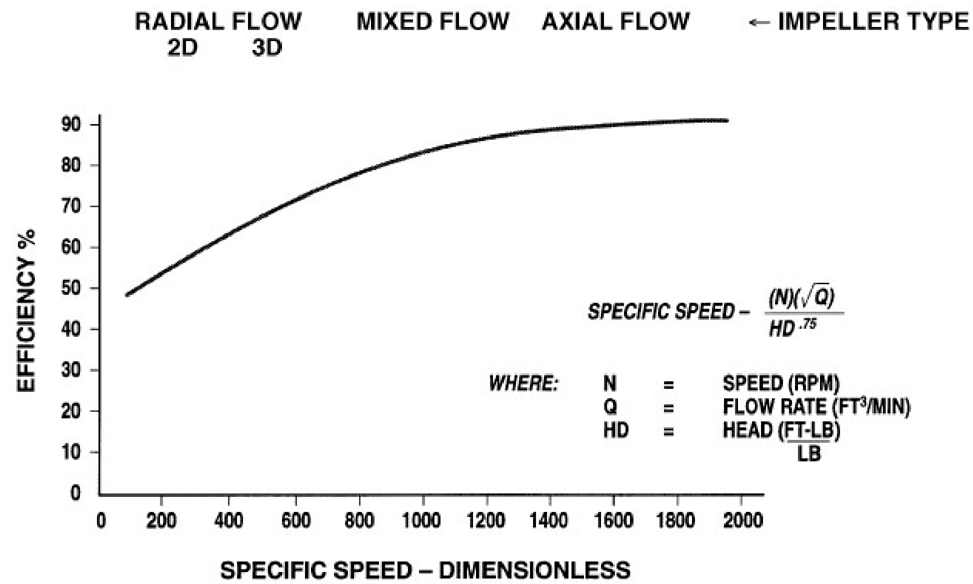


FIGURE 12

The objective of all compressor vendors is to design their impellers or blades for the highest specific speed that the mechanical design parameters can accept. The parameters are:

- Impeller eye and bore stress
- Axial stage spacing
- Diffuser ratio (diffuser OD/diffuser ID) – typical minimum = 1.5

Although specific speed is a useful parameter for comparison purposes, it is not always a useful tool for making compressor computations. The most commonly used parameters for compressor calculations are flow coefficient and head coefficient. Both these coefficients are non-dimensional, and they can be used to define completely an impeller's performance. The flow and head coefficient are defined in Figure 13.

FLOW & HEAD COEFFICIENT

Flow

$$\emptyset = \frac{3.056Q}{UD^2} = \frac{700Q}{ND^3}$$

Where Q = inlet flow, icfm
U = tip speed, ft/s
D = impeller outside diameter, in
N = r/min

The flow coefficient can be used to put any compressor impeller's flow-handling capability in non-dimensional terms

Head

$$\psi = \frac{H_g}{U^2}$$

Where H = head, ft (per stage)
G = gravitational constant (32 FT/Sec^2)
U = tip speed, ft/s

The head coefficient relates the head output of an impeller non-dimensionally

FIGURE 13

Figure 13 presents a comparison of four compressor impellers, showing both specific speeds and flow coefficients. Figures 14, 15, 16 and 17 show three different compressor rotors illustrating various impeller types.

IMPELLER SPECIFIC SPEEDS & FLOW COEFFICIENTS

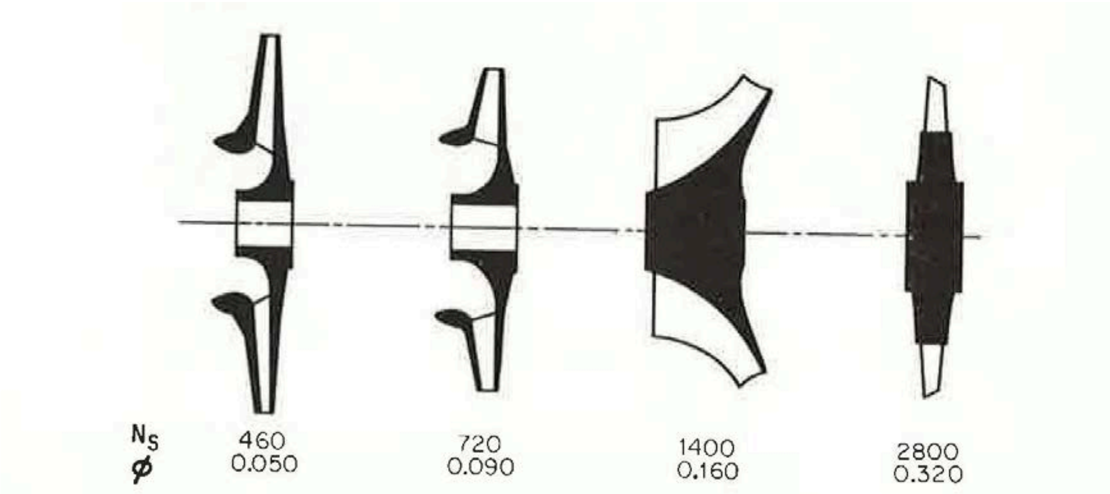


FIGURE 14
Courtesy of DeLaval

TYPICAL SMALL-CAPACITY LOW-SPECIFIC-SPEED MULTISTAGE ROTOR

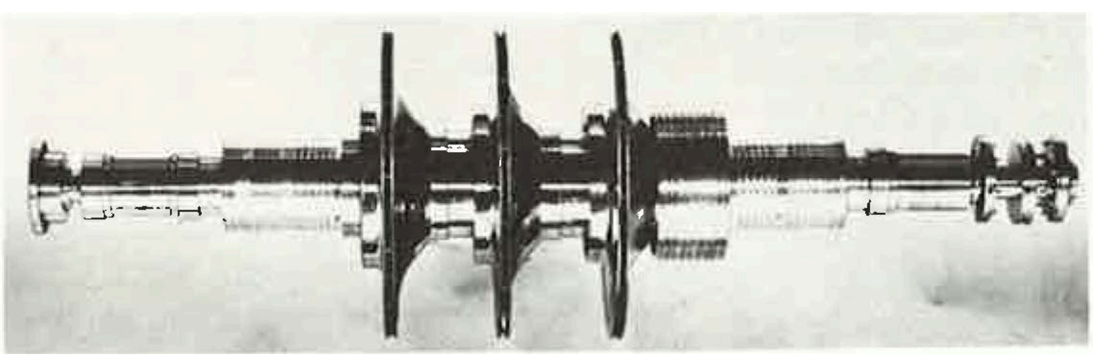


FIGURE 15
Courtesy of DeLaval

COMBINED MIXED-FLOW & MEDIUM-SPECIFIC-SPEED CENTRIFUGAL-COMPRESSOR ROTOR

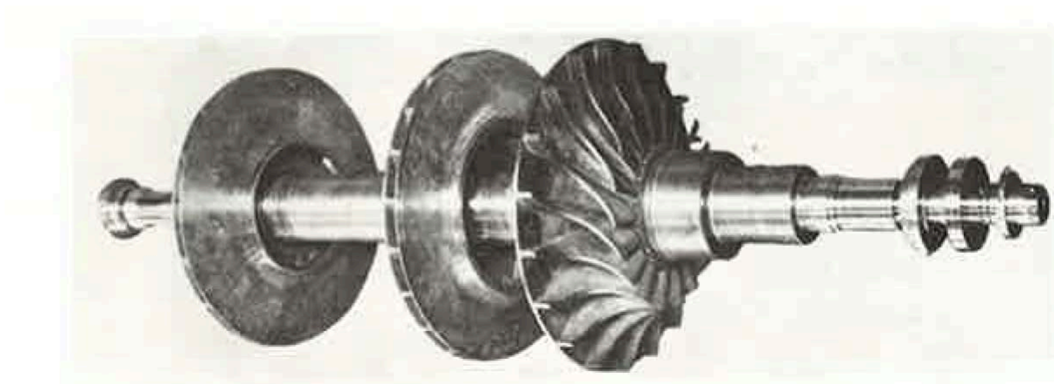


FIGURE 16
Courtesy of DeLaval

TYPICAL AXIAL-COMPRESSOR ROTOR

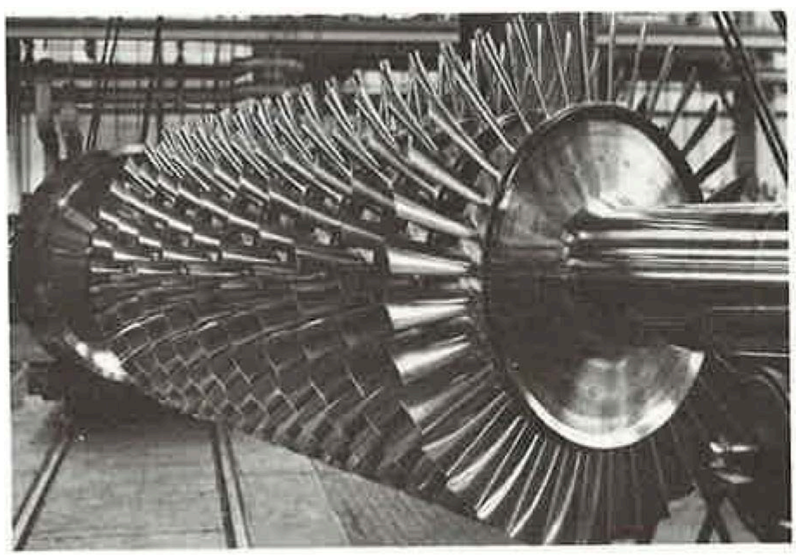


FIGURE 17
Courtesy of DeLaval

By using a combination of flow coefficient and head coefficient, any impeller is defined in terms that can be easily manipulated.

Figure 18 shows a present day rotor with “mega flow” coefficient impellers (flow coefficients >0.16 up to 0.25) used for heavy gas applications (propane, propylene, butane). These high specific speed impellers are “mixed flow” type-backward learning and approach axial blades in design (high flow, lower head produced).

MEGA HEAVY GAS IMPELLERS

FIGURE 18
Courtesy of MHI

The concept of flow coefficient can also be expressed by Q/N (actual volume divided by speed).

The concept of head coefficient can also be expressed by volume ratio (actual outlet flow divided by actual inlet flow). See Figure 19.

$$Q/N = \frac{\text{Actual Flow}}{\text{Rotational Speed}}$$

Normalizes flow for a given speed like flow coefficient

$$\frac{Q_2}{Q_1} = \text{Volume Ratio}$$

Defines the compression of an individual impeller like head coefficient

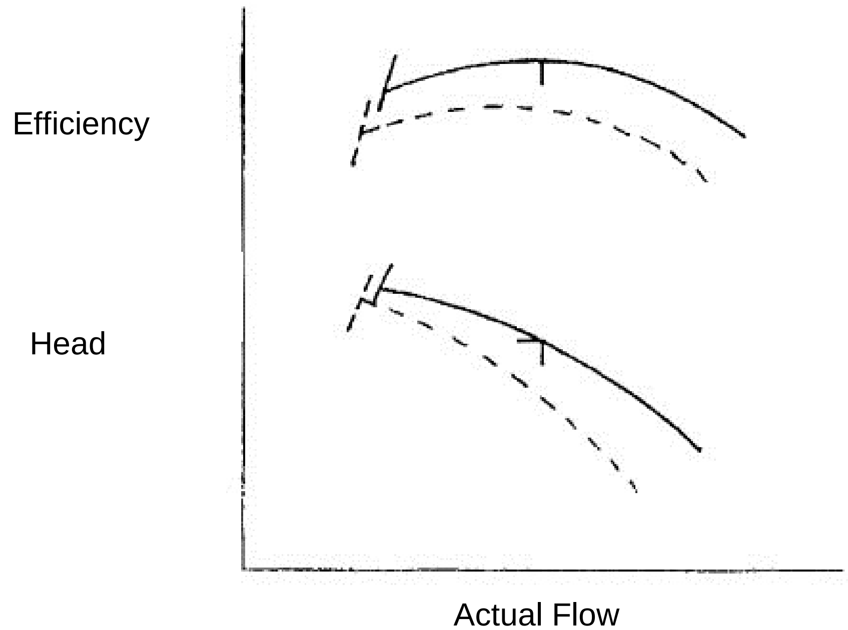
FIGURE 19

THE PREDICTED (EXPECTED) CURVE AND TESTED (ACTUAL) CURVE

The predicted (estimated or expected) curve is contained in the vendor proposal and represents the expected performance based on the vendor's data base (previous actual individual impeller tests). The predicted curve shape, head rise (head at surge divided by head at design point) will not be identical to the test curve results. Per API compressor specifications, only the overall compressor power (head x mass flow divided by efficiency) is guaranteed at rated point to not exceed +4%.

Therefore, the predicted curve should never be used to measure field performance - only the test curve should be used. This applies to centrifugal pumps as well as compressors. See Figure 20 for an explanation of predicted vs. test curve.

PREDICTED VS. TEST CURVE



Always determine field performance using "test curves"

_____ = Predicted

--------- = Test

FIGURE 20

COMPRESSOR INTERSTAGE AND BALANCE DRUM LEAKAGES

The efficiency of any dynamic impeller or axial blade is a function of the leakage thru the interstage seals. See Figure 21 for an example of interstage leakage paths for a closed centrifugal impeller.

<u>A STAGE</u> CONSISTS OF AN IMPELLER, STAGE SEALS, DIFFUSER (CROSSOVER & RETURN PASSAGE-IF IN A MULTISTAGE CONFIGURATION)

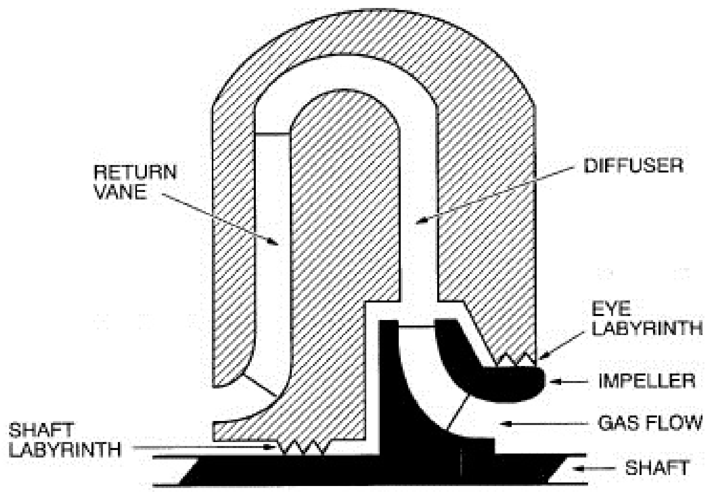


FIGURE 21

Compressor vendors are constantly reviewing inter-stage seal and balance drum seal designs to minimize leakages to optimize individual impeller/blade efficiency.

Today "abradable" seals are typically used to minimize leakage and increase individual impeller/blade efficiency as much as 3%

Typical abradable seal material is fluorosint (mica impregnated Teflon) with knife edges on the rotating component. See Figure 22 for an example of an abradable separation seal used in a dry gas seal application (a FAI Best Practice).

ABRADABLE DRY GAS SEAL SEPARATION SEAL (FAI BEST PRACTICES)

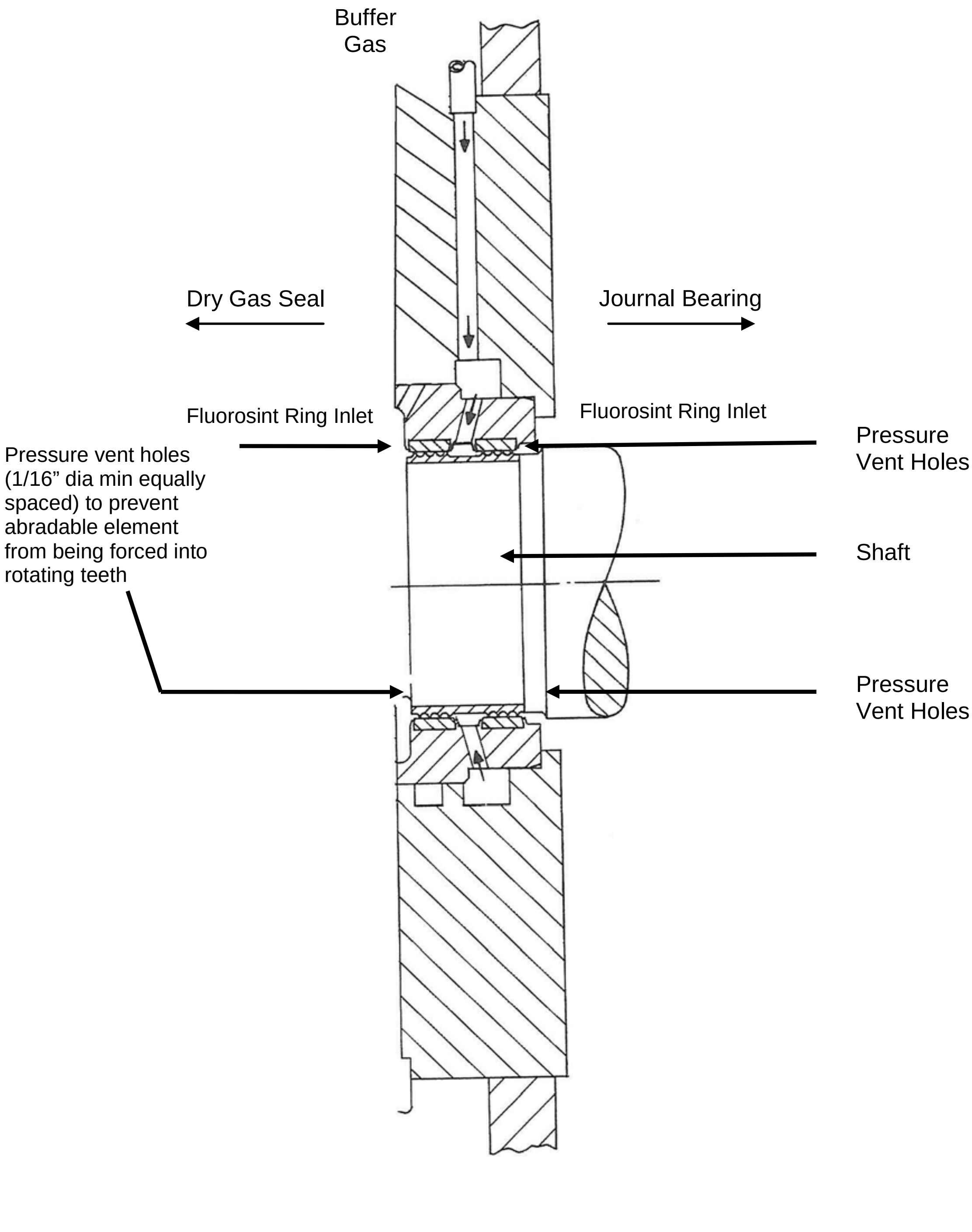


FIGURE 22

It is very important to properly pressure balance the stationary abradable element in its holder to prevent it from being forced into the rotating component knife edges.

At this point, the instructor's will use "Labyflex" software to demonstrate typical leakages.

RE-RATING EXISTING COMPRESSORS

When re-rating existing dynamic compressors, the following items must be considered to assure optimum compressor reliability. See Figure 23.

RE-RATE "KEY FACTS"

- Proven flow coefficients
- Proven head coefficients
- Head rise (head @ surge/head at rated point)
- Location of each impeller rated point to its BEP
- Stage spacing
- Critical speeds (lateral and torsional)
- Suction drum/demister entrainment velocity
- Piping Velocity

FIGURE 23

Since there usually will not be an opportunity to factory performance test the compressor, vendor confirmation of flow and head coefficient experience head rise and location of the rated point to best efficiency point for each impeller is a must to assure optimum field performance.

Mechanically, the stage spacing (axial space for each impeller stage) and critical speeds must be determined and confirmed.

From a process perspective suction drum and demister entrainment velocity and pipe velocity must be confirmed.