Executive Summary

This report represents the culmination of work that has been done over the past Fall 2008 semester towards the design and construction of a test rig whose sole purpose is to determine the most effective labyrinth seal design. The project is sponsored by Danfoss Turbocor, an industrial compressor company. In their compressors are several different designs and sizes of labyrinth seals, however they are unsure on which design is the most effective at stopping the leakage of refrigerant 134a. The purpose of the test rig is to determine which design most effectively does this.

The test rig utilizes a vertical "top-down" design in which there are two chambers on either side of a seal mounting plate. The chamber on top will be pressurized to approximately 400kPa and the flow across the seal into the un-pressurized chamber will be monitored. To do this a redundant measurement system will be utilized to ensure accuracy. The three methods of measuring the mass flow across the seal are: an Omega brand flowmeter, a mass balance calculation based on the conditions of the gas cylinder (which is supplying the fluid), and finally pressure transducers which will monitor the conditions inside the high pressure chamber.

Another aspect of the design is the request that the concentricity between the shaft and the seal be adjustable. This is to determine what effect if any concentricity has on a seals ability to prevent leakage. Concentricity adjustment will be accomplished through the use of a differential threaded mechanism which allows the seal to be moved in relation to the shaft on a level of microns.

Finally, it was requested the air replace R134a as the working fluid for health, safety, and monetary reasons. However, it is still important that conditions inside the test rig accurately represent those in a Danfoss Turbocor compressor. To do this working labyrinth seal relations developed by Dr. Egli were used to determine the predicted mass flow rate and calculate the Reynolds numbers of the two fluids in hopes that a relationship between the two could be found.

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Introduction

Problem Definition & Objective

The objective of this project is to design and build a test rig that has the ability to quantitatively determine the amount of leakage across a labyrinth seal. Multiple seal designs of various sizes must be able to be tested. The test rig must also accurately replicate conditions inside a typical Danfoss Turbocor compressor while using air in place of R134a. Finally a study must be performed using the test rig in order to determine which seal design is the most effective at stopping fluid flow.

Background Information

According to Flitney and Brown [1], a labyrinth seal operates on following two methodologies: rotating radial faces cause centrifugal separation of liquid or solid from air and a series of restrictions followed by a clear volume creates expansion of a gas and hence reduces the pressure. These seals use a very small gap in between the seal and the rotating shaft, and then grooves are machined into the seal in order to disrupt the flow. A general design of a labyrinth seal is shown in figure 1 [2]. The fluid is prevented from leaking through the seal by the grooves which induce turbulence and misdirect the flow into the small gaps between each tooth. According to Boyce [2], a labyrinth seal has the following advantages: simplicity, reliability, tolerance to dirt, system adaptability, very low shaft power consumptions, material selection flexibility, minimal effect on rotor dynamics, back diffusion reduction, integration of pressure, lack of pressure limitations, and tolerance to gross thermal variations. Boyce [2] further claims disadvantages associated with this type of seal are the following: high leakage, loss of machine efficiency, increased buffering costs, tolerance to ingestion of particulates with resulting damage to other critical items such as bearings, the possibility

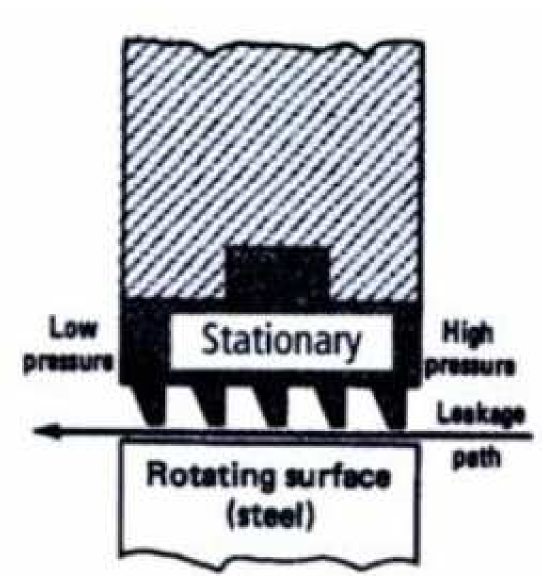


Figure 1: A generic Design of a Traditional Labyrinth Seal

of the cavity clogging due to low gas velocities or back diffusion, and the inability to provide a simple seal systems that meets OSHA or EPA standards.

There are several variations of the generic seal design (discussed above) currently in use at Danfoss - Turbocor. The designs vary in tooth number, tooth size and spacing, step number, and sizing. Much research has been preformed regarding the labyrinth seal designs, however engineers at Danfoss-Turbocor are uncertain as to what combination of variants will produce the least amount of leakage through the seal.

An experiment was conducted at Texas A&M University in order to determine the most effective configuration of teeth in a labyrinth seal. Figure 2 [3] represents the test rig used in their study. Despite the fact that the Texas A&M study had a more specific focus (tooth size), it will still provide valuable insight as well as numbers to which the results of this study may be compared to.

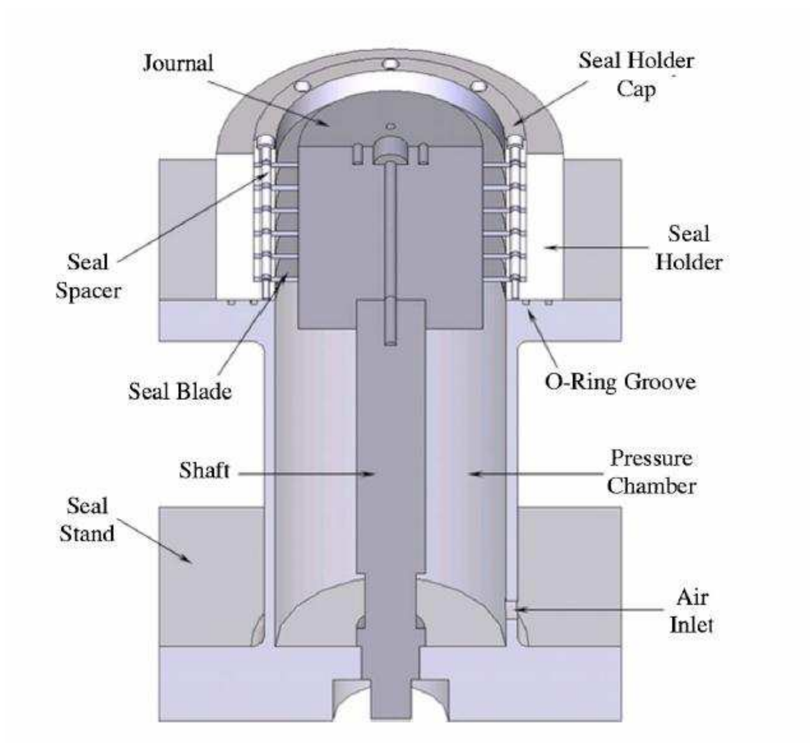


Figure 2: Test Rig used in the Texas A&M University study to determine most effective tooth configuration

Needs Assessment

Danfoss Turbocor manufactures state of the art compressors for air conditioning systems, and a crucial part of their compressors is a labyrinth seal that prevents the refrigerant from leaking from the high pressure compression stage into the low pressure portion of the compressor. The company has implemented different labyrinth seal designs; however, they have failed to determine conclusively which design yields the most efficient results (I.e. least amount of leakage through the seal). Danfoss Turbocor needs a test rig which will be able to provide quantitative results on the amount of leakage that is encountered at this labyrinth seal. The test rig should be adjustable to fit various seal sizes, shaft alignments, and experience different pressurized testing conditions. It has also been requested that the working fluid of the test rig be air instead of R134A in order to provide a safer test rig and minimize test costs. Danfoss Turbocor also inquired about a possible use of a CFD (Computational Fluid Dynamics) analysis of the seal, but this analysis was revoked from the requirements, due to a lack of experience of this type of software use.

Product Specifications

In order to make the design process more manageable a coupling matrix was created to match the customer needs with prospective so that product specifications can be created. The coupling matrix shows a connection between what the customer wants and how the design will reflect those needs. The parameters in the horizontal direction indicate what the customer needs where as the vertical direction contains the product specifications. The intersection of the two shows how each specification is relevant to the various needs. The customer has indicated that they need the product to be able to test multiple seal designs, sizes, to vary the shaft concentricity. Simulation of an actual seal conditions inside a Danfoss Turbocor compressor is vital while maintaining relatively low cost. They also would like the rig to be eco-friendly as well as safe to the operator.

		Product Specifications			
		Use air instead of R134a	Differential thread mechanism	Flowrmeter, Pressure Transducers, mass balance systm	Pressure Gradient – Gas Cylinder
Customer Needs	Test Multiple Seal Designs		X	X	X
	Test Multiple Seal Sizes		X	X	X
	Vary Shaft Concentricity		X		
	Eco-Friendly	X			X
	Accurate Internal Compressor Conditions	X		X	X
	Low cost	X			

Table 1: Coupling Matrix. Listed along the left side are the customer needs, and listed along the top are the product specifications. The "X" indicates where the needs are met

Over the course of designing the test rig several different design options have been considered, but one trait was shared by all options: the need for high and low pressure sides in order to create a differential across the seal. Since pressure is the driving force behind the fluid’s movement, it is essential that the test rig incorporate a pressure gradient across the seal in order to accurately recreate compressor conditions.

High Pressure Housing

In order to create leakage, a pressure difference must be created on either side of the provided seal, and to ensure proper calculations of this leakage, the high pressure air being supplied to the system must be held constant. It will be important to know that the high pressure housing does not leak anywhere but the seal itself. This creates the important specification that the high pressure housing will have the ability to attach to the

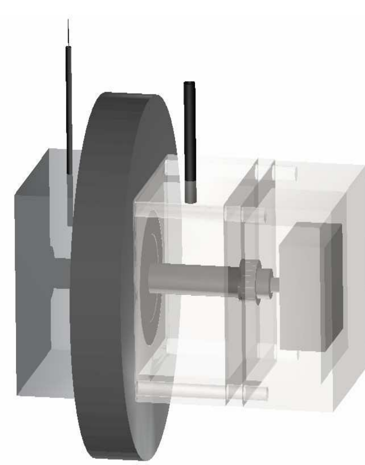


Figure 3: Early Generic Design of the test rig, incorporating a high and low pressure box on either side of the seal.

seal and provide an airtight connection. Using information provided by Danfoss-Turbocor, the highest pressure seen inside the compressor is near 200psi. It can then be assumed that the high pressure housing of the test rig would not be exposed to pressures higher than those seen at the maximum values of the Danfoss-Turbocor compressor. The design of the test rig should be accompanied by calculations of the forces due to pressure, and should then be able to maintain structural integrity during testing. When considering materials for the high pressure housing, it should be determined which option is most likely to resist deformation and leakage and in order to prevent leakage, the construction of this subsystem should utilize gaskets/o-rings, chemical welding for plastic applications, or standard welding for metal applications.

Low Pressure Housing

This subsystem will not be subjected to the same pressure as the high pressure housing, and therefore will be more flexible in the material selection process. However, this portion of the test rig will be responsible for containing the leaking flow, and therefore it is required that this subsystem be able to eliminate any leakage within the housing.

While there are several other subsystems of the test rig that were involved in making the coupling matrix, Aspects of each subsystem has become more refined than the original broad ideas that were initially used to generate the matrix. As such these ideas will be discussed in detail in later section in this report. These topics include but are not limited to: concentricity adjustments, measurement of flow and concentricity, calculations containing to the Reynolds numbers, and the method used to maintain a pressure gradient.

Design Selection

Design 1

The key component of this concept that differs from others is a high-pressure gas cylinder. From here on the concept will be referred to as “the gas-cylinder concept” for simplicity. For the fluid to flow across the seal, it is dependent upon a pressure difference between the two sides of the seal. In this concept, the high-pressure side of the seal will be maintained by a pressure regulator attached to a fixed-volume high pressure reservoir: a gas cylinder. The gas cylinder will start the experiment with a pressure that is much higher than that required to run the experiment (for example: experiment pressure = 150 psi, cylinder pressure = 2500 psi). The cylinder will contain an initial fixed, measurable mass. The mass of the gas inside the cylinder can be found by making the assumption that the gas inside is an ideal gas, and then by applying the ideal gas law.

$$P \cdot V = m \cdot R \cdot T$$

Equation 1: The Ideal Gas Law

In equation 1, the variable P is pressure inside the cylinder, V is the fixed, geometric volume of the cylinder, m is the total mass of the gas inside the cylinder, R is a universal gas constant, and T is the absolute temperature of the gas inside the cylinder. The volume of the cylinder is known. By rearranging the equation, the mass of the gas inside the cylinder can be found if the state of the gas is fixed by pressure and temperature simply by rearranging equation 1.

$$m = \frac{P \cdot V}{R \cdot T}$$

Equation 2: The Ideal Gas law rearranged to solve for mass

By Applying the Ideal Gas Law before and after the test the mass of air inside the cylinder may be found at both times. Subtracting the final mass from the initial mass in the cylinder will yield a change in mass, Δm:

$$\Delta m = m_{start} - m_{end}$$

Equation 3: Change in mass of air inside the gas cylinder (kg)

Timing the duration of the test the change in mass over time can be found. By definition this would yield the average mass flow rate.

$$m' = \frac{\Delta m}{\Delta t}$$

Equation 4: Average mass flow rate over time (kg/s)

One problem with this version of the time-averaged calculation is that the test has a start-up time. The start-up time is a result of the development of steady-state conditions inside the high-pressure reservoir on the high-pressure side of the seal, which is located after the gas regulator which is connected to the high-pressure cylinder. This "mid-pressure" zone holds the operating pressure, the high-side pressure, of the labyrinth seal. As an important note: throughout our design we refer to this mid-pressure zone as the high pressure reservoir, but in this document it will be referred to as the mid-pressure zone, or chamber. The mid-pressure is set as a standard operating pressure at which all of the seals will be tested for performance comparisons. During the start of the test, the mid-pressure zone (the high-pressure side of the seal that is maintained/supplied by the regulator and the cylinder) will be at standard atmosphere pressure and will need to be "charged" (pressurized) to the operating pressure. During the charging stage of the experiment, the mass flow rate through the seal is not the same as the steady-state mass flow rate, and will be lower than the steady state flow. The *unsteady flow rate* can be compared across the different types of seals, but this comparison is complicated and beyond the scope of this project.

A more accurate way to find a time-averaged mass flow rate at a standardized pressure would be to wait until a steady-state flow condition has been achieved. At this condition, mass flow rate into the system from the cylinder is equal to the mass flow rate through the labyrinth seal. The related equation is defined as:

$$dm = \frac{P \cdot V}{R \cdot T} \cdot dt \qquad \frac{dm}{dt} = \frac{P \cdot V}{R \cdot T}$$

Equation 5: Change in mass flow with respect to time and the ideal gas law relations

The above equation can be applied twice: it can be applied to the gas cylinder and to the mid-pressure chamber. The volume used in the equation, and for both applications, is constant. The volume is known for the cylinder, but will be difficult to calculate for the mid-pressure chamber due to a complex geometry. Fortunately, the volume the mid-pressure chamber does not need to be known, it is only important that the volume does not change. If volume and the universal gas constant are both constants and do not change with time, then they play an insignificant role in equation 5 (listed below) and the critical parameters become pressure and temperature (which can change with time). If pressure and temperature do not change inside the mid-pressure chamber, then dm/dt (the change in mass inside the chamber with respect to time) is zero, and the flow state can be considered "steady," hence the term "steady-state condition." For the purposes of this design concept, it is an accurate assumption to treat the mass flow as if it were in steady state.

Graphically plotting the mass inside the cylinder versus time will yield a straight, diagonal line reflecting a constant decrease in mass.

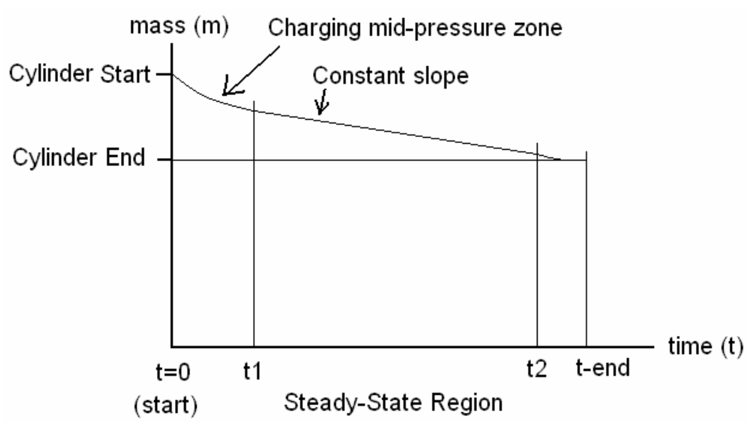


Figure 4: Prediction of the relationship between air mass inside the cylinder and the time over which the test is run

In this graph, t1 and t2 correspond to the start and end time across the steady-state condition. The zone corresponding to the constant slope is the steady-state zone where dm/dt in the mid-pressure chamber is zero. Keep in mind that dm/dt is not a measure of the mass flow rate, it is a measure of the mass capacitance of the mid-pressure chamber: as pressure increases and temperature is held constant, the chamber will take in more mass and dm/dt will be a positive value. Below is a graph predicting the mass flow rate through the seal based upon the discussed concept:

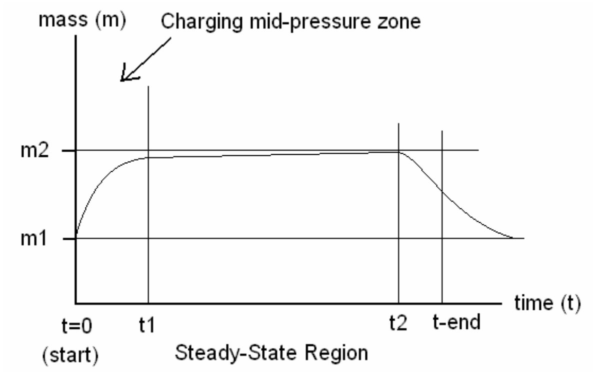


Figure 5: Prediction of mass change inside the "Mid-Pressure" Chamber

In this graph, the slope in the steady state region is shown to be slightly non-constant (exaggerated) to display the asymptotic relation as the experiment attempts to pressurize the chamber while simultaneously leaking through the labyrinth seal. For the purpose of this concept, it would be assumed that the change in mass flow rate during this time period would be negligible, or would be acknowledged and accounted for in error analysis.

Benefits and Drawbacks of the Gas Cylinder Concept

A major benefit of the cylinder method is that it can be used in conjunction with other instantaneous mass flow testing methods such as with a Venturi meter, Orfice meter, or Pitot-probe-utilizing these other methods as a check. If a graph is created that depicts mass flow rate calculated from other methods versus time, numerical integration methods can be used to determine the total mass measured by those meters (The area under the graph would represent the total mass passing through the meter). This total mass can be calculated across a steady-state condition and compared to a result obtained from the gas cylinder.

One of the drawbacks to the gas cylinder method is cost. While the cylinder itself can be rented, the purchase of the compressed air and of the measurement devices is a concern. From one source, Mr. Bill Starch, Machine Shop Manager at the Applied Superconductivity Center associated with Florida State University, rental fees are on the range of 6 to 8 dollars per month, and the cost of purchasing compressed air is close to 6 dollars for a full cylinder. It is not expected that renting to Danfoss Turbocor (DTC) would cost much more, despite these numbers representing a special FSU-Airgas pricing. The gas cylinder would also require a digital, high-pressure, pressure transducer to monitor the pressure in the gas cylinder, if DTC is unable to provide one, procuring the device could become expensive. This pressure transducer could cost as much as $125, according to Industrial Automation’s online store. We would also need a temperature probe, but may be able to work with DTC to avoid having to make a purchase.

Final Design

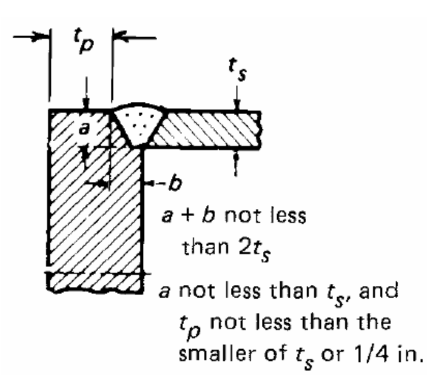
The labyrinth seal test rig assembly is comprised of four main subsystems: the high pressure housing, the adjustable seal mount, the low pressure housing, and the structural components. These subsystems work together to perform an analysis of the leakage rates of air through a gap located between the labyrinth seal and the balancing piston. The balancing piston will be rigidly fixed to the shaft, and it will serve as the fixed reference frame for the concentricity adjustments. Each subsystem has components which must operate effectively in order for the entire rig to operate successfully, and the following descriptions will explicitly outline how each of these components must be constructed in order to perform in such a manner

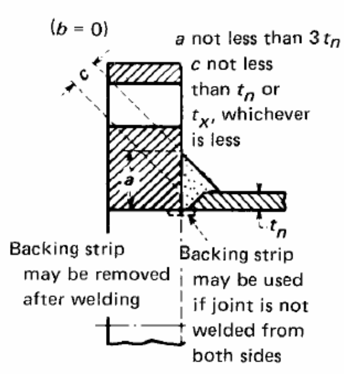
High Pressure Housing

To properly conduct testing on numerous seals, it is important that each seal be tested under identical conditions. The pressure conditions of a single test must be consistent throughout the entirety of that test, and the conditions should also be consistent over the entire range of tests preformed. To ensure these conditions remain constant, the high pressure housing must be formed into one solid airtight component. For cost purposes, the ideal method for forming this subsystem into an airtight container will be to weld circular plates onto either side of a cylinder. The following components which are being discussed will be assembled to one another, and form a subsystem which will be referred to as the high pressure housing.

The first component of this subsystem is the high pressure cylinder, and in this application, it will be constructed from a DOM seamless structural round steel tube with an outer diameter of six inches and a wall thickness of one-quarter inch. This item will initially be two feet in length, but will be cut down to meet a specified length. The circular plates which are attached at either end of this cylinder will be machined from a one-half inch thick rectangular piece of A36 steel. These two plates will be referred to as the high pressure cap (upper plate) and the high pressure mount (lower plate). Once the high pressure cap is machined into a circular shape, the only other required machining is a threaded hole which will allow the addition of a quick connect male adapter. The male adapter will allow for an easy connection to the high pressure air source, whether it is a pressure regulator located on a gas cylinder or a connection to shop air. This male adapter will have a threaded end which will be configured to install directly into the high pressure cap, and in order to ensure that no leaks are created at this junction, a silicon tape will be wrapped around the threads of the male adapter prior to installation of this component. The high pressure mount will create the connection between the high pressure air source and the labyrinth seal being tested. Therefore, the high pressure mount will be responsible for transferring the pressure force from the high pressure housing to the bolts which secure its position. The high pressure mount will have eight holes drilled through it, and each of these will be offset forty-five degrees from each other at a location four and one-half inches from the center. In order to keep this connection airtight, an o-ring must be in place at the mating surfaces of the high pressure mount and the seal mount.

The high pressure mount, cylinder, and cap will be connected to each other through a two circumferential welds. The American Society of Mechanical Engineers (ASME) has outlined welding standards for pressure vessels found in the 1998 Pressure Vessel and Boiler Code. The proper geometry for welding of a flange and flathead are depicted below. The following figures are found in the 1998 Section VIII – Division 1 of the ASME Pressure Vessel and Boiler Codes.





Figures 6&7: Welding geometries for flanged and flat head cylinders

Adjustable Seal Mount

The main purpose of this subsystem is to provide a location for the seal to be connected to both the high pressure housing and the low pressure housing. It also allows the labyrinth seal to be rigidly connected to a component of the test rig, and also allows the seal to be adjustable relative to the location of the fixed position of the shaft. In order to properly satisfy these requirements, the following important components will be utilized within this subsystem: the seal mount, the labyrinth seal, the differential threading mechanism, and the use of various o-rings.

The seal mount will be machined into a circular plate made from one-half inch thick A36 steel, and it will have eight one-half inch diameter holes drilled through the plate at a radial location of four and one-half inches. These holes will be offset forty-five degrees from each other, and will serve as the standard bolt pattern for each of the connecting subsystems. The labyrinth seal will be mounted directly to the seal mount through the use of four three-eighth inch diameter bolts. These bolts will secure the seal onto to the mount, and they will also create the compression needed to ensure the o-ring is functioning properly. In order to properly use o-rings, a groove must be machined into one of the sealing surfaces. This groove is commonly referred to as a gland, and the sizing of the gland depends on which type of pressure the o-ring will be subjected to, either internal (outward pressure direction) or external (internal pressure direction). The following figure and table was provided by the Parker Hannifin Corporation, and was used as the primary source for determining the proper sizing of the gland:

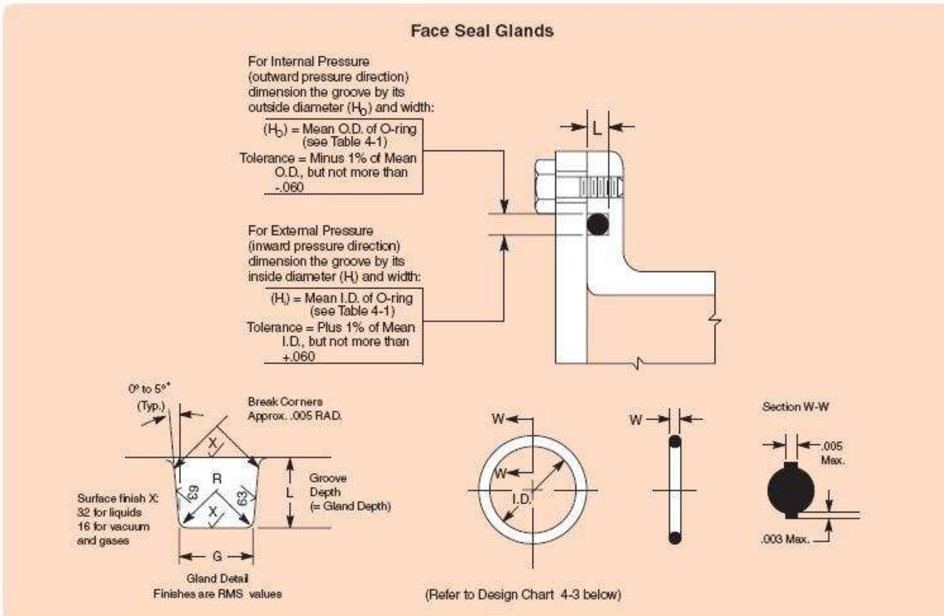


Figure 8: Illustrates location of variables listed in Table 2

O-Ring Face Seal Glands These dimensions are intended primarily for face type O-ring seals and low temperature applications.

O-Ring Size Parker No. 2	W Cross Section Nominal	W Cross Section Actual	L Gland Depth	Squeeze Actual	Squeeze %	G Groove Width Liquids	G Groove Width Vacuum and Gases	R Groove Radius
004 through 050	1/16	.070 ±.003 (1.78 mm)	.050 to .054	.013 to .023	19 to 32	.101 to .107	.084 to .089	.005 to .015
102 through 178	3/32	.103 ±.003 (2.62 mm)	.074 to .080	.020 to .032	20 to 30	.136 to .142	.120 to .125	.005 to .015
201 through 284	1/8	.139 ±.004 (3.53 mm	.101 to .107	.028 to .042	20 to 30	.177 to .187	.158 to .164	.010 to .025
309 through 395	3/16	.210 ±.005 (5.33 mm)	.152 to .162	.043 to .063	21 to 30	.270 to .290	.239 to .244	.020 to .035
425 through 475	1/4	.275 ±.006 (6.99 mm)	.201 to .211	.058 to .080	21 to 29	.342 to .362	.309 to .314	.020 to .035
Special	3/8	.375 ±.007 (9.52 mm)	.276 to .286	.082 to .106	22 to 28	.475 to .485	.419 to .424	.030 to .045
Special	1/2	.500 ±.008 (12.7 mm)	.370 to .380	.112 to .138	22 to 27	.638 to .645	.560 to .565	.030 to .045

Table 2: Used as the primary means to determine the proper sizing of a gland.

The seal mount will have three glands machined into its surface, with two being located on the top surface, and one located on the bottom. The three sealed surfaces will be between the seal mount and the labyrinth seal, the high pressure housing, and the low pressure housing. The sizes of o-rings is governed by the AS 568A standard, and a table of these standard dimension has been attached in the appendix of this report. The data shown in the appendix is a condensed version of the entire catalog, and it shows the dimensions of the o-ring specified in this design. The o-ring at the labyrinth seal is an AS 568A standard size 233, and the o-rings at the high pressure housing and the low pressure housing are both the AS 568A standard size 255.

The labyrinth seals used in this test rig will be manufactured by Danfoss Turbocor, and should each have identical bolt patterns, allowing for interchangeability of different seals for a single seal mount. This bolt pattern will consist of four ¼-28 threaded holes, and bolts will used to fasten the seal to the mount.

Differential Threading

Differential threading will be used in order to make concentricity adjustments, however this topic is addressed in the following section: Detailed Design Analysis.

Low pressure housing

The purpose of the low pressure housing is to capture all the air which leaks through the labyrinth seal, and direct this air through a measurable location. The construction of this subsystem will be very similar to that of the high pressure housing, in that it will consist of the three same major components: a mount, a cylinder, and cap. The low pressure mount and cap have some differences from the high pressure housing, and these will be discussed in detail below. Even though this subsystem will not need to be able to withstand the same pressures as the high pressure housing, it will be constructed with the same procedures.

The low pressure housing has three important functional requirements. It must maintain an airtight connection with the seal mount, it must be supported by the structural components of the rig, and it must channel the collected flow through a measurable location. The mount will be connected directly to the bottom side of the seal mount, and the main purpose for this component is to create an airtight connection between the mount and the low pressure housing. This will be achieved through the use of the o-ring found on the bottom side of the seal mount. The low pressure mount will have the standard hole pattern which align the seal mount and high pressure housing, but four of these holes will have a one inch diameter countersink their bottom surface. These countersinks will allow the mount to be properly positioned on the four structural rods which are responsible for supporting the weight of all the components listed above. The low pressure cap is a critical component in this subsystem. It must provide a location for radial bearings which support the shaft, and it must also provide and outlet for the leaking air to escape. The radial bearing will need to be press fit into the low pressure cap prior to the welding of the low pressure cylinder, as the press fitting of this bearing will be necessary in eliminating any possible air leaks through this location. The cap will have a threaded outlet where an elbow fitting will connect, and this will be responsible for transferring the leaked air from the low pressure housing to the flow meter. The threading of this outlet will require the use of silicon tape to reduce the possibilities of leakage at this junction. From this elbow, a pipe will extend to the connection of the flow meter.

Structural Subsystem

The purpose of this subsystem is to provide support for each component listed above. This support will be provided by three separate components: the legs, the base plate, and the spacers. The construction of the legs and the spacers will be very similar in nature, as both are one inch diameter steel rods with a threaded hole at either one or both ends. The legs will have one threaded end, which will be rigidly attached to the base plate by the use of 3/8-20 bolts, and the unthreaded end of the four legs will rest freely on the ground. The base plate is a critical component for the support of the rig and also the shaft. The support of the low pressure housing, the seal mount, and the high pressure housing each rely on the structural integrity of this component. The spacers will be rigidly connected to the base plate through the use of 3/8-20 bolts, and each spacer will fit into a one inch diameter countersink located on the base plate. This will aid in the stability of the rig during the assembly, and will allow for an easier connection of the bolts. It is very important to consider the amount of axial loading that will occur due to the pressure force. This load will be distributed through the shaft onto the base plate; therefore the test rig will utilize thrust bearings at this location.

Detailed Design Analysis

Flow Measurement Systems & Instrumentation

A major portion of this project involves the ability to accurately measure the amount of flow that is passing through the seal. In order to ensure the accuracy of the readings found by the rig, there will be three separate methods of measuring the flow so that they may serve as checks against each other. These systems are: a flowmeter, mass balance system for the gas cylinder supplying the air, and finally Pressure transducers which will monitor the conditions inside the high pressure chamber for any pressure drop over time.

The primary and simplest way to measure the flow is using a flow-meter. This will attach to the low pressure chamber of the rig and as the flow moves out to atmospheric pressure it will be recorded. This is the most direct way to measure the flow and it utilizes the advances in technology to measure the flow. Problems that were encountered by using a flow-metering is selecting one in the range estimated for this project. A flow-meters price increases by the range it can measure. For the range that is estimated in this project the flow-meters start at approximately half the allowed budget. Because of the price, it is extremely important that the correct flow meter be purchased. After comparing various meters it was decided that an Omega brand flowmeter model FMA-5000 would be used and will cost approximately $648.00.

The secondary measurement system relies upon a mass balance system for the gas cylinder of compressed air. This concept involves the tank being attached to the high pressure side via hose and a pressure regulator. This regulator will help to maintain the constant pressure gradient by keeping the high pressure side at a steady state. This ensures that a constant velocity will be maintained through the seal due to a constant pressure difference across the seal. This difference in pressure is vital to the leakage through the seal. Changing the velocity by allowing the high pressure side to lose pressure would disrupt the steady state condition. This would make measuring the flow much more difficult and in turn could cause that particular seal to be rated incorrectly.

The tank used for this part of the experiment will be compressed air at an initial pressure of 2500 psi. This large pressure is needed to maintain the pressure of 60 psi (400kPa) inside the high pressure chamber as a nominal value. The time for the experiment to reach steady state is currently unknown, however, the large high pressure air tank will be large enough to last through the trial. These tanks can be rented cheaply, especially with air being the internal gas. The mass of the tank before and after the experiment will be known. This value along with the time over which the experiment is run can be used to solve for a mass flow rate for the leakage across the seal. Assuming no gaps or holes anywhere in the high pressure chamber, the fluid must be presumed to flow through the seal, which can be recorded as the flow rate. The flow rate of each seal should vary and the seal that allows the least amount of flow to pass through will be rated more favorably.

The third check on the system will be performed by Pressure transducers placed inside the high pressure chamber. These will monitor the chamber to ensure that a steady state is maintained and alert the testers should the conditions become otherwise. They will also be able to provide information on how quickly the pressure will drop once the supple is stopped. This information can be used to generate pressure curves over time.

It should also be noted that a pressure regulator and transducer will be used to control and monitory the supplied flow, however, they are not discussed in this section due to the fact that they are used mainly for control and not taking measurements.

Shaft Concentricity Adjustment and Measurement

Labyrinth seals are a form of non-contact seal. The nature of this seal requires a gap, in this case the diameter of a shaft is smaller than that of the labyrinth seal into which it is inserted. It is theorized that a cylindrical labyrinth seal's performance is based upon, among other factors, the concentricity of the shaft and seal (concentricity is the centering of two circles). Specifically, if the shaft is closer to the seal on one side, it is theorized that the leak rate will increase. The test rig needs to be designed to allow for the measurement of concentricity and also for the alteration of concentricity so that its affect on a seal's performance may be analyzed.

The term 'concentricity' refers to the centering of two circles, one within the other. Concentricity is measured in two dimensions: in polar coordinates, an angle and radius are required. A rectangular coordinate system will take an x and y measurement. Figure 6 shows the two methods of measuring the concentricity of two circles.

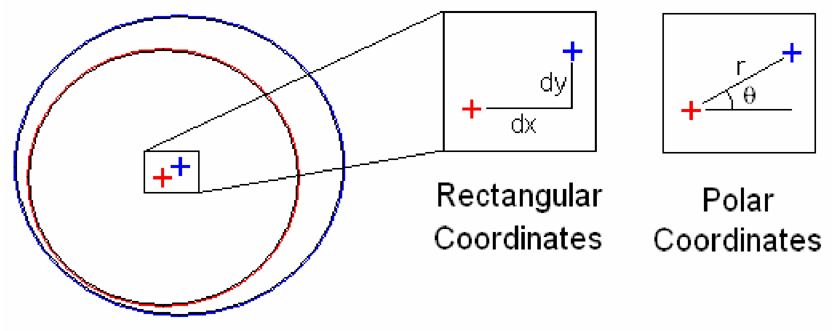


Figure 9: Concentricity measured in both rectangular and polar coordinates

Concentricity Adjustment System

In order to test the effect of concentricity, the capability for adjustment must be designed into the test rig. To do this, either the shaft or the seal must be able to be adjusted and moved in relation to the other. Since the shaft will be fixed to two sets of bearings, it is simpler to adjust the seal position while the shaft remain a stationary reference. In order to facilitate movement, the mounting holes on the seal mounting plate will be larger than the mounting screws protruding from the labyrinth seal design-billet. The labyrinth seal design billet is a cylindrical piece of material with mounting holes in a standardized configuration. The mounting billet will be given to DTC for the machining of a labyrinth seal within specified dimensions. The mounting screws protruding from the labyrinth seal billet will feed through the mounting plate and are tightened with flange washers and nuts to cover the gap from the over-toleranced holes. Designing the holes larger than the screws allows the labyrinth seal a certain amount of freedom, in this case enough freedom to allow for concentricity adjustments.

The shaft and seal concentricity tolerance is on the level of single micron lengths and a movement of that minute magnitude is needed in order meet the needs of the system. The small scale of the tolerance presents a challenge. The devised solution uses a ramp-like system. Essentially, a unit-change in one direction will yield a fractional-change in a perpendicular direction. A highly precise screw-system will be used to accomplish the task of converting the unit changes into fractional moves.

One of the smallest thread pitches, in English units, is the designation 0-80 (zero-eighty), and has 80 threads per inch. A full revolution of this screw gives a displacement of 0.0125 inches, or 317.5 micrometers. Single-degree turns of this screw would yield displacements acceptable for the purposes of this project; however, there are several other factors that must be considered. There is a frictional force between the components that are moving, and calculations predict that with the diameter of a 0-80 screw, an axial deformation on the micron level will be present. In addition to the deformation issue, single-degree turns would be difficult to achieve with human hands. To resolve this problem a smaller thread pitch and larger screw diameter are needed. Unfortunately the next size up had a thread pitch of .355 mm which is too large for the application. To resolve this issue, engineers at Danfoss Turbocor suggested utilizing a ‘differential threaded mechanism.”

The differential threaded mechanism uses two screws of different thread pitches in order to develop a displacement that is equivalent to the difference of the pitches. One of the screws is of a larger diameter and has threads on outer surface as well as a screw hole in the center. This screw is referred to in the mechanism as “the dual-threaded screw,” or DTS for short. The smaller screw, which is inserted into the DTS, has a thread pitch that is slightly different than that of the DTS. The smaller screw is fixed to the part that will be moved (the labyrinth seal), and the dual-threaded screw screws onto it. The dual-threaded screw also screws into the reference (a flange on the shaft) to which the seal is moving in relation to. Below is a sketch of the differential threaded mechanism. In color, the DTS is green, the smaller screw is blue, the labyrinth seal is pink, and the reference is orange. The small screw and the labyrinth seal are fixed and will not unscrew. In the final design, the labyrinth seal is indirectly attached to the smaller screw. When the DTS is turned counter-clockwise it screws out through the reference and at the same time draws out the smaller screw that is fixed to the labyrinth seal. While the DTS is turning counter-clockwise and screwing out of the reference, the smaller screw is turning clockwise with respect to the DTS. This motion causes the smaller screw to unscrew from the DTS. The result is a displacement of the labyrinth seal about the final reference that is a value of the difference of the two screws’ displacements.

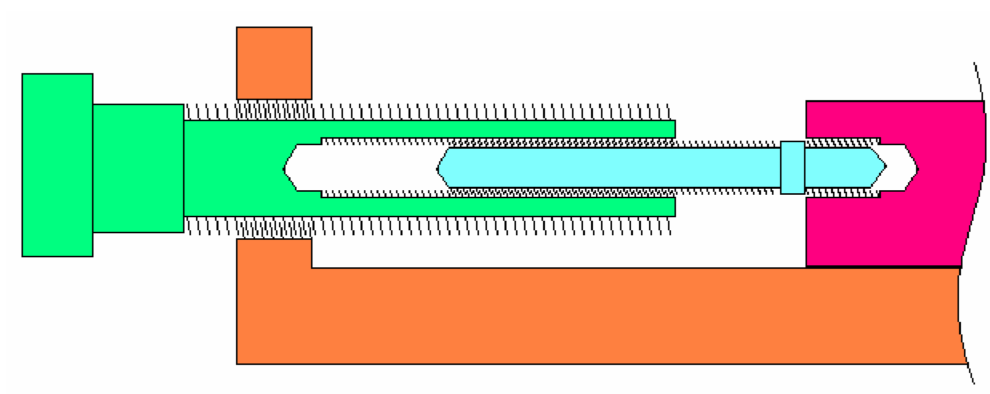


Figure 10: Differential Thread Mechanism Assembly

Dual Thread Dimensions and Assembly

Available resources dictate that the taps and dies used to manufacture the device are in English units. Through iteration and calculations, it has been determined that the mechanism will utilize a larger-diameter screw of thread type 3/8-24 and the smaller screw will be ¼-28. These numbers represent 3/8 inch diameter with 24 threads per inch, and ¼ diameter with 28 threads per inch. The respective thread pitches are 1.058 mm and 0.907mm. For one full revolution, the total displacement is equal to the difference between the two pitches, or 0.151 mm. The total displacement for a turn of 5 degrees would be 2.09 microns.

The multi-threaded system presents unique complications to assembly. The system cannot be assembled with the reference and seal (screw attachment points) pre-fixed. In order to assemble the system, the mounting bracket must be able to detach from the rest of the system. First, the smaller screw will be permanently attached to the adjustor plate to which the labyrinth seal will be attached. Next, the mounting bracket will be loosely put over the smaller screw (they will not attach due to diameter difference). Next, the dual-threaded screw is screwed onto the smaller screw to a preset distance, thus fixing the mounting bracket in between the labyrinth mount and the DTS. The mounting bracket is then screwed onto the DTS to a preset distance. The entire assembly is then loosely attached to the reference, with the mounting bracket attaching to the reference frame. The labyrinth seal is attached to the assembly. This condition is the "ready position" for labyrinth seal adjustment. Once the concentricity is measured, two of the differential threaded mechanisms will be used to adjust the seal position in an x and y-direction until the concentricity is within a target range. Once the concentricity is verified, under-toleranced bolts (loose diameter) will be tightened to fix the position. This is the ready-to-test condition for concentricity concerns.

Safety Calculations

The stresses in the materials were also calculated. If the contents above the moveable plate weigh 25 lbf, and with a coefficient of static friction for lubricated steel-on-steel of 0.16, the net force to overcome static friction is 17.8 Newtons. This force will generate a deformation in the DTS as well as the smaller screw. With the diameters of the two screws known, the cross-sectional area of the two screws can be found (the DTS has a wall-thickness area). Data was found for the yield strengths and modulus of elasticity for over 20 classes of low-carbon steel. The tensile yield stresses of all of the steels were above 170 MPa, and all of the modulus' of elasticity were close to 180 GPa. Using this information, it was found that the total deformation for a small screw exposed length of 2 cm, and a DTS effective length of 2.5 cm, was 0.124

micrometers. The factor of safety in both parts was over 300, so failure is unlikely for unforeseen loads. Sample calculations for the screw sizing and angular rotation, forces experienced, and factor of safety can all be found in Appendix 2.E.

Concentricity Measurement

Due to the minute scale that all of the concentricity adjustments will be made on, it is important that the method used to measure the concentricity be precise. The concentricity will be found by measuring the distance from the shaft's outer diameter to the seal's inner diameter. After reviewing several measurement options, a multi-armed dial gauge micrometer was chosen. Danfoss Turbocor is providing the measuring device from their surplus to ensure that the accuracy needed is met. The gauge is capable of measuring a change in distance between 0 and 1000 microns, which means that if the shaft is off-center by more than 1 centimeter the measuring device will be ineffective. The arms of the gauge are able to be both moved and fixed, with the meter located at the end of the reach. The dial gauge has a strong, magnetic base that can be fixed anywhere so long as the surface is magnetic. The gauge can then take measurements in relation to its position. For instance, If the dial gauge was placed on the stationary rig body and shaft was rotated the gauge would measure any wobble experienced by the shaft.

In the case of this project, the magnetic base will be attached to the shaft, and the arms will position the meter to a cylindrical edge of the seal, perpendicular to the shaft axis. The arms will be fixed in this position. The radius from the center of the magnetic base to the center of the shaft is also fixed. With all of the arms in a locked position, the radius from the center of the shaft to the end of the arm chain is fixed.

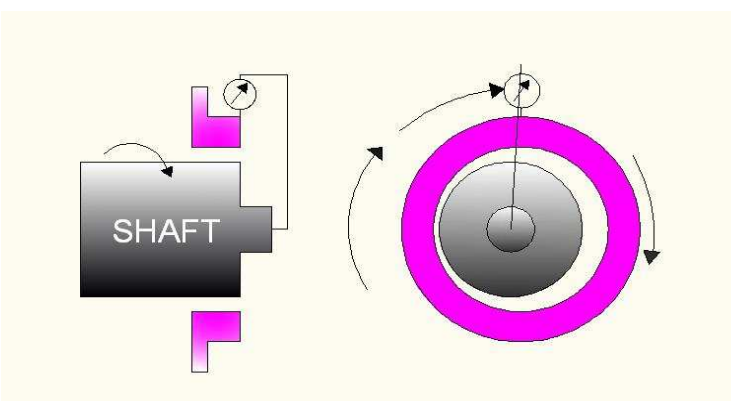


Figure 11: Conceptual diagram of a dial gage. The picture on the left is a side profile while the drawing on the right is from the perspective of looking at the shaft head on

The meter is fixed to the end of the arm chain, and as the shaft is rotated the guage will register any changes in radius in relation to the shaft; I.e. the gauge will measure any changes in the distance from the shaft to the seal. Figure 8 displays a conceptual drawing of how a dial gauge will be utilized in this application.

Material Selection

After completion of the final design, it was decided to use A36 steel to build the test rig. There were several factors that led to this decision, some of which are: magnetic properties, machine-ability, weld-ability, strength, and price. Although the flow rate through the seal does not depend on any type of magnetic field, the test rig must be made out of a magnetic material in order to accommodate the dial gauge which is to be used for concentricity measurements. The best way to ensure a air tight seal on the pressure chambers is to weld them closed (accept where the seal attaches). Due to both of these requirements the materials that could be chosen became limited. A36 steel was able to fill both of these requirements. In addition, it is also relatively easy to machine and when bought from an appropriate vendor, fairly inexpensive. In an effort to further decrease material costs a single material was chosen so that a single vendor could be used.

To create the pressure chambers, a 6" diameter, 2' long and ½" thick piece of steel tube will be purchased. A single tube will be purchased in order to save money and it will be cut to length for each pressure chamber. There is some scrap anticipated to be left over from this process. By purchasing a tube, no further machining needs to be done on the tubes other than cutting them to length. Also a cylindrical pressure chamber is able to withstand higher pressures than a rectangular box. Also, should the need arise for alter the chamber, cylindrical objects can be machined more accurately through use of a lathe over a mill.

In addition to the steel tube, two steel plates and a steel rod will be purchased. The plates of dimensions 2'x 2' x ¼' and 1' x 2' x ½" will be used both to weld shut the pressure chambers. The plates are also ½" thick so that the internal pressure stresses in the chambers are evenly distributed and there are no weak spots that could fail. Finally the steel rod (6ft long) will be cut down and used for legs for the rig body, spacers, and any other miscellaneous parts that may be needed.

Cost Analysis

This project was allocated a total budget of $1500.00 dollars to be used for the purpose of purchasing all materials and instruments that are necessary to build and test the labyrinth seal test rig. The budget was essentially allocated to three different areas:

raw materials needed to build the rig, measurement equipment needed to test the seals, and materials needed to make the differential threaded mechanism. An attempt was made to minimize cost and confusion by buying products from a minimal amount of vendors. Table 3 below contains an itemized list of required products, cost, and the vendor that supplies them.

Itemized Cost Analysis		
Item	Individual Cost	Vendor
Rig Body		
Steel Tube	$138.14	www.metalsdepot.com
Steel Plate 1' x 2' x 1/2"	$84.41	www.metalsdepot.com
Steel Plate 2' x 2' x 1/4"	$84.41	www.metalsdepot.com
Steel Rod 6 ft	$25.98	www.metalsdepot.com
Subtotal	$332.94	
Measurement Equipment		
Flow Meter	$648.00	www.omega.com
Pressure Gauge	$125.00	www.industrialautomation.com
Dial Gauge	$0.00	Danfoss Turbocor
Pressure Regulator	$0.00	Danfoss Turbocor
Pressure Transducers	$0.00	Danfoss Turbocor
Subtotal	$773.00	
Differential Threading Meachanism		
3/8 - 24 threaded rod 3 ft	$6.19	www.drillspot.com
1/4 - 28 threaded rod 3 ft	$6.19	www.drillspot.com
Subtotal	$12.38	

Table 3: Itemized cost analysis, including individual price and vendor

A large portion of the budget was able to be saved by Danfoss Turbocor generously making their spare measurement equipment available for use. They are supplying a dial gauge for concentricity measurements, a pressure regulator for the gas cylinder and pressure transducers for monitoring the conditions in the high pressure chamber. By saving money on these devices a greater amount was able to be allocated towards the purchase of a quality precision flow meter.

Overall Cost Analysis	
Item	Cost
Rig Body	
Steel Tube	$138.14
Steel Platting	168.82
Steel Rod	$25.98
Subtotal	$332.94
Measurement Equipment	
Flow Meter	$648.00
Pressure Gauge	$125.00
Dial Gauge	$0.00
Pressure Regulator	$0.00
Pressure Transducers	$0.00
Subtotal	$773.00
Differential Threading Mechanism	
3/8 - 24 threaded rod	$6.19
1/4 - 28 threaded rod	$6.19
Subtotal	$12.38
S&H Estimate	$95.82
Total	$1,214.14

Table 4: Lists the total allocated budget to date

As can be seen in Table 4 (above) the project is currently coming in under budget, however it is important to note that several necessary purchases have yet to be included in the analysis. The most obvious is the price of the gas cylinder. However, it has been omitted for the time being due to the fact that Danfoss Turbocor may be able to supply one. Should that not be supplied, several vendors are still being considered for selection so that the best price may be found. Also omitted is the cost of hardware: screws nuts, bolts, washers, etc. These are expected to be purchased at a local hardware store at minimal cost. Despite these things (cylinder and hardware) not being included in the cost analysis, research to date has lead to the belief that everything can be purchased at a reasonable low price and that the project will still come in under budget, or right at $1500.00 dollars.

Pertinent Calculations

Fluid Properties: Matching the Reynolds # of R134a and Air

To be the most use to Turbocor the test rig must be able to fit several different designs and sizes of labyrinth seals. For this project three seal designs will be used: Impeller Labyrinths, Main Labyrinths, and Interstage Labyrinths. These three designs represent the various seals that can be found at various locations inside a Turbocor compressor. The figure to the left is a diagram of an impeller Labyrinth seal. As can be see in the diagram there are multiple shaft diameters that vary based on the steps of the seal. For the impeller seals that will be tested there are two different sizes which have gap sizes of 0.8 mm and 0.2 mm. The figure also shows an excellent representation of the teeth on each step which serve a crucial role in the function of the seal. An impeller seal is capable of having 7, 8, or 10 teeth per step. Although not pictured the approximate dimensions for main and interstage seals are also known. The main labyrinth seal design has by far the most possible variations. At its smallest point the shaft diameter is 50 mm through the seal. This is the first of six steps which end in a shaft diameter that is 85.9 mm. The teeth on each step can be varied with the options of 0, 3, 9, or 13 teeth. Also the main seal design comes in several different sizes so that there are several different possible dimensions for the leak gap; they are: 0.091 mm, 0.113 mm, 0.14 mm, and 0.15 mm. Despite the variation in seal diameters, it is, in fact, the dimension of the distance in the gap between the shaft and the seal that carries the most weight. This is the location of the leak and the dimension is the value that will be used to calculate the Reynolds number so that air may accurately be used in place of R134a.

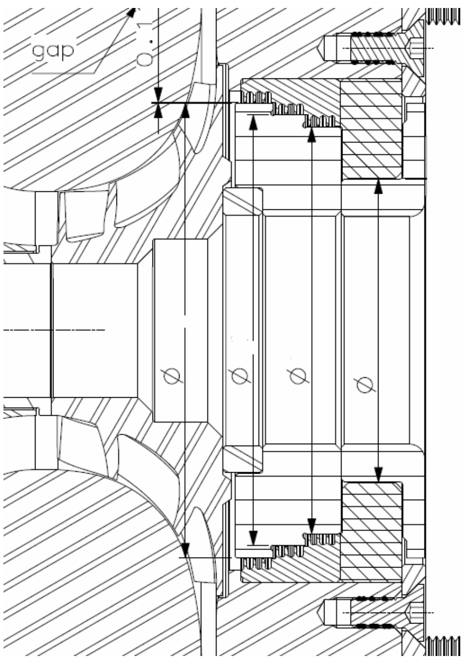


Figure 12: Diagram of impeller seal. The shaft diameters have been omitted in order to preserve their confidentiality

In order to perform accurate analysis on the results yielded by the test rig it is extremely important that a relationship is found between the flow of refrigerant and air.

Since the fluid properties of air and refrigerant are extraordinarily different a relationship will be formed based on the calculation of the Reynolds number for each fluid. Part of the problem Danfoss Turbocor is experiencing lies in the fact that certain values, such as the fluid velocity, are unknown at various points inside the compressor. This presented an obvious challenge due to the Reynolds number's reliance on that value. $\mathrm{Re} = \frac{VD}{\nu}$ An alternate methodology to find the Reynolds numbers was needed.

Calculations Methodology

In 1935 Dr. Egli became the leading authority on labyrinth seals by writing several classical papers which developed working relations that could be used when analyzing flow across the seals. Very little has changed on the topic of labyrinth seals and his relations are applicable to the analysis that needs to be performed for this project. The corner stone of Dr Egli's relations is a formula to find the mass flow rate across a seal with an unspecified number of teeth.

$$\dot{m}_L = \pi d \delta C_t C_c C_r \rho \sqrt{RT}$$

Equation 6: mass flow rate across a seal based on empirical values

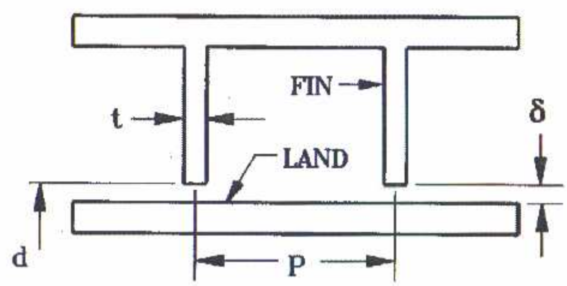


Figure 13: Generic diagram of a labyrinth seal that defines the variables used in Equation 6

The variables referred to in Equation 1 are as follows. "d" refers to the seal diameter, "δ" is the aforementioned gap between the seal and the shaft through which the fluid will be leaking; "t" refers to thickness of each tooth and "p" defines the spacing between individual teeth. Finally the values C_t, C_c, and C_r are all empirically determined values. They are affected by gap distance, number of teeth in a seal, tooth thickness, and tooth spacing. The formulas for these relations as well as the calculations that are outlined and discussed in this section are all available in Appendices 2.A - 2.C. Dr. Egli's relations also call for the fluid properties (ρ- density, R-gas constant, and T-temperature) be taken from the high pressure side of the seal.

Once the mass flow rate was found using Equation 1, the velocity could be determined based on the simple concept of conservation of mass : $\dot{m} = \rho V A$. Using the found velocity, and the gap size for the diameter variable, the Reynolds number could then be calculated. The same process is then performed using the properties of air instead of R134a. The ultimate goal of the calculations is to determine at what operating pressure and temperature the test rig should be run to best match the Reynolds numbers of R134a and air.

While the methodology for calculating and matching the Reynolds numbers appears simple, it is in fact a fairly complex and involved calculation. The complexity appears when the number of possible combinations of diameter, tooth count, and gap size are taken into account. The number of calculations is compounded by analyzing the three possible operating conditions for a Danfoss Turbocor compressor. Conditions 1 and 2 are both common conditions and they correspond to water cooled and air cooled applications respectively. Condition 3 is an extreme condition where the compressor experiences a high pressure ratio, however, this is a rare condition.

	Condition 1	Condition 2	Condition 3
Saturated Suction Temp (degC)	5.5	0	-2
Saturated Discharge Temp (degC)	36.1	50	55
Ps_{uct} (Kpa)	255	192	171
P_{disch} (Kpa)	813	1217	1391
P ratio	2.57	4.50	5.49
ΔP (Kpa)	558	1025	1220
Pi	469	520	536

ΔP main labyrinth	558	1025	1220
ΔP 2nd impeller labyrinth	255	192	171
ΔP1st impeller labyrinth	303	833	1049

Table 5: Summarizes the pressures and temperatures that can be seen in a typical Danfoss Compressor at 3 different operating conditions

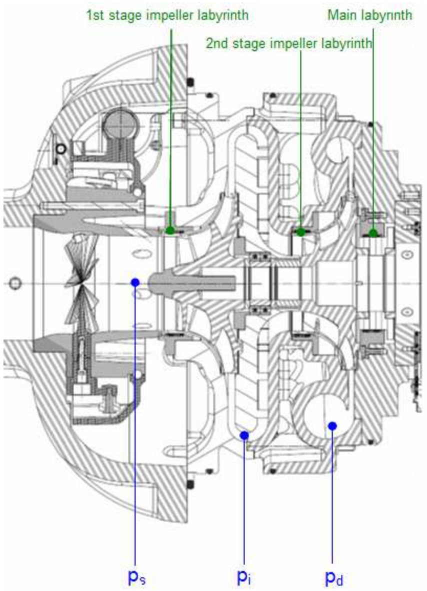


Figure 14: Accompanies Table 2 in displaying location of values

Calculation Results

When performing the necessary calculations to match the Reynolds numbers several factors needed to be taken into account for determining the test rig run conditions. The first is that the air in the high pressure chamber is at atmospheric (or room) temperature. This was done in order to eliminate designing or purchasing a heat exchanger to heat or cool the air to a specified temperature. By doing so the design was able to be made simpler and the budget was able to be conserved. The client also agreed to the temperature restriction. It is also important to take into account the pressure at which the high pressure chamber will be pressurized to due to equipment constraints. For instance, the Reynolds numbers may be able to be matched exactly at 500 MPa (This is an arbitrary number used for illustration), however, the ability to supply that high of pressure from a gas cylinder is unlikely. For this reason, pressures above 1 MPa were not considered in the calculations. Ultimately, it was decided that the best possible operating pressure for the test rig was 400 kPa (58psi). This choice was influenced by a relationship between mass flow and pressure that was discovered while testing the design prototype. This relationship is discussed in greater detail in the section devoted to the prototype.

The interstage labyrinth seal is the smallest of all the seals and also has the least variation in design combinations; these qualities make it an appropriate place to start the analysis of calculation results. The mass flow rates that are listed below are the mass flow rates of air at 400 kPa. Table with values for the Reynolds numbers and mass flow rates of both air and R134a for all seal combinations are available in Appendix 1. The largest mass flow rate that was seen across a seal was 0.023kg/s and it occurred at the seal diameter of 25.64 mm, a gap size of 0.18mm and 10 teeth. The lowest accurately calculated flow rate is .015 and occurs at a seal diameter of 29.64 mm with 7 teeth (there is only 1 possible gap dimension). The qualifier “accurately” is used due to a problem that was encountered when attempting to perform the calculations for 3 teeth. The empirical formulas that are used in the calculations require constants to subtracted from the tooth number (or some variation on the tooth number I.e. ln) which in this were larger than 3 (tooth number). The result of this is that the equations yield a negative number for the mass flow rate and the Reynolds number. While no numerical values can be found for this circumstance it can be inferred that there is a minimum number of required teeth for the interstage seal design, which is in this 4. This information may play an important role in later analysis of the seal performance.

The impeller seal design is larger and had more possible combinations: there are 4 possible diameter, 2 different gap sizes, and 3 different tooth counts. The largest and smallest mass flow rates were: 0.051 kg/s and 0.031 kg/s. The largest flow rate occurs at a seal diameter of 89.2 mm, a gap of 0.18mm, and 8 teeth. Meanwhile the smallest mass flow rate occurs at 2 separate seal diameters, 66mm and 67mm, at a gap of 0.18mm and 7 teeth.

The Main seal calculations were the most intensive due to the fact that there were 6 possible diameters, 4 gap sizes, and 4 different teeth counts. A variation on the previous calculations was for this seal design due to possibility of 0 teeth. One of the empirical formulas (please see Appendix2C) requires taking the natural log of the tooth count. Since doing so for 0 teeth would yield impossibility, it was assumed that the varrable relying upon that particular formula (C_t) was equal to 1. The normal method was used for calculating the mass flow for all other cases. The largest mass flow rate occurred at a seal diameter of 85.9 mm with a gap dimension of 0.15mm and 0 teeth with a value of 0.225 kg/s. Because that value is found using the special condition it is excluded and the next largest flow rate was 0.049kg/s and was located at a seal diameter of 85.9mm, gap dimension of 0.15mm and 9 teeth. The lowest flow rate was 0.011 kg/s located at a seal of 50 mm diameter, gap size of 0.15 mm and 3 teeth.

Although specific numbers are not mentioned in the body of this report, the calculated Reynolds numbers of air and R134a (which can be seen in the tables in Appendix 1) are not on the same order of magnitude and therefore do not match. After analysis of the results it was determined that matching the Reynolds numbers without altering the input temperature would be extremely difficult. Since the Reynolds number of R134a was several orders of magnitude larger than that of air (in most cases), scaling up the test rig was considered as a possible option to increase the reynolds number of air. Unfortunately since most of the dimensions are on a micron level, the scale of the rig would need to be increased many times before any significant effect would be felt by the Reynolds number. Scaling the rig that amount would make it infeasible to build or keep for later use by Danfoss Turbocor. For these reasons it was determined to keep the rig at the original dimensions. Instead of matching the Reynolds number to determine at which pressure and temperature the rig should run at, the test rig will be subjected to a pressure of 400kPa and atmospheric temperature. The relations previously calculated will then be used to perform a numerical analysis on the seal performance and compare the actual results to the theoretically projected results. It is believed that so long as each seal is tested under the same conditions the results will still be valid so long as an accurate numerical analysis is performed.

Internal Pressure Vessel Loading

It was determined that for safety reasons the forces experienced by the pressure inside each pressure chamber should be analyzed so as not to exceed the test rigs capabilities. Each chamber is made out of A36 steel, .635cm (½”) thick tubes and 0.14 m (5.5”) in diameter. A36 Steel has a tensile strength of 180MPa. With the pressures of the high and low sides known, 400kPa and 101kPa, it becomes a very simple matter to analyze the hoop and longitudinal stresses. $\sigma_1 = \frac{\Pr}{t}$ and $\sigma_2 = \frac{\Pr}{2t}$ are the formulas used to find the hoop and longitudinal stresses respectively. Based on these values, the high pressure chamber will experience a hoop stress of 5.51MPa and a longitudinal stress of

2.76MPa. The low pressure side will experience a hoop stress of 2.22 MPa and a longitudinal stress of 1.11MPa.

The factors of safety are found by dividing the tensile strength by the hoop or longitudinal stresses. $FS = \frac{\tau}{\sigma}$. Based on this formula the high pressure chamber has a factor of safety of 32.6 and 65.3 in the hoop and longitudinal directions. The low pressure side also has very high factors, 80.8 (hoop) and 161.5 (longitudinal). Based on these calculations it is determined that the test rig will be able to withstand all foreseen testing conditions. Complete calculations of the stresses and factors of safety can be seen in Appendix 2.D.

Bearing Load Analysis

While testing of the rig will be done without the rotation of the shaft, the client would like the option for rotation to be included in the design should they choose to add a motor at a later date. Rotation of the shaft will require the support of two bearings in order to withstand the forces generated. The bearings will be located towards the bottom of the test rig; one located on the outside and one inside the low pressure chamber. The bearing furthest from the seal will serve as the thrust bearing while the other will be the radial bearing. Ball bearings were selected for this application. Due to the possibility of very high speed rotations (approximately 10,000 RPMs) ball bearings were selected over roller bearings as they are better suited for high speed applications.

The process for selecting a bearing is a lengthy one. The first factor needed for the bearing information is the fatigue life. The projected lifetime of the bearings is solved for in millions of revolutions by multiplying time by the rotational speed. The time was estimated to be 10 hours. The projected lifetime is only a reference value. However, the bearings themselves will run for much longer than the 6 million revolutions predicted. This factor is to account for any possible fatigue loading the bearings will endure.

The loads on the bearings were the next factors that needed to be calculated. This was done by finding out the surface area exposed to the high pressure side of the rig. The maximum pressure reached inside the rig is estimated to be 100 pounds per square inch

(400 kPa). Once known, the area was multiplied by this maximum projected pressure inside the rig to find the force acting in the axial direction. This value was found to be around 5.6 kilo Newtons. Since the test rig is set up in a vertical arrangement, the weight of the shaft is also supported by the bearings. This weight was added to the force of the pressure to find a total axial pressure, however, the weight of the shaft was insignificant in comparison to the force caused by the pressure chamber. The bearing must meet this force requirement in order for the rig to maintain stability. The radial force has a negligible effect on the total force exerted on the bearings due to the rig's vertical orientation. A radial force was arbitrarily chosen to be a non-zero value of 90 Newtons.

The total force exerted can be used to solve for the dynamic load factor 'C', which is one of the two dominant force factors in bearing selection. The other factor to consider is the static load, 'C_0', which represents the amount of load that the bearing supports with no rotation of the rings before dimpling on the bearings will occur. Once the static and dynamic loads have been determined, a bearing must be chosen with a dynamic load close to the value given by the total force and projected lifetime. The dynamic load was calculated to be 18.775 kilo Newtons. The bearing chosen was 63/22, which has a dynamic load of 18.6 kilo Newtons.

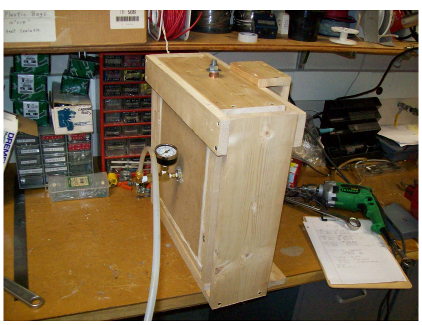
An iterative process was used to determine whether or not the bearings would meet the indicated requirements. Before the equation for equivalent load can be applied, several unknown factors must be determined. The unknowns include the rotation, thrust, radial factors and the radial and axial loads. The rotation factor was found to be 1 because the inner ring, not the outer ring, of the bearing rotates. The radial and axial loads are the values that were previously calculated. To solve the radial and thrust factors the axial load must be divided by the static load. The value given by the division of these two numbers yields a reference value "e" from the SKF bearing reference tables from the referenced text (Machine Design: an Integrated Approach). The ratio of axial to static load was found to have a value of .602. An interpolation can then be performed to find the value of "e" corresponding to the correct location in the table. Once "e" is known, it can be compared to the ratio of the axial force to the radial force. Depending on the comparison a different set of radial and thrust factors may be needed. In this application the thrust force is much greater than the radial force so the ratio of forces is larger. The column used gives a value of .56 for the radial factor and roughly 1 for the thrust factor. With all values in the equivalent equation solved for the load was found to be 5.6 kN. This load was used in the dynamic force load equation incorporating projected lifetime and the end result was 18.9 kN.

The final calculated dynamic load is .116 kN, which is more than the original dynamic load of the selected bearing. There is a 0.6% difference in the two values making the selected bearing appropriate for this application.

Another important aspect of this bearing selection is the diameter of the inner ring. The bearings chosen has an inside diameter of 22mm which is smaller than the shafts diameter of 25.4mm. This will require machining of the shaft in order to accommodate the bearings. As the diameter of the bearings increase the load they can withstand increases as well, however, the rotation they can be subjected too decreases. As stated before, this application requires a maximum speed of ten thousand revolutions. The decision was made that acquiring the correct load and allowing it to safely rotate at the maximum anticipated speed took precedent to the inner ring diameter. The shaft can be easily machined to comfortably fit the bearings in place. It must also be realized that the dynamic load calculated for the bearings is not the load the bearings will endure. This is merely a safeguard for the bearings incorporating the life cycle of the bearings and the rotation that they will subjected to. The calculations that correspond to the method described in this section are available in Appendix 2.F

Proof of Concept: A Prototype

After the final design had been decided upon, there were several concerns on whether the theory would actually work. In an effort to confirm the hypothesized behavior a prototype of the test rig was built out of wood using the scrap parts (seals) that had been provided by Danfoss Turbocor. Specifically, it was hoped that the prototype would confirm methodology of finding the mass flow rate. By verifying the mass flow calculations, an accurate approximation of the flow rate of air through the test rig could be made. Using this information a flowmeter could be decided upon. Due to the precision and cost that is typically associated with flow meters it was important to know the range over which it would be taking reading in order to avoid purchasing a meter that would prove ineffective. Confirmation of the mass flow rates would also provide insight into the feasibility of measuring the pressure drop over time in the high pressure chamber. There were concerns that the leak might be great enough to prevent steady state pressurization from being achieved in the chamber before the seal. It is important that the air leaves the chamber slow enough to pressurize the chamber as well as allow the pressure to drop in a measurable period of time.



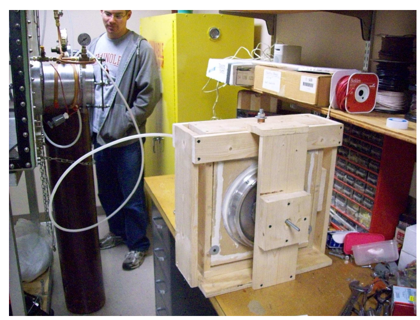


Figure 15 & 16: Shows the front and back of the prototype.

The prototype measured the mass flow across the seal by measuring the mass change inside the gas cylinder. Also Nitrogen was used in place of air due to the availability of nitrogen cylinders. The test was conducted at several pressure differentials, with the low pressure being ambient air pressure. During the prototype testing, several dynamic side effects were noticed. First, the gas cylinder became very cold. The temperature of the gas inside the cylinder is hypothesized to decrease with decompression, and to reach an equilibrium value where the temperature of the air in the cylinder is constant (do to non-adiabatic conditions of the cylinder). The point at which a hypothetical temperature equilibrium is reached is still under investigation. The second occurrence noticed pertains to pressurizing the high-pressure side of the labyrinth seal. The flow was increased from zero to a certain unknown value, m_0, before the pressure inside began to increase above atmosphere pressure. After m_0 was attained, a small increase in supply produced an immediate increase in pressure. This seemed to indicate that there is a "bottle-neck," or limiting factor to the flow. Analysis of test data from the final design tests is expected to show a correlation. The last, and perhaps the most important, is the required mass change in the cylinder to obtain statistically-relevant, time-averaged analysis of the mass flow: four tests with the prototype used a total of 64% of the total mass in the cylinder. Note that there is a mass loss during non-steady-state stabilization.

Prototype Test Procedure

Before beginning the test, a check to insure all of the necessary components were attached to the test rig was performed. The pressure was then turned on and the fluid began to flow into the high-pressure chamber. Flow was increased until the high-pressure chambers' pressure gauge began to indicate an increased pressure. The flow was then carefully adjusted to the operating pressure (this adjustment is small compared to initiating the flow to read a pressure increase). Steady-state conditions inside the high-pressure chamber are created as quickly as one can adjust the pressure regulator on the gas cylinder (with exception of temperature).

Once the operating pressure was reached data acquisition could begin. For the prototype, this included writing down simultaneously the time and pressure inside the high-pressure cylinder. The pressure graduations on the cylinder were in kilogram-force per square centimeter (kgf/cm^2), and one kilogram-force is equal to 9.80665 Newtons. The graduations on the gauge were every 5 kgf/cm^2. The pressure gauge on the high-pressure chamber had graduations at every 1 psi. The high-pressure chamber pressure gauge starts at 1 psi, so it is assumed that the gauge is not accurate at or below 1 psi. The tests were conducted across a change of 15 kgf/cm^2, according to the high-pressure cylinders' pressure gauge. Sample calculations of the work performed to determine the time averaged mass flow rate (during steady-state conditions) is available in Appendix 2.G

Prototype Test Results and Analysis

Below are the technical results from the prototype test.

Prototype Supply Gas Cylinder US DOT Designation: 3AA2400

Standard Internal Volume: 49.9 Liters

	Test 1	Test 2	Test 3	Test 4
Steady-State Pressure	2 psi	3 psi	4 psi	5 psi
Temperature (est.)	0° C	0° C	0° C	0° C
Cylinder Gas Mass Start	7.133 kg	5.992 kg	4.851 kg	3.424 kg
Cylinder Gas Mass End	6.277 kg	5.136 kg	3.995 kg	2.568 kg
Cylinder Gas Mass Change	0.856 kg	0.856 kg	0.856 kg	0.856 kg
Elapsed Test Time	140 s	185 s	110 s	147 s
Measured Mass Flow	0.00611 kg/s	0.00463 kg/s	0.00778 kg/s	0.00582 kg/s
Predicted Mass Flow	0.003625 kg/s	0.004202 kg/s	0.004635 kg/s	0.004976 kg/s
Measured Volumetric Flow	311.469 L/min	235.706 L/min	396.415 L/min	296.637 L/min
Predicted Volumetric Flow	183.712 L/min	212.946 L/min	234.891 L/min	252.139 L/min

Table 6: results of tests performed on the wooden prototype

Overall the prototype was very successful in achieving its intended purpose. Upon looking at the data several concepts were able to be confirmed. Perhaps the most substantial, is the accuracy of the mass flow equations in predicting the flow rate through the seal. This allows a flowmeter to be chosen with a high degree of confidence in the selection's ability to perform under the conditions that the test will be conducted under.

Also it was proven that the high pressure chamber can be pressurized and reach a steady state before starting data acquisition.

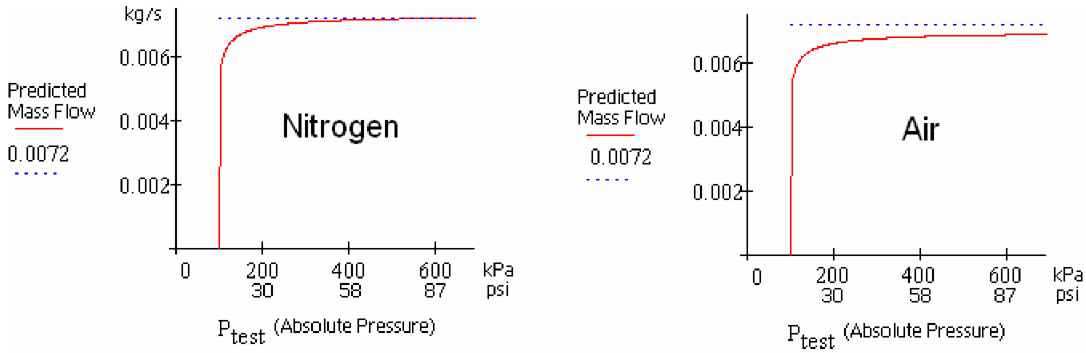


Figure 17: Graphs of the mass flow rate vs pressure for Nitrogen and Air

The above graphs illustrate two very important concepts. There is a minimum flow rate needed to generate a pressure gradient and there is also a maximum flow rate that can be achieved; once this flow rate is achieved it will not increase despite increases in pressure. Based on the results found during this test, the minimum required flow rate is approximately 0.005 kg/s. The mass flow rate also appears to asymptote at approximately 400 kPa (approximately 58 psi). It is partially due to these results that the decision to run the actual test rig at 400 kPa was made.

Error sources in the measured numbers come from unmeasured temperature changes, pressure gauge accuracy, misalignment of the shaft and seal, and a single knot in one of the planks in the wood that allowed a small leak. An exact error analysis was not performed due to the knot leak generating an unknown error.

References & Vendors

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- http://www.vxb.com/page/bearings/PROD/Kit7437?gclid=CNjThKX_nZcCFQJvswodVUaj9g

Vendors

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- http://www.drillspot.com/products/376896/Approved_Vendor_5MY72_Low_Carbon_Steel_Threaded_Rod
- http://www.omega.com
- http://www.shender4.com/thread_chart.htm

Appendix 1: Summary tables of Re and flow rates

Interstage Seal Results:

Seal:	Interstage			
Condition:	1		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.18		-34.191	
# Teeth:	3			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
29.64	-1.555E+04	-1.408E+03	-0.018	-2.410E-03
33.4	-1.554E+04	-1.407E+03	-0.020	-2.716E-03
35.64	-1.553E+04	-1.407E+03	-0.021	-2.898E-03
AVG	-1.554E+04	-1.407E+03	-0.020	-2.675E-03

Seal:	Interstage			
Condition:	1		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.18		217.289	
# Teeth:	7			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
29.64	9.894E+04	8.960E+03	0.111	0.015
33.4	9.887E+04	8.954E+03	0.126	0.017
35.64	9.884E+04	8.950E+03	0.134	0.018
AVG	9.888E+04	8.955E+03	0.124	0.017

Seal:	Interstage			
Condition:	1		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.18		268.416	
# Teeth:	10			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
29.64	1.216E+05	1.101E+04	0.137	0.019
33.4	1.215E+05	1.100E+04	0.154	0.021
35.64	1.214E+05	1.100E+04	0.165	0.023
AVG	1.215E+05	1.100E+04	0.152	0.021

Seal:	Interstage			
Condition:	2		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.18		-34.191	
# Teeth:	3			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
29.64	-2.705E+04	-1.408E+03	-0.032	-2.410E-03
33.4	-2.704E+04	-1.407E+03	-0.036	-2.716E-03
35.64	-2.703E+04	-1.407E+03	-0.039	-2.898E-03
AVG	-2.704E+04	-1.407E+03	-0.036	-2.675E-03

Seal:	Interstage			
Condition:	2		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.18		217.289	
# Teeth:	7			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
29.64	1.722E+05	8.960E+03	0.205	0.015
33.4	1.720E+05	8.954E+03	0.231	0.017
35.64	1.720E+05	8.950E+03	0.247	0.018
AVG	1.721E+05	8.955E+03	0.228	0.017

Seal:	Interstage			
Condition:	2		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.18		268.416	
# Teeth:	10			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
29.64	2.115E+05	1.101E+04	0.252	0.019
33.4	2.113E+05	1.100E+04	0.284	0.021
35.64	2.113E+05	1.100E+04	0.303	0.023
AVG	2.114E+05	1.100E+04	0.280	0.021

Seal:	Interstage			
Condition:	3		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.18		-34.191	
# Teeth:	3			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
29.64	-3.296E+04	-1.408E+03	-0.040	-2.410E-03
33.4	-3.294E+04	-1.407E+03	-0.045	-2.716E-03
35.64	-3.293E+04	-1.407E+03	-0.048	-2.898E-03
AVG	-3.294E+04	-1.407E+03	-0.044	-2.675E-03

Seal:	Interstage			
Condition:	3		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.18		217.289	
# Teeth:	7			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
29.64	2.097E+05	8.960E+03	0.256	0.015
33.4	2.096E+05	8.954E+03	0.288	0.017
35.64	2.095E	8.950E+03	0.307	0.018
AVG	2.097E+05	8.955E+03	0.284	0.017

Seal:	Interstage			
Condition:	3		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.18		268.416	
# Teeth:	10			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
29.64	2.577E+05	1.101E+04	0.314	0.019
33.4	2.575E+05	1.100E+04	0.354	0.021
35.64	2.574E+05	1.100E+04	0.378	0.023
AVG	2.575E+05	1.100E+04	0.349	0.021

Impeller Seal Results:

Seal:	Impeller			
Condition:	1		Avg Vol. flow Rate (L/min)	
Gap Size(mm):	0.18		511.269	
# Teeth:	7			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
66	9.86E+04	8.93E+03	0.248	0.034
67	9.861E+04	8.929E+03	0.252	0.035
82.2	9.856E+04	8.925E+03	0.309	0.043
89.2	9.854E+04	8.923E+03	0.336	0.046
AVG	9.858E+04	8.927E+03	0.286	0.040

Seal:	Impeller			
Condition:	1		Avg Vol. flow Rate (L/min)	
Gap Size(mm):	0.2		460.142	
# Teeth:	7			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
66	8.871E+04	8.033E+03	0.223	0.031
67	8.871E+04	8.033E+03	0.227	0.031
82.2	8.866E+04	8.028E+03	0.278	0.038
89.2	8.864E+04	8.027E+03	0.302	0.042
AVG	8.868E+04	8.030E+03	0.258	0.036

Seal:	Impeller			
Condition:	1		Avg Vol. flow Rate (L/min)	
Gap Size(mm):	0.18		562.396	
# Teeth:	8			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
66	1.096E+05	9.925E+03	0.276	0.038
67	1.096E+05	9.925E+03	0.280	0.039
82.2	1.095E+05	9.920E+03	0.344	0.047
89.2	1.095E+05	9.918E+03	0.373	0.051
AVG	1.096E+05	9.922E+03	0.318	0.044

Seal:	Impeller			
Condition:	1		Avg Vol. flow Rate (L/min)	
Gap Size(mm):	0.2		549.614	
# Teeth:	8			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
66	1.058E+05	9.583E+03	0.266	0.037
67	1.058E+05	9.583E+03	0.270	0.037
82.2	1.058E+05	9.577E+03	0.332	0.046
89.2	1.057E+05	9.575E+03	0.360	0.050
AVG	1.058E+05	9.580E+03	0.307	0.043

Seal:	Impeller			
Condition:	1		Avg Vol. flow Rate (L/min)	
Gap Size(mm):	0.18		626.305	
# Teeth:	10			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
66	1.211E+05	1.097E+04	0.305	0.042
67	1.211E+05	1.097E+04	0.310	0.043
82.2	1.211E+05	1.096E+04	0.380	0.052
89.2	1.211E+05	1.096E+04	0.412	0.057
AVG	1.211E+05	1.097E+04	0.352	0.049

Seal:	Impeller			
Condition:	1		Avg Vol. flow Rate (L/min)	
Gap Size(mm):	0.2		639.087	
# Teeth:	10			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
66	1.255E+05	1.136E+04	0.316	0.043
67	1.255E+05	1.136E+04	0.332	0.044
82.2	1.254E+05	1.136E+04	0.393	0.054
89.2	1.254E+04	1.135E+04	0.427	0.059
AVG	9.724E+04	1.136E+04	0.367	0.050

Seal:	Impeller			
Condition:	2		Avg Vol. flow Rate (L/min)	
Gap Size(mm):	0.18		511.269	
# Teeth:	7			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
66	1.716E+05	8.93E+03	0.457	0.034
67	1.716E+05	8.929E+03	0.464	0.035
82.2	1.715E+05	8.925E+03	0.569	0.043
89.2	1.715E+05	8.923E+03	0.617	0.046
AVG	1.716E+05	8.927E+03	0.527	0.040

Seal:	Impeller			
Condition:	2		Avg Vol. flow Rate (L/min)	
Gap Size(mm):	0.2		460.142	
# Teeth:	7			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
66	1.544E+05	8.033E+03	0.411	0.031
67	1.543E+05	8.033E+03	0.417	0.031
82.2	1.543E+05	8.028E+03	0.511	0.038
89.2	1.542E+05	8.027E+03	0.555	0.042
AVG	1.543E+05	8.030E+03	0.474	0.036

Seal:	Impeller			
Condition:	2		Avg Vol. flow Rate (L/min)	
Gap Size(mm):	0.18		562.396	
# Teeth:	8			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
66	1.907E+05	9.925E+03	0.508	0.038
67	1.907E+05	9.925E+03	0.515	0.039
82.2	1.906E+05	9.920E+03	0.632	0.047
89.2	1.906E+05	9.918E+03	0.686	0.051
AVG	1.907E+05	9.922E+03	0.585	0.044

Seal:	Impeller		Avg Vol. flow Rate (L/min)	
Condition:	2			
Gap Size(mm):	0.2		549.614	
# Teeth:	8			
Diameter (mm)	Re of R134a	Re of Air	Flow Rate R134a (kg/s	Flow Rate Air (kg/s)
66	1.841E+05	9.583E+03	0.490	0.037
67	1.841E+05	9.583E+03	0.497	0.037
82.2	1.840E+05	9.577E+03	0.610	0.046
89.2	1.840E+05	9.575E+03	0.662	0.050
AVG	1.841E+05	9.580E+03	0.565	0.043

Seal:	Impeller		Avg Vol. flow Rate (L/min)	
Condition:	2			
Gap Size(mm):	0.18		626.305	
# Teeth:	10			
Diameter (mm)	Re of R134a	Re of Air	Flow Rate R134a (kg/s	Flow Rate Air (kg/s)
66	2.108E+05	1.097E+04	0.561	0.042
67	2.108E+05	1.097E+04	0.569	0.043
82.2	2.107E+05	1.096E+04	0.699	0.052
89.2	2.106E+05	1.096E+04	0.758	0.057
AVG	2.107E+05	1.097E+04	0.647	0.049

Seal:	Impeller		Avg Vol. flow Rate (L/min)	
Condition:	2			
Gap Size(mm):	0.2		639.087	
# Teeth:	10			
Diameter (mm)	Re of R134a	Re of Air	Flow Rate R134a (kg/s	Flow Rate Air (kg/s)
66	2.183E+05	1.136E+04	0.581	0.043
67	2.183E+05	1.136E+04	0.590	0.044
82.2	8.182E+05	1.136E+04	0.723	0.054
89.2	2.181E+05	1.135E+04	0.785	0.059
AVG	3.682E+05	1.136E+04	0.670	0.050

Seal:	Impeller			
Condition:	3		Avg Vol. flow Rate (L/min)	
Gap Size(mm):	0.18		511.269	
# Teeth:	7			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
66	2.090E+05	8.93E+03	0.569	0.034
67	2.080E+05	8.929E+03	0.578	0.035
82.2	2.089E+05	8.925E+03	0.709	0.043
89.2	2.267E+05	8.923E+03	0.769	0.046
AVG	2.132E+05	8.927E+03	0.656	0.040

Seal:	Impeller			
Condition:	3		Avg Vol. flow Rate (L/min)	
Gap Size(mm):	0.2		460.142	
# Teeth:	7			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
66	1.880E+05	8.033E+03	0.512	0.031
67	1.880E+05	8.033E+03	0.520	0.031
82.2	1.879E+05	8.028E+03	0.637	0.038
89.2	2.039E+05	8.027E+03	0.692	0.042
AVG	1.920E+05	8.030E+03	0.590	0.036

Seal:	Impeller			
Condition:	3		Avg Vol. flow Rate (L/min)	
Gap Size(mm):	0.18		562.396	
# Teeth:	8			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
66	2.323E+05	9.925E+03	0.633	0.038
67	2.323E+05	9.925E+03	0.642	0.039
82.2	2.322E+05	9.920E+03	0.788	0.047
89.2	2.520E+05	9.918E+03	0.855	0.051
AVG	2.372E+05	9.922E+03	0.730	0.044

Seal:	Impeller		**Avg Vol. flow Rate (L/min)**	
Condition:	3			
Gap Size(mm):	0.2		549.614	
# Teeth:	8			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
66	2.243E+05	9.583E+03	0.611	0.037
67	2.243E+05	9.583E+03	0.620	0.037
82.2	2.242E+05	9.577E+03	0.760	0.046
89.2	2.433E+05	9.575E+03	0.825	0.050
AVG	2.290E+05	9.580E+03	0.704	0.043

Seal:	Impeller		**Avg Vol. flow Rate (L/min)**	
Condition:	3			
Gap Size(mm):	0.18		626.305	
# Teeth:	10			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
66	2.568E+05	1.097E+04	0.699	0.042
67	2.568E+05	1.097E+04	0.710	0.043
82.2	2.567E+05	1.096E+04	0.871	0.052
89.2	2.785E+05	1.096E+04	0.871	0.057
AVG	2.622E+05	1.097E+04	0.945	0.049

Seal:	Impeller		**Avg Vol. flow Rate (L/min)**	
Condition:	3			
Gap Size(mm):	0.2		639.087	
# Teeth:	10			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
66	2.268E+05	1.136E+04	0.724	0.043
67	2.568E+05	1.136E+04	0.735	0.044
82.2	2.567E+05	1.136E+04	0.902	0.054
89.2	2.785E+05	1.135E+04	0.978	0.059
AVG	2.547E+05	1.136E+04	0.835	0.050

Main Seal Results:

Seal:	Main			
Condition:	1		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.091		588.235	
# Teeth:	0			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	1.227E+05	1.156E+04	0.234	0.034
62	1.227E+05	1.155E+04	0.291	0.042
67	1.227E+05	1.155E+04	0.312	0.045
71	1.227E+05	1.155E+04	0.333	0.048
75	1.227E+05	1.155E+04	0.351	0.050
85.9	1.226E+05	1.155E+04	0.403	0.058
AVG	1.227E+05	1.155E+04	0.321	0.046

Seal:	Main			
Condition:	1		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.113		1036.000	
# Teeth:	0			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	2.175E+05	2.048E+04	0.415	0.059
62	2.174E+05	2.047E+04	0.515	0.074
67	2.174E+05	2.047E+04	0.556	0.080
71	2.173E+05	2.047E+04	0.589	0.085
75	2.173E+05	2.047E+04	0.623	0.089
85.9	2.173E+05	2.046E+04	0.713	0.102
AVG	2.174E+05	2.047E+04	0.568	0.081

Seal:	Main			
Condition:	1		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.14		1893.000	
# Teeth:	0			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	3.952E+05	3.722E+04	0.754	0.108
62	3.950E+05	3.720E+04	0.935	0.134
67	3.949E+05	3.719E+04	1.010	0.144
71	3.948E+05	3.718E+04	1.070	0.153
75	3.948E+05	3.718E+04	1.131	0.162
85.9	3.947E+05	3.717E+04	1.295	0.185
AVG	3.949E+05	3.719E+04	1.032	0.148

Seal:	Main			
Condition:	1		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.15		2289.000	
# Teeth:	0			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	4.805E+05	4.525E+04	0.916	0.131
62	4.802E+05	4.522E+04	1.136	0.162
67	4.801E+05	4.522E+04	1.228	0.176
71	4.801E+05	4.521E+04	1.301	0.186
75	4.800E+05	4.520E+04	1.374	0.197
85.9	4.799E+05	4.519E+04	1.574	0.225
AVG	4.801E+05	4.522E+04	1.255	0.179

Seal:	Main			
Condition:	1		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.091		294.118	
# Teeth:	3			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	6.278E+04	5.787E+03	0.120	0.017
62	6.275E+04	5.785E+03	0.149	0.021
67	6.275E+04	5.784E+03	0.161	0.023
71	6.274E+04	5.784E+03	0.170	0.024
75	6.274E+04	5.784E+03	0.180	0.025
85.9	6.273E+04	5.783E+03	0.206	0.029
AVG	6.275E+04	5.784E+03	0.164	0.023

Seal:	Main			
Condition:	1		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.113		306.905	
# Teeth:	3			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	6.513E+04	6.004E+03	0.124	0.017
62	6.510E+04	6.002E+03	0.154	0.022
67	6.510E+04	6.001E+03	0.167	0.023
71	6.509E+04	6.000E+03	0.177	0.025
75	6.508E+04	6.000E+03	0.186	0.026
85.9	6.507E+04	5.999E+03	0.214	0.030
AVG	6.510E+04	6.001E+03	0.170	0.024

Seal:	Main			
Condition:	1		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.14		242.967	
# Teeth:	3			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	5.076E+04	4.679E+03	0.097	0.014
62	5.073E+04	4.676E+03	0.120	0.017
67	5.072E+04	4.676E+03	0.130	0.018
71	5.071E+04	4.675E+03	0.138	0.019
75	5.071E+04	4.675E+03	0.145	0.020
85.9	5.070E+04	4.673E+03	0.166	0.023
AVG	5.072E+04	4.676E+03	0.133	0.019

Seal:	Main			
Condition:	1		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.15		191.816	
# Teeth:	3			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	3.963E+04	3.653E+03	0.076	0.011
62	3.961E+04	3.651E+03	0.094	0.013
67	3.960E+04	3.650E+03	0.103	0.014
71	3.959E+04	3.650E+03	0.107	0.015
75	3.959E+04	3.650E+03	0.113	0.016
85.9	3.958E+04	3.649E+03	0.130	0.018
AVG	3.960E+04	3.650E+03	0.104	0.015

Seal:	Main			
Condition:	1		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.091		294.118	
# Teeth:	9			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	6.318E+04	5.824E+03	0.121	0.017
62	6.316E+04	5.822E+03	0.150	0.021
67	6.315E+04	5.821E+03	0.162	0.023
71	6.314E+04	5.821E+03	0.171	0.024
75	6.314E+04	5.821E+03	0.181	0.025
85.9	6.313E+04	5.820E+03	0.207	0.029
AVG	6.315E+04	5.821E+03	0.165	0.023

Seal:	Main			
Condition:	1		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.113		383.632	
# Teeth:	9			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	8.147E+04	7.511E+03	0.156	0.022
62	8.144E+04	7.507E+03	0.193	0.027
67	8.143E+04	7.506E+03	0.208	0.029
71	8.142E+04	7.506E+03	0.221	0.031
75	8.141E+04	7.505E+03	0.233	0.033
85.9	8.140E+04	7.504E+03	0.267	0.037
AVG	8.143E+04	7.506E+03	0.213	0.030

Seal:	Main			
Condition:	1		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.14		473.146	
# Teeth:	9			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	1.006E+05	9.270E+03	0.192	0.027
62	1.005E+05	9.265E+03	0.238	0.033
67	1.005E+05	9.264E+03	0.257	0.036
71	1.005E+05	9.262E+03	0.272	0.038
75	1.005E+05	9.262E+03	0.288	0.040
85.9	1.004E+05	9.259E+03	0.330	0.046
AVG	1.005E+05	9.264E+03	0.263	0.037

Seal:	Main			
Condition:	1		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.15		498.721	
# Teeth:	9			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	1.062E+05	9.789E+03	0.205	0.028
62	1.061E+05	9.784E+03	0.251	0.035
67	1.061E+05	9.782E+03	0.271	0.038
71	1.061E+05	9.781E+03	0.288	0.040
75	1.061E+05	9.780E+03	0.304	0.043
85.9	1.061E+05	9.777E+03	0.348	0.049
AVG	1.061E+05	9.782E+03	0.278	0.039

Seal:	Main			
Condition:	1		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.091		255.754	
# Teeth:	13			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	5.595E+04	5.158E+03	0.107	0.015
62	5.593E+04	5.156E+03	0.133	0.019
67	5.593E+04	5.156E+03	0.143	0.020
71	5.592E+04	5.155E+03	0.152	0.021
75	5.592E+04	5.155E+03	0.160	0.022
85.9	5.591E+04	5.154E+03	0.184	0.026
AVG	5.593E+04	5.156E+03	0.146	0.020

Seal:	Main			
Condition:	1		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.113		345.269	
# Teeth:	13			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	7.468E+04	6.884E+03	0.143	0.020
62	7.464E+04	6.881E+03	0.177	0.025
67	7.463E+04	6.880E+03	0.191	0.027
71	7.463E+04	6.879E+03	0.202	0.028
75	7.462E+04	6.879E+03	0.214	0.030
85.9	7.461E+04	6.878E+03	0.245	0.034
AVG	7.463E+04	6.880E+03	0.195	0.027

Seal:	Main			
Condition:	1		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.14		447.570	
# Teeth:	13			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	9.707E+04	8.949E+03	0.185	0.026
62	9.702E+04	8.944E+03	0.230	0.032
67	9.701E+04	8.942E+03	0.248	0.035
71	9.699E+04	8.941E+03	0.263	0.037
75	9.698E+04	8.940E+03	0.278	0.039
85.9	9.696E+04	8.938E+03	0.318	0.045
AVG	9.701E+04	8.943E+03	0.254	0.035

Seal:	Main			
Condition:	1		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.15		485.934	
# Teeth:	13			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	1.049E+05	9.667E+03	0.200	0.028
62	1.048E+05	9.661E+03	0.248	0.035
67	1.048E+05	9.660E+03	0.268	0.038
71	1.048E+05	9.658E+03	0.284	0.040
75	1.048E+05	9.657E+03	0.300	0.042
85.9	1.047E+05	9.655E+03	0.344	0.048
AVG	1.048E+05	9.660E+03	0.274	0.038

Seal:	Main			
Condition:	2		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.091		588.235	
# Teeth:	0			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	2.128E+05	1.156E+04	0.429	0.034
62	2.127E+05	1.155E+04	0.533	0.042
67	2.127E+05	1.155E+04	0.575	0.045
71	2.127E+05	1.155E+04	0.610	0.048
75	2.127E+05	1.155E+04	0.644	0.050
85.9	2.126E+05	1.155E+04	0.738	0.058
AVG	2.127E+05	1.155E+04	0.588	0.046

Seal:	Main			
Condition:	2		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.113		1036.000	
# Teeth:	0			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	3.771E+05	2.048E+04	0.761	0.059
62	3.769E+05	2.047E+04	0.943	0.074
67	3.769E+05	2.047E+04	1.019	0.080
71	3.768E+05	2.047E+04	1.080	0.085
75	3.768E+05	2.047E+04	1.141	0.089
85.9	3.767E+05	2.046E+04	1.307	0.102
AVG	3.769E+05	2.047E+04	1.042	0.081

Seal:	Main			
Condition:	2		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.14		1893.000	
# Teeth:	0			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	6.852E+05	3.722E+04	1.381	0.108
62	6.848E+05	3.720E+04	1.713	0.134
67	6.847E+05	3.719E+04	1.851	0.144
71	6.846E+05	3.718E+04	1.962	0.153
75	6.845E+05	3.718E+04	2.072	0.162
85.9	6.844E+05	3.717E+04	2.373	0.185
AVG	6.847E+05	3.719E+04	1.892	0.148

Seal:	Main			
Condition:	2		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.15		2289.000	
# Teeth:	0			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	8.331E+05	4.525E+04	1.679	0.131
62	8.326E+05	4.522E+04	2.082	0.162
67	8.325E+05	4.522E+04	2.250	0.176
71	8.324E+05	4.521E+04	2.385	0.186
75	8.323E+05	4.520E+04	2.519	0.197
85.9	8.321E+05	4.519E+04	2.885	0.225
AVG	8.325E+05	4.522E+04	2.300	0.179

Seal:	Main			
Condition:	2		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.091		294.118	
# Teeth:	3			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	1.092E+05	5.787E+03	0.220	0.017
62	1.092E+05	5.785E+03	0.273	0.021
67	1.092E+05	5.784E+03	0.295	0.023
71	1.092E+05	5.784E+03	0.313	0.024
75	1.092E+05	5.784E+03	0.331	0.025
85.9	1.091E+05	5.783E+03	0.379	0.029
AVG	1.092E+05	5.784E+03	0.302	0.023

Seal:	Main			
Condition:	2		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.113		306.905	
# Teeth:	3			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	1.133E+05	6.004E+03	0.229	0.017
62	1.133E+05	6.002E+03	0.283	0.022
67	1.133E+05	6.001E+03	0.306	0.023
71	1.133E+05	6.000E+03	0.325	0.025
75	1.132E+05	6.000E+03	0.343	0.026
85.9	1.132E+05	5.999E+03	0.393	0.030
AVG	1.133E+05	6.001E+03	0.313	0.024

Seal:	Main			
Condition:	2		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.14		242.967	
# Teeth:	3			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	8.831E+04	4.679E+03	0.178	0.014
62	8.827E+04	4.676E+03	0.221	0.017
67	8.825E+04	4.676E+03	0.239	0.018
71	8.824E+04	4.675E+03	0.253	0.019
75	8.823E+04	4.675E+03	0.267	0.020
85.9	8.821E+04	4.673E+03	0.306	0.023
AVG	8.825E+04	4.676E+03	0.244	0.019

Seal:	Main			
Condition:	2		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.15		191.816	
# Teeth:	3			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	6.895E+04	3.653E+03	0.139	0.011
62	6.891E+04	3.651E+03	0.172	0.013
67	6.890E+04	3.650E+03	0.186	0.014
71	6.889E+04	3.650E+03	0.197	0.015
75	6.888E+04	3.650E+03	0.208	0.016
85.9	6.887E+04	3.649E+03	0.239	0.018
AVG	6.890E+04	3.650E+03	0.190	0.015

Seal:	Main			
Condition:	2		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.091		294.118	
# Teeth:	9			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	1.099E+05	5.824E+03	0.222	0.017
62	1.099E+05	5.822E+03	0.275	0.021
67	1.099E+05	5.821E+03	0.297	0.023
71	1.099E+05	5.821E+03	0.315	0.024
75	1.099E+05	5.821E+03	0.333	0.025
85.9	1.098E+05	5.820E+03	0.381	0.029
AVG	1.099E+05	5.821E+03	0.304	0.023

Seal:	Main			
Condition:	2		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.113		383.632	
# Teeth:	9			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	1.418E+05	7.511E+03	0.286	0.022
62	1.417E+05	7.507E+03	0.355	0.027
67	1.417E+05	7.506E+03	0.383	0.029
71	1.417E+05	7.506E+03	0.406	0.031
75	1.417E+05	7.505E+03	0.429	0.033
85.9	1.416E+05	7.504E+03	0.491	0.037
AVG	1.417E+05	7.506E+03	0.392	0.030

Seal:	Main			
Condition:	2		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.14		473.146	
# Teeth:	9			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	1.750E+05	9.270E+03	0.353	0.027
62	1.749E+05	9.265E+03	0.437	0.033
67	1.748E+05	9.264E+03	0.473	0.036
71	1.748E+05	9.262E+03	0.507	0.038
75	1.748E+05	9.262E+03	0.529	0.040
85.9	1.748E+05	9.259E+03	0.606	0.046
AVG	1.749E+05	9.264E+03	0.484	0.037

Seal:	Main			
Condition:	2		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.15		498.721	
# Teeth:	9			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	1.848E+05	9.789E+03	0.372	0.028
62	1.847E+05	9.784E+03	0.462	0.035
67	1.846E+05	9.782E+03	0.499	0.038
71	1.846E+05	9.781E+03	0.529	0.040
75	1.846E+05	9.780E+03	0.559	0.043
85.9	1.845E+05	9.777E+03	0.640	0.049
AVG	1.846E+05	9.782E+03	0.510	0.039

Seal:	Main			
Condition:	2		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.091		255.754	
# Teeth:	13			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	9.736E+04	5.158E+03	0.196	0.015
62	9.732E+04	5.156E+03	0.244	0.019
67	9.731E+04	5.156E+03	0.263	0.020
71	9.730E+04	5.155E+03	0.279	0.021
75	9.730E+04	5.155E+03	0.295	0.022
85.9	9.728E+04	5.154E+03	0.338	0.026
AVG	9.731E+04	5.156E+03	0.269	0.020

Seal:	Main			
Condition:	2		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.113		345.269	
# Teeth:	13			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	1.299E+05	6.884E+03	0.262	0.020
62	1.299E+05	6.881E+03	0.325	0.025
67	1.299E+05	6.880E+03	0.351	0.027
71	1.298E+05	6.879E+03	0.372	0.028
75	1.298E+05	6.879E+03	0.393	0.030
85.9	1.298E+05	6.878E+03	0.450	0.034
AVG	1.299E+05	6.880E+03	0.359	0.027

Seal:	Main			
Condition:	2		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.14		447.570	
# Teeth:	13			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	1.689E+05	8.949E+03	0.341	0.026
62	1.688E+05	8.944E+03	0.422	0.032
67	1.688E+05	8.942E+03	0.456	0.035
71	1.688E+05	8.941E+03	0.484	0.037
75	1.687E+05	8.940E+03	0.511	0.039
85.9	1.687E+05	8.938E+03	0.585	0.045
AVG	1.688E+05	8.943E+03	0.467	0.035

Seal:	Main			
Condition:	2		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.15		485.934	
# Teeth:	13			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	1.825E+05	9.667E+03	0.368	0.028
62	1.824E+05	9.661E+03	0.456	0.035
67	1.823E+05	9.660E+03	0.493	0.038
71	1.823E+05	9.658E+03	0.522	0.040
75	1.823E+05	9.657E+03	0.552	0.042
85.9	1.822E+05	9.655E+03	0.632	0.048
AVG	1.823E+05	9.660E+03	0.504	0.038

Seal:	Main			
Condition:	3		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.091		588.235	
# Teeth:	0			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	2.591E+05	1.156E+04	0.535	0.034
62	2.590E+05	1.155E+04	0.663	0.042
67	2.590E+05	1.155E+04	0.717	0.045
71	2.589E+05	1.155E+04	0.760	0.048
75	2.589E+05	1.155E+04	0.802	0.050
85.9	2.589E+05	1.155E+04	0.919	0.058
AVG	2.590E+05	1.155E+04	0.733	0.046

Seal:	Main			
Condition:	3		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.113		1036.000	
# Teeth:	0			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	4.591E+05	2.048E+04	0.947	0.059
62	4.589E+05	2.047E+04	1.178	0.074
67	4.588E+05	2.047E+04	1.270	0.080
71	4.588E+05	2.047E+04	1.345	0.085
75	4.588E+05	2.047E+04	1.421	0.089
85.9	4.587E+05	2.046E+04	1.628	0.102
AVG	4.589E+05	2.047E+04	1.298	0.081

Seal:	Main			
Condition:	3		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.14		1893.000	
# Teeth:	0			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	8.342E+05	3.722E+04	1.721	0.108
62	8.338E+05	3.720E+04	2.133	0.134
67	8.336E+05	3.719E+04	2.306	0.144
71	8.335E+05	3.718E+04	2.443	0.153
75	8.334E+05	3.718E+04	2.581	0.162
85.9	8.332E+05	3.717E+04	2.956	0.185
AVG	8.336E+05	3.719E+04	2.357	0.148

Seal:	Main			
Condition:	3		Avg Vol. flow Rate (L/min) Air	
Gap Size(mm):	0.15		2289.000	
# Teeth:	0			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	1.014E+06	4.525E+04	2.092	0.131
62	1.014E+06	4.522E+04	2.594	0.162
67	1.014E+06	4.522E+04	2.803	0.176
71	1.013E+06	4.521E+04	2.970	0.186
75	1.013E+06	4.520E+04	3.137	0.197
85.9	1.013E+06	4.519E+04	3.593	0.225
AVG	1.014E+06	4.522E+04	2.865	0.179

Seal:	Main			
Condition:	3		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.091		294.118	
# Teeth:	3			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	1.331E+05	5.787E+03	0.275	0.017
62	1.330E+05	5.785E+03	0.341	0.021
67	1.330E+05	5.784E+03	0.368	0.023
71	1.330E+05	5.784E+03	0.390	0.024
75	1.330E+05	5.784E+03	0.412	0.025
85.9	1.330E+05	5.783E+03	0.472	0.029
AVG	1.330E+05	5.784E+03	0.376	0.023

Seal:	Main			
Condition:	3		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.113		306.905	
# Teeth:	3			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	1.381E+05	6.004E+03	0.285	0.017
62	1.380E+05	6.002E+03	0.353	0.022
67	1.380E+05	6.001E+03	0.382	0.023
71	1.380E+05	6.000E+03	0.405	0.025
75	1.380E+05	6.000E+03	0.427	0.026
85.9	1.379E+05	5.999E+03	0.489	0.030
AVG	1.380E+05	6.001E+03	0.390	0.024

Seal:	Main			
Condition:	3		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.14		242.967	
# Teeth:	3			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	1.076E+05	4.679E+03	0.222	0.014
62	1.075E+05	4.676E+03	0.275	0.017
67	1.075E+05	4.676E+03	0.297	0.018
71	1.075E+05	4.675E+03	0.315	0.019
75	1.075E+05	4.675E+03	0.333	0.020
85.9	1.075E+05	4.673E+03	0.381	0.023
AVG	1.075E+05	4.676E+03	0.304	0.019

Seal:	Main			
Condition:	3		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.15		191.816	
# Teeth:	3			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	8.401E+04	3.653E+03	0.173	0.011
62	8.396E+04	3.651E+03	0.215	0.013
67	8.394E+04	3.650E+03	0.232	0.014
71	8.393E+04	3.650E+03	0.246	0.015
75	8.392E+04	3.650E+03	0.260	0.016
85.9	8.390E+04	3.649E+03	0.298	0.018
AVG	8.394E+04	3.650E+03	0.237	0.015

Seal:	Main			
Condition:	2		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.091		294.118	
# Teeth:	9			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	1.339E+05	5.824E+03	0.276	0.017
62	1.339E+05	5.822E+03	0.343	0.021
67	1.339E+05	5.821E+03	0.370	0.023
71	1.339E+05	5.821E+03	0.393	0.024
75	1.338E+05	5.821E+03	0.415	0.025
85.9	1.338E+05	5.820E+03	0.475	0.029
AVG	1.339E+05	5.821E+03	0.379	0.023

Seal:	Main			
Condition:	3		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.113		383.632	
# Teeth:	9			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	1.727E+05	7.511E+03	0.356	0.022
62	1.726E+05	7.507E+03	0.442	0.027
67	1.726E+05	7.506E+03	0.478	0.029
71	1.726E+05	7.506E+03	0.506	0.031
75	1.726E+05	7.505E+03	0.535	0.033
85.9	1.725E+05	7.504E+03	0.612	0.037
AVG	1.726E+05	7.506E+03	0.488	0.030

Seal:	Main			
Condition:	3	Avg Vol. flow Rate (L/min) Air		
Gap Size(mm):	0.14	473.146		
# Teeth:	9			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	2.132E+05	9.270E+03	0.440	0.027
62	2.131E+05	9.265E+03	0.545	0.033
67	2.130E+05	9.264E+03	0.589	0.036
71	2.130E+05	9.262E+03	0.624	0.038
75	2.130E+05	9.262E+03	0.659	0.040
85.9	2.129E+05	9.259E+03	0.755	0.046
AVG	2.130E+05	9.264E+03	0.602	0.037

Seal:	Main			
Condition:	2	Avg Vol. flow Rate (L/min) Air		
Gap Size(mm):	0.15	498.721		
# Teeth:	9			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	2.251E+05	9.789E+03	0.464	0.028
62	2.250E+05	9.784E+03	0.576	0.035
67	2.249E+05	9.782E+03	0.622	0.038
71	2.249E+05	9.781E+03	0.659	0.040
75	2.249E+05	9.780E+03	0.696	0.043
85.9	2.248E+05	9.777E+03	0.797	0.049
AVG	2.249E+05	9.782E+03	0.636	0.039

Seal:	Main			
Condition:	3	Avg Vol. flow Rate (L/min) Air		
Gap Size(mm):	0.091	255.754		
# Teeth:	13			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	1.186E+05	5.158E+03	0.245	0.015
62	1.186E+05	5.156E+03	0.304	0.019
67	1.186E+05	5.156E+03	0.328	0.020
71	1.185E+05	5.155E+03	0.348	0.021
75	1.185E+05	5.155E+03	0.367	0.022
85.9	1.185E+05	5.154E+03	0.421	0.026
AVG	1.186E+05	5.156E+03	0.336	0.020

Seal:	Main			
Condition:	3		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.113		345.269	
# Teeth:	13			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	1.583E+05	6.884E+03	0.327	0.020
62	1.582E+05	6.881E+03	0.405	0.025
67	1.582E+05	6.880E+03	0.438	0.027
71	1.582E+05	6.879E+03	0.464	0.028
75	1.582E+05	6.879E+03	0.490	0.030
85.9	1.581E+05	6.878E+03	0.561	0.034
AVG	1.582E+05	6.880E+03	0.448	0.027

Seal:	Main			
Condition:	3		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.14		447.570	
# Teeth:	13			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	2.058E+05	8.949E+03	0.424	0.026
62	2.057E+05	8.944E+03	0.526	0.032
67	2.056E+05	8.942E+03	0.569	0.035
71	2.056E+05	8.941E+03	0.603	0.037
75	2.056E+04	8.940E+03	0.637	0.039
85.9	2.055E+05	8.938E+03	0.729	0.045
AVG	1.748E+05	8.943E+03	0.581	0.035

Seal:	Main			
Condition:	3		**Avg Vol. flow Rate (L/min) Air**	
Gap Size(mm):	0.15		485.934	
# Teeth:	13			
Diameter (mm)	**Re of R134a**	**Re of Air**	**Flow Rate R134a (kg/s**	**Flow Rate Air (kg/s)**
50	2.223E+05	9.667E+03	0.458	0.028
62	2.222E+05	9.661E+03	0.568	0.035
67	2.221E+05	9.660E+03	0.614	0.038
71	2.221E+05	9.658E+03	0.651	0.040
75	2.221E+05	9.657E+03	0.688	0.042
85.9	2.220E+05	9.655E+03	0.787	0.048
AVG	2.221E+05	9.660E+03	0.628	0.038

Appendix 2: Pertinent Calculations

Appendix 2.A: Interstage Seal Calculations

Appendix 2: Pertinent Calculations

Appendix 2.B: Impeller Seal Calculations

Appendix 2: Pertinent Calculations

Appendix 2.C: Main Seal Calculations

*Note: Due to the lengthiness for the calculations for this particular seal, only a sample of the calculations are seen here in this appendix. The first set of calculations deals with the case of zero teeth where it became necessary to assume that the empirical coefficient Ct was equal to 1 in order to avoid a nonreal number. The second set of calculations are the calculations performed for the case of 3 teeth. Both samples are performed at condition 1

Appendix 2: Pertinent Calculations

Appendix 2.D: Internal Pressure Vessel Loading Calculations

Appendix 2: Pertinent Calculations

Appendix 2.E: Concentricity & Differential Threading Sample Calculations

Appendix 2: Pertinent Calculations

Appendix 2.F: Bearing Load Analysis

Appendix 2: Pertinent Calculations

Appendix 2.G: Prototype Sample Calculations

Test 2 - Operating Pressure: 3 psi

Gas constant for diatomic nitrogen: $R_{Nitrogen} := 0.3141 \frac{kJ}{kg \cdot K}$

Assumed Operating Pressure: $T := 273K$

Volume inside the gas cylinder: $TankVolume := 69.9L$

Pressure readings on the cylinder at the start and end of the test (steady-state):

$$P_{start} := 105 \frac{kgf}{cm^2} = 1493\,psi \qquad P_{end} := 90 \frac{kgf}{cm^2} = 1280\,psi$$

Mass of nitrogen inside the cylinder at the start and end of the test (steady-state):

$$m_{start} := \frac{P_{start} \cdot TankVolume}{R_{Nitrogen} \cdot T} = 8.394\,kg \qquad m_{end} := \frac{P_{end} \cdot TankVolume}{R_{Nitrogen} \cdot T} = 7.195\,kg$$

$$\Delta m := m_{start} - m_{end} = 1.199\,kg$$

Change in mass during the test (steady-state):

The pressure readings were taken at 8:51:10 AM and 8:54:15 AM, from a standard wall clock. The test duration is calculated below:

$$T_{start} := 51 \cdot 60s + 10s = 3070\,s \qquad T_{end} := 54 \cdot 60s + 15s = 3255\,s$$

$$\Delta t := T_{end} - T_{start} = 185\,s$$

The test (steady-state) lasted 185 seconds.

Divide the change in mass by the change in time during the steady-state interval to get a time-averaged mass-flow value:

$$m' := \frac{\Delta m}{\Delta t} = 0.00648 \frac{kg}{s}$$

Common to direct flow-measurement systems is volumetric flow rate. Find the volumetric flow rate at one atmosphere, 0 degrees Celsius:

$$\rho = \frac{P}{RT} \qquad \rho := \frac{101kPa}{R_{Nitrogen} \cdot T} = 1.178 \frac{kg}{m^3} \qquad Vol' := \frac{m'}{\rho} = 330.177 \frac{L}{min}$$

Appendix 3: Pro-E Detailed Drawings