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Design of Oil-less Compressors and Pumps

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INTRODUCTION

This paper will describe the technology of designing oil-less reciprocating, rotary vane, and diaphragm compressors and pumps.

Since the widespread use of oil-less air is relatively new, particular emphasis will be stressed on the design features and considerations which distinguish the oil-less unit from a lubricated unit. For discussion purposes "oil-less" compressors and pumps are units that operate without the aid of oil lubrication. The media being pumped does not come in contact with an oil lubricant.

RECIPROCATING COMPRESSORS AND VACUUM PUMPS

GENERAL ARRANGEMENT

Discussion will be limited to units 3 HP and smaller. The units are usually horizontal or vertical machines, single and two cylinder. The arrangement is shown in Figure 1.

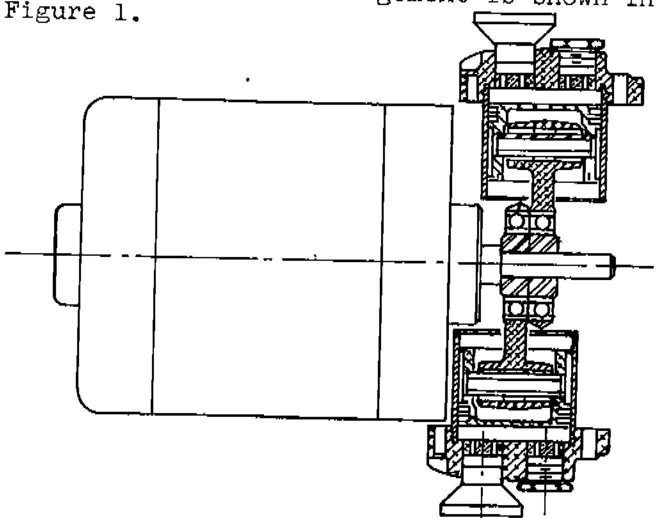

Figure 1

Usually the compressors are manufactured with an electric motor using an extended motor shaft over which the eccentrics and connecting rods are pressed.

Two cylinder machines are usually directly opposed, whereby the forces counteract each other and the basic unbalance is a couple created by the force acting upon the piston and the distance between the connecting rods.

Basically an oil-less compressor or vacuum pump is a machine that operates with out oil lubrication. The piston rings and skirts are usually of a dry lubricant material such as carbon or a P.T.F.E. filled material. The connecting rod and wrist pin bearings are grease filled and sealed for life.

DESIGN DETAILS

The actual design of a compressor and the attention given to detail parts are directly related to its intended application.

Marketing and Engineering Design Parameters must be used as the guidelines for any specific design. The main parameters most affecting the overall design are: displacement, air flow at rated pressure, maximum continuous duty, horsepower limitations, noise level, physical dimensions, and unit cost.

With a fixed or known air flow requirement at the operating point, the displacement can be calculated using the following formulas:

$$\text{Displacement(CFM)} = \frac{A \times S \times N}{1728}$$

$$\text{Delivered air} = \text{Delivered (CFM)} \times VE$$

$$\text{Volumetric efficiency} = \frac{\text{Delivered Air}}{\text{Displacement}}$$

A = Area of Piston (square inches)
S = Stroke (inches)
N = Speed Revolutions per Minute
VE = Volumetric Efficiency

Volumetric efficiencies vary with different piston diameters, strokes, and speeds selected.

Figure 2 is for a typical two cylinder compressor with a nominal 2-1/4 diameter piston running at 1750 RPM.

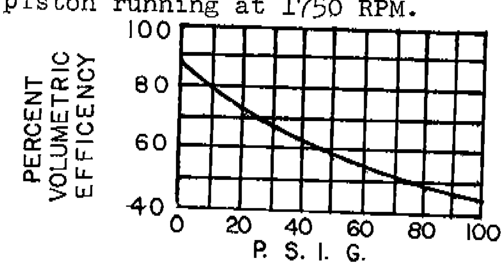

Figure 2

SELECTION OF BORE AND STROKE

The overall physical size of the compressor design is established mainly by the bore and stroke combination.

Refer to Figure 3 for displacements per cylinder for a given bore size and various strokes.

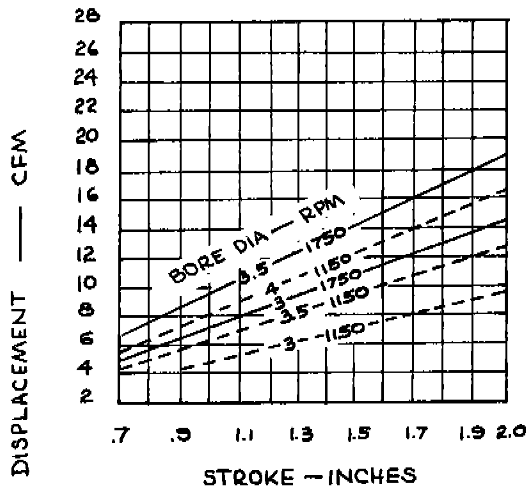

Figure 3

Since an oil-less compressor is designed for several HP ratings, the displacement is varied by changing the stroke while keeping the bore and speed constant. The choice of the piston diameter and stroke will depend on experience and a thorough analysis of many factors. This analysis of the piston diameter reveals the following influence on other compressor design criteria:

Large Piston Diameters
- Small strokes
- Higher clearance volume
- Lower volumetric efficiency
- Less vibration
- Lower ring wear

Small Piston Diameters
- Longer stroke
- Lower clearance volume
- Higher volumetric efficiency
- Higher ring wear
- Less valve opening area

LOAD X VELOCITY LIMITS

Another design consideration greatly influencing the overall design of the unit is load x piston velocity (PV) value of the piston rings and skirts.

Filled P.T.F.E. materials are limited to a maximum PV of approximately 12,000. A greater value will usually result in premature failure or wear.

By adding fillers to the P.T.F.E. materials the PV value can be increased. (See Figure 4)

The ultimate selection of a bore and stroke is directly related to an allowable PV value on the rings and piston skirts. Fig.5

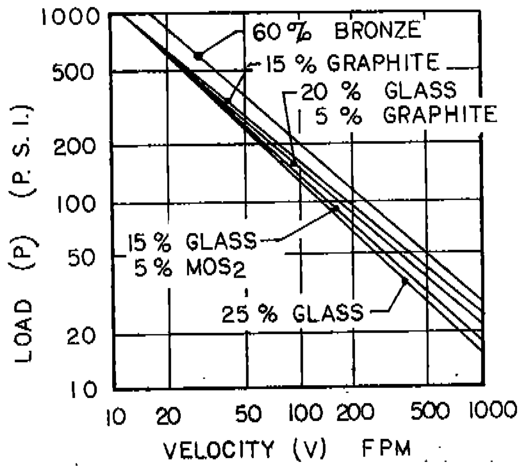

Figure 4

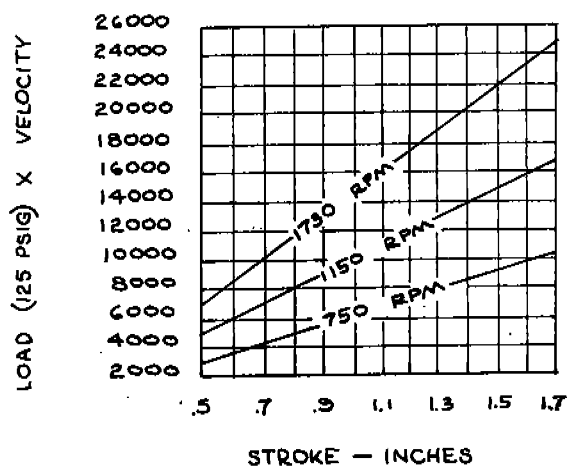

Figure 5

RINGS

The ring design is governed by two basic factors-sealing and life. The more common types are butt joint, radial cut, and lap joint (Figure 6).

The butt joint is the least expensive type to manufacture, but has certain disadvantages. Because of the high thermal expansion of P.T.F.E., a large gap must be designed into the ring for expansion, thereby permitting blow by losses during warm up.

The radial cut ring is the most efficient of the ring types. The flexible portion bisected by the radial cut expands against the cylinder wall and effectively seals the cylinder. The rings after warm up will seal and permit nearly zero blow by.

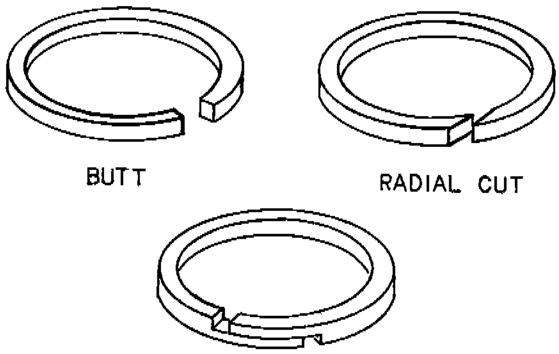

Figure 6

The lap joint configuration is a design compromise between the butt joint and radial cut. The lap cut when used in conjunction with two or more rings provides an effective seal. The gap between the ends of the ring provides a bleed off of compressed air in the cylinder in a static condition. The elimination of the compressed air (Figure 7) aids in starting the compressor and in some cases permits a lower starting torque design motor.

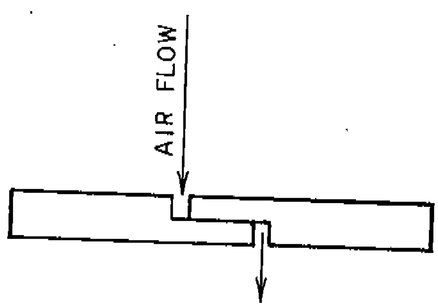

Figure 7

PISTON SKIRTS

The piston skirt material selection is similar to the rings, whereas the rings are pressurized with full cylinder pressure, the piston skirts only see the thrust load imposed by piston connecting rod mechanism. Since the skirt acts as the guide for the piston, the skirt can be of a rigid material to withstand the thrust load.

The more common types of skirt materials are P.T.F.E. with bronze, carbon, glass, and coke fillers. See (Figure 4) for chart of various fillers and PV (load velocity) values.

The basic design criteria should be low coefficient of friction, have long axial length to provide guidance for piston, and have high PV rating. The thrust load acting upon the skirt is the maximum crank force plus friction.

RING BACK-UP SPRING

The back-up spring's basic function is to apply a slight amount of pressure (2-3 PSI) and assist the piston ring in establishing the initial seal between the ring and cylinder wall. The springs are usually constructed in a circular form (Figure 8) and are either blue clock spring or stainless steel material.

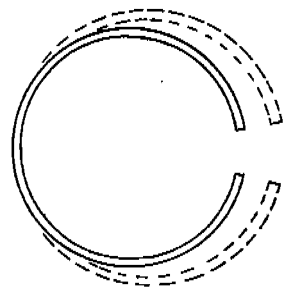

Figure 8

The springs in a free condition have a free gap of the fitted dia/4.

The fitted O.D. of the spring is = inside diameter of piston ring + fitted gap of approximately .060".

CONNECTING ROD BEARING

The connecting rod bearings are sized to carry the load imposed by the piston. The bearings are sealed and lubricated for life. The grease should have a 300°F rating and be non-channeling type to prevent loss of ball and race lubrication. The bearings are packed with approximately 1/3 full value of grease.

The bearings are selected based upon piston load, required bearing life, and speed.

Required bearing capacity =

$$\text{Load}\left[\frac{\text{Life}}{3} \times \frac{\text{Speed}}{16667}\right]^{1/3}$$

A ball bearing is selected to carry the basic dynamic capacity required.

NEEDLE BEARINGS

The wrist pin needle bearings are contained in the connecting rod and are sized to a slip fit over the wrist pin (Figure 9). Since the needle bearings have very limited grease storage capacity the connecting rods are usually designed with an integral grease reservoir between the bearings.

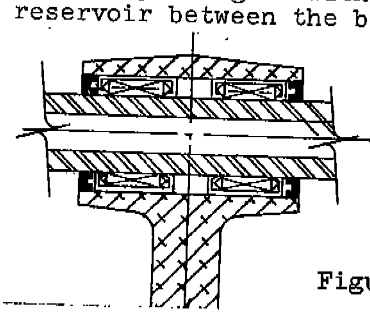

Figure 9

Since the bearings are not full rotating bearings the loading calculations are based on load/life formulas and then factored based on practical experience.

Required bearing dynamic capacity =
Load x SF x LF x HF

SF = Speed factor
LF = Life factor
HF = Wrist pin hardness factor

Consult bearing manufacturers for factors.

The bearing B_{10} life can be calculated using the empirical formula -
Life B_{10} =

$$2\left[\frac{\text{Basic Dynamic Capacity (C)}}{\text{Radial Load (P)}}\right]^{3.33} \times \left[\frac{16667}{\text{Speed RPM}}\right]$$

B_{10} life should be corrected because the bearing does not rotate 360°. From experience the B_{10} life should be reduced by an approximate factor of 6.

PISTON
The piston design is usually influenced by the number of rings required. The pistons are of die cast aluminum alloy and should be light weight to reduce unbalance. The material thickness and wall section should be designed to eliminate porosity. If the casting design is porous, a plastic impregnation will seal small minute holes.

CONNECTING RODS
The connecting rod is also made from a die cast aluminum alloy to reduce weight. The design must be structurally strong enough to carry the load imposed by the piston and motor shaft. The crank end of the rod is pressed over the ball bearing outer race and then staked to provide additional retention.

VALVES
The valves are the most difficult part of the unit to design.

The valve must operate without the aid of lubrication, must be large enough to handle the free air flow of the unit and have a low sound level. The valve's lift area should be large enough to permit the air velocity to not exceed 9,000 feet/min.

The overall sound level is affected by the weight of the valve and the shape of the inlet port.

Due to the absence of lubrication the valve design is usually of a reed or finger type (Figure 10). It is fastened or secured on one end.

Because of the stresses imposed on the valve, the edges of the valve must be free of burrs and manufacturing defects.

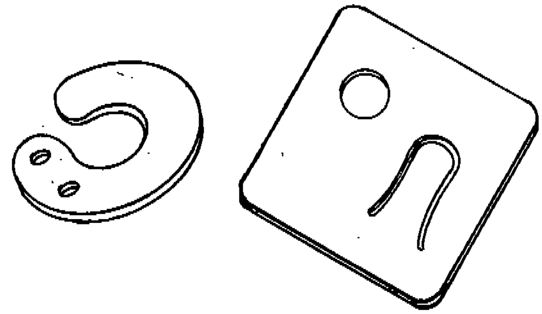

Figure 10

Most valves are tumbled to smooth the edges. Therefore, the more simple shapes are desirable for tumbling.

The material used is usually Swedish stainless steel with mechanical properties of 292,000 PSI tensile strength.

CYLINDER SLEEVE
The cylinder sleeve's basic function is to guide the piston during the compression and intake strokes and act as a reservoir to contain the compressed air.

The material selection can be either cast iron or hard coated aluminum. Cast iron provides an excellent long life bearing surface. The porous surface of the cast iron furnishes a receptive base to accept the transfer of the P.T.F.E. film from the rings and skirts. The cast iron material should be 190-220 Brinnell hardness to improve machining and provide the long wear resistance required.

If a compressor should stand idle for several weeks, a slight film of rust may occur on the cast iron surface. New platings such as porous nickel infused with P.T.F.E. can be applied and provide excellent corrosion protection.

The aluminum sleeve must be hard coated to prevent excessive wear.

The cylinder bore should be honed to a finish of 10 - 20μ for use with P.T.F.E. rings. The cylinder wall surface finish is very important in that too rough a finish will result in rapid wear of the P.T.F.E. rings and skirts, and a finish too smooth will not permit the rings and skirts to depart a mating film on the metal surface.

MOTOR
The motor selected to drive an oil-less compressor should be capable of providing the running torque and also the starting torque required to compress the air to its maximum pressure.

The motor ventilating systems are designed to induce a flow of air through the motor from the outboard end rather than pull the heated air off the compressor and into the motor. The bearings are sized to the compressor load and must withstand the high temperatures developed.

The HP required to compress one CFM of air is based upon the general equation -

$$.0153\, P_1\left[\left(\frac{P_2}{P_1}\right)^{.285} - 1\right]$$

P_1 = inlet pressure

P_2 = discharge pressure

The friction load of the compressor and motor can amount to as much as 50% of the total load.

MOTOR TORQUE

The theoretical torque required for compression versus crank angle (0° at top dead center) as shown in Figure 11 was calculated to determine average torque and peak torque required by the motor. The curves can also be used to aid in determining the flywheel required.

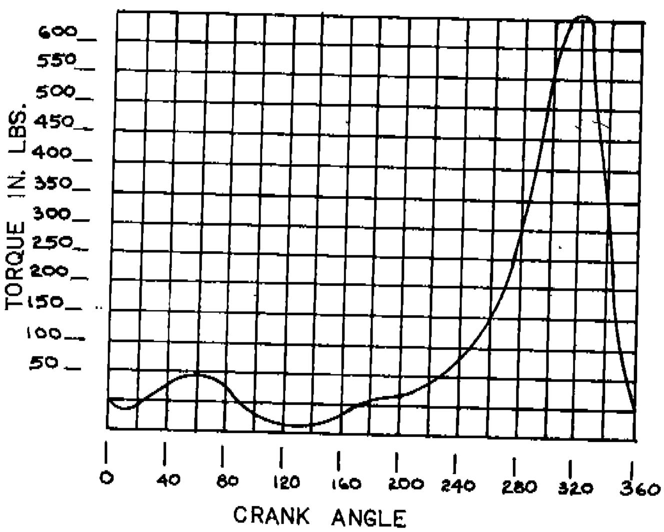

Figure 11

The calculation used assumes adiabatic compression from suction pressure (minus suction valve pressure drop) to discharge pressure (plus discharge valve pressure drop). A clearance volume of a predetermined value is used in the calculation.

The calculation assumes a constant angular velocity of the crankshaft for the kinomatic analysis of the slider crank mechanism. The angular and linear acceleration (including the magnitude and direction of the connecting rod center of gravity acceleration) are used to determine the magnitude and direction of all the inertia forces involved.

A force analysis using D'Alembert's Principle of Superposition used to determine the forces (magnitude and direction) on the piston, connecting rod and crank arm, from which the required theoretical torque can be calculated.

The friction force involved consists of two parts: (1) the ring friction force $(\pi DW)(P)(M)$ which is calculated using the area of contact (for one ring), between piston and cylinder wall, and the pressure in the cylinder for the crank angle in question. (2) the skirt friction force which is the calculated normal force at given crank angle between the piston and cylinder wall, times the coefficient of friction of the skirt material.

D = Ring diameter

W = Ring width

P = Cylinder Pressure

M = Coefficient of friction (Ring)

Analyzing torque curves helps in the design and selection of the motor.

If peak torque (especially for 2 cylinders firing at the same time) is greater than the motor break down torque, then the energy difference must be made up by flywheel addition (or motor redesign). If not, the motor will heat up resulting in higher amperage.

The average torque of the compressor must match the motor torque at the speed in question.

DIAPHRAGM COMPRESSOR

The diaphragm compressor is similar to the reciprocating compressor in that each uses an eccentric or crank to reciprocate a connecting rod.

Due to the diaphragm being an elastomer and very flexible, the diaphragm pumps are usually limited to low pressure (50 PSIG). Their main advantage is simplicity, low manufacturing cost, and low noise level. Since the same design calculations apply to the diaphragm compressor as a reciprocating compressor, this discussion will be limited to description only.

The diaphragm is clamped between the crankcase and the cylinder head and seals the compression chamber.

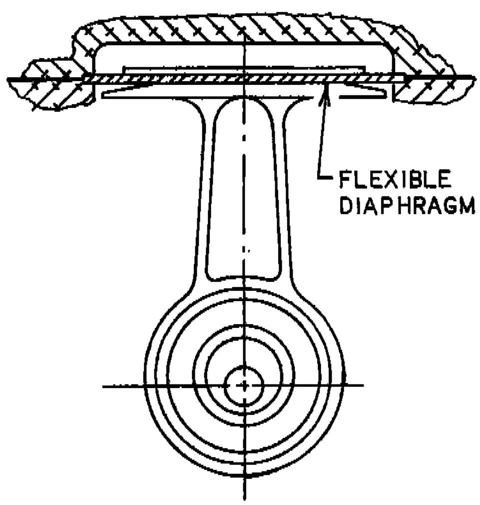

Figure 12

The selection of a diaphragm material is based on the amount of flexing and life required. A commonly used material is a Neoprene outer covering with a nylon type fabric inner member.

OIL-LESS ROTARY VANE VACUUM PUMP AND COMPRESSOR

The oil-less rotary vane compressors and vacuum pumps are usually designed for low compression ratios. Rotary units are capable of pumping large volumes of air, are free of mechanical vibrations, require no valves or lubrication, and generally are smaller in physical size than a reciprocating compressor.

Oil-less rotary vacuum pumps and compressors are built in multivane construction and operate without valves. The pump can be driven by a belting arrangement or driven directly off the motor shaft.

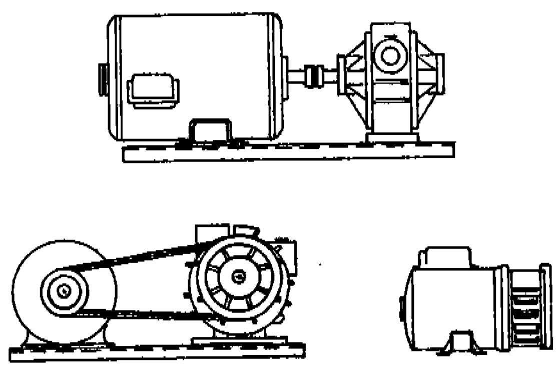

Figure 13

DESCRIPTION

The oil-less rotary vane pump has a concentric rotor and vanes which rotate inside a housing, mounted eccentric to its axis. Relatively high volumetric displacement per revolutions provides fast pumping speeds. The sliding vanes and concentric rotor design provide minimum vibration and extremely quiet operation.

The rotor is free to turn by means of a shaft driven by an electric motor. The sliding vanes in the rotor, moved by centrifugal force, follow the inner wall of the cylinder, and as the rotor revolves, the vanes slide freely in the rotor slots, thus sweeping the air or other gases in the crescent shaped space from intake toward the exhaust. The air or gases are exhausted without the aid of valves, provided the pump has a minimum of 3 vanes. The suction and pressure side is divided and sealed at the point where the rotor and housing are closest to each other.

The pump can be applied as a vacuum pump or as a compressor, single stage or multistage.

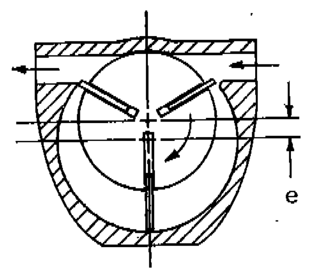

Figure 14

Small and medium capacity pumps are air cooled (100 CFM). Larger units are preferably water cooled.

Current designs are now used in applications up to 50 SCFM and produce a vacuum of 26" Hg or pressure to 20 PSIG. Units of this capacity rating are designed without shaft seals. The operating RPM range is from 500 to 3600 RPM, depending on capacity and design.

PUMP DISPLACEMENT

The displacement or swept volume of a rotary pump is a function of rotor and housing diameter, rotor and housing length, rotational speed and number of vanes.

The relation of rotor diameter to housing diameter is a selected design parameter based on the ultimate application of the pump. As a rule, rotor-housing ratio of 0.8 to 0.9 is generally applied.
The lower ratio is applied to vacuum pumps and compressors up to 15 PSIG and the higher ratio for compressors up to 80 PSIG.

The displacement or swept volume of a multivane rotary pump with radial vane slots is as follows:

$$V_d = 2(R-r) .L.N (2R\pi - t.z)$$

or

$$V = 2 eL N (2R\pi - t.z)$$

Where V_d= Displacement - The relationship (e - R - r) is a design characteristic and depends on the end use of pump

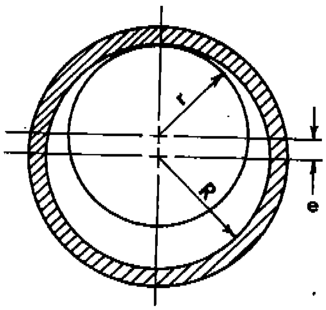

Figure 15

$R - r = e$

$\frac{r}{R}$ = .8 to .9 = design factor
or
e = .2 R (For Low Pressure)
e = .1 R (For High Pressure)
r = Rotor Radius
R = Housing Radius
e = Eccentricity of Rotor In Relation to Housing

(t) The vane thickness is governed by vane material used and the pressure differential the vane has to withstand. On oil-less vacuum pumps or compressors carbon material is successfully used.

(L) Length of rotor and housing is selected in combination with eccentricity (e) for the best cooling efficiency and rigid mechanical construction. The vane material selection will influence the rotor length. A large rotor diameter to length ratio will increase the volumetric efficiency over a smaller rotor diameter ratio. This is due to rotor side leakage loss.

(N) RPM is limited to the allowable surface speed of the vane material against housing surface. Higher surface speed will increase the wear on the vane tip which is a function of pump life. With carbon vanes the maximum recommended surface speed is 2400 ft/min.

(R) Housing radius - The diameter of the housing should be selected in respect to the recommended surface speed of the vanes in contact with the housing inner diameter.

(Z) Number of vanes are selected for:
a. maximum displacement
b. pressure differential acting on vane
c. allowable leakage tolerated

The utilization for maximum capacity can be achieved by proper selection of the number of vanes.

The space enclosed by housing-rotor and the vanes becomes smaller with increasing number of vanes. However, this space is filled times the number of vanes per revolution. As the number of vanes increases, the quantity pumped increases, but the relative gain in this respect steadily diminishes, since the volume of the vanes must be considered.

Illustrations of Pump Displacement With Multiple Vanes.

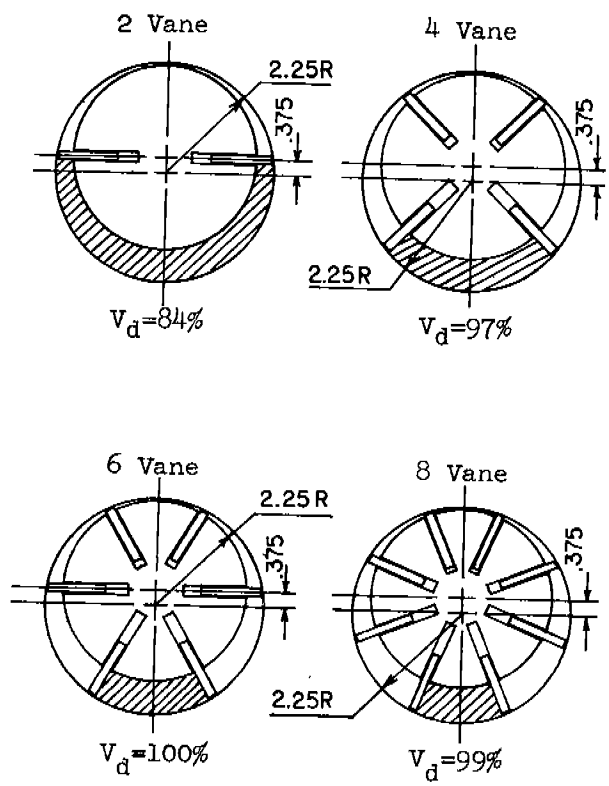

Figure 16

An advantage of having a large number of vanes is that the pressure differential between the adjacent spaces is smaller, resulting in smaller leakage losses ; however, each vane adds to undesirable friction heat, therefore, a maximum of four vanes are applied on vacuum and low pressure pumps. Six vane pumps are dominate in the pressure range of 20 to 60 PSIG. Higher pressure pumps require a larger amount of vanes. The pressure differential between leading and trailing side of the vane should not exceed 20 PSIG to keep leakage across the vanes within practical dimensions.

DESIGN DETAIL OF MAIN COMPONENTS

Rotor

The rotors for pumps with capacities up to 100 CFM are preferably machined from a solid piece of cast iron. Cast iron slugs produced by permanent iron mold processes are preferred. For smaller diameter rotors, slugs produced by continuous cast process are widely used. Certain rotor configurations can be manufactured by the metal sintering method.

The cast iron is heat treated to improve machinability, relieve stresses, and to improve wear resistance. For certain applications the rotor is machined from carbon material. Rotors for pumps with emphasis on long life and greatest reliability, rather than economics, are made of corrosion resistant metal.

The machining of the rotor has to be held within close tolerances. Heat absorbed by the rotor due to compression and friction, causes thermal expansion at a greater rate than in the surrounding housing. Since high volumetric efficiency depends upon minimum clearance between rotor-vane-housing and end plates, precise manufacturing control is required. The depth of the rotor slots should be a minimum of four times the eccentricity between the rotor and housing. A rotor surface finish of 32 microinches is desirable except for the finish in the rotor slots which requires 16 microinches.

The rotor slot edges are machined with a radius. The trailing side of the rotor slot is perfectly blended with the edge radius to achieve a polished finish in the transition area, since service life of a rotary carbon vane pump depends mainly on the surface finish of the mating parts.

Housing

Cast iron is widely used for the cylinder housing, however, sintered metal materials are successfully being applied to smaller sizes.

The pump housing is designed to assist in cooling since it represents the major component for convenient heat removal created by the pump. As a common rule the intake and exhaust porting form an integral part with the pump housing.

The inner surface of the housing should be honed to a minimum of 16 microinches. This surface can also be improved for the benefit of vane life, leakage and smooth operation by hard-coating, hard metal spraying, induction hardening as well as treating by a hot quenching method.

Vane

In oil-less rotary pumps a carbon composition vane material is successfully used. Vanes are heated not only by the compressed gas, but also by friction.

The outstanding advantages of carbon vanes over other materials are the self-lubricating properties and low specific gravity.

For vacuum or low pressure applications other solid lubricants are included in the composite basic vane material. The low expansion rate of carbon material contributes to dimensional stability, which is a remarkable advantage. The low coefficient of thermal expansion makes it possible to maintain desired clearances between vane end-plate and vane rotor slot. The vane has to be machined to precise dimensions.

End clearance (gap between vane and end-plate) should be .0005 to .0015 depending on length and vane thickness, in order to obtain minimum leakage. The leading edge of the vane should be radiused to prevent chipping and to reduce carbon dusting during run in. The maximum differential pressure acting on the vane should not exceed 20 PSIG. Surface speed at the vane tip should be less than 2400 ft/min. Since the vanes are moved outward by centrifugal force, care should be taken not to exceed a specific vane tip load rating of 12 PSIG.

SUMMARY

The design of oil-less compressors and vacuum pumps involves the correct selection and use of dry lubricant materials, dissipation of heat, and the selection of parts than can operate with out oil or grease lubrication.