COMPRESSOR TYPES, CLASSIFICATIONS, AND APPLICATIONS

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ABSTRACT

The compressor industry has emerged from the decade of the 1980s right sized, streamlined, and computerized. Management trends include a broadening of responsibility for all departments. In order to satisfy these new responsibilities, maintenance, operations, and engineering personnel need continuous review of compressor types, classifications, and applications.

Companies are discovering the void in talent that right sizing has created and most organizations retain a core group of experienced professionals who are utilized as a reference resource. The turbomachinery grass roots introduction seeks to present elementary compressor concepts to all interested parties. This group will also certainly need a working knowledge of aerodynamics, blade design (and repair), magnetic bearing theory, and advanced thermodynamic concepts. The compressor types introductory program does not address these more advanced topics.

Compressor Types starts at the beginning of the user/manufacturer relationship with applications. Moreover, utilizing Balje's [1] work on specific speed as a focal point, the following concepts will be covered at entry level:

- Fan, blower, compressor differentiation
- Curve shape and where to operate
- Head, flow, horsepower; calculations
- The specific speed of positive displacement machinery
- The specific speed of single stage centrifugals
- Applications
  - Vents
  - Flares
  - Oxidizers
  - Overhead recompression
  - Process recompression
  - Transmission

The Compressor Types Program is an ongoing program offered by the Turbomachinery Laboratory at Texas A&M University to assist industry professionals in gaining exposure to elementary compressor concepts to help them assume their broadening compressor responsibilities in the 1990s.

DEFINITION AND OPERATION

Fans, blowers and compressors are machines designed to deliver gas at a pressure higher than that originally existing. Pressure rise, working pressure, specific speed and mechanical design form the basis of differentiation and classification. Initially, these machines can be divided into positive displacement and dynamic categories (Figure 1).

The methods employed to achieve compression are:

- Trap consecutive quantities of gas in some type of enclosure, reduce the volume, thus increasing the pressure, then push the compressed gas out of the enclosure.
- Trap consecutive quantities of gas in some type of enclosure, carry it without volume change to the discharge opening, com-

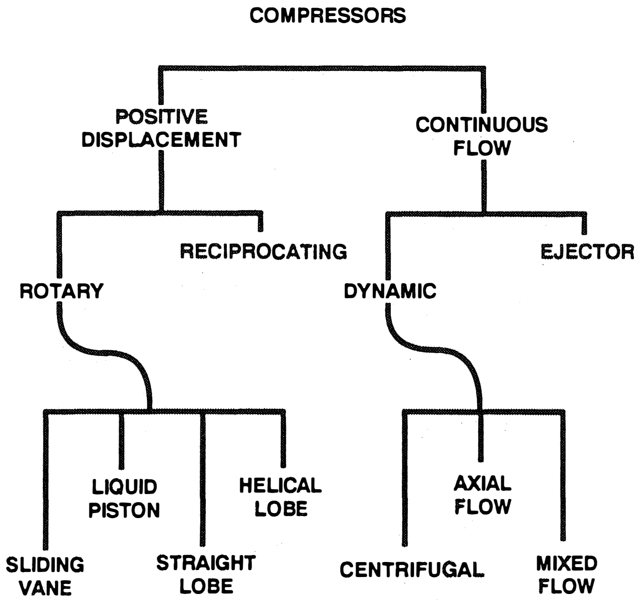

Figure 1. Compressor Types.

pressing the gas by overcoming back flow from the discharge system and pushing the compressed gas out of the enclosure.

• Compress the gas by the mechanical action of rotating impellers or bladed rotors that impart velocity and pressure to the flowing gas. Additional velocity energy in the gas is converted to pressure in an adjacent stationary diffuser or blade.

• Entrain the gas in a high velocity jet of another compatible gas and convert the high velocity of the mixture into pressure via a diffuser.

COMPRESSOR TYPES DESCRIPTION

Positive displacements machines work by mechanically changing the volume of the working fluid. Dynamic machines work by mechanically changing the velocity of the working fluid.

Reciprocating Piston

Reciprocating compressors are positive displacement machines in which the compressing and displacing element is a piston having a reciprocating motion within a cylinder. The overall cycle is shown in Figure 2 with the four typical phases of intake, compression, discharge, and expansion. Inlet valves are open from 4 to 1, and discharge valves are open from 2 to 3.

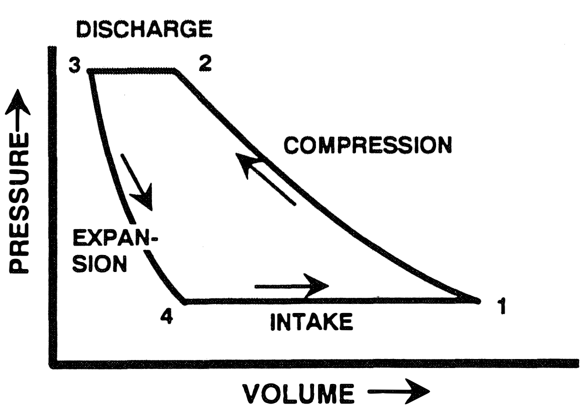

Figure 2. Reciprocating Compressors.

Sliding Vane

Sliding vane compressors are rotary positive displacement compressors (Figure 3). A slotted cylinder is fitted with nonmetallic vanes and placed eccentric inside a tube. As the slotted cylinder is turned, the vanes slide along the inner wall of the tube forming regions of changing volume.

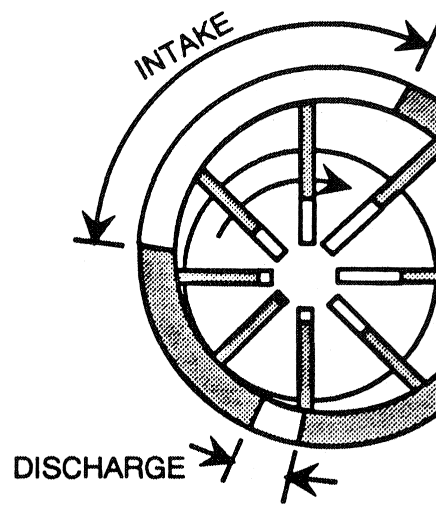

Figure 3. Sliding Vane Compressor.

Liquid Ring

Liquid ring compressors utilize a squirrel cage fan type impeller which is placed eccentric inside a tube (Figure 4). A compatible liquid is introduced into the chamber along with the gas to be compressed. Because of the centrifugal force and the shape of the internal cavity, the liquid forms an eccentric shape producing regions of changing volume. The liquid must be separated from the compressed gas after the compression process and recirculated.

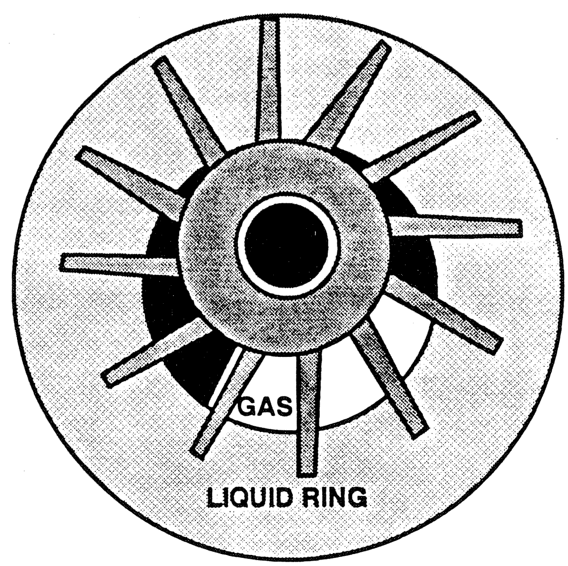

Figure 4. Liquid Ring Compressor.

Rotary Lobe

Two straight mating lobed impellers trap the gas and carry it from intake to discharge (Figure 5). There is no internal compression.

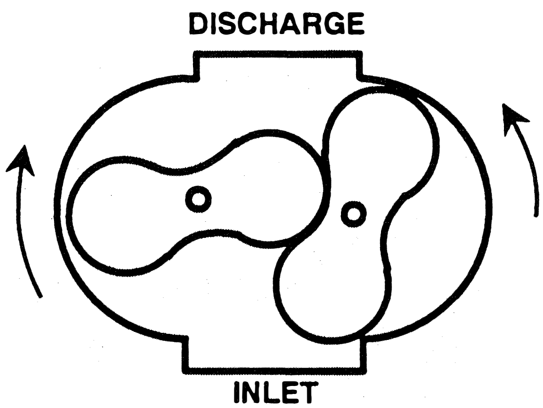

Figure 5. Straight Lobe, Two Rotor Compressor.

Helical Screw

Two intermeshing rotors compress and displace the gas (Figure 6). The gas is trapped in the rotor pockets at one end; it is compressed between the intermeshing rotors and discharged at the opposite end. Some helical screw compressors operate with fluid and these are called flooded screw compressors. The fluid provides a liquid seal around the rotors, absorbs the heat of compression, allowing the machine to produce a greater pressure rise. The fluid must be removed from the gas after the compression process.

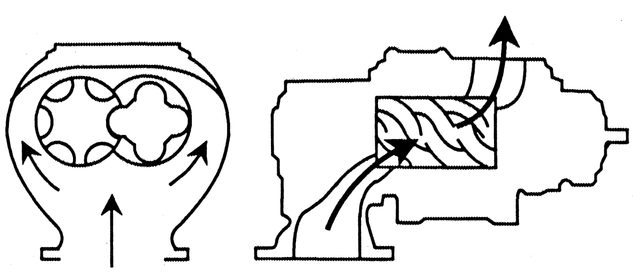

Figure 6. Helical or Spiral Lobe Compressors.

Centrifugal Compressors

Centrifugal compressors are dynamic machines in which the rapidly rotating impeller accelerates the gas (Figure 7). The process flow propagates from axial to radial (perpendicular to shaft centerline) into a stationary diffuser converting velocity to pressure.

Axial Compressors

Axial compressors are dynamic machines in which the gas flow is accelerated in an axial and peripheral direction by the rotation of specially shaped blades (Figure 8). The process flow is parallel to shaft centerline. Stator blades allow the recovery of velocity to pressure.

Compressor Performance

A variety of factors such as type, discharge pressure, capacity, speed, power range, compression ratio, and specific speed can be used to differentiate fans, blowers and compressors. Performance

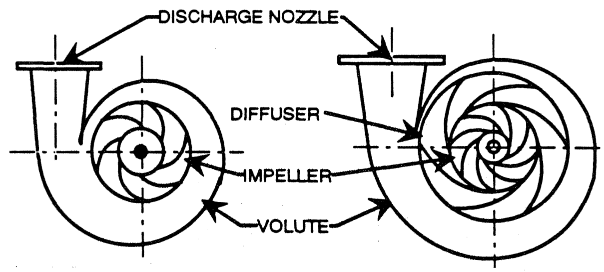

Figure 7. Centrifugal Compressors.

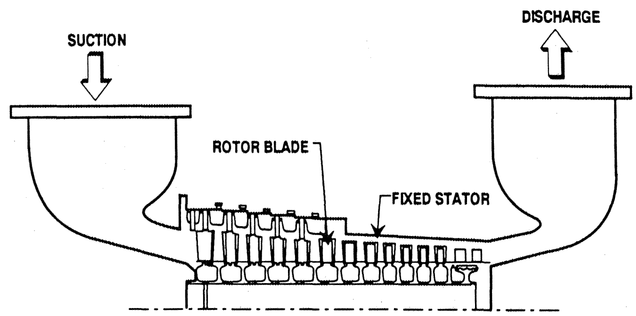

Figure 8. Axial Compressors.

is generally expressed through graphs of pressure produced, power required and efficiency *vs* flow. Organizing the family tree of compressor types by pressure and flow in Figure 9 shows the relative performance position of the various compressor types and makes visible the dilemma of application engineering; which compressor type is suitable for actual process problems. For flowrates greater than 10,000 cfm, the choice of a multistage axial is relatively simple. For pressure ratios greater than 20, the choice of a multistage reciprocating compressor is simple as well. However, most compressor applications have a pressure ratio less than 20 and a flowrate less than 10,000 cfm, which requires a compression method evaluation for a given process. All of the positive displacement and dynamic compressor types make contributions in this performance zone. For a given flow and pressure rise, each type of compressor can be satisfactory yet accomplish the task in a much different way. The graph of compressor type via curve shape in Figure 10 highlights the differences in curve shape between positive displacement and dynamic compressor designs. Positive displacement designs tend to produce variable head at a

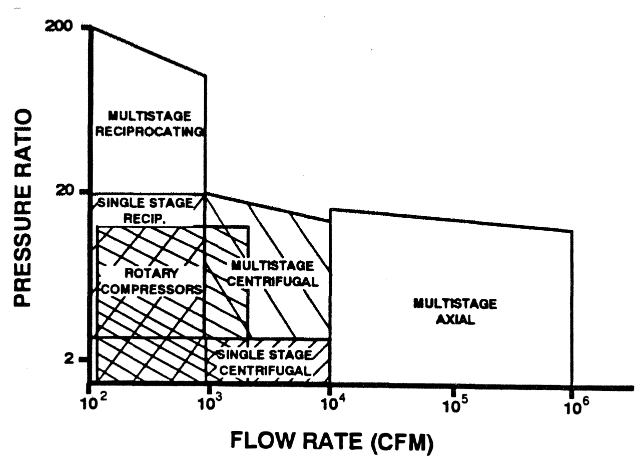

Figure 9. Compressor Types.

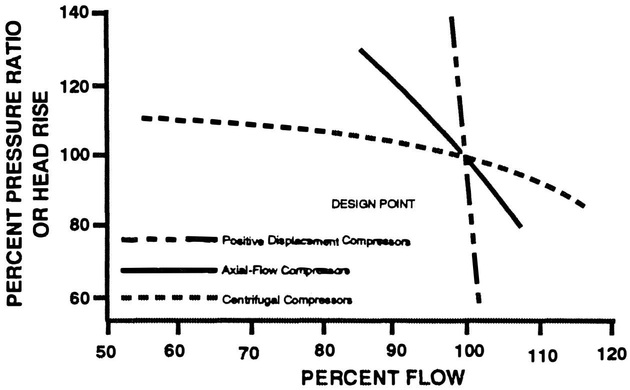

Figure 10. General Characteristic.

relatively constant flow while dynamic designs tend to produce variable flow at a relatively constant head. For the purposes of the turbomachinery program, the interest here is in dynamic designs and when to choose a dynamic fan, blower, or compressor.

Fan, Blower, Compressor Differentiation

While differentiation of dynamic compressor types is possible utilizing dimensional analysis it is more common to loosely classify these machines by the pressure rise they produce.

- Fans produce a pressure rise of 0.25 psi to 3.0 psi
- Blowers produce a pressure rise of 1.0 psi to 8.0 psi
- Compressors produce a pressure rise greater than 5.0 psi

Curve Shape and Where to Operate

Equipment selected to operate at, or near, its absolute design limits generally requires a more aggressive maintenance program. It would seem prudent, then, to operate at the extremities of the centrifugal compressor curve only when absolutely necessary or when adequate system controls are included in the design. While the positive displacement compressor is traditionally pressure controlled centrifugal compressors are flow controlled. At the extreme right side of the centrifugal performance curve the pressure rise tends to drop. This cutoff region is called stone wall. For a given impeller, diffuser and housing design, energy losses increase dramatically as the gas flow through the machine increases. The ability to predict the pressure rise of a given design is difficult and therefor unattractive in this unstable flow region. As system resistance increases flow through the centrifugal machine decreases and at low flow, which is particular to the given hardware design, the centrifugal will surge. Surge occurs when the process flow reverses around the impeller and is aerodynamically and mechanically undesirable.

HEAD, FLOW, HORSEPOWER, SPECIFIC SPEED: CALCULATIONS

Required Data

P-1 psia, suction pressure	kg/cm2
P-2 psia, discharge pressure	kg/cm2
T-1 °R, Suction temperature	°K
MW, mol weight	–
k, adiabatic ratio of specific heat	–
SCFM, Rate of Flow Std ft³/min	Nm3/min

Z, compressibility; if unknown, use 1 for estimating only

Calculate:

Pressure ratio (ratio of compression) = p2 / p1 (1)

K/K(K−1)

(K−1)/K

Required Head = foot pound force per pound mass at design

$$H = Z \times \frac{1545}{MW} \times T_1 \times \frac{K}{K-1} \times \left(\left(\frac{P_2}{P_1}\right)^{(K-1)/K} - 1 \right) \tag{2}$$

Actual flow rate at compressor inlet = ACFM act ft³/minute

$$ACFM = SCFM \times \frac{T_1}{T_{std}} \times \frac{P_{std}}{P_1} \tag{3}$$

Mass flow rate = Q* = SCFM/379 × MW pounds mass per minute

$$Q^* - SCFM \times \frac{1 mol}{379 scf} \times \frac{MW}{1 mol} \tag{4}$$

$$Hp = \frac{Q^* \times H}{33{,}000 \times EFF} \tag{5}$$

$$\text{Disch. temp} = T_2 = T_1 \left(\frac{\left(\frac{P_2}{P_1}\right)^{(K-1)/K}}{EFF} + 1 \right) \tag{6}$$

Assume EFF = 0.6 or, estimate EFF from specific speed graph. Calculations are based upon mol weight range of 16 to 50.

DIMENSIONAL ANALYSIS AND SPECIFIC SPEED

Mathematicians have confirmed the notions of machine designers that for each type of machine there is an optimum geometry which allows the machine to operate in its best efficiency range. For compressors, dimensional analysis has been used to identify the speed in rpm a given machine geometry must operate in order to raise one cubic foot of gas one foot in head. Moreover, by generalizing geometries into specific diameters a plot of specific speed *vs* specific diameter for the compressor family tree in Figure 11 reveals that for each machine design there are ideal operating ranges where compressor efficiencies are optimum.

Required Data

N rpm, operating speed
V1 actual cubic feed per second, inlet flow

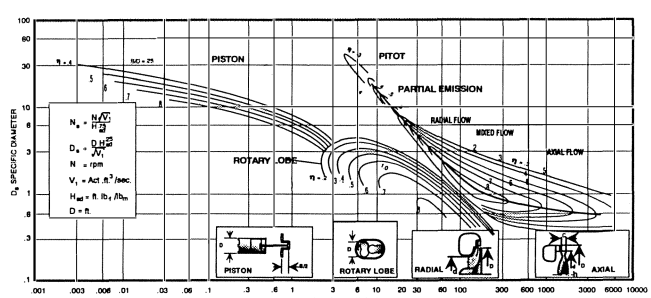

Figure 11. Generalized N_S D_S Diagram for Single Stage Pumps and Compressors at Low Pressure Ratios. From Turbomachines, O.E. Balje. Reprinted by permission of John Wiley & Sons, Inc., Copyright ©1981 [1].

H foot pound force per pound mass, adiabatic head
D feet, diameter

SPECIFIC SPEED AND DIAMETER EQUATIONS

$$Ns = N \times \frac{V1^{.5}}{H^{.75}} \quad (7)$$

$$Ds = \frac{D \times H^{.25}}{V1^{.5}} \quad (8)$$

SPECIFIC SPEED OF POSITIVE DISPLACEMENT MACHINERY

Piston type compressors generally operate in the 300 to 1,800 rpm range utilizing 4.5 in to 26.0 in diameter pistons and stroke lengths of 3.0 in to 13.0 in. Piston compressor designers tend to think in terms of linear stroking velocities of 500 ft/min to 1,000 ft/min. The optimum specific speed and diameter ranges for these machines is:

Ns = 0.003 to 3.0 Ds = 3.0 to 30.0

Rotary lobe compressors operate in the 600 to 3,600 rpm range utilizing 4.0 in to 14.0 in diameter lobes. The optimum specific speed and diameter ranges for these machines is:

Ns = 2.5 to 150 Ds = 0.3 to 5.0

SPECIFIC SPEED OF SINGLE STAGE CENTRIFUGALS

Single stage centrifugals operate in the 3,600 to 50,000 rpm range utilizing 4.0 in to 30.0 in diameter impellers. Centrifugal designers tend to think in terms of impeller rotational tip speeds of 500 to 1,000 ft/sec. The optimum specific speed and diameter ranges for these machines are:

Ns = 7.0 to 150 Ds = 1.0 to 10

Mixed flow and axial flow machines operate at similar speeds to centrifugals, utilizing diameter wheels and stators. The optimum specific speed and diameter ranges for these machines are:

Ns = 150 to 6000 Ds = .4 to 3.0

APPLICATIONS

Vapor containment process problems produce the need, to analyze and apply various types of compressors. Vapor compression can be divided by technique into two groups:

Destruction	*Recovery*
Vents	Overhead recompression
Flares	Process compression
Oxidizers	Transmission

Recovery techniques have substantial visibility because of the required capital investment along with payback potential and usually command the attention of a more senior engineering management team. Vapors which have a high sales value, or are present in sizable quantities, are recovered, compressed, and sold in a liquid or gaseous state.

Historically, low pressure, dilute vapors, and aromatics, such as those present at your convenience store gasoline pump, as well as the oil wellhead, gas plant, chemical plant and refinery, have been vented to atmosphere. Environmental and regulatory issues confirm the public notion that:

If you can't breathe it, you shouldn't vent it.
If you can't drink it, you shouldn't spill it.

To meet these types of demands, destruction techniques are being utilized to dispose of noncommercial (nonbreathable) vapors. These low pressure, compliance projects are of interest from an application perspective.

VENTS

Vent applications include: coal mine methane, bio-pond vapor feed, landfill gas, barge vapors, tank, tanker, rail car vapors, pump tandem seal vent drum vapors, and aeration.

From Figure 8, the reader is reminded that there are numerous mechanical choices for these applications. The following procedure may prove helpful in making a compressor selection:

- Obtain the process conditions and gas quality.
- Calculate the design point head flow, horsepower and discharge temperature.
- Utilize specific speed and diameter criteria to narrow the choice of compressors.
- Compare available seals, metallurgy, control requirements installation costs, operating costs, price, delivery, and vendor service reputation.
- Select the compressor.

EXAMPLES

P_1 = 3 in Hg Vacuum, P_2 = 15 psig, MW = 30, k = 1.39, Z = 1
T_1 = 100°F, Q = 504 MSCFD @ 14.7 & 60 F, Elevation; Sea Level.
Atmospheric pressure - 14.65 psia

P_1 = 14.65 psia - (3 in Hg × 0.491 psi/in Hg)	= 13.18 psia
P_2 = 14.65 + 15	= 29.65 psia
MW	= 30
k	= 1.39
T_1 = 460 + 100	= 560°R

From Equation (1)

Pressure Ratio = P_2/P_1 = 29.65/13.18	= 2.25
(k−1)/k	= .281
k/(k−1)	= 3.56

From Equation (2)

$$H - 1 \times \frac{1545}{30} \times 560 \times 3.56 \times (2.25^{.281} - 1) - 26{,}276 \text{ ft lbf/lbm}$$

From Equation (3)

$$ACFM = \frac{504{,}000 \text{ SCF}}{1 \text{ day}} \times \frac{1 \text{ day}}{1440 \text{ min}} \times \frac{560°R}{520°R} \times \frac{14.65 \text{ psia}}{13.18 \text{ psia}} = 419 \text{ACFM}$$

From Equation (4)

$$\text{Mass Flow Rate } Q^* = \frac{504{,}000 \text{ SCF}}{\text{day}} \times \frac{1 \text{ day}}{1440 \text{ min}} \times \frac{1 \text{ mol}}{379 \text{ SCFM}} \times \frac{30 \text{ lbm}}{1 \text{ mol}} = 27.7 \text{ lbm/min}$$

From Equation (5)

$$\text{Horsepower} = \frac{27.7 \times 26276}{33{,}000 \times 0.6} = 36.76 \text{ est. HP at design}$$

From Equation (6)

$$\text{Disch. temp} = 560\left(\frac{2.25^{.281} - 1}{0.6} + 1\right) = 798°\text{R} - 460 = 338°\text{F}$$

From Equation (7)

	Piston/Vane	*Lobe*	*Centrifugal*
Typical rpm	1170	2,880	35,000
Note: For Balanced Opposed Piston Use 1/2 ACFM	Direct Connect	Belt Drive	Gear Drive

$$Ns = \frac{1170\left(\frac{209.5}{60}\right)^{.5}}{26{,}276^{.75}} \quad \frac{2880\left(\frac{419}{60}\right)^{.5}}{26{,}276^{.75}} \quad \frac{35{,}000\left(\frac{419}{60}\right)^{.5}}{26{,}276^{.75}}$$

Ns =	1.059	3.69	44.82

From Figure 11

Optimum Ds	4.7	2.0	3.0

From Equation (8)

Piston/Vane *Lobe* *Centrifugal*

$$D = \frac{4.7\left(\frac{209.5}{60}\right)^{.5}}{26{,}276^{.25}} \times \frac{12}{ft} \quad \frac{2 \times \left(\frac{419}{60}\right)^{.5}}{26{,}276^{.25}} \times \frac{12}{ft} \quad \frac{3.0\left(\frac{419}{60}\right)^{.5}}{26{,}276^{.25}} \times \frac{12}{ft}$$

D =	(2) – 8.28"	4.98"	7.47"

From Figure 11

Eff. Estimate	0.8	0.45	0.65

From Equation (5)

$$\text{Recalculate Horsepower} \quad \frac{27.7 \times 26{,}276}{33{,}000 \times 0.8} \quad \frac{27.7 \times 26{,}276}{33{,}000 \times 0.45} \quad \frac{27.7 \times 26{,}276}{33{,}000 \times 0.65}$$

	Piston/Vane	*Lobe*	*Centrifugal*
HP at Design	27.57	49.01	33.93

From Equation (6)

Recalculate Discharge T	268.6	418	338

Prepare installation, operation, life cycle cost matrix and complete evaluation.

This example was created for comparison only. The choices are more obvious if, for example, the flowrate is increased to 2.0 mmscfd, and the pressure is decreased to 5.0 psig.

	(2) Pistons @ 1,200 RPM	*Lobe @ 3,600 RMP*	*Centrifugal @ 15,000 RPM*
For 2 MMSCFD; NS and 5 psig discharge	3.75	126	527

Q = 1663 Hd = 12,194

A specific speed of 3.75 is outside of the nominal piston design range and a better choice would be centrifugals or rotary lobes.

Conversely, if the flowrate is decreased to 504 mmscfd and the discharge pressure is increased to 30 psig:

	(2) Pistons @ 1,200	*Lobe @ 2,000*	*Centrifugal @ 35,000*
Ns	.728	1.76	31

Q = 419 Hd = 43,294

In this case, the rotary lobe is clearly not the best selection, and while the centrifugal appears satisfactory, the specific diameter calculation coupled with the 1000 to 2000 fps nominal tip speed limit probably forces the centrifugal into a series (two stage) configuration making the piston or sliding vane design a more attractive selection.

REFERENCE

1. Balje, O. E., *Turbomachines,* New York: John Wiley & Sons, Inc. (1981).