12-2003

Design and Development of a Lubrication Pump for a Horizontally Mounted Air-Conditioning Compressor.

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Design and Development of a Lubrication Pump for a Horizontally Mounted Air-Conditioning Compressor

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A thesis

presented to

the faculty of the Department of Technology

East Tennessee State University

In partial fulfillment

of the requirements for the degree

Master of Science in Technology

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by

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December 2003

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Keywords: oil pump, lubrication, compressor, horizontally mounted, rotating vane pump, dry priming

ABSTRACT

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Horizontally mounted compressors offer the advantage of reduced height in central air-conditioning units but prove difficult to produce economically due to costs associated with the manufacture of acceptable lubrication systems for the compressors. This study develops an effective, affordable oil pump for use on a horizontal compressor. Concepts are proven through testing of prototype assemblies. Test results drive modifications for future prototypes, and prototypes demonstrating adequate performance are modified for ease of manufacture.

Research in this study proves that the most suitable design results from a modification of a rotating vane pump. The pump's modifications enable it to pump oil in the same direction, regardless of the direction of shaft rotation and to prime itself when totally dry of oil. However, extensive use of horizontal compressors hinges upon the development of a satisfactory suspension system.

CONTENTS

Chapter Page

Chapter Page

Chapter Page

LIST OF FIGURES

CHAPTER 1

INTRODUCTION

The highly competitive nature of the refrigeration and air-conditioning industry creates an atmosphere in which systems manufacturers must continuously improve their products in order to maintain or improve their current market shares. Environmental concerns and higher energy costs persuade governments in many parts of the world to require more efficient systems. Thus, customers demand systems that cost less at purchase and operate at lower costs than in the past. The challenge faced by compressor manufacturers is producing products that reduce the system manufacturers' applied costs. A winning compressor design reduces the sum of the costs of purchase of the compressor and the purchase and installation of other required components in the air conditioning or refrigeration system.

One method of reducing the system manufacturers' applied costs is to provide them with an affordable horizontally mounted compressor. The height of the compressor determines the height of most air conditioning systems. Therefore, the system manufacturers either add more coil to the condensers than necessary or add sheet metal to the sides of their units in order to properly direct air flow through the condenser coils. In either case, the added cost is not desirable and is evidence of a design that cannot use component sizes to maximize efficiency of manufacture and eliminate waste. A horizontal compressor reduces the height of the compressor to the point that overall unit height is no longer determined by the height of the compressor, and the amount of expensive copper coil for the condensing unit is reduced to the minimum required for the unit's cooling capacity. The use of a horizontal compressor also promises to reduce or completely eliminate sheet metal skirting around the perimeter of the unit.

Statement of the Problem

Horizontal compressors are not in widespread use at present due to the design challenges inherent in changing a compressor's natural vertical orientation. The most significant problem with orienting a compressor horizontally is the expense of using a different method of delivering lubrication to the bearings. Current horizontal lubrication systems make the compressors prohibitively expensive and limit the use of horizontal compressors to small capacity refrigeration units and other applications in which space limitations justify the added cost to the compressor.

Purpose of the Study

The purpose of this study is to design and develop a cheap and effective lubrication system in such a way as to avoid massive assembly and manufacturing line overhauls. The manufacture of a truly viable horizontal compressor that promises to be competitive in almost every market requires such a lubrication system. By matching the cost, efficiency, and sound levels of other designs, the product can gain a substantial portion of the market by taking advantage of its orientation to reduce the required size of the system manufacturers' units, as well as their applied costs.

Assumptions, Limitations, and Procedures

Assumptions

The assumptions listed below are made in this study:

1. It is assumed that units using horizontal compressors will be installed on a surface that is level within five degrees.

2. It is assumed that a horizontal compressor's oil volume will be large enough to compensate for mixing with the refrigerant of any unit in which it is installed.

3. It is assumed that the oil used in a horizontal compressor will be compatible with the refrigerant to be compressed.

4. It is assumed that the finished compressor's mechanism will not be exposed to air for extended periods of time.

5. It is assumed that the finished compressor's mechanism will not be exposed to water.

6. It is assumed that the compressor will be correctly installed into the unit.

7. It is assumed that the compressor will be powered by a source having the frequency and voltage required by the compressor's motor.

<u>Limitations</u>

This study is constrained by the following limitations:

1. The lubrication system must be fabricated in such a way as to allow fabrication and assembly on the current facilities at Bristol Compressors.

2. Materials that are used in the system must be inexpensive.

3. The system must be inexpensive to fabricate.

4. Efficiency must be comparable to or better than that of a vertical compressor of the same model.

5. Sound levels must be comparable to or better than those of a vertical compressor of the same model.

6. Cost must be comparable to or better than that of a vertical compressor of the same model.

7. The lubrication system must pump oil in the same direction, independent of the direction of shaft rotation.

Procedures

The following procedures will be used in this study:

1. A review of related patents is to be conducted and steps are to be taken to secure patents on original ideas as they are developed in the design process.

2. New designs are to be conceptualized and analyzed.

3. Prototypes of the most favorable design are to be fabricated and tested for performance.

4. Data collected during testing will be compiled and analyzed.

5. If a prototype fails to perform adequately, another prototype is fabricated and tested.

6. Prototypes that perform adequately will be redesigned for feasibility of manufacture.

7. Conclusions will be stated.

8. Recommendations will be stated.

CHAPTER 2

LEGAL CONSIDERATIONS

Patent Search

In the process of developing a feasible design for the horizontal oil pump, one must ensure that no infringements are made on patents belonging to entities other than Bristol Compressors, Incorporated. Consequently, the services of a patent attorney are necessary. Don Spurrell, Attorney at Law (Johnson City, Tennessee, Phone: (423)929-1700) bears the responsibility of conducting patent searches pertinent to any designs proposed during this study.

Patent Application and Approval

Due to the originality of the designs proposed in this study, prudence dictates obtaining some form of legal protection from the theft of the inventor’s intellectual property. Accordingly, an idea record has been submitted and a patent has been granted by the United States Patent Office (United States Patent 6,422,346). Don Spurrell, in close association with the author, conducted all legal work associated with the patent.

CHAPTER 3

DESIGN CONCEPTUALIZATION

Rotating Vane Pump

Rotating vane pumps work well in horizontal, semi-hermetic refrigeration compressors. Such pumps reverse the direction in which oil is pumped when the direction of shaft rotation is reversed. Thus, rotating vane pumps in conventional configurations are not acceptable for consideration in this study. However, a rotating vane pump may be feasible if the mechanism can be redesigned in such a way as to make its pumping direction independent of the direction of shaft rotation.

Standard Rotating Vane Pump Operation

The suction and discharge pressures in a rotating vane pump are generated by changing the volume constraining the pumped fluid. As the shaft depicted as Item 1 in Figure 1 rotates (clockwise in this example), centrifugal force slings the vanes (Items 2 and 3 in Figure 1) away from the center of the shaft and forces the end of each vane farthest from the center of the shaft to maintain contact with the inside surface of the pump chamber (Item 4 in Figure 1). Referring to the volume designated as Item 5 in Figure 1, one observes that the volume possessed by Item 5 increases as the shaft rotates from the position depicted in Frame 1 to those depicted in Frames 2 and 3. Consequently, the pressure in Item 5 reduces, and the pumped fluid flows into Item 5 through the suction port (Item 6 in Figure 1). Item 5's volume begins to reduce from Frame 3 through Frame 4, and the pressure in Item 5 increases. The increase in pressure in Item 5 causes the pumped fluid to flow out of the pump chamber through the discharge port (Item 7 in Figure 1). A rotating vane pump may have one or more vanes, but the concept of operation remains unchanged. The number of vanes equals the number of pockets of fluid to be pumped during one shaft rotation. By observing Figure 1, one easily comprehends that Item 7 becomes the suction port and Item 6 the discharge port upon reversal of the direction of shaft rotation. Such a characteristic renders the pump depicted in Figure 1 unacceptable for use in the application associated with this research.

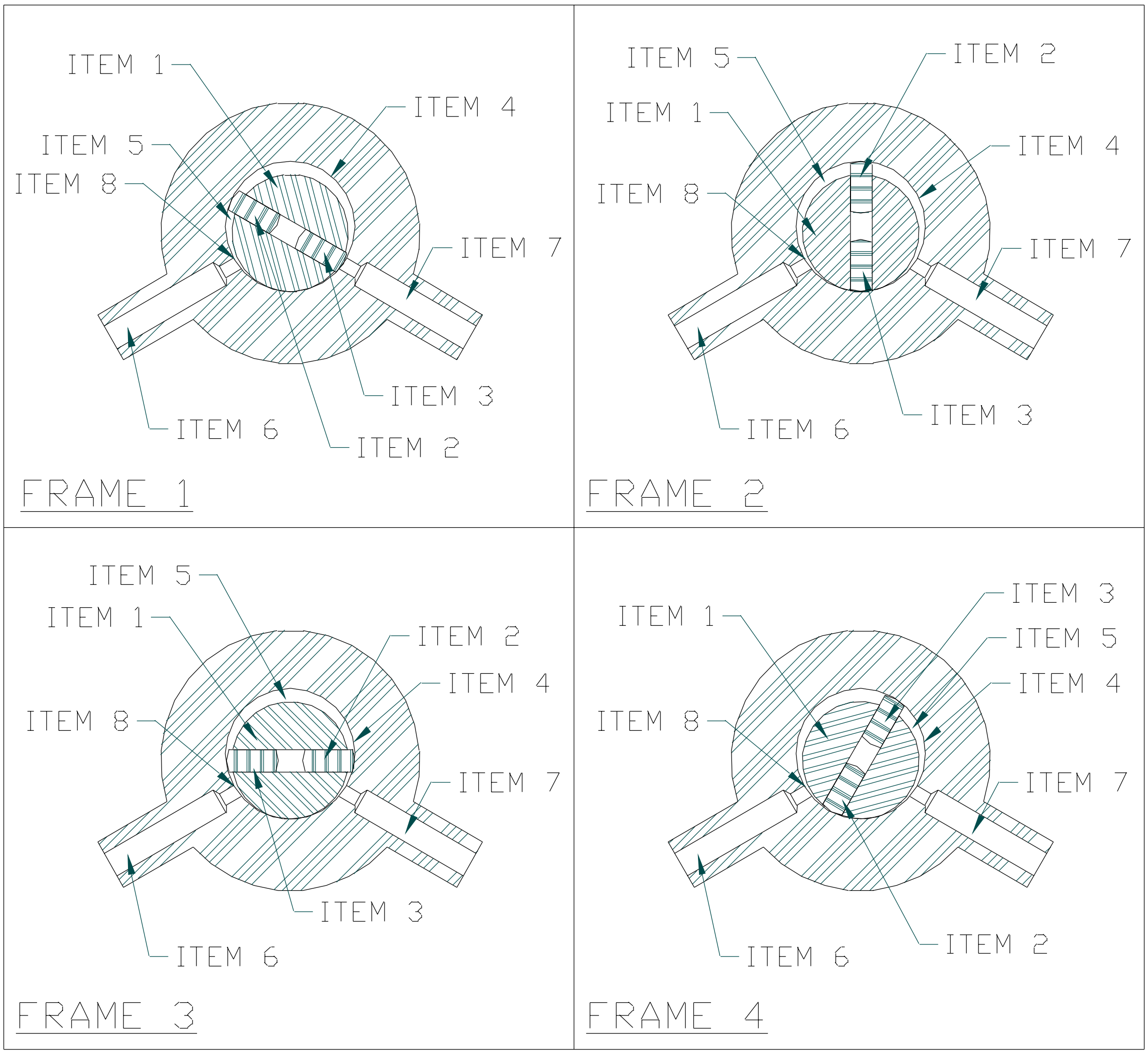


Figure 1. Schematic of Operation of a Traditional Rotating Vane Pump.

Pump Design Modifications for Rotationally Independent Pumping Direction

Twin Vane Pump

Concept of the Modifications. Suction and discharge ports are normally positioned at the end of the crescent formed by the pump chamber and the shaft (Item 8 in Figure 1). The port position causes the direction of shaft rotation to dictate which port conducts suction flow and which port conducts discharge flow. Thus, a feasible means of developing a rotating vane pump whose pumping direction is independent of the direction of shaft rotation may be to devise a mechanism which reverses the positions of the ports themselves when the direction of shaft rotation changes.

One possible method of repositioning a port in a rotating vane pump is to locate the port on the shaft. Cutting the slot in which the vanes ride (Item 1 in Figure 2) to a width substantially larger than the width of the vane (Item 2 in Figure 2) places the pump chamber (Item 7 in Figure 2) in direct communication with holes drilled through the center of the shaft to supply bearings (Item 4 in Figure 2). Each vane (Item 2 in Figure 2) is free to position its trailing edge against the trailing edge of the slot in which it is installed (Item 5 in Figure 2) and leave a gap between its leading edge and the slot (Item 6 in Figure 2). Upon a reversal in shaft rotation, the vane (Item 2 in Figure 2) moves to the opposite side of the slot, automatically placing the gap in front of the vane's leading edge. Because the pump's high pressure side (Item 7 in Figure 2) always locates to the front of the vane's leading edge, the hole that supplies oil to the bearings (Item 4 in Figure 2) is always exposed to discharge from the pump, regardless of the direction of shaft rotation.

The configuration depicted in Figure 2 allows for a discharge port to be repositioned when the direction of shaft rotation changes, but the suction port must also be constructed in such a way as to function correctly when the rotation of the shaft reverses. Although the momentum of the pump's components during shaft rotation does not lend itself to constructing a suction port in the manner used for discharge ports, the suction port can be made to supply oil to the pump when the shaft rotates in either direction. Placing the suction port (Item 8 in Figure 2) at the pump chamber's widest point, the port supplies oil to the pump, regardless of the direction of shaft rotation. One must note that such a configuration forces the pump to attempt to draw a vacuum behind each vane for half of the vane's travel around the pump chamber, resulting in a loss of pump efficiency.

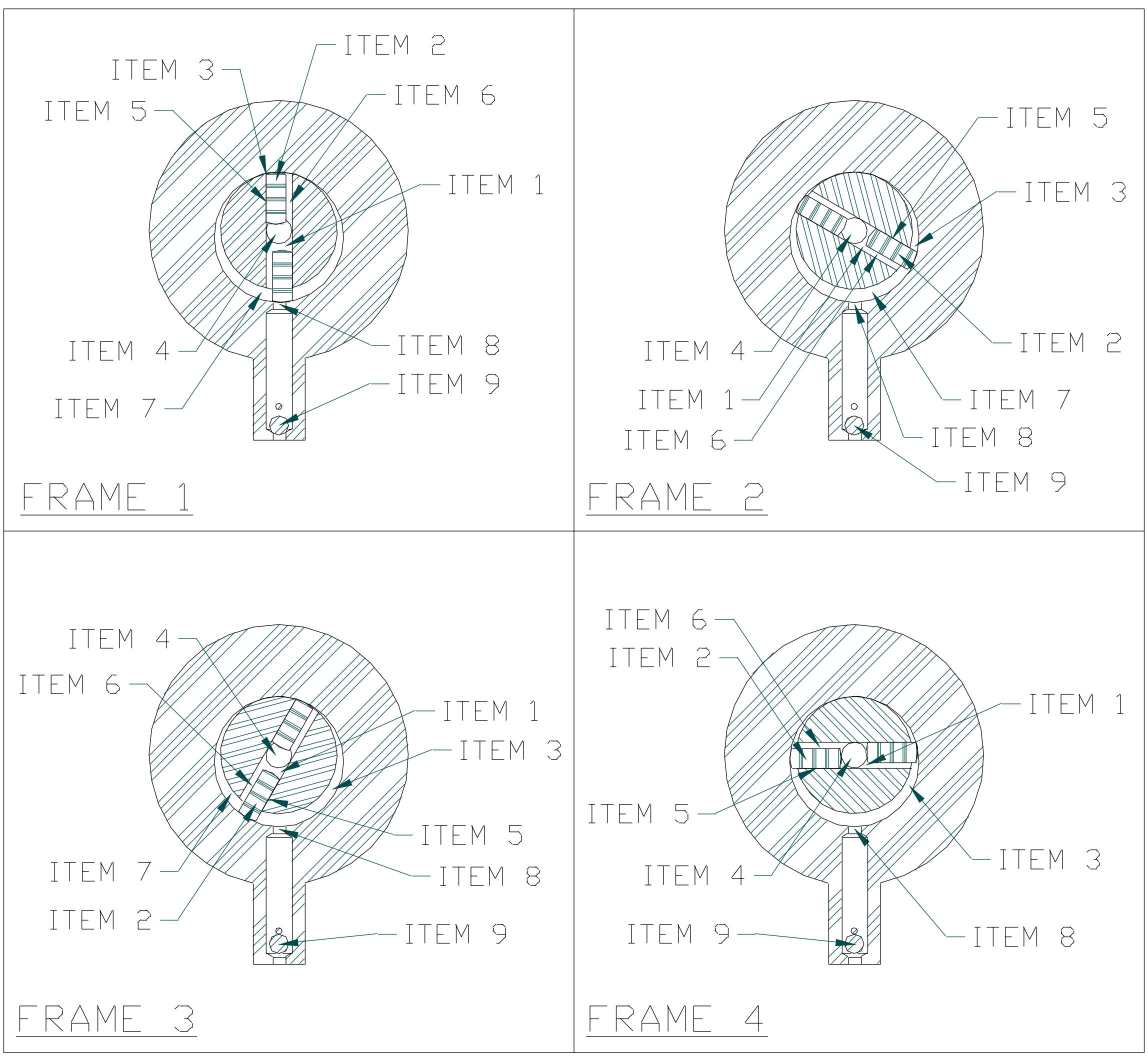


Figure 2. Schematic of Operation of the Suction Phase of a Proposed Twin Vane Rotating Vane Pump.

Concept of Operation. Referring to Figure 2, one notes that the volume denoted as Item 3 continues increasing as the shaft rotates from the position shown in Frame 1 to those shown in Frames 2, 3, and 4, respectively. From Frame 1 to Frame 2 the pump attempts to draw a vacuum in Item 3, for no suction port communicates with Item 3 in either frame. In Frame 3 one of the vanes (Item 2 in Figure 2) has passed over the suction port (Item 8 in Figure 2), allowing direct communication between the suction port and Item 3. Thus, pressure pushes up on the ball check valve (Item 9 in Figure 2), and fluid flows through the suction port and fills Item 3. As the shaft continues to rotate and the volume of Item 3 increases, the suction port continues to fill Item 3 until reaching the position depicted by Frame 4. Suction activity ceases in Item 3 at Frame 4.

Progression beyond Frame 4 of Figure 2 results in a reduction in the volume of Item 3 and is illustrated in Figure 3. As shaft rotation continues from Frame 1 to Frame 2 in Figure 3, the pressure in Item 3 increases and forces the ball check valve (Item 9 in Figure 3) to shut, preventing return flow through the suction port. Therefore, all fluid in Item 3 must flow through the discharge port created by the gap between the pump vane and the slot in the shaft (Item 6 in Figure 3) and into the lubrication hole at the center of the shaft (Item 4, Figure 3).

Single Vane Design

Figure 4 depicts a pump using only one vane. This design operates in a manner identical to the pump shown in Figures 2 and 3 but realizes a cost savings by eliminating the need for purchasing one of the vanes. Additionally, the single vane design further reduces cost by reducing the amount of metal to be cut from the shaft in order to make the slot; the slot in the shaft on the single vane design measures half the length of that in the twin vane design. The single vane pump also offers the possibility of improving on the reliability of the twin vane pump, for the single vane pump is composed of fewer moving parts that can wear, break, or become misplaced during assembly.

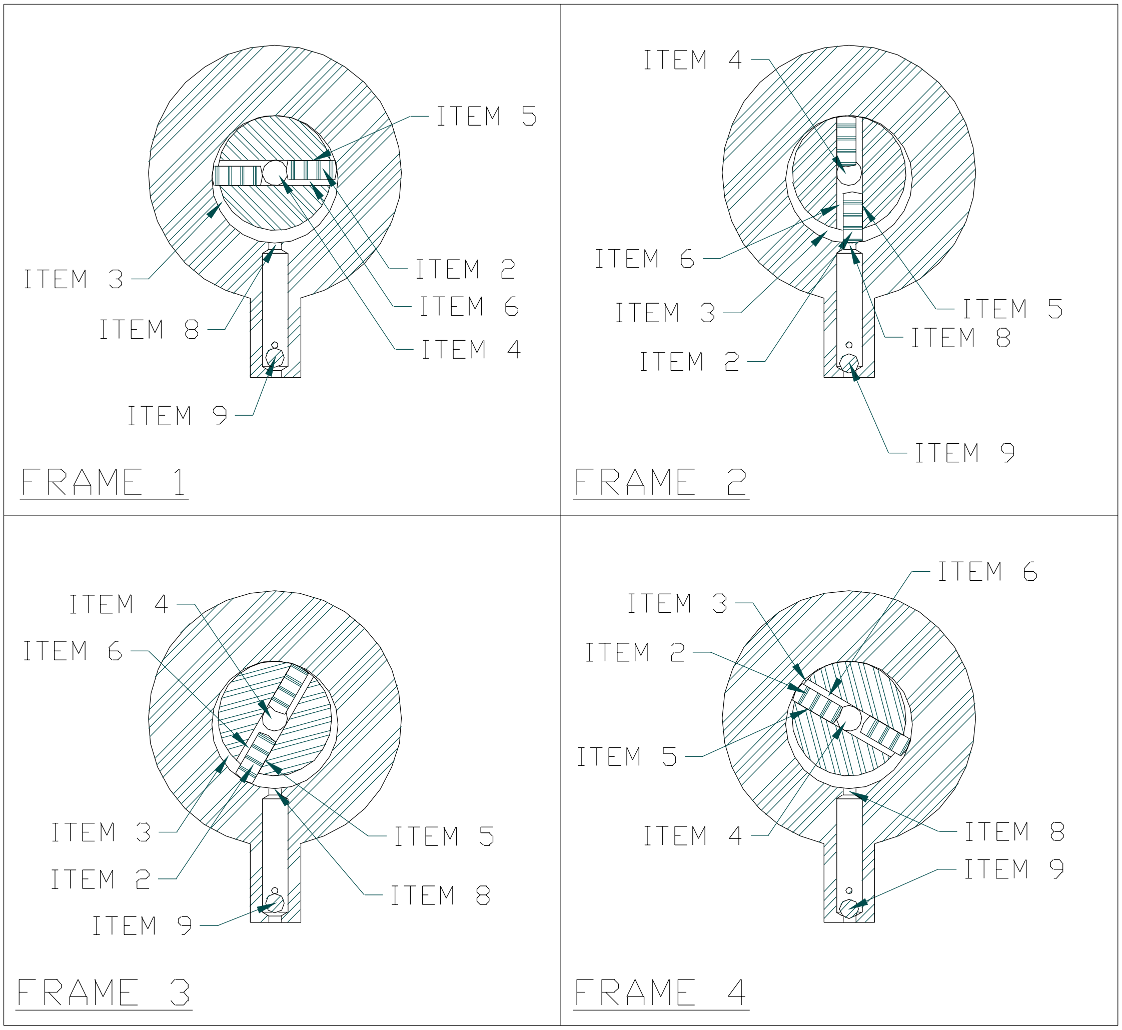


Figure 3. Schematic of Operation of the Discharge Phase of a Proposed Twin Vane Rotating Vane Pump.

A single rotation of the shaft in the single vane pump moves one volume of fluid from the suction port to the discharge port. The twin vane pump transports two smaller volumes of fluid through the same locations, and the significance of this difference is not apparent. If the small clearance between the shaft and the pump chamber (Item 1 in Figure 4) fails to provide adequate resistance to flow from Item 2 to Item 3 in Figure 4, then equalization of pressure on each side of the vane will occur and cause the pump to lose suction.

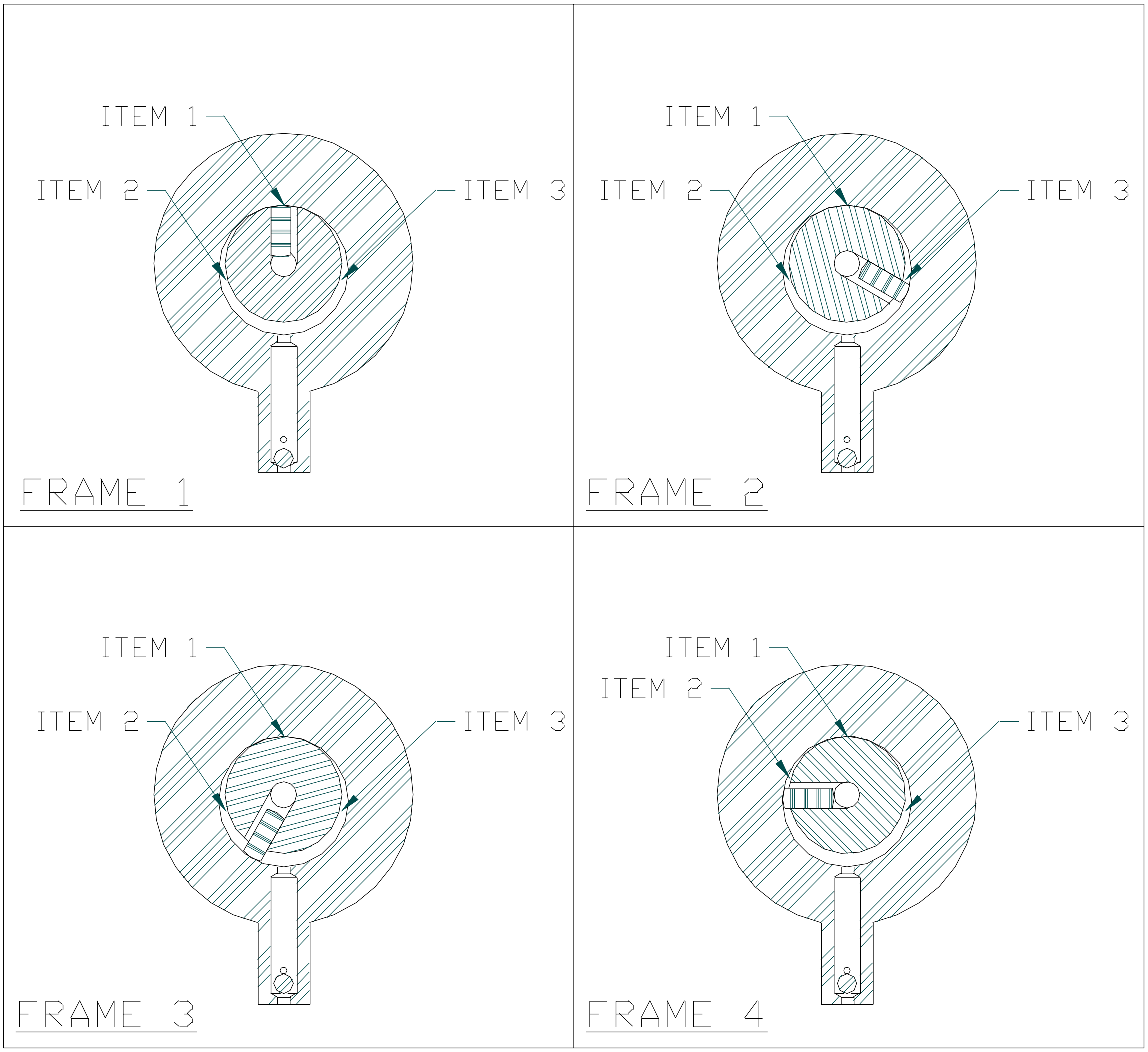


Figure 4. Schematic of Operation of a Proposed Single Vane Rotating Vane Pump.

Valveless Design

Another possibility exists for reducing cost and simplifying assembly for the rotating vane pump. By removing the ball check valve from the assembly, fewer moving parts exist to malfunction or get misplaced during assembly. Reliability then improves as cost reduces, if the pump continues to function adequately without a check valve installed. Referring to the description of the pump's operation above, one notices that removing the check valve does not prevent flow from entering the suction port during the suction stage of the pump's operation (Frames 1 through 4 of Figure 2). However, the valve does come into play during part of the discharge stage (Frames 1 and 2 of Figure 3) by preventing return flow through the suction port.

Some attention must now be focused on the action taking place at the suction port during the discharge stage of operation in a valveless pump. As the pump progresses from the position depicted in Frame 1 to that depicted in Frame 2 of Figure 5, the volume of Item 3 reduces and forces fluid to flow through the discharge port (Item 6 in Figure 5) and the suction port. The flow of fluid back through the suction port results in lost work in the process, but the open discharge port allows a small amount of fluid to flow to the lubrication hole in the shaft (Item 4 in Figure 5). Therefore, the capacity of the pump must be made large enough to absorb the lost work and ensure adequate flow to the bearings. From Frame 2 through Frames 3 and 4, a vane (Item 2 in Figure 5) prevents communication between Item 3 and the suction port. During this stage of operation, the pressure in Item 3 continues to increase and forces all fluid flow to progress through the discharge port and into the lubrication hole in the center of the shaft.

Although the potential exists for preserving the functionality of the pump without using a check valve, the valve's absence creates a leak path from the lubrication hole to the suction port and the atmosphere outside the pump. As mentioned previously, fluid flow reverses in the suction tube as the shaft rotates from the position in Frame 1 to that in Frame 2 of Figure 5. Such an open path sets an upper limit on the pressure that the pump develops and reduces the volume of fluid it can move in a given time. In the case of the twin vane pump, the leak path is open through each discharge port for one half of a shaft revolution. Placement of the vanes dictates that a leak path through one discharge port exists at any time during shaft rotation. The single vane design exhibits a similar leak path through its discharge port, but the existence of only one discharge port in the pump guarantees that no leak path exists for one half of one revolution of the shaft (Compare Figure 4 with Figure 5). Thus, the single vane pump appears capable of producing greater discharge pressure than the twin vane design.

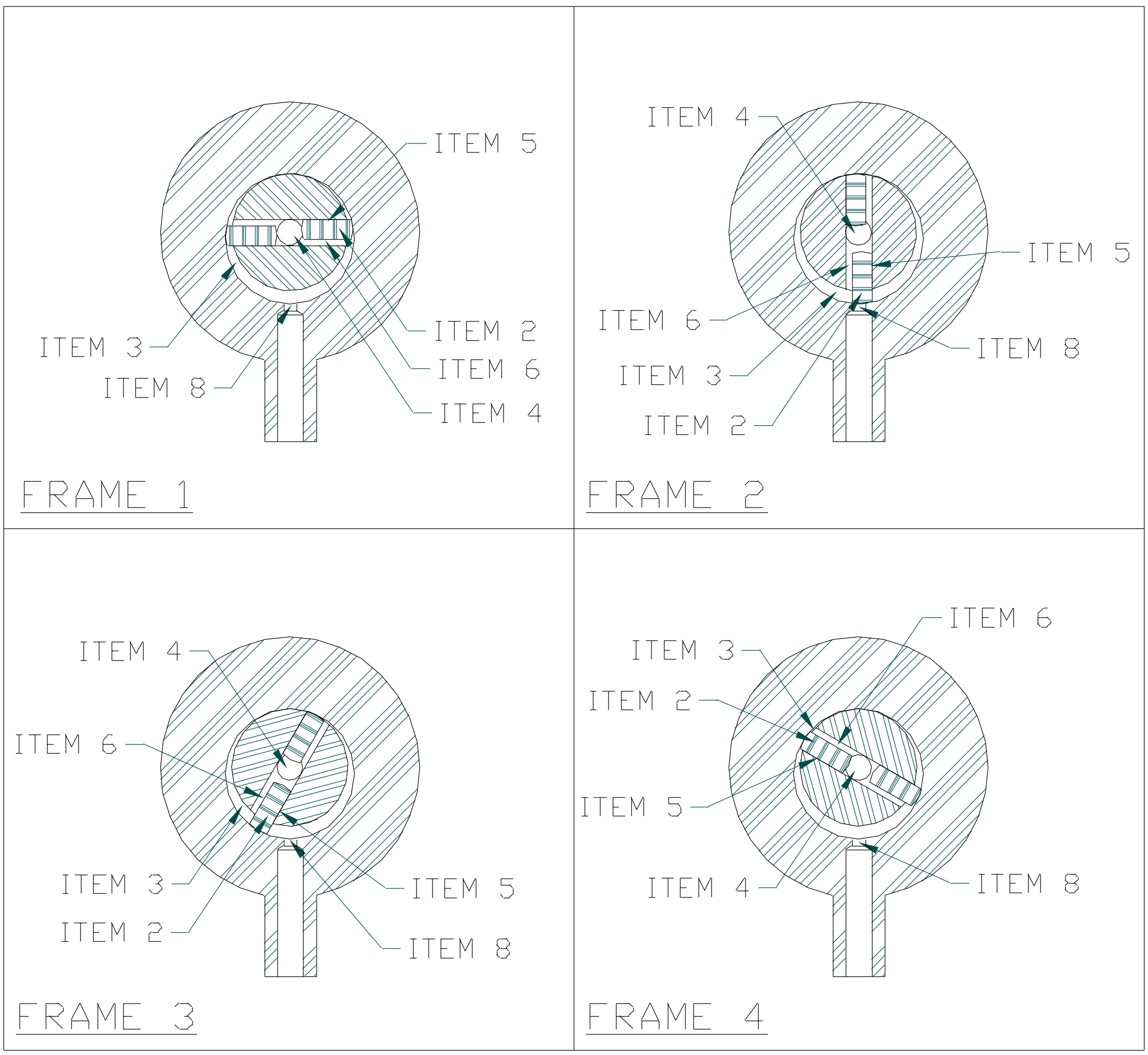


Figure 5. Schematic of Operation of a Proposed Valveless Twin Vane Rotating Vane Pump.

Dry Priming Considerations

At this point in the design conceptualization, the engineer must consider the requirement of designing a pump that primes itself while its moving parts are totally dry. In order for the pump to prime itself while dry, it must possess the capability to pump a gaseous fluid. Pumping a compressible fluid poses challenges that require modification of a pump that generally pumps incompressible liquid fluids. Moreover, the pump must not only pump a gas, it must simultaneously develop sufficient vacuum in the suction tube to allow the ambient pressure outside the oil pump to push oil through the tube and into the pump chamber. One must also note that the pump has no valves with which to prevent reverse flow through its mechanism, and the discharge port allows free communication between the pump chamber volume located in front of a vane's leading edge (see Figure 2) and the lubrication hole. Thus, some part of the pump chamber is open to the entire volume of the compressor's lubrication system at all times.

Consequently, the pump chamber pumps a compressible fluid into the lubrication system, which is of a much larger volume than that of the pump chamber. The tight bearing tolerances in the lubrication system upstream of the pump offer enough flow resistance to cause the pump to compress the gas pumped into the lubrication system, but the difference between the pump chamber volume and the lubrication system volume prevents adequate compression to force gas through the bearing clearances in significant amounts. Hence, the gas simply rushes back into the pump chamber when shaft rotation carries the vane beyond the narrowest part of the pump chamber.

The expanding gas raises the chamber pressure and inhibits formation of a vacuum in the suction tube, and the pump is forced to repeatedly compress the same small volume of gas with no hope of pulling oil into its mechanism. Although some reverse flow does occur while pumping liquids, the incompressible liquids produce higher pressures when forced into the lubrication system's volume than those produced by gases. When pumping liquids, the pump produces sufficient pressure to force significant amounts of fluid through the bearing clearances, thereby reducing the amount of fluid in the lubrication system to the point that the incompressible liquid cannot expand sufficiently to reverse flow and refill the pump chamber.

One method of enabling a pump to prime from a dry state involves providing an avenue of escape for gases in the lubrication system. Drilling a small hole in the lubrication system at any point upstream of the pump vanes allows any gas pumped into the system to escape through the hole (see Figure 6). Such a vent hole prevents gas compression in the lubrication system; lack of compression leads to reduced expansion of the gas into the pump chamber and results in increased vacuum in the suction tube as gas continues to travel through the suction tube and the pump. Eventually, the vacuum reaches such a value that oil is forced up the suction tube and into the pump by the ambient pressure of the atmosphere outside the oil pump. At this point, the pump is primed and begins pumping oil. Oil leaks through the bleed hole after the lubrication system fills with oil, but allowing gas to escape the system does not require a bleed hole of a sufficient diameter to prevent the oil pump from supplying both the lubrication system and the bleed hole leak.

Prudent location and sizing of the bleed hole allow its use in more functions than that of priming the pump. If the hole is of a large enough diameter and is drilled in the correct location, it can be used to aid in priming, regulate pressure, and filter oil. Pressure regulation is dependent only on the size of the bleed hole. As bleed hole diameter increases, oil leakage through the bleed hole increases, and the pressure in the lubrication system decreases. The use of the bleed hole as an oil filtration system is sufficient in and of itself to warrant a separate study, and such a system is not fully developed in this publication. However, allowance is made for the addition of an oil filtration system to be added to the pump, and a proposed filtration system design using a bleed hole in the oil pump is discussed below.

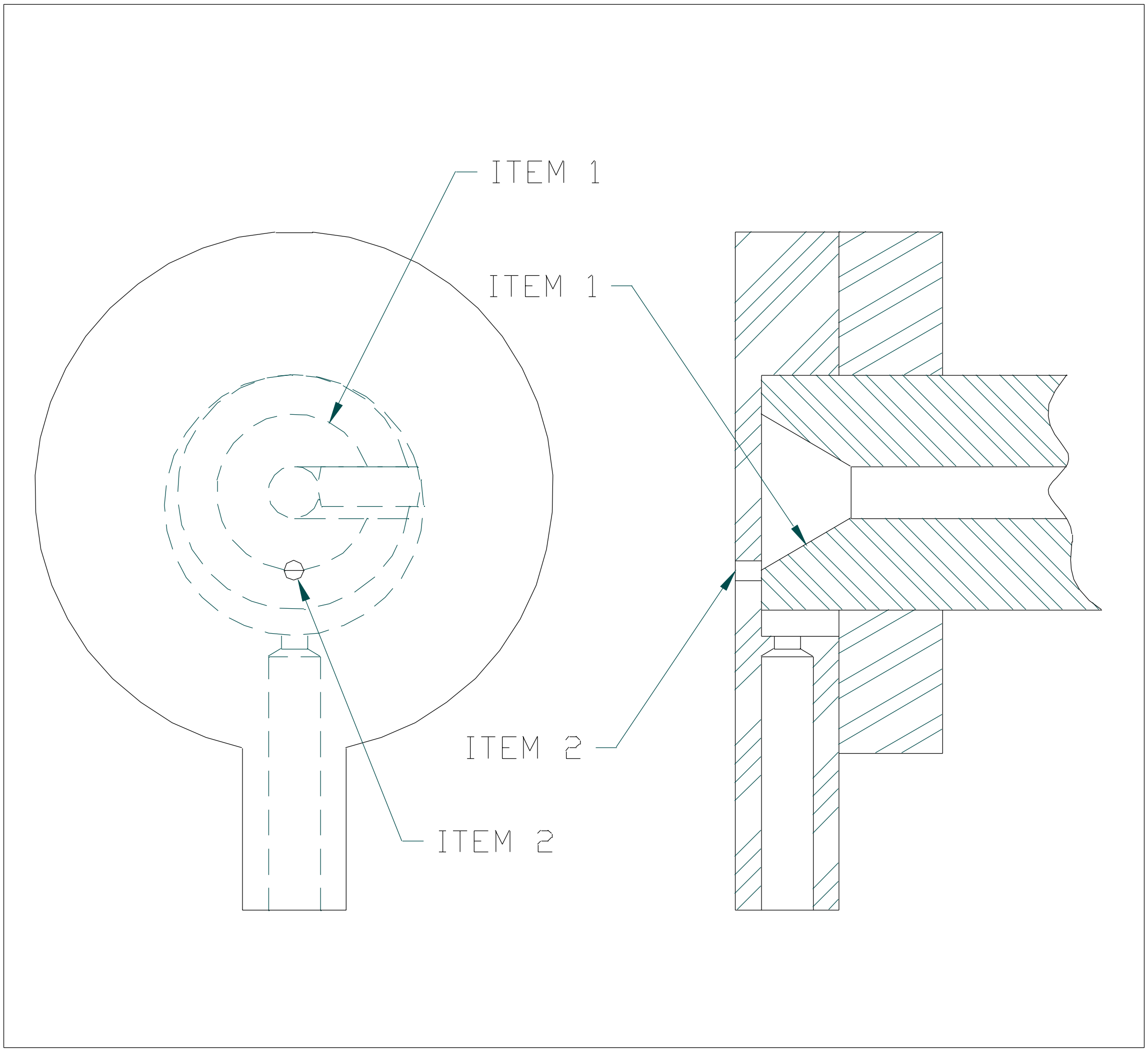


Figure 6. Diagram of the Centrifuge and Bleed Hole Locations.

Oil Filtration System

Journal bearings depend upon a clean supply of oil to function properly, but some debris occasionally finds its way into air-conditioning systems during the assembly process. Preventing such debris from elevating compressor failure rates necessitates the installation of a system to remove debris from the oil while it is outside the mechanism.

The system devised for this project depends upon the depth of the end of the suction tube below the surface of the oil to prevent debris from entering the pump. If the suction tube opening is deep enough in the oil, any floating debris remains in the oil reservoir as the suction tube pulls oil from the depths of the oil reservoir. However, the tube opening must sit at a position high enough above the bottom of the oil reservoir to prevent the suction tube oil flow from entraining debris from the bottom of the reservoir and pulling it into the pump. Ideally, a suction tube of the correct length and a reservoir of the correct depth result in a condition which leaves all material more dense than oil on the bottom of the reservoir for the life of the compressor. Furthermore, any material less dense than oil remains floating on top of the oil.

A mechanical separator provides further protection for the bearings in the compressor. The separator designed for use with the pumps described above uses centrifugal force to generate a false gravity field in order to separate oil from more dense materials. Cutting a conical cavity in the bottom of the shaft (Item 1 in Figure 6) creates a volume which acts as a centrifuge to any fluid traversing the distance between the vane and the lubrication hole at the center of the shaft. Oil and any entrained debris entering the centrifuge spins at a rate of speed sufficient to force the more dense debris to move to the outside of the centrifuge. Seeking the outermost point from the axis of revolution, the debris moves along the conical wall of the centrifuge and away from the lubrication hole. A small bleed hole (Item 2 in Figure 6) in the pump housing corresponds with the largest diameter of the cone (its base) and releases the debris from the pump. Oil, as well as debris, flows through the bleed hole, and the capacity of the pump must be great enough to supply oil flow to the bleed hole and the bearings.

Figure 7 demonstrates the utility of placing a small volume as a dirt trap immediately outside the bleed hole in the pump. Provided that neither the bleed hole (Item 1 in Figure 7) nor the exit vent (Item 2 in Figure 7) resides at the lowest point in the trap, debris sinks to the bottom of the trap and remains so indefinitely. An adequately designed dirt trap of sufficient volume promises to remove all debris not residing on the surface of the oil or in the bottom of the reservoir.

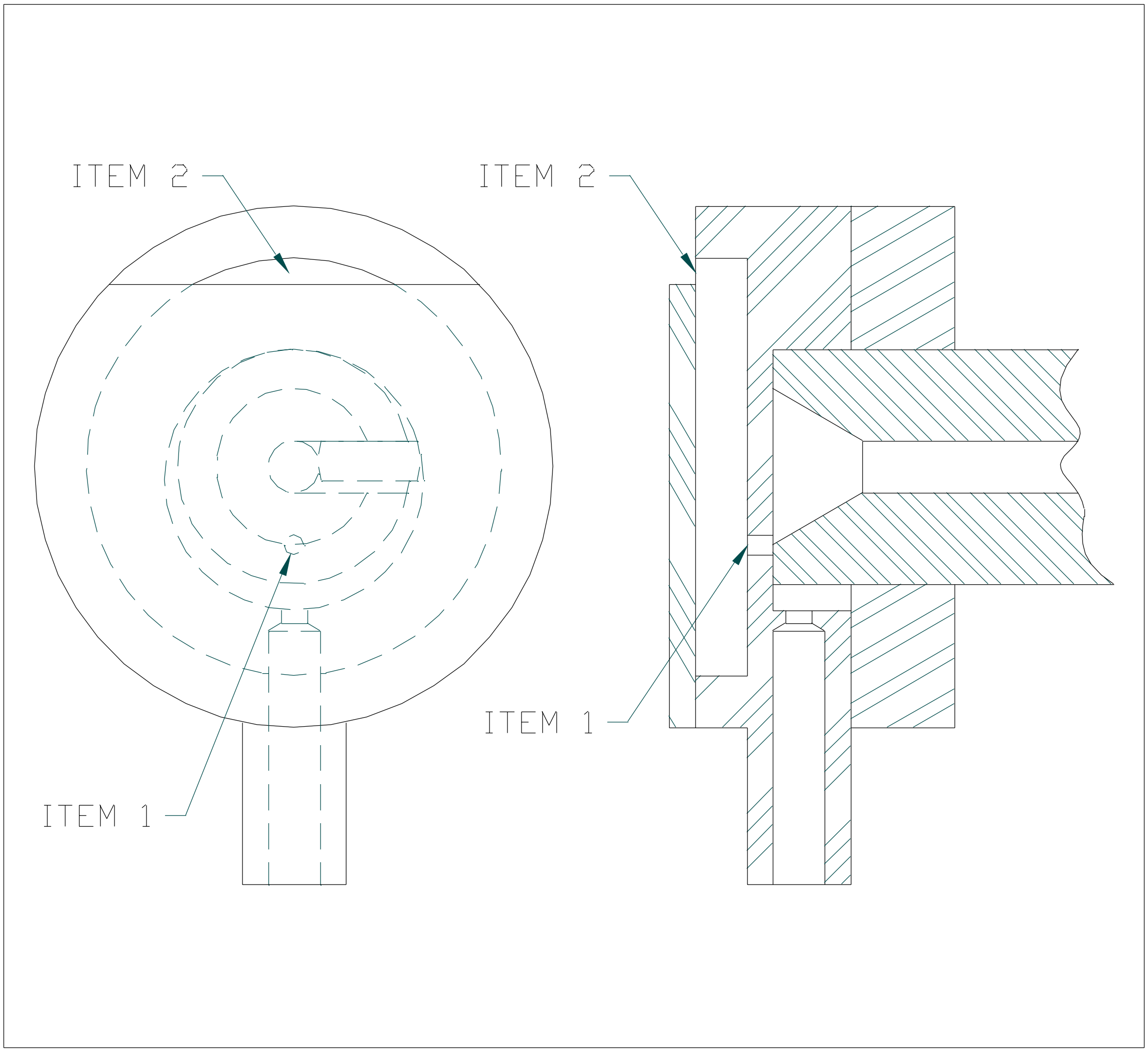


Figure 7. Diagram of the Dirt Trap Showing the Vent Location.

CHAPTER 4

MATERIAL SELECTION

Cast Iron

The low purchase cost of cast iron enables it to compete as a material for various components in compressors. Cast iron also exhibits favorable vibration damping characteristics and wear resistance. Additionally, cast iron machines more rapidly than steel and requires no cutting fluid to aid in machining. Thus, the use of cast iron eliminates the cost of cutting fluid. However, the expense of salvaging cast iron scrap renders recovery of chips unprofitable in many situations, and the inability to cast intricate shapes using cast iron often adds machining costs not possessed by parts produced using other materials. The yield strength of cast iron prohibits its use as a shaft, and casting the ball in the check valve is difficult. Therefore, cast iron will be considered only as a last resort for use as a material for the pump housing or vanes.

Die-cast Aluminum

Die-cast aluminum (380 series) is one of the cheapest manufacturing materials for large volumes of parts. Furthermore, die-cast aluminum machines almost three times as fast as cast iron and produces recyclable chips. Journals made from die-cast aluminum also perform well as bearings when mated with hardened steel. The limited capabilities for die casting intricate shapes to tight tolerances using die-cast aluminum necessitates machining, and the surface finishes desired for use in aluminum bearings require the use of cutting fluid and expensive diamond cutting tools. Aluminum's low yield strength prevents its use as a shaft, and the line contact between the housing and the vanes (see Figure 8) prevents the use of a soft metal such as aluminum as a vane material. However, the pump housing can be made of aluminum, provided that a vane of a relatively soft material is chosen.

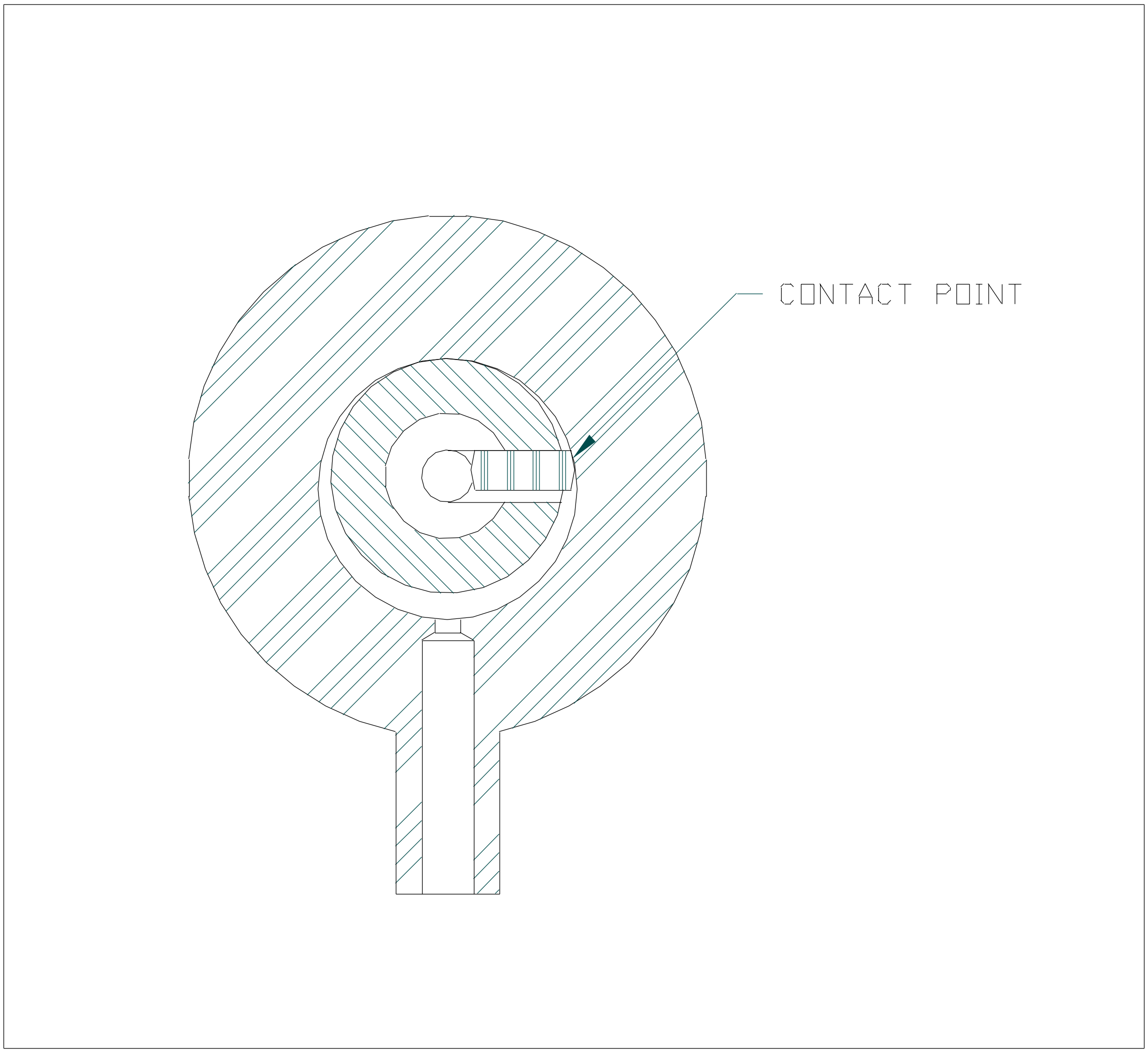


Figure 8. Location of Line Contact Between the Vane and the Pump Chamber.

Steel

Steel's versatility lends itself well to the production of parts that must possess high strength or hardness. Thus, steel is the material of choice for the compressor shaft, which serves well as the pump shaft when lengthened to extend from the lower bearing into the pump housing. Furthermore, steel balls may be easily obtained for use in the check valve.

Powdered metal processes also allow the fabrication of pump housings in large volumes, at reasonable cost, and in dimensions accurate enough to possibly eliminate most or all machining of the pump housing. Even the pump vanes can be made of powdered metal steel, if used with a housing of sufficient hardness (probably steel) to prevent the vanes from wearing the inside of the pump chamber. The pump housing also contains the compressor's thrust bearing (see Figure 9) in the pumps considered in this study, and a thrust washer must be used in the design if the pump housing is not made of steel. The flatness and uniformity of thickness inherent in some grades of sheet steel qualify them as excellent candidates for thrust washer materials.

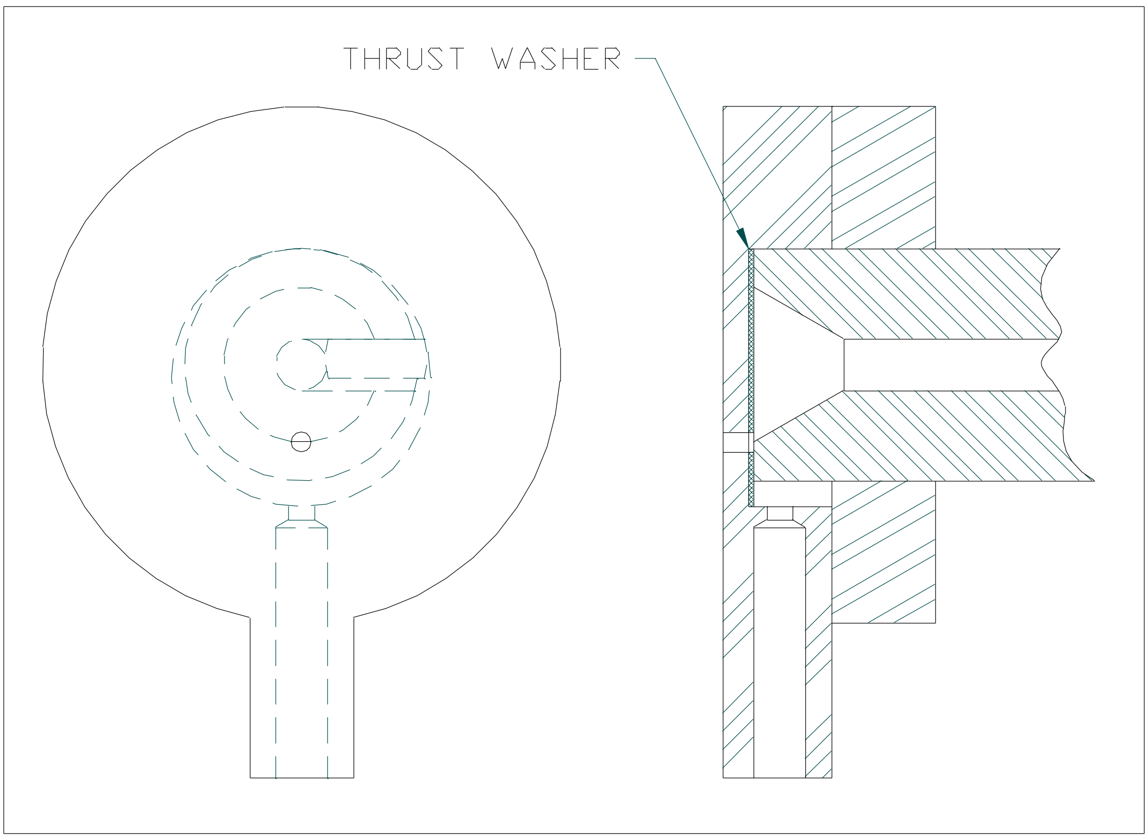


Figure 9. Diagram Showing the Thrust Washer Location.

Polymers

Polymers generally dissolve in refrigerants or mineral oils. Of the few refrigerant-compatible polymers on the market, most are prohibitively expensive. Some of the PEEK, poly-imides, and poly-amide-imides offer potential for use in the oil pump, but only if they are used in small quantities. Considering the difficulty of fabricating such materials into spheres of sufficient roundness to serve as balls or check valve bodies, the pump vanes and thrust washers present the only viable possibilities for plastic components. Some plastics exhibit excellent wear properties and are not abrasive to metals.

CHAPTER 5

EFFECTS OF CLEARANCES ON PUMP OPERATION

Part Sizing for Design Clearances

The Need for Clearances

The discussion above dealing with the conceptualization of the various designs and modifications assumes perfect fits between the parts of the pump assemblies. In reality, a perfect fit between assembly components is neither possible nor desirable. Friction between the vanes, the slot in the shaft, and the floor of the pump chamber prevents the vanes from sliding in the slot in a pump assembly that fits together with no clearances. Likewise, a shaft with no bearing clearance allows no oil film to develop between the shaft and the bearing journal. Such a situation inevitably results in friction, intense heat, and destruction of the bearing seconds after the shaft begins to rotate. Furthermore, the parts must be aligned perfectly in order to assemble an assembly with no clearance between components. Even the most expensive manufacturing equipment lacks the accuracy necessary to fabricate parts to exact dimensions; and cost, reject rate, part rework, and manufacturing time increase with reductions in clearances between components and dimensional tolerances for individual parts.

Minimum Clearances Between Parts

With the exception of the space between the vanes and the sides of the slot in the shaft (discharge ports), clearances between pump components need only allow movement of the parts with respect to each other. For purposes of theoretical function, no minimum design clearances exist for most of the parts in the assembly. However, the existence of manufacturing tolerances compels the designer to set minimum clearances. Thus, manufacturing tolerances dictate minimum clearances between parts such that an assembly made up of parts yielding the worst case tolerance stack (greatest material condition) possess the smallest easily measurable clearances (not less than 0.0005 inch). The clearance between the vanes and the sides of the slot determine the size of the discharge port and must be determined by experiment.

<u>Maximum Clearances Between Parts</u>

Fluid leakage around the vanes and between the shaft and the wall of the pump chamber determine the maximum clearances between parts in the oil pumps considered in this project. The requirement that the pump prime itself when dry of oil reduces the allowable clearances to the extent that the pump components must seal sufficiently to allow pumping of gaseous refrigerant until sufficient vacuum is developed in the pump chamber to pull fluid through the suction tube and into the pump. Such dry conditions rarely occur in practice, but designing products for the most severe conditions insures against product failure and agrees with accepted engineering practice. This study makes use of prototype testing to determine the maximum allowable clearances between parts of the pump assembly.

The only maximum clearance not determined by leakage is the clearance between each vane and the wall of the slot in the shaft, which is governed by the needs to allow the pump to prime itself and to prevent the vane from turning sideways in the slot (see Figure 10). Although conventional wisdom dictates that the discharge ports be as large as possible, this pump cannot prime itself (even under ideal conditions with no leakage) if the volume of each discharge port is too large. During priming, the pump actually pumps a gas from the chamber (Item 1 in Figure 10) to the discharge port (Item 2 in Figure 10) as the shaft rotates one half of one revolution from the suction port (Item 3 in Figure 10). If the discharge port is too large, the compressibility of the gas allows centrifugal force to compress the gas enough to prevent any of it from reaching the lubrication hole (Frames 1 and 2 of Figure 10). As the next half of shaft rotation progresses, the compressed gas is free to flow back into the volume of the pump chamber (Frames 3 and 4 of Figure 10). Thus, if the volume of each port is large enough, it causes the pump to do nothing more than trade the same compressible flow between two volumes.

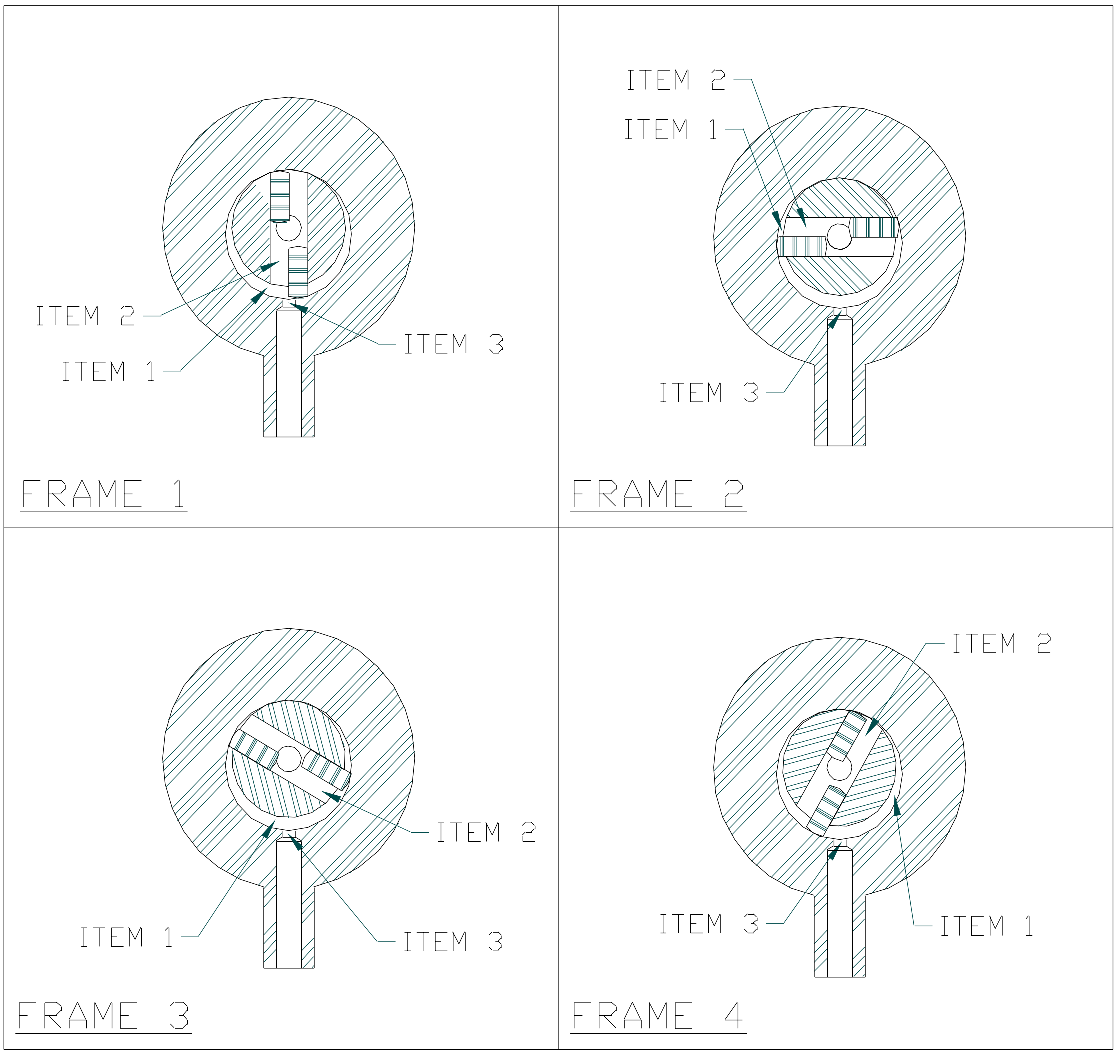


Figure 10. Schematic Depicting Re-expansion in Excessively Large Discharge Ports.

Leak Paths

Vane and Pump Housing

The primary leak paths between the vanes and the pump housing occur between the top and bottom surfaces of each vane and the top and bottom surfaces of the pump chamber (see Figure 11). This leak path allows fluid to travel across the vane from the high pressure side to the low pressure side of the pump chamber. Such a leak allows discharge pressure to drop as fluid escapes the discharge section of the pump chamber. As this fluid then enters the suction portion of the pump chamber, the suction pressure rises, resulting in a loss of suction. If the pump loses enough suction to prevent lifting fluid from the bottom to the top of the suction tube, pump failure occurs.

No significant leakage occurs between the end of the vane and the pump chamber wall, for centrifugal force maintains contact between the surfaces in question. However, the vane crosses the suction port at one point in its travel around the pump chamber. As the vane crosses the suction port, the leak around the end of the vane enlarges enough to become significant. Fortunately, the velocity possessed by the vane in its travel around the pump chamber limits the time of interaction between the vane and the suction port to the extent that the leakage poses no threat to pump operation.

Vane and Shaft Slot

Potential leak paths exist between the vanes and the shaft along the trailing edge of each vane in the interface between the side of the vane and the side of the slot (see Figure 11). Friction between the vane and the bottom of the pump chamber and friction between the vane and the top of the thrust surface force the vane to contact the side of the slot and reduce leakage to the point of insignificance.

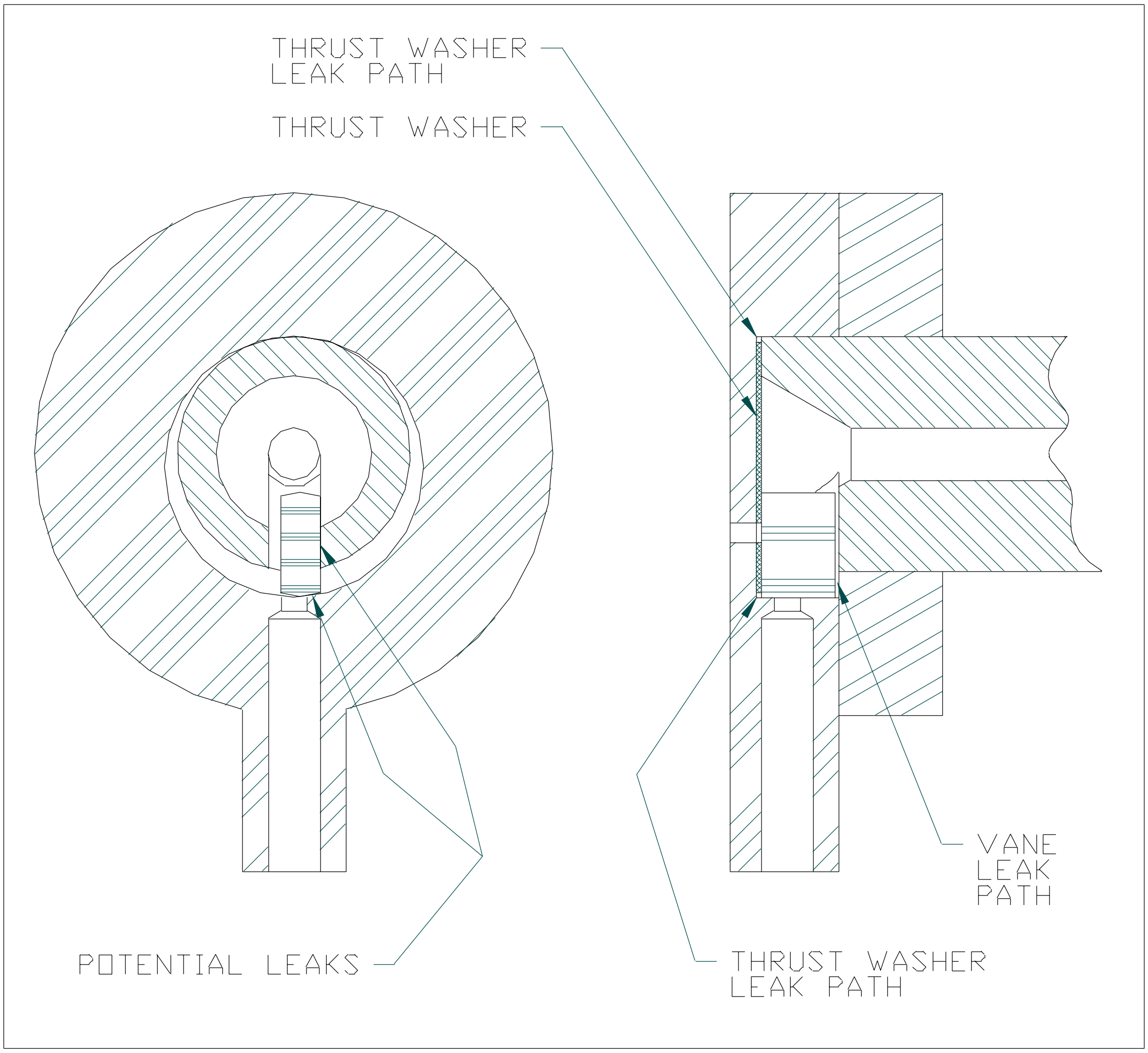


Figure 11. Potential and Actual Leaks Through Gaps Between Assembly Components.

Vane, Shaft, and Thrust Washer

Any clearance between the outside diameter of the thrust washer and the pump chamber creates a leak path under the vanes (see Figure 11). Furthermore, thrust washer flatness directly affects the leak rate from the discharge to the suction portion of the pump chamber. A warped thrust washer can lift the shaft, creating a leak across the bottom of the shaft. Deliberate misalignment between the rotor and stator in the compressor's motor applies axial force to maintain contact between the shaft and the thrust surface but may not flatten an excessively stiff warped thrust washer.

Designs using a pump chamber that is integral to the lower bearing must use thrust washers of a diameter less than or equal to the diameter of the shaft in order for the thrust washer to be installed (see Figure 12). In contrast, designs using a bearing as a separate part from the pump housing use thrust washers matching the diameter of the pump chamber (see Figure 12). Warped surfaces on thrust washers in either design create leaks across the bottom of the shaft between the high and low pressure sides of the pump and may cause the vanes to bind between the thrust washer and the ceiling of the pump chamber. Even if one assumes perfect flatness on the top surface of the smaller thrust washer, the difficulty of accurately placing the thrust washer's top surface coplanar to the bottom of the pump chamber guarantees unwanted leakage or interference between the vane and other parts of the assembly. The manufacturing issues listed above dictate avoiding designs using the lower bearing and the pump housing in a single part.

Pump Housing and Shaft

A critical clearance between the pump housing and shaft exists at the tangent point between the shaft and the pump chamber (see Item 1, Figure 4). Leakage across the tangent point referred to above results in direct flow from the discharge side to the suction side of the pump and results in pump failure, if excessive leakage exists. The existence of manufacturing tolerances forces the engineer to account for them in the design.

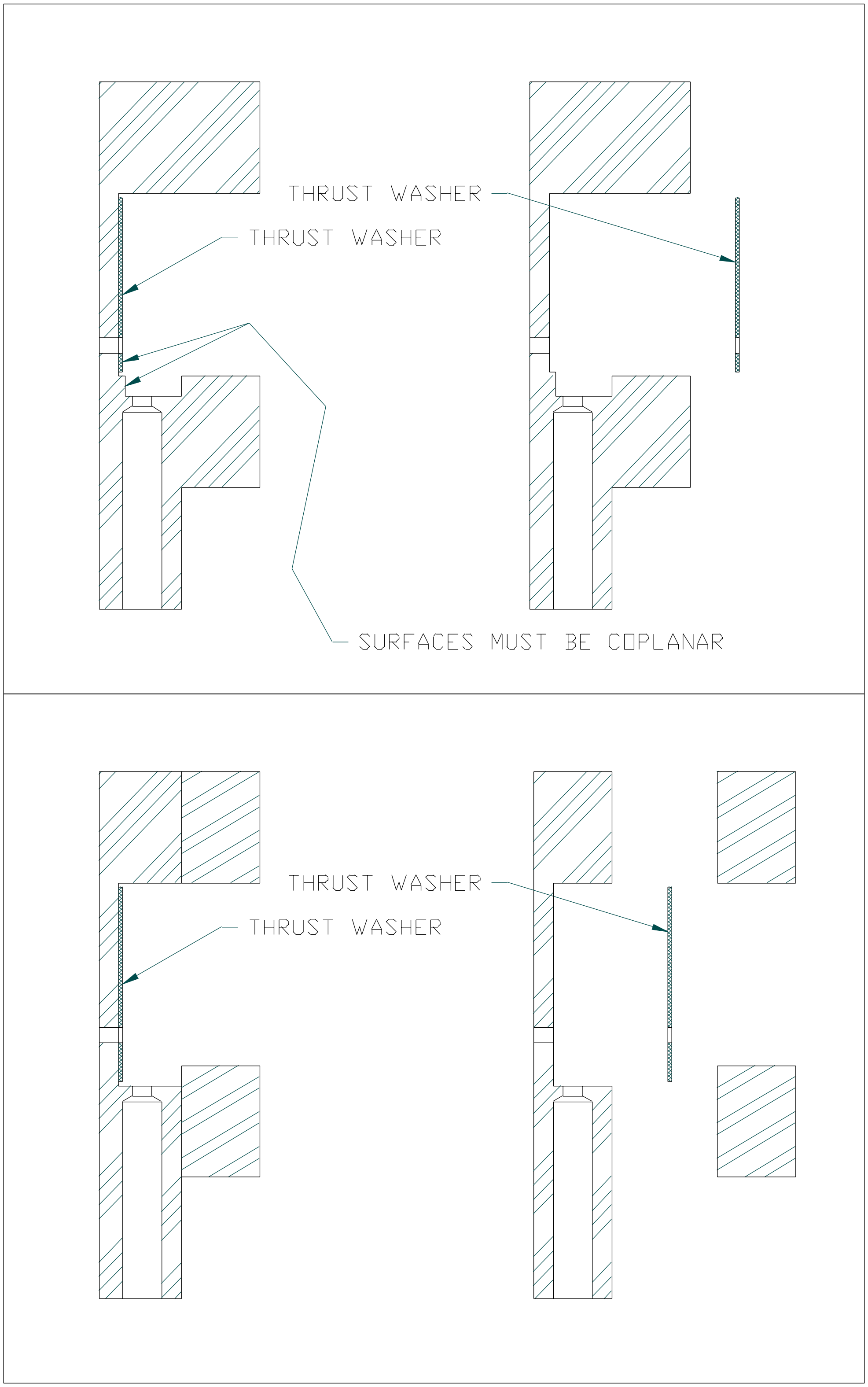


Figure 12. Comparison of Different Pump Housing and Lower Bearing Assemblies.

CHAPTER 6

PROTOTYPE PART FABRICATION

Pump Housing

Theoretically, the pump chamber must contact the shaft at a point opposite the suction port (see Figure 13) in order to prevent leakage from the discharge cavity to the suction cavity during pump operation. In reality, such contact results in excess friction and catastrophic failure. The shaft and the pump chamber weld together under the influence of the extreme heat produced by friction due to contact between the rotating shaft and the stationary pump chamber. Thus, an oil film must separate the shaft from the chamber and requires a slight clearance. Adequate clearance to maintain an oil film exists between the crank shaft and lower bearing in the compressor. Using a standard bearing clearance in this application requires that the top edge of the pump chamber share a tangent point with the bottom edge of the lower bearing (see Figure 13). In reality, such a tangent point is impossible to create with any degree of repeatability because manufacturing tolerances force the placement of tolerances on the eccentricity and the diameters of the bearing and the chamber.

Tolerance stack prevents creating the tangent point intersection, but allowing the pump chamber wall near the tangent point to move too far toward the center of the bearing causes interference with the shaft and prevents assembly of the pump. Therefore, the wall of the pump chamber nearest the tangent point must never cross inside the radius of the bearing's surface. Allowing movement of the same portion of the chamber wall too far to the outside creates leakage that the pump cannot overcome. The allowable minimum radial distance between the pump chamber surface and the bearing surface must be determined by prototype testing and is minimized in initial tests.

Pump chamber depth and the distance between the chamber's axis and that of the shaft (eccentricity) determine the capacity of the pump (see Figure 13). Maintaining a tangent point common to the pump chamber and the shaft requires that the shaft diameter determine the pump chamber diameter for a given eccentricity. The eccentricity must be large enough to provide the pump with adequate capacity but small enough to prevent the vane from tilting in the shaft slot while traveling at its maximum extension at the suction port (see Figure 13). Prototype testing determines the correct eccentricity, and chamber depth (see Figure 13) is then set to achieve adequate capacity. At this point in the study, one must note that the existence of more than one viable combination of eccentricity and vane depth may exist, but for any given eccentricity, one and only one chamber diameter exists.

Prototype testing determines the correct capacity of the pump and involves trial and error. Therefore, choosing the first prototype's eccentricity of 0.0625 inch is an educated guess (see Appendix for complete engineering drawings of all prototype parts). The acceptable tolerance on this value and the values and tolerances of the remaining pump chamber dimensions are set by the dimensions and tolerances of other components in the design and are documented in later sections of this chapter.

The bleed hole must be sized and located in such a fashion as to vent gas and reduce pressure during priming without starving the lubrication hole of the quantity of oil necessary to lubricate the mechanical components of the compressor during normal operation. Although the capacity of the bleed hole can be controlled by its diameter alone, designs used in this study also use the location of the bleed hole to control its capacity by overlapping the bleed hole and the edge of the countersink in the shaft (see Figure 6). Changing the capacity of the bleed hole by changing the overlap it shares with the shaft countersink allows a constant hole diameter for different bleed hole capacities.

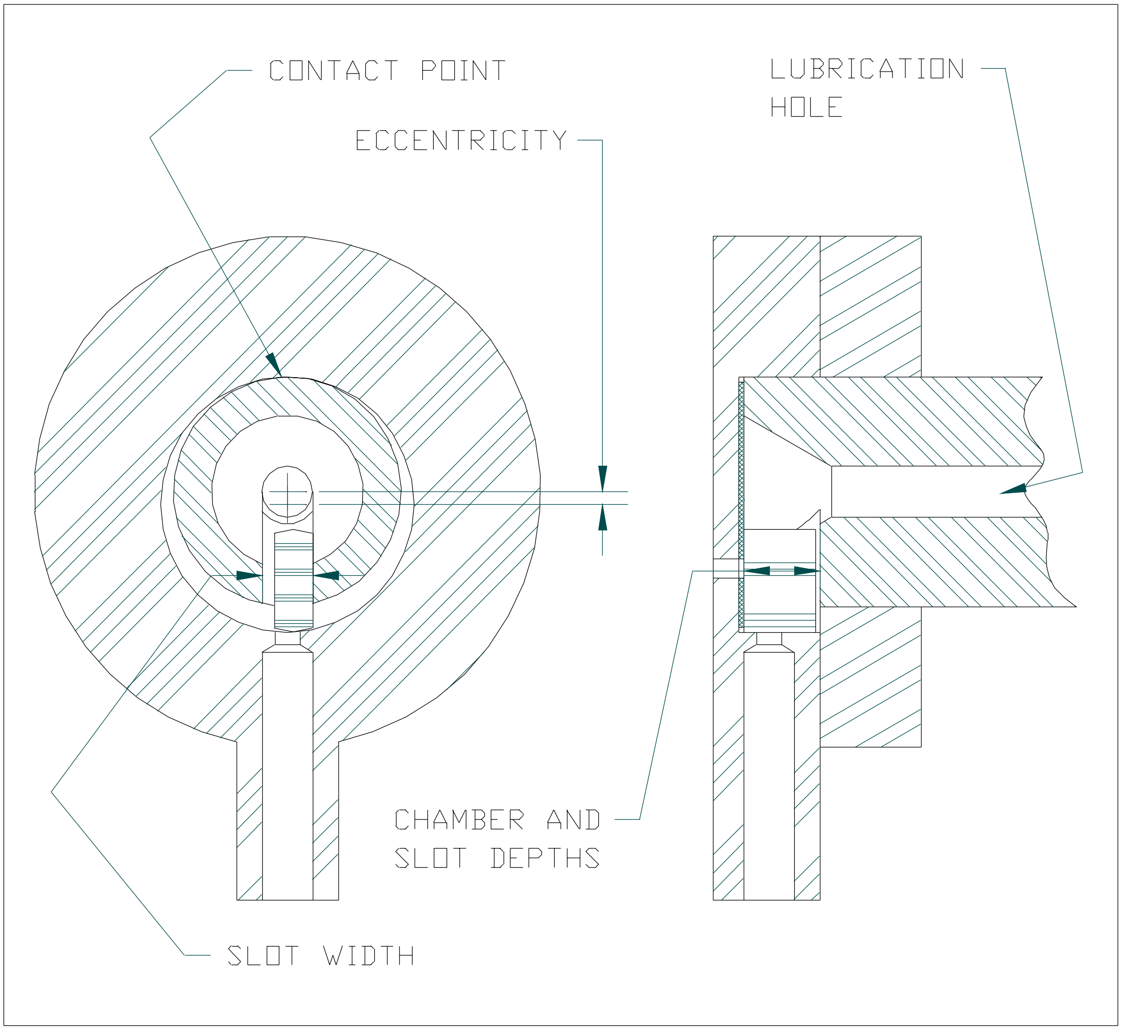


Figure 13. Critical Dimensions of the Pump Chamber.

As hole diameters become smaller, the bits required to drill those holes become smaller. Rigidity and cutting speed decrease with decreasing bit diameter, and manufacturing time and tool breakage increase with decreasing bit diameter. The smallest bit diameter used in shaft production at Bristol Compressors is 0.096 inch. Therefore, a 0.096 inch diameter bleed hole located 0.275 inch from the center of the pump housing is used in the initial prototype. Because the shaft axis is coaxial to the center of the pump housing and the shaft countersink diameter is 0.500 inch at the base of the shaft, the bleed hole communicates with the countersink through an area consisting of less than one quarter of the hole's cross sectional area. Thus, optimizing the effective area of the bleed hole need not minimize tool size.

Pump chambers for the prototypes are machined from steel because it is readily available and facilities exist for fabrication of the parts at Bristol Compressors. Steel provides a low cost with properties that allow machining to tolerances close enough to duplicate those of any other process considered in this study. Prototype production allows parts to be tailored to extremely tight dimensions, although a relatively high part scrap rate of up to 50% must be accepted. Unless otherwise noted, tolerances on all prototype part dimensions are held to within 0.0005 inch.

Shaft and Lower Bearing

Shaft Diameter and Bearing Bore

As discussed above, maintaining a constant eccentricity and requiring the pump chamber and the lower bearing to share a common tangent point dictates that the shaft diameter determine the pump chamber diameter. Increasing the shaft diameter forces the chamber diameter to increase, yielding a larger chamber and pump capacity. For a given motor speed, the linear velocity along the outer perimeter of a shaft's cross section increases with increasing shaft diameter, and the vanes are coupled to the shaft by the slot and travel near the shaft's outside diameter. Consequently, the smaller shaft moves the vanes slower in a smaller pump.

If the clearances between pump components are held constant, similar leak paths exist, regardless of the pump capacity. However, a higher vane speed results in increased pump capacity and increased flow resistance through leak paths. Accordingly, larger pumps are better able to make up for losses due to leakage and are less likely to fail. Consideration of the principles discussed above leads one to conclude that developing a feasible design using the smallest acceptable shaft diameter eliminates the need to test shafts of larger diameters.

The standard crankshaft diameter used on most of the compressors manufactured at Bristol Compressors is 1.124 inch at one end and 0.876 inch at the other end. In order to retain commonality with the compressor crankshaft forgings and to minimize time during manufacture, the pumps used in these designs must use shaft diameters that are within 0.005 inch of the diameter at one end of the shaft. Considering the advantage of using the smallest acceptable shaft diameter (discussed in the previous paragraph), one concludes that the shaft diameter should be between 0.871 and 0.881 inch. Using a nominal radial bearing clearance of 0.0005 inch (somewhat of a standard for Bristol Compressors) implies the use of a bearing bore diameter of between 0.872 and 0.882 inch. Thus, setting the bearing bore diameter to 0.875 inch takes advantage of the availability of standard tool sizes while remaining within the allowable bore size. Subtracting the radial bearing clearance from the bearing diameter yields the nominal diameter of 0.874 inch for the pump shaft.

Machining is simplified by using the same equipment and methods to produce all machined surfaces on the diameter of a crank shaft. Bristol Compressors' standard bearing tolerances are maintained on the shaft diameters because the pump shaft is merely an extension of the lower bearing surface of the steel crank shaft in all the designs studied in this publication. Accordingly, the shaft diameter varies from 0.8736 to 0.8744 inch.

Use of a steel shaft necessitates the use of a lower bearing of a softer material. Bristol Compressors uses die-cast aluminum as a lower bearing material, and the low cost and ease of manufacture of die-cast aluminum nominates it as the prime material for the oil pump's lower bearing. The machining characteristics of die-cast aluminum allow reliable machining of the lower bearing bore to within 0.0004 inch. Thus, the lower bearing bore varies from 0.8746 to 0.8754 inch.

As discussed previously, the shaft diameter and the eccentricity determine the chamber diameter for the pump. Particularly, the nominal chamber radius is the sum of the nominal values for eccentricity and lower bearing radius. In the case of a pump using an eccentricity of 0.0625 inch and a bearing having a diameter of 0.875 inch, the nominal chamber diameter is calculated as follows:

$$\text{Nominal chamber diameter} = 2\ (0.0625 + (0.875/2)) = 1.000 \text{ inch}$$

Both the chamber diameter and the lower bearing bore can be held to within 0.0005 inch of their nominal values. By assuming the same tolerance value for the eccentricity and the shaft diameter, a total of 0.0005 + 0.0005 + 0.0005 = 0.0015 inch must be added to the nominal chamber diameter to allow for the tolerance stack between it, the eccentricity, and the bearing bore. Consequently, the nominal chamber diameter must be set to 1.0015 inch in order to render interference between the shaft and the pump chamber impossible in all cases. The value of the eccentricity's tolerance is assumed as the minimum acceptable but may be enlarged, if demonstrated during prototype testing.

Lubrication Hole Diameter

A hole of 0.25 inch diameter allows for an adequate cutting speed to drill swiftly and maintain enough tool stiffness to prevent the gun drill from deflecting in the hole. Deflection tends to cause tools to break, especially in applications in which the hole depths are orders of magnitude larger than the hole diameters. Such applications require high cutting speed, stiff tools, sharp tools, and high flow rates of cutting fluid. With the proper set up, gun drills create deeper holes more rapidly than conventional drill bits. Bristol Compressors, Inc. employs gun drills to drill lubrication holes and uses a standard diameter of 0.25 inches. Therefore, the pump designs examined in this study adhere to the proven standard lubrication hole diameter and a tolerance of $\pm$ 0.010 inch. The pump lubrication hole is merely an extension of the hole supplying lubrication to the compressor mechanism (see Figure 13).

Vane Slot

Slot Width. Manufacturing time governs the width of the vane slot (Figure 13). Time of manufacture reduces as the slot widens, for larger slot widths allow the use of cutting tools of larger diameters. Compared to a cutting tool of a smaller diameter, a cutting tool of a larger diameter possesses a greater cutting speed for a given spindle speed due to the greater linear velocity at the perimeter of the larger tool. A nominal slot width of 0.260 inch allows machining in two passes with a standard 1/4 inch end mill or a horizontal milling cutter of 1/4 inch width and makes adequate allowance for oversized cuts. Both the proposed upper tolerance of +0.010 inch and the lower tolerance of -0.010 inch for production parts are large enough to practically eliminate part rejects due to the noncritical slot width dimension being out of tolerance.

Slot Depth. In order for the vane to align with the pump chamber and maintain contact with the chamber's outer perimeter, it must not get caught on the edge of the chamber's ceiling or be pinched by the slot in the shaft. Thus, the depth of the slot must closely match that of the pump chamber. One must also consider that raw material and manufacturing time for all components increase with increasing slot depth. Therefore, the slot depth only slightly exceeds that of the pump chamber (0.001 inch in excess of the chamber depth) in order to maximize the allowable thickness of the vane, and the lower tolerance is set such that the chamber depth never exceeds that of the slot (-0.001 inch). The relatively small + 0.001 inch upper tolerance (assisted by small radii along the edges of the vane) ensures that the vane never catches on the edge of the chamber's ceiling while sliding into the pump chamber.

Suction Tube

Suction tube cross section must remain large enough to allow ease of manufacture but small enough to prevent the vanes from falling into the tube. For any given flow rate, flow resistance and velocity increase with decreasing tube cross sectional area, resulting in reduced pump efficiency. Furthermore, the risk of entraining debris into the suction tube increases as fluid velocity increases at the tube inlet. Thus, the end of the tube nearest to the pump chamber is manufactured to dimensions small enough to prevent the vane from falling into the tube while the remainder of the tube is manufactured to a dimension large enough to eliminate significant risk of pulling debris into the pump. The principals governing the length of the suction tube are discussed on Page 24 of this publication. Prototype testing sets the length and cross sectional area for the suction tube, while the chosen manufacturing process for fabricating the pump housing governs the tolerances for those dimensions. Initial prototypes make use of a suction port of rectangular cross section measuring 0.125 inch wide by 0.075 inch deep (see Appendix). The remainder of the suction tube is split between the lower bearing and the pump chamber and is of a rectangular cross section measuring 0.250 inch wide by 0.448 inch deep when the pump chamber and the lower bearing are assembled. The somewhat unique rectangular cross section of the tube need not concern the engineer, because each half of the tube is machined before assembly, and the tube is formed upon assembly of the pump housing and the lower bearing (see Appendix). All tolerances for the suction tube are ranges within those which standard tools can machine.

Check Valve

The ball check valve need only be constructed using a standard size steel ball. The lift of the ball and the clearance around it when lifted must allow the same or more flow area than that required by the oil pump (see Figure 3) in order to prevent starving the pump. Initial prototypes omit the check valve in an attempt to reduce the parts, resulting in a simpler, less expensive design.

Thrust Washer

The thrust washer may be of any practical thickness, but its diameter must closely match that of the shaft or the pump chamber, depending upon which pump design is used. Furthermore, the shape and thickness of the thrust washer must be such that it remains flat when the pump is assembled. The use of machined steel pump housings in the initial prototypes allows the omission of the thrust washer. Thus, determining the thrust washer's dimensions and tolerances is delayed until a design using a thrust washer is worthy of further exploration.

Vane

A pump vane must be thick enough to adequately span the depth of the pump chamber to prevent excess leakage. Furthermore, a vane must possess a width allowing adequate clearance between the leading edge of the vane and the side of the slot in the shaft (see Figure 13) in order to permit adequate flow through the discharge port of the pump. For assembly purposes, the thickness and the width of the production pump vane must differ enough to prevent assembly of the vane in the wrong orientation. Prototype parts are not required to possess such assembly failsafes because the parts are assembled under the direct supervision of the designer. Although the polymer vanes produced as prototypes adhere to relatively tight tolerances ($\pm$ 0.0005 inch), testing must verify that the pump functions using vanes possessing tolerances more characteristic of injection molded parts.

Vane Width

Recalling that the clearance between the slot in the shaft and the vane creates the discharge port for the pump, the vane width must allow adequate clearance to provide acceptable oil flow from the pump chamber to the lubrication hole (see Figure 13). A somewhat arbitrary vane width of 0.200 inch for the first prototype allows a discharge port width of 0.060 inch when used in a slot of 0.260 inch width. 0.060 inch is probably an acceptable value for the discharge port width because the pump chamber itself varies in width from 0.124 inch (twice the eccentricity) to 0.000. The proposed tolerance on the width of production vanes is set at 0.005 inch above or below the nominal value and is based on tolerances conducive to injection molding of polymer components.

Vane Thickness

Vane thickness dictates chamber depth, and chamber depth and eccentricity are the primary dimensional factors affecting pump capacity. Using a vane thickness of 0.186 inch on the first prototype provides a starting point for experimenting with pump capacity and sets the pump chamber depth at 0.188 inch. The values cited above are acceptable as nominal values if the tolerances on both are set at 0.0005 inch, which is satisfactory for the machine tools used to make the parts. Initial prototype production uses a shaft slot depth of 0.190, with a tolerance allowing the depth to vary from 0.189 to 0.191. Thus, the vane thickness is never great enough to interfere with the shaft.

Vane Length

Vane length is not a critical dimension as long as the vane is not so short as to allow its center of mass to slide over the center of the shaft. If the shaft begins rotating in such a situation, centrifugal force holds the vane in the slot instead of forcing it into the pump chamber. The vane must also be short enough to enable its movement into and out of the pump chamber without interfering with another vane (twin vane design) or the end of the slot in the shaft (single vane design). Initial prototypes use a vane length of 0.500 inch with a tolerance of 0.020 inch above or below the nominal value. One should also note the presence of a 0.250 inch radius on each end of the vane, which allows for assembly with either end of the vane facing the pump chamber. The end radii also offer the advantage of easily manufacturing prototype vanes from ½ inch bar stock.

CHAPTER 7

PROTOTYPE TESTING

Proof of Concept

Prototype Characteristics

Table 1 lists the critical part dimensions used in the prototype. As stated in the previous chapter, a thrust washer is not used in the initial prototype. Furthermore, use of the single vane design concept and the omission of the check valve allow testing of the simplest and most quicky fabricated of the proposed designs. If the simplest design is tested first and proves to perform adequately, tests of other designs will be unnecessary.

Test Results

Testing the prototype in the fixture depicted in Figure 14, the pump primes itself from a dry state and provides adequate supply to the fixture's mechanical components in either direction of shaft rotation. Thus, the theory behind the concept proves to be true. The test also proves that eliminating the check valve and using the single vane design result in a functional pump, as well as a simpler, more inexpensive assembly.

Future Tests

Future tests must determine the effects of part tolerances on pump performance by modifying one dimension for each test in order to determine acceptable tolerances for all critical dimensions in the assembly. The combined tolerance stack found through prototype part testing is then used to determine which material and manufacturing options are applicable to the manufacture of each component in the pump. Tolerances for dimensions deemed noncritical to the function of the design use tolerances accepted as standard for the process used to manufacture the components.

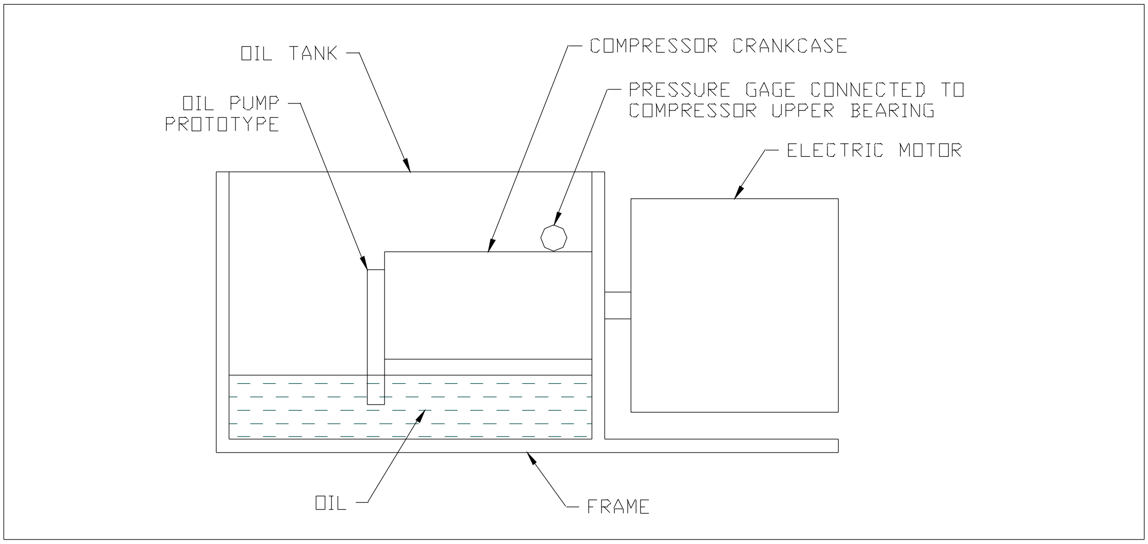


Figure 14. Diagram of the Fixture Used to Test the Oil Pump Prototypes.

Table 1. Initial Prototype Critical Dimensions and Tolerances.

Dimension	Nominal Value, inches	Tolerance, inches
Suction Port Width	0.125	0.0005
Suction Port Depth	0.075	0.0005
Eccentricity	0.0625	0.0005
Shaft Diameter	0.874	0.0004
Countersink Diameter	0.500	0.005
Bleed Hole Diameter	0.096	0.002
Bleed Hole Location	0.275	0.005
Slot Width	0.260	0.010
Slot Depth	0.190	0.001
Chamber Diameter	1.0015	0.0005
Chamber Depth	0.188	0.0005
Vane Width	0.200	0.005
Vane Thickness	0.186	0.0005

Vane Thickness Variation

In order to find the acceptable tolerance for the thickness of the pump vanes, the vane thickness in the prototype tested above is reduced to 0.168 inch and increased by 0.002 inch with each successive test until the pump primes itself from a dry state. The vane thicknesses tested are listed in Table 2, along with test results for each vane.

Table 2. Effect of Vane Thickness on Priming in the Initial Prototype.

Vane Thickness, inches	Prime from a Dry State
0.168	No
0.170	No
0.172	Yes
0.174	Yes
0.176	Yes
0.178	Yes
0.180	Yes
0.182	Yes
0.184	Yes
0.186	Yes

Table 2 shows promise for the use of polymer vanes by indicating that the maximum acceptable axial clearance between the vane and the pump chamber is 0.014 inch. Only the 2 thinnest of the 10 vanes prepared for the test prove incapable of priming the pump from a dry state, and the tolerance of the thickness of an injection molded polymer vane is $\pm$ 0.007 inch. Therefore, an injection molded polymer vane having a nominal thickness of 0.180 inch proves acceptable for use in the prototype. However, any variance in another critical dimension, such as chamber depth, renders the thickness tolerance of an injection molded vane unacceptable for use in the design. Thus, some modification of the initial prototype is necessary in order to develop a feasible design.

Chamber Depth Modification

Theorem of the Modification

The variation in thickness of injection molded polymer vanes, combined with variations in pump chamber depth, creates a situation in which priming the pump from a dry state is not sufficiently repeatable for use in a production assembly. If the capacity of the pump is increased, then the acceptable amount of leakage also increases, allowing a larger maximum axial clearance between the pump chamber and the vane. The preferred method of increasing pump capacity involves increasing the thickness of the vane and the depths of the pump chamber and vane slot. By increasing the pump capacity in such a way, greater tolerances are allowed on the modified dimensions, for the acceptable leak rate increases. Therefore, the chamber depth is increased to 0.250 inches, and the vane thickness and slot depth are adjusted accordingly. All tolerances remain unchanged.

Test Results

Table 3 lists the critical dimensions for the prototype using the 0.250 inch chamber depth. Testing in the fixture depicted in Figure 14 demonstrates that the pump primes itself from a dry state using a vane of 0.230 inch thickness. By maintaining a tolerance of 0.001 inch on the chamber depth and slot depth, a tolerance of $\pm$ 0.009 inch is allowable on the vane thickness with no possibility of priming failure or interference between the vane and the chamber.

Variation of Clearance between Shaft and Pump Chamber

In order to determine the allowable clearance between the shaft and the pump chamber wall, a series of tests is conducted in which the diameter of the pump chamber is increased by 0.0010 inch after each test. The pump fails to prime from a dry state using a chamber diameter of 1.0100 inch, which corresponds to a clearance of 0.0050 inch between the shaft and the chamber wall at the point shown in Figure 13. Such a clearance satisfies the requirements of the design, if the tolerance stack determining the maximum clearance is 0.0040 inch. The minimum acceptable tolerances are calculated and found to fix the clearance at a value of 0.0035 inch.

Table 3. Critical Dimensions and Tolerances for a Prototype Using a 0.250 Inch Chamber Depth.

Dimension	Nominal Value, inches	Tolerance, inches
Suction Port Width	0.125	0.0005
Suction Port Depth	0.075	0.0005
Eccentricity	0.0625	0.0005
Shaft Diameter	0.874	0.0004
Countersink Diameter	0.500	0.005
Bleed Hole Diameter	0.096	0.002
Bleed Hole Location	0.275	0.005
Slot Width	0.260	0.010
Slot Depth	0.252	0.001
Chamber Diameter	1.0015	0.0005
Chamber Depth	0.250	0.0005
Vane Width	0.200	0.005
Vane Thickness	0.240	0.0005

Recommendations for Material Changes

Based upon the tolerances of injection molding processes and the allowable clearances between components in the assembly, the author recommends the use of a material other than a polymer for the vanes. Although acceptable clearances allow for tolerances achievable using injection molded plastics, little room is left for error. Part reject rate can be reduced by using powdered metal vanes, which can be fabricated to within 0.002 inch of nominal thickness. Tighter vane tolerances reduce the size of the leak path between the vane and the chamber ceiling, allowing other tolerances in the assembly to be relaxed somewhat while maintaining the same total leak rate as was present using the larger vane tolerances.

Power Requirements

The viability of the design depends upon low energy consumption. Measuring the power consumption of the pump requires comparing the power required to operate a compressor equipped with a horizontal oil pump to that of the same type of compressor equipped with a standard oil pump. However, unavoidable differences in calorimeter conditions from one test to the next allow a variation of approximately one percent in efficiency and capacity. Using identical compressors minimizes such differences, but such differences exist, nonetheless. Normalizing the test data allows a more exact comparison of the power consumption of the compressor with the horizontal oil pump to that of the standard compressor and is demonstrated immediately following the test data in Table 4.

Table 4. Power Comparison Between Standard and Horizontal Oil Pumps.

Horizontal Oil Pump	Power, W	Capacity, Btu/W/hr
No	3251	35,462
Yes	3259	34,934

Normalization of the test data consists of taking a ratio of the capacity of the compressor with the standard oil pump to that of the compressor with the horizontal oil pump. The actual power consumption of the compressor with the horizontal oil pump is then multiplied by the calculated ratio as shown below:

Normalized Power = (3259 W)*(35,462 Btu/hr) / (34,934 Btu/hr) = 3308 W

Although the normalized power consumed by the compressor using the horizontal oil pump exceeds that of the standard model by 57 Watts, the design shows merit. The compressor using the horizontal oil pump is in a vertical orientation in the power consumption test. In the horizontal orientation, the capacity is within 100 Btu/hr and the power consumption is within 50 Watts of the vertical compressor. Placing the compressor in the horizontal orientation forces the oil level in the housing to fall below the moving parts in the compressor's mechanism. Thus, the performance of the horizontal design compares well to the vertical design.

Sound Radiation

System manufacturers demand that compressors meet both performance and sound goals. Consequently, the horizontal oil pump is rendered unacceptable as a design if orienting a compressor horizontally raises the sound to an unacceptable level. Sound testing for this study involves performing sound tests in both the vertical and horizontal orientation on the same compressor. The horizontal compressor tested for this study shows an increase of 2.0 decibel in total sound power, and sound instrumentation error is within a decibel. Although the horizontal compressor does increase sound power, the change in sound power is probably caused by a poor suspension system. Thus, further research is needed to discover and correct the cause of the increased sound power.

Production Tolerances

The prototypes used in this study are machined to excessively tight tolerances in order to test the allowable clearances between components of the assembly. Tolerances of production parts can be relaxed to a large degree if powdered metal vanes are used in the design. The lower bearing, pump housing, and vane are shown in the engineering drawings located in the appendix.

Manufacturing processes at Bristol Compressors allow the production lower bearing to retain the dimensions and tolerances of the prototype. Powdered metal parts are capable of holding the tolerances of the prototype vane and pump chamber and need not be adjusted. Although the parts shown in the appendix are suitable for production, the pump housing used in production may be modified. Such modifications must not affect any of the dimensions or tolerances in the drawings, but material can be removed from the exterior of the pump housing in order to reduce the cost of the part.

The crank shaft used in production is a standard compressor crank shaft having the centrifuge, lubrication hole, and slot machined into its lower end. Production tolerances of standard crank shafts agree with those of the prototypes tested and need not be changed.

CHAPTER 8

CONCLUSIONS

Tests conducted in the previous chapter demonstrate the viability of the cheapest rotating vane pump design considered in the study. The single vane pump design meets the requirements and lies within the constraints of the study. Additionally, it eliminates one of the vanes present in the initial design concept. Omitting the check valve results in a functional pump with even fewer parts. Thus, the general configuration of the pump should be that of the valveless single vane rotating vane pump shown in Figure 6.

The success of the crank shaft and bearing materials in current compressor designs warrant use of the same materials in the pump. Manufacturing processes in place at Bristol Compressors machine bearing bores and diameters to tolerances identical to those held in the pump prototype components. Consequently, the prototype tolerances on the shaft diameter and bearing bore need not be relaxed for production parts. With the exception of machining the slot in the shaft, the horizontal compressor's shaft and lower bearing can be manufactured using equipment currently set up at Bristol Compressors.

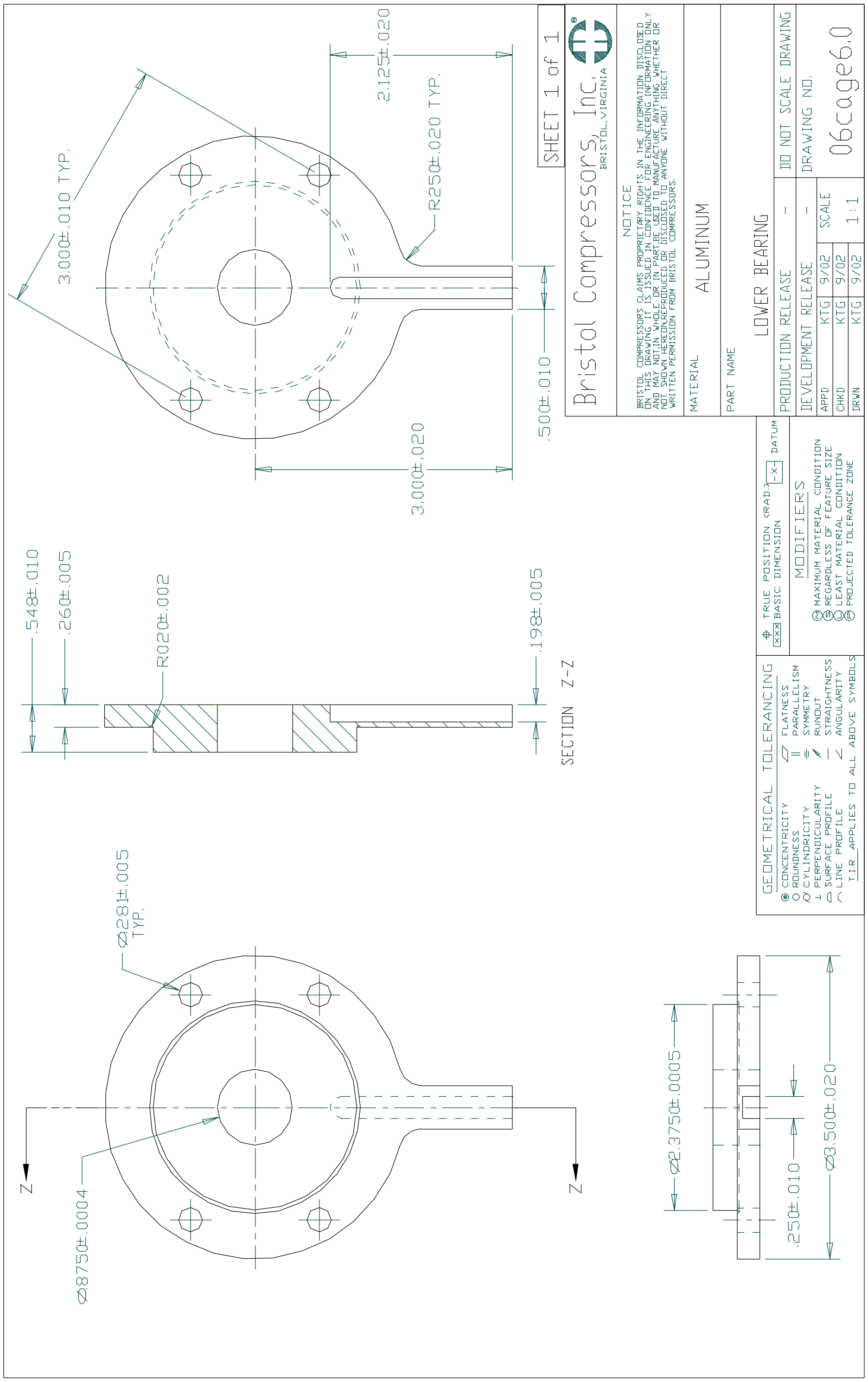
Powdered metal pump housings can be purchased as finished parts with bolt patterns and dimensions that allow assembly on current production lines with no tooling modifications. Using powdered metal vanes allows part tolerances on the vanes and the pump housing to be tighter than those required to guarantee pump operation. Also, the inherent thrust surface present in the powdered metal pump housing eliminates the need for a separate thrust washer.

CHAPTER 9

RECOMMENDATIONS

Based upon the findings of this study, the horizontal compressor concept is worthy of further exploration. Engineering drawings for the lubrication system recommended for use in the horizontal compressor are located in the appendix of this publication. With the exception of the powdered metal vanes, the pump is identical to the prototype tested in this study using a chamber depth of 0.250 inch. Although this research has developed a feasible oil pump, development should continue by subjecting the pump to a full battery of life tests. Furthermore, oil pump parts need to be quoted by suppliers, though all the parts developed in this project are similar in size and use materials and manufacturing processes common to compressor components currently in use. Also, an adequate suspension system must be developed. Until such time as issues with the mounting system are overcome, the goal of a commercially attractive horizontal air-conditioning compressor remains unrealized.

APPENDIX



SHEET 1 of 1
Bristol Compressors, Inc.
BRISTOL, VIRGINIA
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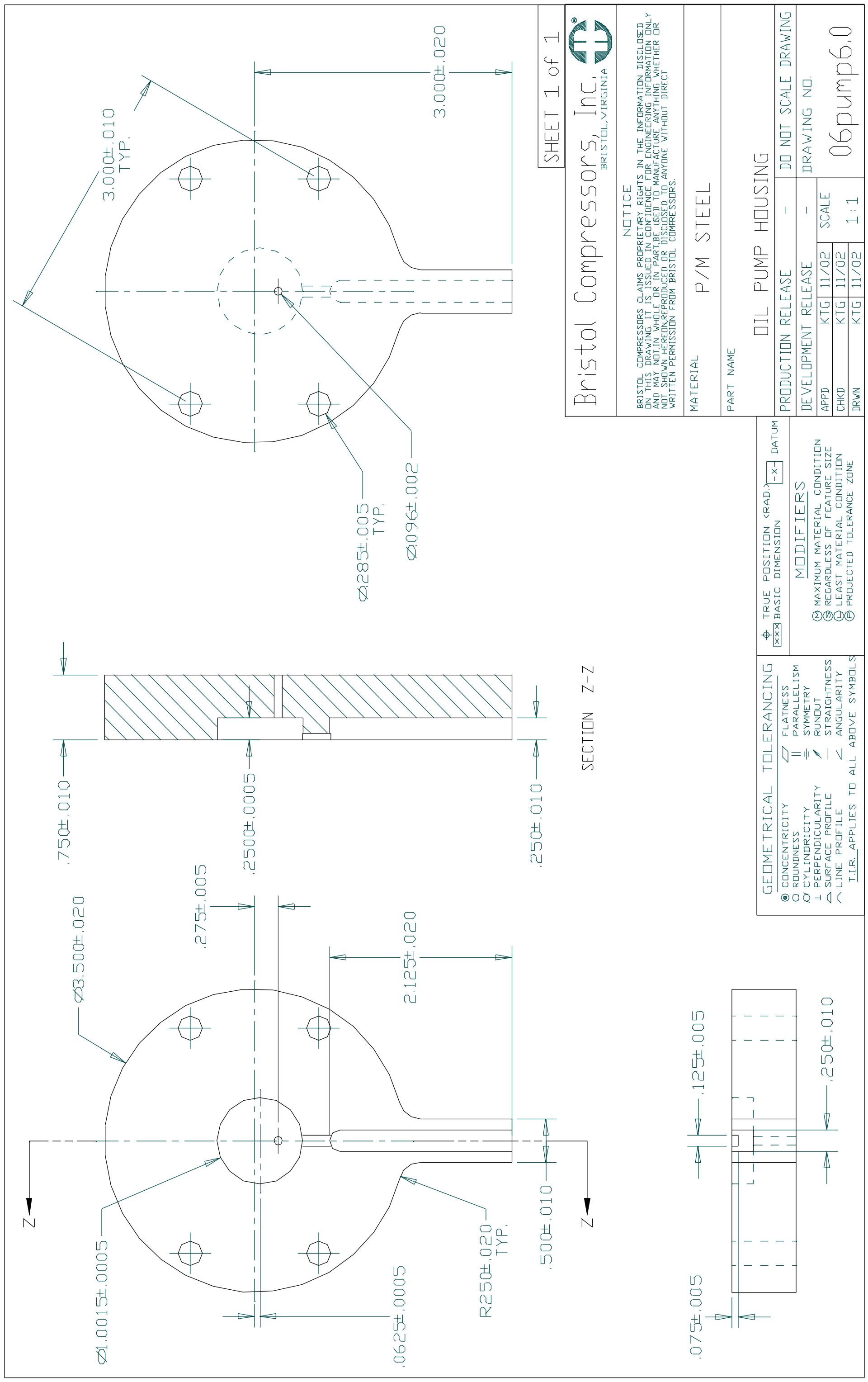
PRODUCTION RELEASE -
DEVELOPMENT RELEASE -
APPD KTG 9/02
CHKD KTG 9/02
DRWN KTG 9/02
DO NOT SCALE DRAWING
DRAWING NO.
06cage6.0
SCALE
1:1
GEOMETRICAL TOLERANCING
CONCENTRICITY
ROUNDNESS
CYLINDRICITY
PERPENDICULARITY
SURFACE PROFILE
LINE PROFILE
FLATNESS
PARALLELISM
SYMMETRY
RUNOUT
STRAIGHTNESS
ANGULARITY
T.I.R. APPLIES TO ALL ABOVE SYMBOLS
TRUE POSITION (RAD.)
BASIC DIMENSION
DATUM
MODIFIERS
MAXIMUM MATERIAL CONDITION
REGARDLESS OF FEATURE SIZE
LEAST MATERIAL CONDITION
PROJECTED TOLERANCE ZONE
2.125±.020
R250±.020 TYP.
3.000±.010 TYP.
.500±.010
3.000±.020
.548±.010
.260±.005
R020±.002
.198±.005
SECTION Z-Z
Ø.281±.005 TYP.
Ø.8750±.0004
Ø2.3750±.0005
.250±.010
Ø3.500±.020
Z
Z



SHEET 1 of 1
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BRISTOL, VIRGINIA
NOTICE
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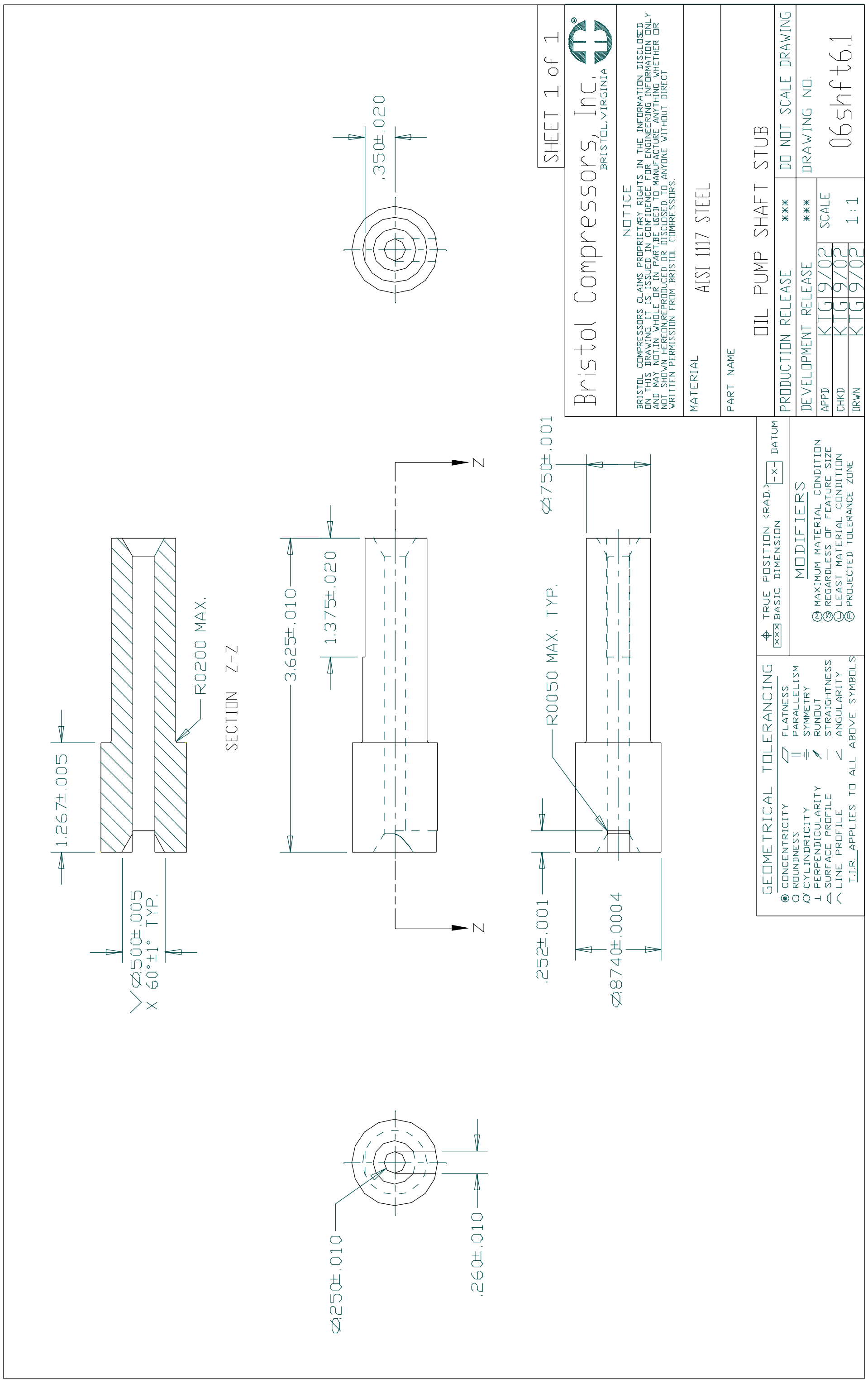
MATERIAL
P/M STEEL
PART NAME
OIL PUMP HOUSING
PRODUCTION RELEASE -
DEVELOPMENT RELEASE -
DO NOT SCALE DRAWING
DRAWING NO.
06pump6.0
APPD KTG 11/02
CHKD KTG 11/02
DRWN KTG 11/02
SCALE 1:1
GEOMETRICAL TOLERANCING
CONCENTRICITY
ROUNDNESS
CYLINDRICITY
PERPENDICULARITY
SURFACE PROFILE
LINE PROFILE
FLATNESS
PARALLELISM
SYMMETRY
RUNOUT
STRAIGHTNESS
ANGULARITY
T.I.R. APPLIES TO ALL ABOVE SYMBOLS
TRUE POSITION (RAD.)
BASIC DIMENSION
DATUM
MODIFIERS
MAXIMUM MATERIAL CONDITION
REGARDLESS OF FEATURE SIZE
LEAST MATERIAL CONDITION
PROJECTED TOLERANCE ZONE
3.000±.010 TYP.
3.000±.020
Ø.285±.005 TYP.
Ø.096±.002
.750±.010
.2500±.0005
.250±.010
SECTION Z-Z
Ø3.500±.020
.275±.005
2.125±.020
Ø1.0015±.0005
.0625±.0005
R.250±.020 TYP.
.500±.010
Z
Z
.125±.005
.250±.010
.075±.005



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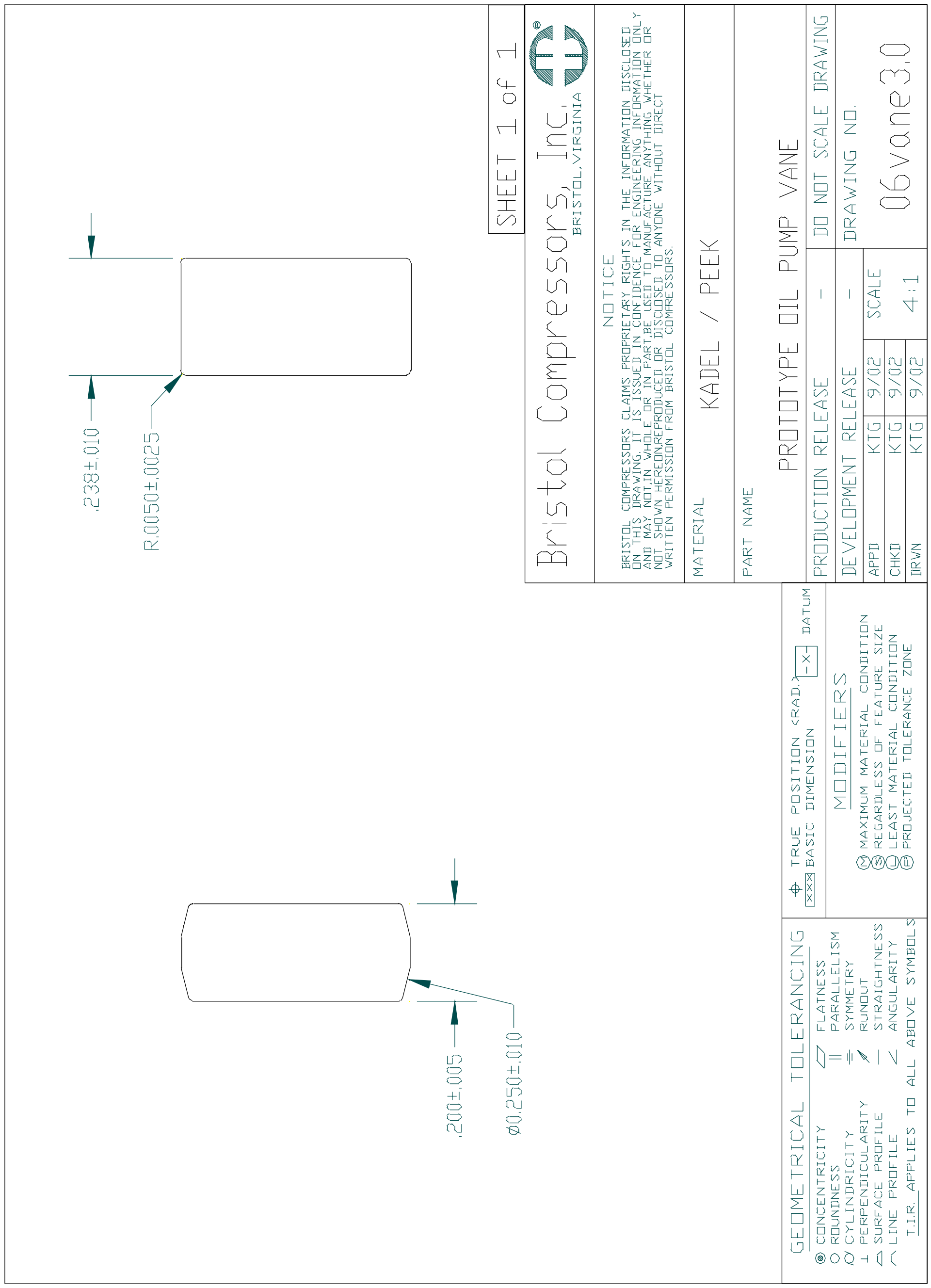
MATERIAL
AISI 1117 STEEL
PART NAME
OIL PUMP SHAFT STUB
PRODUCTION RELEASE ***
DEVELOPMENT RELEASE ***
DO NOT SCALE DRAWING
DRAWING NO.
06shft61
SCALE 1:1
APPD
CHKD
DRWN
GEOMETRICAL TOLERANCING
CONCENTRICITY
ROUNDNESS
CYLINDRICITY
PERPENDICULARITY
SURFACE PROFILE
LINE PROFILE
FLATNESS
PARALLELISM
SYMMETRY
RUNOUT
STRAIGHTNESS
ANGULARITY
T.I.R. APPLIES TO ALL ABOVE SYMBOLS
TRUE POSITION (RAD.)
BASIC DIMENSION
DATUM
MODIFIERS
MAXIMUM MATERIAL CONDITION
REGARDLESS OF FEATURE SIZE
LEAST MATERIAL CONDITION
PROJECTED TOLERANCE ZONE
.350±.020
Z
SECTION Z-Z
R0200 MAX.
1.267±.005
Ø.500±.005
X 60°±1° TYP.
3.625±.010
1.375±.020
Ø.750±.001
R0050 MAX. TYP.
.252±.001
Ø.8740±.0004
Ø.250±.010
.260±.010



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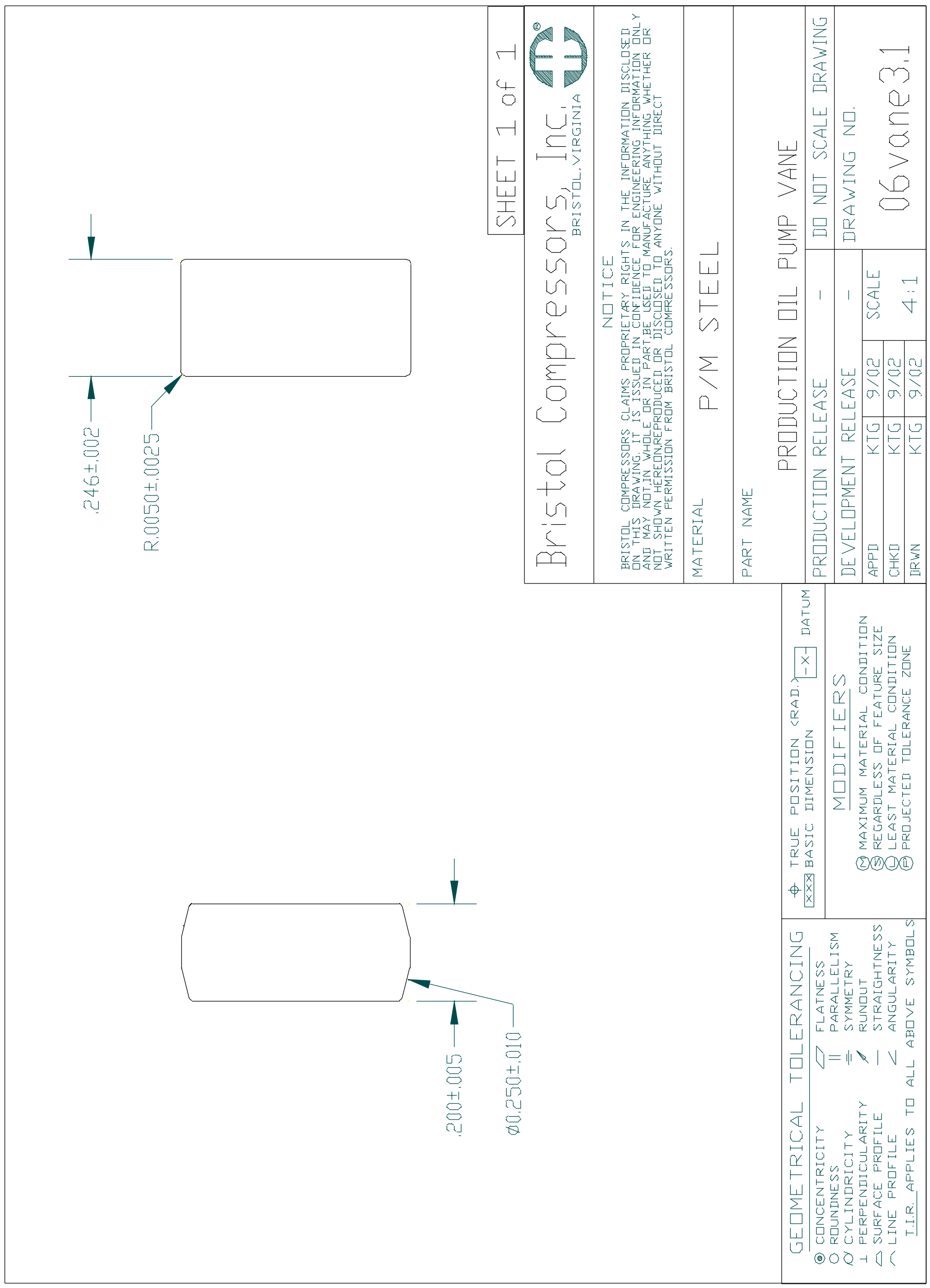
MATERIAL
KADEL / PEEK
PART NAME
PROTOTYPE OIL PUMP VANE
PRODUCTION RELEASE -
DEVELOPMENT RELEASE -
DO NOT SCALE DRAWING
DRAWING NO.
06vane3.0
APPD KTG 9/02
CHKD KTG 9/02
DRWN KTG 9/02
SCALE 4:1
.238±.010
R.0050±.0025
.200±.005
Ø0.250±.010
GEOMETRICAL TOLERANCING
CONCENTRICITY
ROUNDNESS
CYLINDRICITY
PERPENDICULARITY
SURFACE PROFILE
LINE PROFILE
FLATNESS
PARALLELISM
SYMMETRY
RUNOUT
STRAIGHTNESS
ANGULARITY
T.I.R. APPLIES TO ALL ABOVE SYMBOLS
TRUE POSITION (RAD.)
BASIC DIMENSION
DATUM
MODIFIERS
MAXIMUM MATERIAL CONDITION
REGARDLESS OF FEATURE SIZE
LEAST MATERIAL CONDITION
PROJECTED TOLERANCE ZONE



.246±.002
R.0050±.0025
.200±.005
Ø0.250±.010
SHEET 1 of 1
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MATERIAL
P/M STEEL
PART NAME
PRODUCTION OIL PUMP VANE
PRODUCTION RELEASE
-
DEVELOPMENT RELEASE
-
DO NOT SCALE DRAWING
DRAWING NO.
06vane3.1
APPD KTG 9/02
CHKD KTG 9/02
DRWN KTG 9/02
SCALE
4:1
GEOMETRICAL TOLERANCING
CONCENTRICITY
ROUNDNESS
CYLINDRICITY
PERPENDICULARITY
SURFACE PROFILE
LINE PROFILE
FLATNESS
PARALLELISM
SYMMETRY
RUNOUT
STRAIGHTNESS
ANGULARITY
T.I.R. APPLIES TO ALL ABOVE SYMBOLS
TRUE POSITION (RAD.)
BASIC DIMENSION
-X- DATUM
MODIFIERS
MAXIMUM MATERIAL CONDITION
REGARDLESS OF FEATURE SIZE
LEAST MATERIAL CONDITION
PROJECTED TOLERANCE ZONE

VITA

KENNETH T. GILBERT

Personal Data:	Date of Birth: June 7, 1968 Place of Birth: Oneida, Tennessee Marital Status: Married
Education:	Public Schools, Oneida, Tennessee Tennessee Technological University, Cookeville, Tennessee; mechanical engineering, B.S., 1991 Naval Nuclear Power School, Orlando, Florida; 1992
Professional Experience:	Naval Officer, United States Navy, 1991-1993 Senior Draftsman, Fruehauf Trailer Corporation; Huntsville, Tennessee, 1994-1995 Graduate Assistant, East Tennessee State University, College of Applied Science and Technology, 1996-1997 Development Engineer, Bristol Compressors, Incorporated; Bristol, Virginia, 1998-2003