SUPERCHARGER FOR HEAT PUMPS IN COLD CLIMATES

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Abstract: The on-going development of a compact centrifugal compressor as a first stage or pre-compressor for heat pumps is described. Positioned in the low pressure vapor portion of the refrigerant loop, this compressor will, on cold days, boost refrigerant pressure, in a manner similar to the way the an automobile or aircraft engine supercharger pressurizes air. Directly driven by a high-speed motor, the single stage centrifugal compressor runs on oil-free bearings. It is being configured to work in concert with a traditional heat pump compressor. The goal of this work is to enable air source heat pumps to efficiently extract heat from even the coldest ambient air without backup heat and without changing refrigerant type or the basic design of the positive displacement style compressors widely used in today's heat pumps. Doing so will allow air source heat pumps to operate effectively in virtually all major markets. To date, a first article supercharger has been designed and built, and its performance mapped. Work to integrate it with a commercially available positive displacement (PD) heat pump compressor is underway.

Key Words: Air source heat pumps, centrifugal refrigerant vapor compressor, high-speed motor-driven compressor, gas foil bearings

1 INTRODUCTION

Electrically driven heat pumps are an effective method of extracting heat from ambient air. As air temperature falls, however, heat pump performance falls off, essentially limiting their year-round usefulness to warmer climates. In colder climates heat pump-equipped heating systems typically have a secondary means to provide heat, usually electric strip heaters. This adds complexity, duplication, cost and pollution. An air source heat pump system that maintains both capacity and coefficient of performance (COP) in the coldest climates could have a significant energy-savings impact, particularly where natural gas is not available.

For the US Department of Energy (DOE 2012), the author's company is developing elements of an integrated heat pump compressor system to extract heat from subzero ambients. The goal is a system that requires zero backup heat in even the coldest climates, in effect achieving a viable heat pump furnace. The core component is a second compressor, in the low pressure portion of the refrigerant loop that will, on cold days, boost the refrigerant pressure. The goal is a compact pre-compressor that works in concert with a traditional heat pump compressor. Once developed, the market objective is heat pumps that can be applied to virtually all major markets without changing the type of refrigerant employed or the basic design of existing heat pump compressors. As sources of electric power become cleaner, expanded heat pump use offers the ability to produce heat with minimal increase in greenhouse gases. A heat pump that can efficiently function as a furnace, with no requirement for backup heat, will revolutionize major portions of the HVAC industry. The following sections describe work accomplished to date and planned.

2 SIZING THE SUPERCHARGER

Table 1 shows the DOE's performance targets for a cold climate heat pump (DOE 2010). To be viable in the marketplace, a motor-driven pre-compressor must be compact, durable, efficient, inexpensive, and capable of hermetic sealing. There is no known off-the-shelf device that meets these needs.

Table 1: DOE's Cold Climate Heat Pump Performance Targets

Ambient Temperature	Minimum COP	Max Capacity Degradation from Nominal, %
8°C (47°F) - Nominal Rating Point	4	0
-8°C (17°F)	3.5	10
-25°C (-13°F)	3	25

For a system with a nominal rating point of 8°C (47°F) and sized to provide 63 kW (216,000 Btu/hr) at minus -25°C (-13°F), Figure 1 shows a typical portion of dwelling heat a traditional heat pump may extract from the outside air. As outdoor temperatures fall so does the pressure of the refrigerant's vapor, and the conventional compressor (typically a positive displacement (PD) style device) must work harder. The result is degrading capacity with falling ambients. Plus, the dwelling needs additional heat – as much as 55 kW (190,000 Btu/hr) for the Figure 1 example. In traditional heat pumps, this heat must come from a backup source, usually resistance strip heaters. The overarching objective therefore is to come up with a pre-compressor that runs on cold days to A) boost the refrigerant vapor's pressure back up to nominal PD suction pressures, and B) "make up the difference" as temperature falls further, so the PD unit is always presented with inlet pressures that are equivalent to warmer day conditions, where the PD unit's efficiency is higher. As described below, a 10 kW (34,000 Btu/hr) variable speed centrifugal compressor that runs faster as ambient temperatures fall is being developed for this service. If successful, investing a smaller amount of energy (say 10 kWh) to extract a larger amount (say 55 kWh) from the ambient air has enormous savings potential.

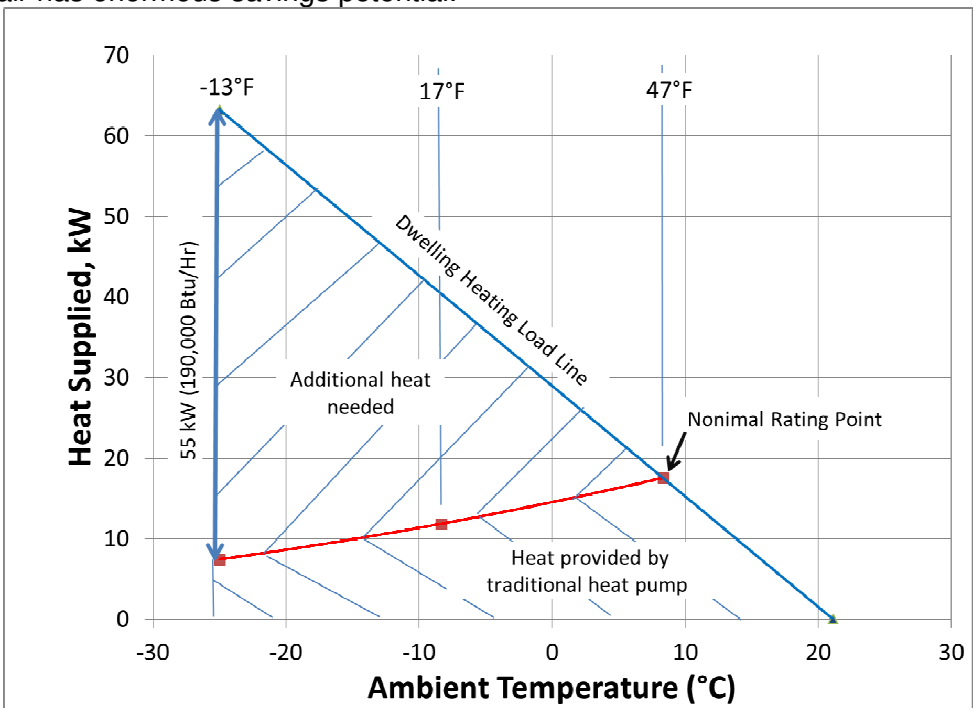


Figure 1: Sample showing supplemental heat needed as ambient temperature decreases

3 SYSTEM SCHEMATIC

An overall system configuration is shown in Figure 2, with added components in blue.

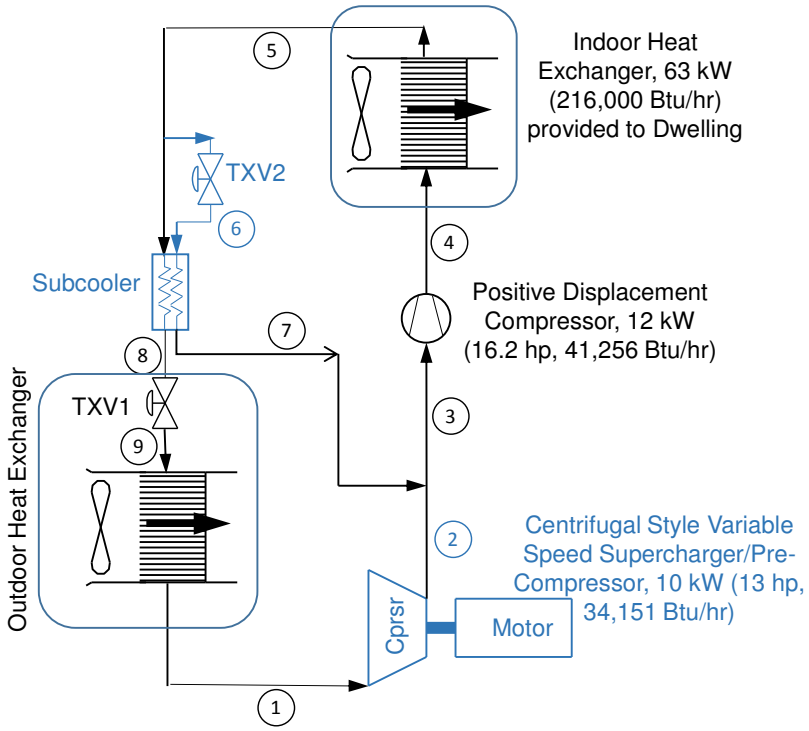


Figure 2 – System schematic for a heat pump furnace

To augment overall system performance, the concept includes a subcooler similar to the economizer arrangement used in higher pressure ratio refrigeration systems. The subcooler is a small heat exchanger with a surface requirement of only about 0.9 m^2 (3 ft^2), due to the high heat transfer coefficients achieved (with boiling on the cold side and high velocity liquid on the hot side). In this regard, approximately 25 percent of the condensed refrigerant exiting the indoor heat exchanger is expanded to the inter-stage location between the two compressors. This expansion further cools the main refrigerant circuit entering the outdoor evaporator. This arrangement produces substantial sub-cooling with an attendant performance increase. Furthermore, some PD compressors also have internal porting which can be used to achieve the same effect when operating on the PD compressor alone. In a similar manner, a small separator can be invoked at state point 1 to assure that only refrigerant and oil vapors enter the compressors. Since the pre-compressor is oil-less, it does not add to the amount of oil in the system.[1] Figure 3 shows the process path on P-H coordinates, with the state points shown in Figure 2. The pre-compressor includes porting to preclude oil accumulation.

Using the values shown in Figures 2 and 3, the sample system's anticipated heating COP is:

COP_h = Heat Provided ÷ Electrical Energy Consumed

COP_h = 63 kW ÷ (12+10) kW

COP_h = 2.86, close to the DOE goal of 3 (see Table 1)

[1] Most PD compressors use a small amount of oil for sealing and lubrication.

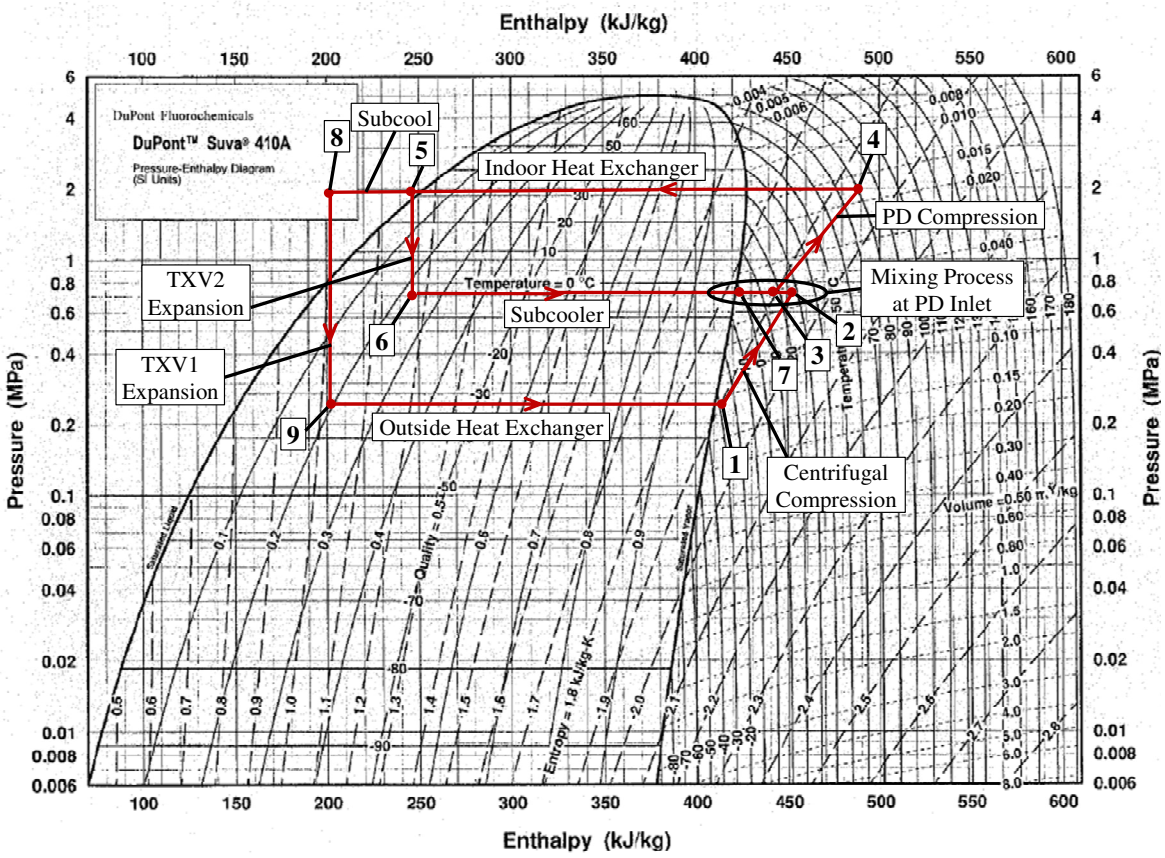


Figure 3 – Process Path for a heat pump furnace using a variable speed centrifugal pre-compressor as a supercharger, to boost the refrigerant's pressure on cold days

4 PROTOTYPE SUPERCHARGER

The pre-compressor consists of a single stage centrifugal compressor driven by a permanent magnet motor. Gas foil bearings (no moving parts, oil-free) support the rotating components. It is driven by a variable frequency drive (VFD). Figure 4 shows the principal parts.

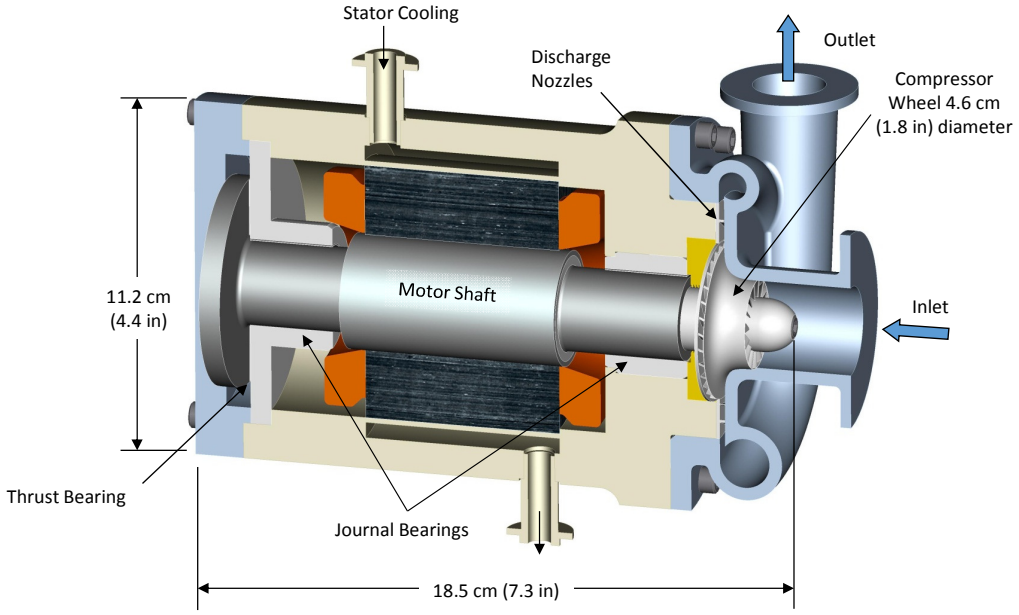


Figure 4 – Supercharger for pre-compressing refrigerant vapors

Based on the cycle described above, these conditions were selected for compressor operation: Inlet Total Pressure = 234 kPa (34 psia); Inlet Total Temperature = -27°C (-16°F); Total Pressure Ratio ≈ 3.3; Mass Flow Rate = 11 kg/min (24.8 lbm/min). Based on these correlations and loss models, a single stage compressor operating at a rotational speed of 100,000 rpm was selected. The compressor stage was modeled using a centrifugal impeller, vaned diffuser, and elliptical volute. This resulted in an impeller with 18 full height blades and zero partial height splitter blades, and a diffuser with 13 vanes. Based upon the analysis, the compressor was predicted to have a Total Adiabatic Efficiency = 0.72. Figure 5 shows the aero components.

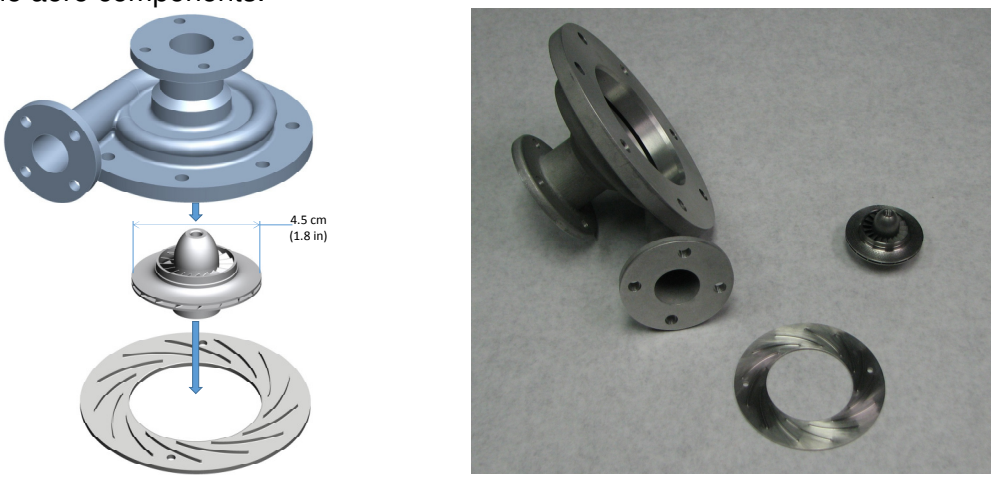


Figure 5 - Final Design and Prototype Parts - Volute Housing (top) Impeller (middle) and Diffuser (bottom)

The impeller, shaft, and attachment bolt were analyzed using finite element-based software to evaluate operating stress and modal characteristics, with satisfactory results.

4.1 Manufacturing Methods for Compressor Aero Components

Historically, manufacturing prototypes for these components is expensive and time-consuming, with components made using traditional machining, casting, or a combination of both processes. This involves initial development cost and lead time to produce the first article components, which were prohibitive for this project. To address this issue, the author's company internally funded an effort to identify and evaluate alternative manufacturing techniques that are faster for prototypes and less expensive. *Additive manufacturing* and *rapid investment casting* were identified as two advancing technologies that offered much promise

Also known as 3D printing, additive manufacturing is growing in popularity and capability. Originally developed to create 3D shapes using thermoplastics and photopolymers, the technology has broadened to include a variety of metals. The process involves the use of a high energy beam, either laser or electron, which is focused on a bed of powder metal. Three processes currently used to fabricate metal components are Direct Metal Laser Sintering (DMLS), Selective Laser Melting (SLM), and Electron Beam Melting (EBM). Of the three processes, DMLS is the most commonly used and offers the best dimensional control and finish quality for small parts, and therefore was selected for manufacturing the impeller.

Rapid Investment Casting: Though the technology has been around for centuries, investment casting has recently been able to make improvements in reducing initial cost and lead times by taking advantage of additive manufacturing technology. The process involves building the pattern for the molds directly from a CAD file using a 3D printing process, such as Stereolithography (SLA) or Selective Laser Sintering (SLS). Lead times are reduced to

days versus 8 to 10 weeks for standard investment castings. Parts are limited in size to those that can be built with the additive manufacturing process, but in some cases sections can be built and joined together to produce larger parts. Because of its size, the volute housing was manufactured using the rapid investment casting process.

Impeller Fabrication: Although the DMLS process possesses many advantages, concerns exist regarding the strength and surface quality of the sintered power metal. For the impeller, a precipitation hardened stainless steel, 17-4PH heat treated to the H900 condition was selected for its high tensile strength. To evaluate the tensile properties of the sintered material, three tensile test specimens were fabricated and processed identically along with the impellers. These specimens were then tested to failure and the results compared to the specifications for wrought and cast material, as listed in Table 2. The results show the sintered material exceeded the minimum strength requirements for both wrought and cast material. The sintered material exhibits slightly lower ductility than wrought material, but better ductility than cast material. In addition, one of the finished impellers was successfully spin-tested to 115,000 rpm (15% over-speed) without failure.

Table 2 - Tensile Test Results Comparing DMLS Produced Samples to Other Forms

DMLS Data

Specimen ID	Ultimate Tensile Strength, MPa (ksi)	0.2% Offset Yield Strength, MPa (ksi)	% Elongation	% Reduction in Area
SN1	1,434 (208)	1,282 (186)	5	11
SN2	1,447 (210)	1,358 (197)	7	8.5
SN3	1,454 (211)	1,330 (193)	9	15

Published Data (minimum values)

Wrought	1,310 (190)	1,172 (170)	10	35
Investment Cast	1,241 (180)	1,103 (160)	4	12

Impeller Surface Finish: Ideally, surfaces exposed to high velocity gas flows should be as smooth as possible to minimize turbulence and friction losses. A surface finish of 0.8 µm (32 µin) or better is preferred. As processed, the DMLS material exhibits a surface finish of 3.3 µm (130 µin). Accessible surfaces can be improved through bead blasting or polishing, however, the presence of the shroud prevents direct access to the blades. Alternate polishing methods were explored to improve the surface finish of these inaccessible areas, including electro-polishing, electro-plating, and extrusion honing. Of these, electro-polishing was determined to be the most likely to improve the surface finish most uniformly with minimal impact on dimensional tolerance. One of the impellers was sent to an electro-polishing vendor, who adjusted process parameters to optimize the process but with marginal results. While improvement was realized in some sectors, other areas were worsened, with the process creating localized pitting. The vendor attributed the varying result to the non-uniform nature of the sintered material. It was therefore decided to proceed with the as-manufactured surface finish and accept the aero performance degradation for initial testing. Methods of producing the impeller in two pieces (i.e., a separate shroud), enabling full access to the blade surfaces prior to attaching the shroud, can be explored later. As related in Section 5, initial testing revealed an adiabatic efficiency of 0.68 vs. the above-noted prediction of 0.72. We view this result as quite acceptable for a first prototype, with impeller surface finish and the type of seals implemented (see Section 4.3) deemed the principal contributors to the difference observed.

Volute Housing Fabrication: Initially we planned to make the volute housing from 17-4PH stainless steel, the same material as the impeller. However, the casting vendor ran into difficulties getting uniform flow through the thin-walled sections of the casting. Since the strength of 17-4PH is not required for the volute housing, the material was changed to A-356

aluminum, an alloy with superior casting properties. The result was a complete success, with the end product shown in Figure 5. To verify the aluminum casting had adequate strength, the housings were successfully hydro-tested to 150 psi (150% of the working pressure).

Diffuser Fabrication: We used traditional machining methods for this part, using Grade 304 stainless steel.

4.2 Motor and Controller

Two motor types (Induction and Permanent Magnet) were evaluated. A PM style was selected for the prototype since it results in a smaller, higher efficiency motor. Based on an assessment of parasitic losses, a 100,000 rpm 4-pole motor with nominal rating of 10 kW (13 hp) was selected. It contains stator laminations that are 0.018 cm (0.007 in) thick and copper windings, with a predicted efficiency of 96.9 percent. Figure 6 shows a completed stator and shaft. A commercially available digital drive variable frequency drive (VFD) which operates without the need for a Hall Effect sensor was selected.

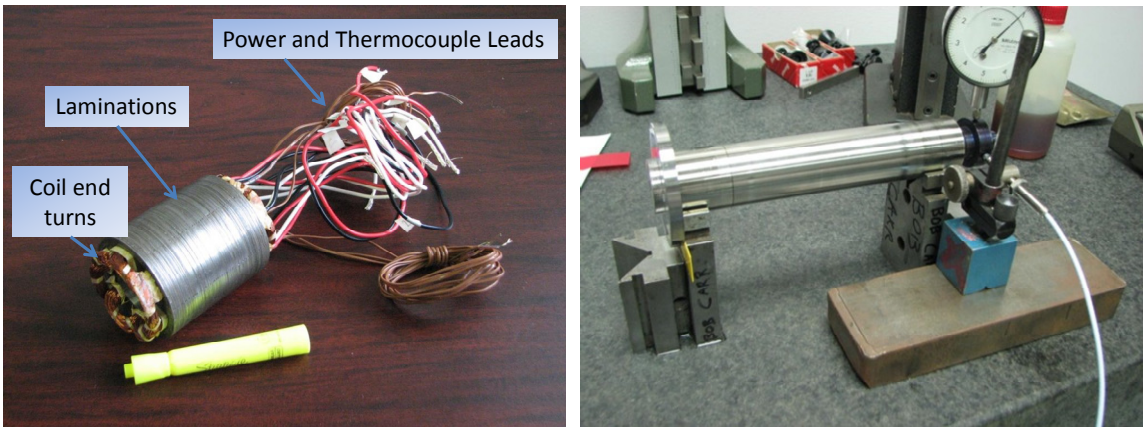


Figure 6 - Motor Stator and Shaft

4.3 Bearings and Seals

A parametric design study was conducted to size the gas foil bearings. Load capacity was determined over a range of bearing lengths and diameters at 100,000 rpm. Once the final designs related above were complete, the bearing sizes were finalized as follows: Both radial bearings are 3.3 cm diameter x 2.5 cm long (1.3 in x 1 in). This results in a more-than-adequate 58 N (13 lbf) load capacity per bearing for the vertically mounted machine. The single acting thrust bearing has a 6.3 cm OD and 3.8 cm ID (2.5 in and 1.5 in) with 6 pads of approximately 60° arc angle per pad. The load capacity of this bearing is 245 N (55 lbf), which is ample to carry the weight of the rotor and anticipated compressor thrust load.

In small centrifugal compressors, sealing around the impeller is an especially key part of overall machine performance. As shown in Figure 4, there are two principal dynamic seals in the supercharger: At the impeller inlet, and on the shaft behind the impeller. For initial operation, labyrinth seals have been implemented at both locations. Labyrinth seals are commonly used in turbomachinery. They provide a close-clearance but non-contacting seal, so there is a small amount of flow (leakage) through the seal. In a large compressor such leakage is generally of no consequence. In small machines however, this leakage can be a significant portion of the total flow. At the shaft seal, a small amount of purposeful flow through the seal cools the motor and bearings by passing refrigerant through the bearings and motor-stator gap. As operational experience is gained, we will consider alternate inlet seal configurations that may moderately improve supercharger efficiency. A potential

downside associated with some higher-performance seals is their potential to introduce small forces on the shaft that could degrade the dynamic stability of the supercharger's suspension system. It should be noted that all flow through the seals is retained within the sealed refrigerant system and merely recompressed; there is no "leakage" in the traditional sense. Figure 7 shows a set of bearings and seals.

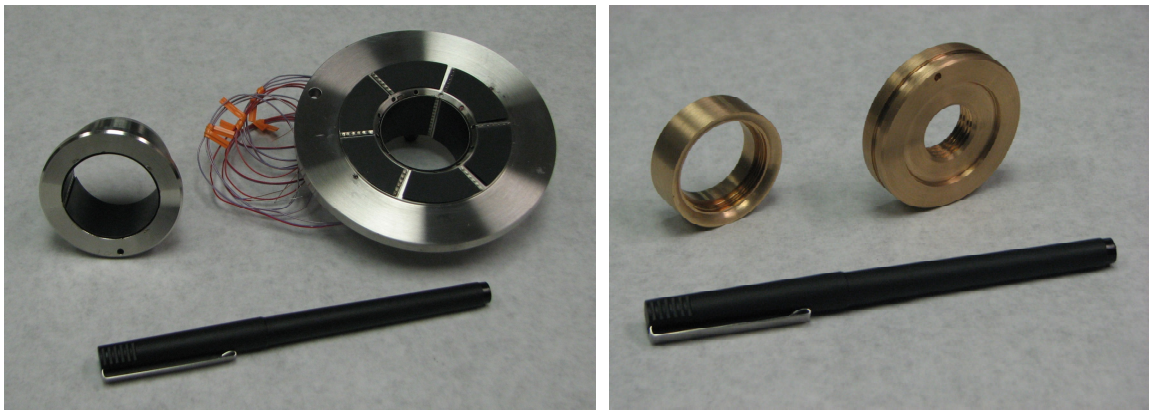


Figure 7 – One set of bearings with thermocouples attached (left) and seals (right)

5 INITIAL TESTING AND RESULTS

An in-house vapor-only test loop to assess compressor performance was designed and built, see Figure 8. In addition to flow meters, accumulators and instrumentation to measure performance and vibration, this loop includes heat exchangers to remove the compression heat added by the supercharger. An automated data acquisition system gathers 30+ channels of process and dynamic data in real time.

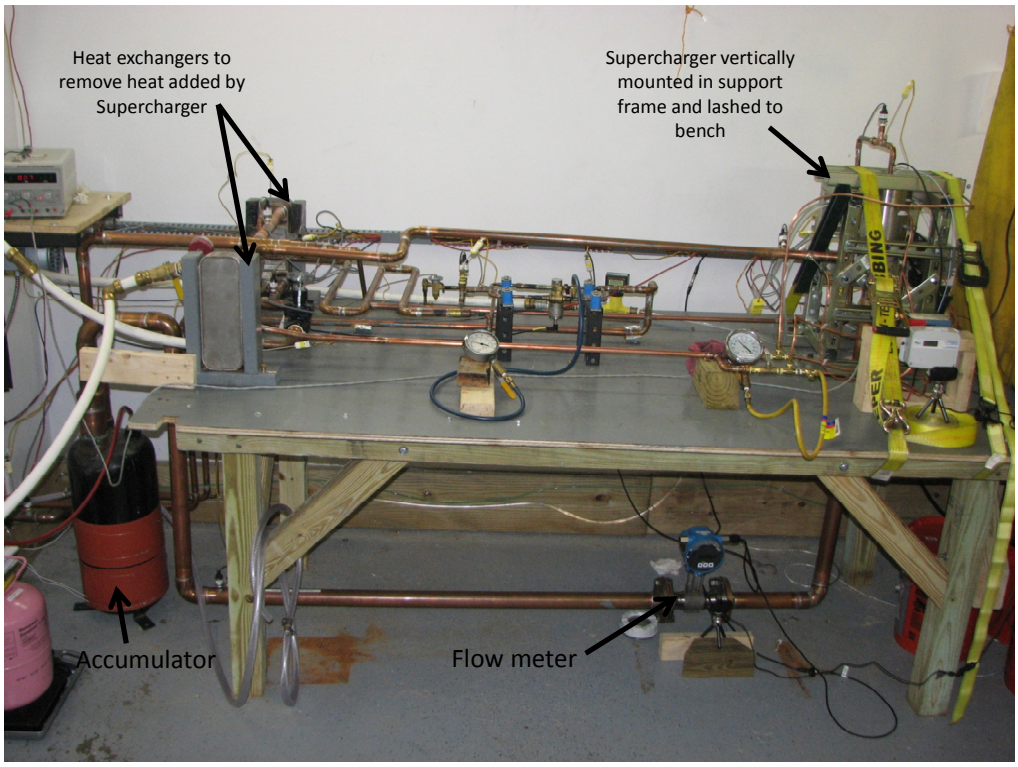


Figure 8: Test loop used to measure supercharger aerodynamic performance

Initial tests involved grooming the motor controller. This was followed by scoping runs with a vane-less impeller, dummy housing, and air not refrigerant in the loop, to confirm rotordynamic performance. For these tests, additional instrumentation was in place, including eddy current displacement sensors to observe shaft motion at all speeds. Next the impeller, diffuser and volute were installed and the loop was charged with refrigerant. During this changeover, the supercharger was opened and inspected, with minimal wear observed.

Test runs to map as-built performance were conducted in late 2013 and early 2014. Figures 9 and 10 relate performance data from all of the runs conducted to date. Figure 9 shows the results as generally depicted for centrifugal compressors. For this style compressor, optimum operation is obtained when running through the center of the "islands" of constant efficiency. Figure 10 shows the corresponding head vs. flow coefficients, which can be used to assess how the compressor will perform in other applications. Both figures show encouraging data, with consistent results, reasonably shaped performance curves, and excellent data point clustering. Although somewhat lower than predicted, both head and flow coefficients are quite acceptable for a first prototype. As noted in Section 4, this initial build included purposeful simplifications regarding seal type and impeller finish. The Operating Line shown in Figure 9 represents current plans for supercharger operation when combined with a commercially available PD compressor. The region above the top end of the Operating Line is being reserved as margin to fine tune compressor interoperation, if needed.

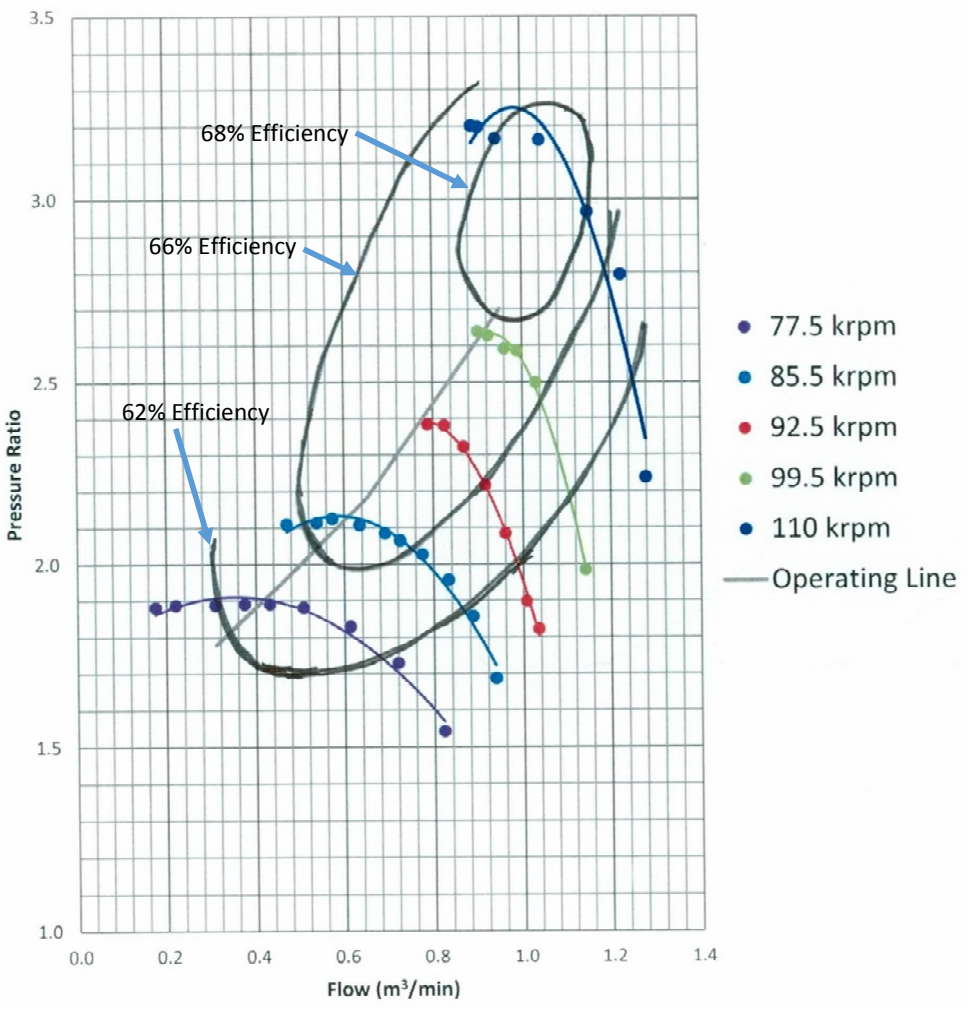


Figure 9: Compressor Performance Map for First Article, and anticipated Operating Line

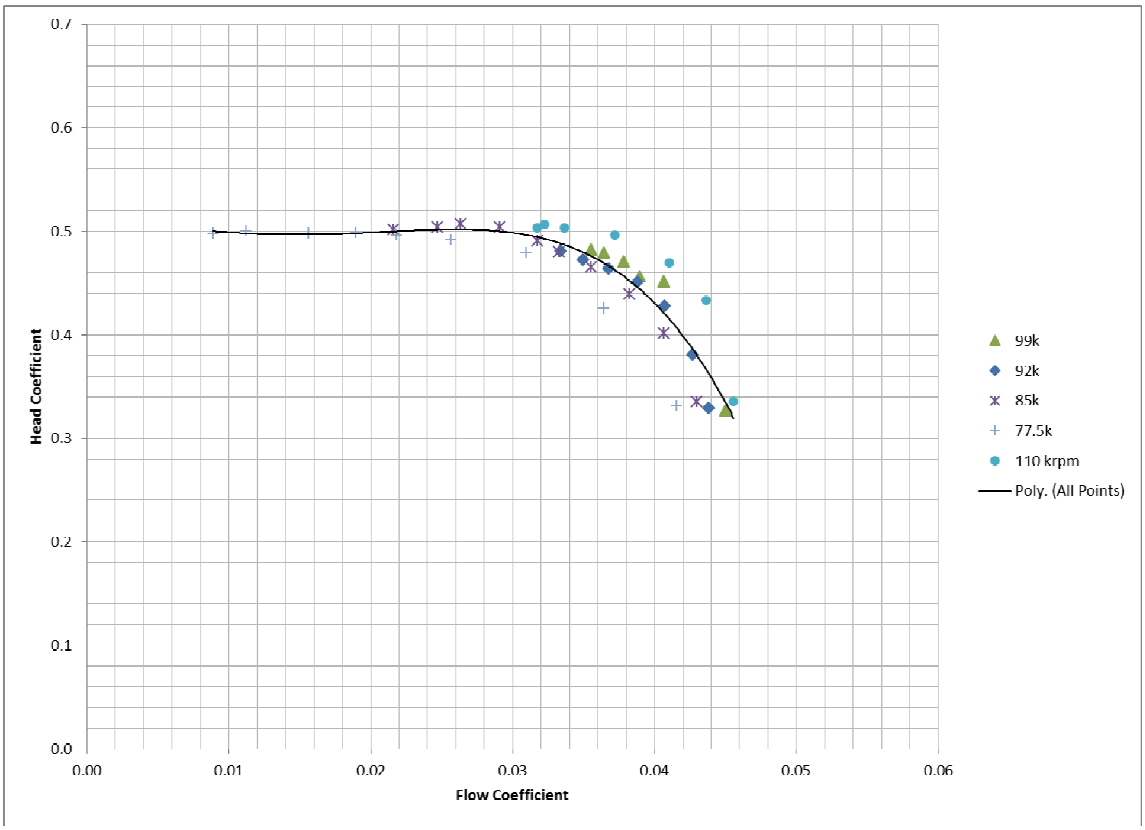


Figure 10: Head and Flow Coefficients for First Article Compressor

Initial assessment of vibration, acoustic, and thermal performance have also been made, to good effect. Shaft displacements were generally 5 $\mu m_{p\text{-}p}$ (0.2 $mils_{p\text{-}p}$) or less at all speeds, with minimal increase when traversing the rigid body critical speeds, which occurred in the 15,000-20,000 rpm range, well below the supercharger's running speed. Impact tests during operation showed reasonable damping for a gas foil bearing-supported shaft, with these typical values: Log decrement = 0.64; damping ratio = 0.1; amplification factor = 5. Motor housing vibration levels were typically 0.05 cm/sec_{rms} (0.02 in/sec_{rms}). Acoustic (airborne) sound levels were not measured but observed as principally a low-level tone, likely associated with impeller vane or blade pass, at speeds above 50,000 rpm. Thermally, bearing losses were in line with predictions. Windage and stator losses were somewhat higher than expected.

6 CONCLUSIONS AND PLANNED WORK

Centrifugal compressors have fewer rubbing parts, are relatively energy efficient, and give higher flow than similarly sized reciprocating or positive-displacement compressors. However, they cannot achieve comparable compression ratios without multiple stages. In many respects, the application discussed herein makes optimum combined use of both compressor types. It is concluded that work conducted to date has verified the underlying design of a motor driven supercharger to improve the cold-day performance of air source heat pumps.

As presently configured, the prototype supercharger is suited for a heat pump in the 45 kW (150,000 Btu/hr) class, with a reasonable COP_h of 2.32. Work is underway to complete the DOE SBIR grant work scope by marrying the supercharger to a properly sized PD compressor and demonstrating effective solo and simultaneous operation of both

compressors on our in-house test loop. Within the constraints of remaining schedule and budget, we are also implementing hardware modifications to reduce windage and motor losses.

Planned work includes discussions with potential commercializers, such as heat pump compressor original equipment manufacturers (OEMs), who bring built-in marketing and sales capabilities, along with service and support. The idea here is to integrate both compressors into a single package with overarching controller, for sale under their brand name.

Follow on work includes an anticipated grant from the State of New York to configure the compressors with associated heat exchangers, fans, and associated controls to fashion the components of a cold climate heat pump for demonstration to interested third parties, including heat pump OEMs, who have direct access to end users and are well suited to establish the bottom line economics needed for a viable cold climate heat pump (NYSERDA 2014). This work is expected to commence in spring 2014.

Other potential uses for the refrigerant pre-compression concept described herein include industrial and commercial coolers, chillers and freezers as a compact and energy-efficient alternative to existing methods, with the potential for increased capacity or lower temperature operation. Outreach to suppliers of this type of equipment is planned as related system demonstration work continues.

7 REFERENCES

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