Title: HIGH TECHNOLOGY CENTRIFUGAL COMPRESSOR FOR COMMERCIAL AIR CONDITIONING SYSTEMS

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EXECUTIVE SUMMARY

R&D Dynamics, Bloomfield, CT in partnership with the State of Connecticut has been developing a high technology, oil-free, energy-efficient centrifugal compressor called CENVA for commercial air conditioning systems under a program funded by the U.S. Department of Energy.

The CENVA compressor applies the foil bearing technology used in all modern aircraft, civil and military, air conditioning systems. The CENVA compressor will enhance the efficiency of water and air cooled chillers, packaged roof top units, and other air conditioning systems by providing an 18% reduction in energy consumption in the unit capacity range of 25 to 350 tons of refrigeration.

The technical approach for CENVA involved the design and development of a high-speed, oil-free foil gas bearing-supported two-stage centrifugal compressor. CENVA encompassed the following high technologies, which are not currently utilized in commercial air conditioning systems:

- Foil gas bearings operating in HFC-134a
- Efficient centrifugal impellers and diffusers
- High speed motors and drives
- System integration of above technologies

Extensive design, development and testing efforts were carried out. Significant accomplishments achieved under this program are described below:

- A total of 26 builds and over 200 tests were successfully completed with successively improved designs.
- Use of foil gas bearings in refrigerant R134a was successfully proven.
- A high speed, high power permanent magnet motor was developed.
- An encoder was used for signal feedback between motor and controller. Due to temperature limitations of the encoder, the compressor could not operate at higher speed and in turn at higher pressure. In order to alleviate this problem a unique sensorless controller was developed.
- This controller has successfully been tested as stand alone; however, it has not yet been integrated and tested as a system.

- The compressor successfully operated at water cooled condensing temperatures. Due to temperature limitations of the encoder, it could not be operated at air cooled condensing temperatures.

- The two-stage impellers/diffusers worked well separately but combined did not match well.

Additional work is needed in the following areas:

- The sensorless controller requires testing in a system.

- The two impellers and diffusers must be modified to improve their performance as a matched set

- The development effort should be continued after making the above listed changes.

PROJECT SUMMARY

A project for development of a two-stage centrifugal compressor supported by totally oil-free, foil gas bearings, powered by high-speed permanent magnet motor and controlled by a variable frequency drive (VFD) was carried out.

A DOE peer review concluded that "the project is highly relevant to industry's expectations for energy reduction of HVAC&R systems and components. The oil-free concept has obvious advantages for better product design". It was further noted that "efficiency and reliability are major objectives" for the industry and that manufacturers would "love to have higher efficiency compressors in addition with oil free compressors as well as the size reduction that this project brings out."

The following tasks were completed:

- Designed and built two breadboard compressors supported by foil gas bearings to prove concept

- Designed and produced foil gas bearings for handling radial and axial loads and compatible with Refrigerant R134a.

- Developed two 50-ton 2-stage compressor supported by foil gas bearings and with stages separated by the electric motor.

- Developed a concept compressor with both stage impellers mounted back-to-back and cantilevered on one side of electric motor.

- Developed two 13-ton 2-stage compressor supported by foil gas bearings and with stages mounted back-to-back cantilevered on one side of electric motor.

- Designed and built permanent magnet motor for speed up to 70,000 rpm.

- In close cooperation with control manufacturers, developed a sensorless variable frequency drive controller capable of operating a permanent magnet motor at speed in excess of 100,000 rpm. The permanent motor and the sensorless VFD controller represented totally new technology not previously commercially available.

- Designed and built gas cycle test rig for testing compressors with refrigerant R134a.

- Developed in conjunction with vendors a process for producing shrouded compressor impellers applying rapid prototype modeling and extrude honing.

- Developed computer based analytical model for studying flow path into and out of impellers, diffusers, return channels and volutes.

- Conducted mechanical testing of compressors to assure dynamic integrity of rotating elements including bearings and subsequent testing with refrigerant

- Testing with R134a was successful up to 105°F saturated discharge temperature.

- Overall compressor efficiency based on combination of efficiency of impeller, motor, VFD controller, friction and windage reached 40 %.

The compressor was successfully demonstrated at water-cooled condensing conditions. However, the project did not achieve the air cooled conditions. More development effort is needed to incorporate sensorless controller, improve matching of two impellers and diffusers to make the design commercially viable.

Design & Build

The compressors were designed to operate with environmentally friendly refrigerant R134a and to be applied in both water-cooled and air-cooled condensing air conditioning systems.

The design point was:

Saturated suction temperature	40°F
Saturated condensing temperature	105°F – water cooled condensing
	125°F – air cooled condensing
Capacity of	50 TR or 13 TR

The basic design of the initial development compressor consisted of two compressor impellers both rated to produce a design pressure ratio of 2:1, mounted on a common shaft. The shaft was integrated with the motor rotor of an electric 50-kW induction motor rated for up to 50,000 rpm.

The drive shaft was supported radially by two foil gas journal bearings and axially by a foil gas thrust bearing. Axial thrust on the shaft was minimized as the two impellers were thrusting in opposite direction. The rotating elements were held together by a 3/8-inch diameter tie rod. The first stage discharged gas through a transition pipe and into the suction of the second stage.

An initial "breadboard" 2-stage compressor design (Figure 1) rated for 50TR employed open-shroud impellers, collectors, channel diffusers and 50kW induction motor. A cutaway view is shown in Figure 2. The compressor was successfully bench tested at speeds up to 33,000 rpm and various mechanical problems relating to component design, tolerances and manufacture were resolved. The compressor also operated successfully with refrigerant at speeds up to 29,000 rpm on a gas cycle rig, which was designed to provide various air-conditioning operating conditions at the suction port. Critical speed instability was detected at speeds exceeding 33,000 rpm. However, the tests provided proof of concept of the compressor.

A total of 10 builds and 54 tests were made with testing efforts concentrated on solving dynamic problems.

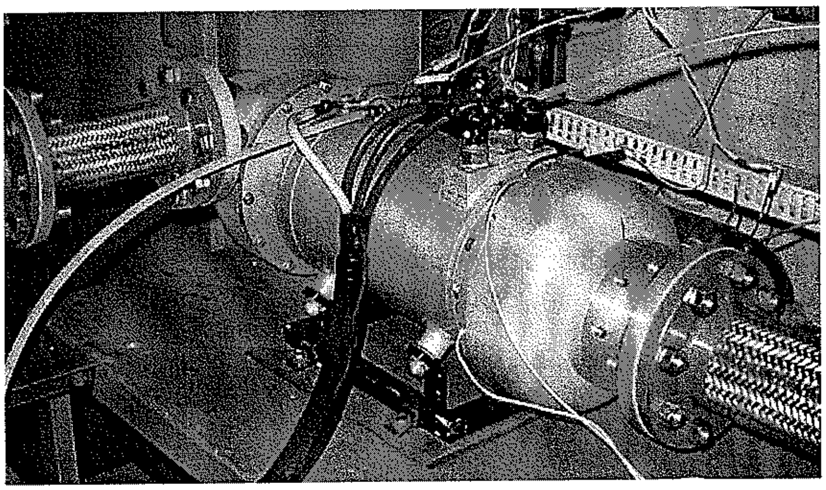

Figure 1: Breadboard Compressor Installed on Gas Test Rig

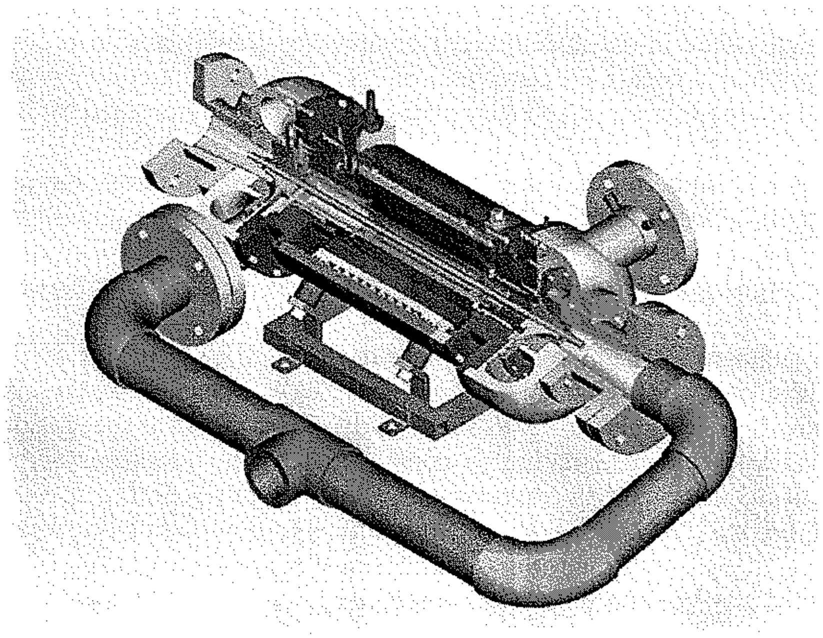

Figure 2: Cutaway View of Breadboard Compressor

The "breadboard" design was replaced by an optimized prototype design A (Figure 3). The design point was the same as for the breadboard compressor: capacity of 50 tons of refrigeration at 40°F saturated suction temperature, 125°F saturated discharge temperature in an air-cooled condensing refrigeration system.

The objectives of the prototype were:

- Optimize aerodynamic and thermodynamic performance
- Improve dynamic stability throughout operating range by making rotating elements lighter, shorter and stiffer
- Reduce cost by reducing the number of parts, improve producibility and ease of assembly

Aerodynamic design

The aerodynamic components were specified using computational fluid dynamics (CFD) computer code for impellers, diffusers and volutes. Input to the CFD was separate for first and second stage. Table 1 shows the basic input data to the code specifying refrigerant, speed, pressure ratio, operating condition at design point according to ARI standard and expected isentropic efficiency. The design strategy worked from reference design conditions consistent with good design-practice, achievable performance goals and focused on obtaining flow coefficients close to 0.1, which yielded the highest efficiency. Table 2 shows comparison of prototype to development design. Figure 4 and 5 show the CFD design output for each stage.

Design Point Specification		1st Stage	2nd Stage
Gas		R-134a	R134a
Isentropic Head	Btu/lbm	6.31	6.51
Volume Flow Rate	Cfm	133.88	81.22
Mass Flow Rate	lbm/min	137.84	159.74
Pressure Ratio		2.0018	2.0018
Speed	rpm	42,000	42,000
Suction Specific Volume	ft^3/lbm	0.9713	0.5090
Suction Temperature	°F	47.00	97.3
Suction Pressure	psia	49.77	99.63
Suction Enthalpy	Btu/lbm	110.26	118.24
Discharge Isentropic Enthalpy	Btu/lbm	116.56	124.76
Discharge Pressure	psia	99.63	199.44
Discharge Enthalpy (target)	Btu/lbm	117.76	126.0
Discharge Temperature (target)	°F	90.26	150.92
Isentropic Efficiency (target)	%	84	84

Table 1: Basic Input Data for CFD Computer Code

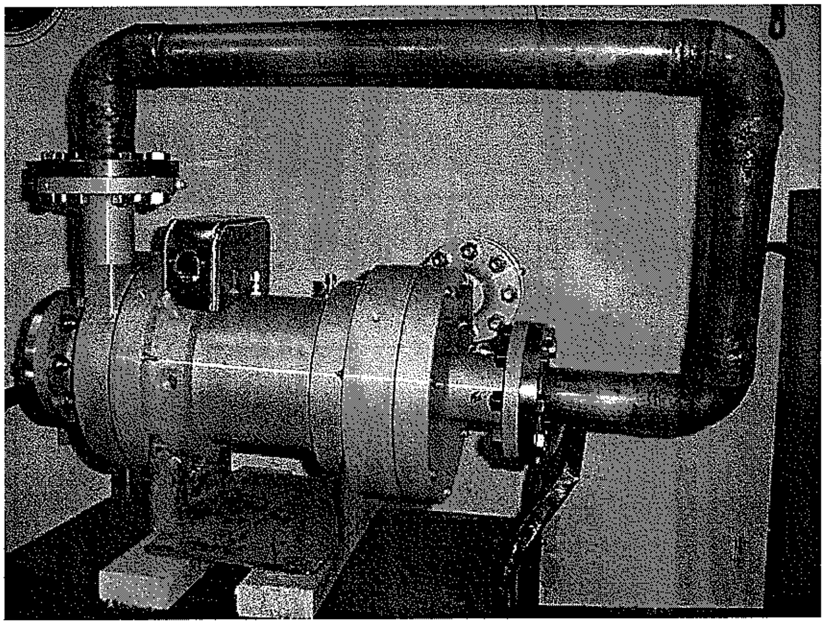

Figure 3: Prototype Design A Compressor with Stage 1 (left) Discharging Via Transition Tube to Stage 2

	Stage 1 Cenva Dev. Rotor	Stage 1 Prototype Rotor	Stage 2 Cenva Dev. Rotor	Stage 2 Prototype Rotor
Speed rpm	42000	42000	42000	42000
Inlet min diameter in	0.75	1.00	0.75	1.00
Inlet max diameter in	1.75	2.012	1.49	1.772
Blade thickness at suction	0.015	0.015	0.015	0.015
Inlet cross section in^2	1.7985	2.2270	1.1798	1.5534
No blades	11	11	11	13
Tip Diameter in	3.50	2.84	3.40	2.684
Tip blade height in	0.14	0.17	0.088	0.099
Blade thickness at discharge	0.03	0.03	0.03	0.03
Tip discharge area in^2	1.4932	1.4607	0.9109	0.7962
Backsweep degrees	-30	-10	-32	-3
Tip Speed ft/sec U_2	641.41	520.46	623.08	491.87
Design massflow lbm/min	137.84	137.84	159.76	159.76
Spec volume R134a ft^3/lbm	0.9713	0.9713	0.509	0.509
Q ft^3/sec	2.2314	2.2314	1.3553	1.3553
F Flow Coefficient $Q/(p \times r^2 \times U_2)$	0.0521	0.0975	0.0345	0.0701

Table 2: Design Parameters and Flow Coefficients for Prototype and Breadboard Compressor

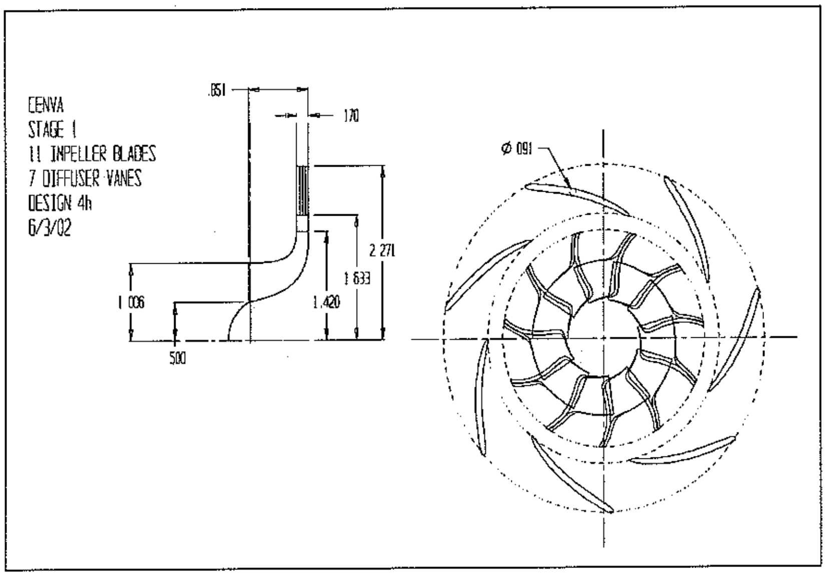

Figure 4: CFD Design of First Stage Impeller and Diffuser

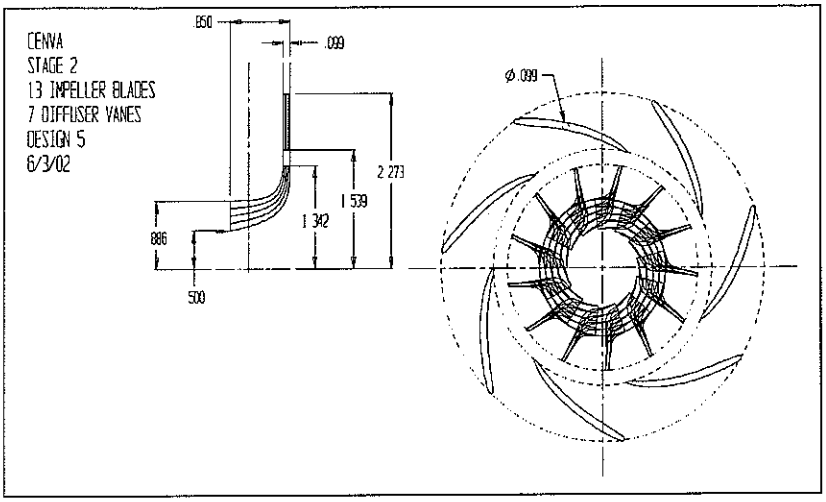

Figure 5: CFD Design of Second Stage Impeller and Diffuser

On the basis of the basic input data the CFD code generated performance tables which were converted into performance maps for each compressor stage. Chart 1 and 2 shows predicted head pressure as a function of corrected flow and corrected speed for each stage. Chart 3 and 4 shows predicted pressure ratio as function of corrected flow and corrected speed for each stage. The maps were guides for tracking performance during tests.

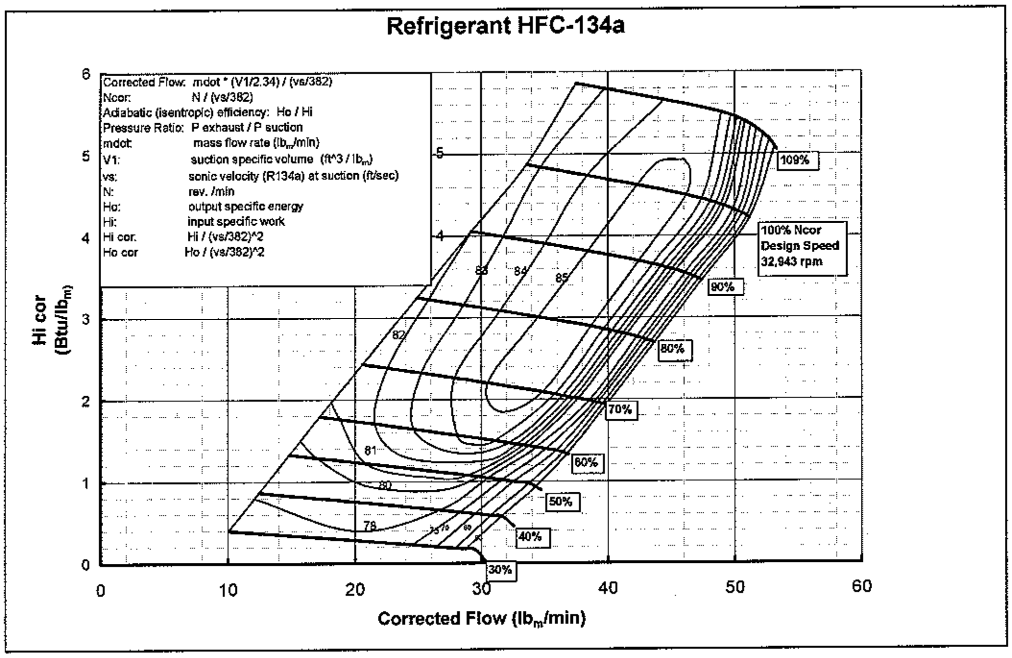

Chart 1: Stage 1 Compressor Map: Predicted Head Pressure Vs Corrected Flow

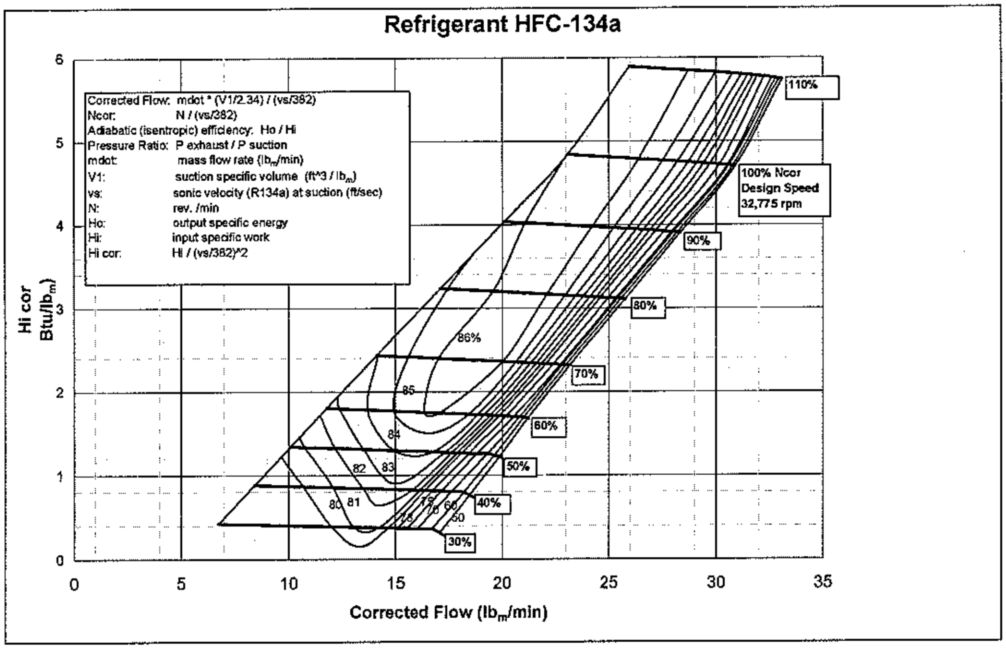

Chart 2: Stage 2 Compressor Map: Predicted Head Pressure Vs Corrected Flow

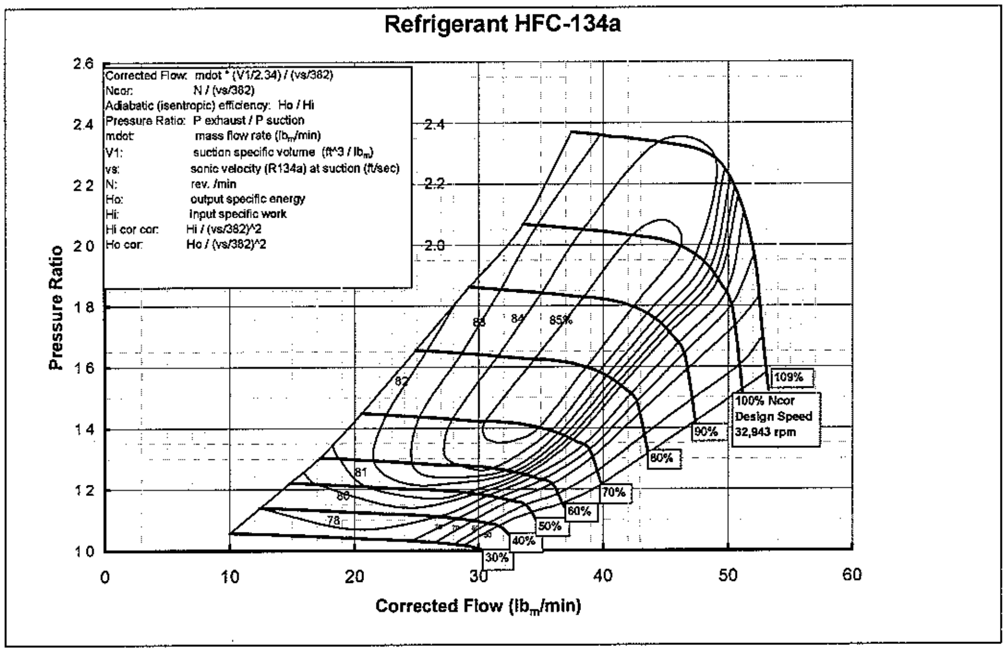

Chart 3: Stage 1 Compressor Map: Predicted Pressure Ratio Vs Corrected Flow

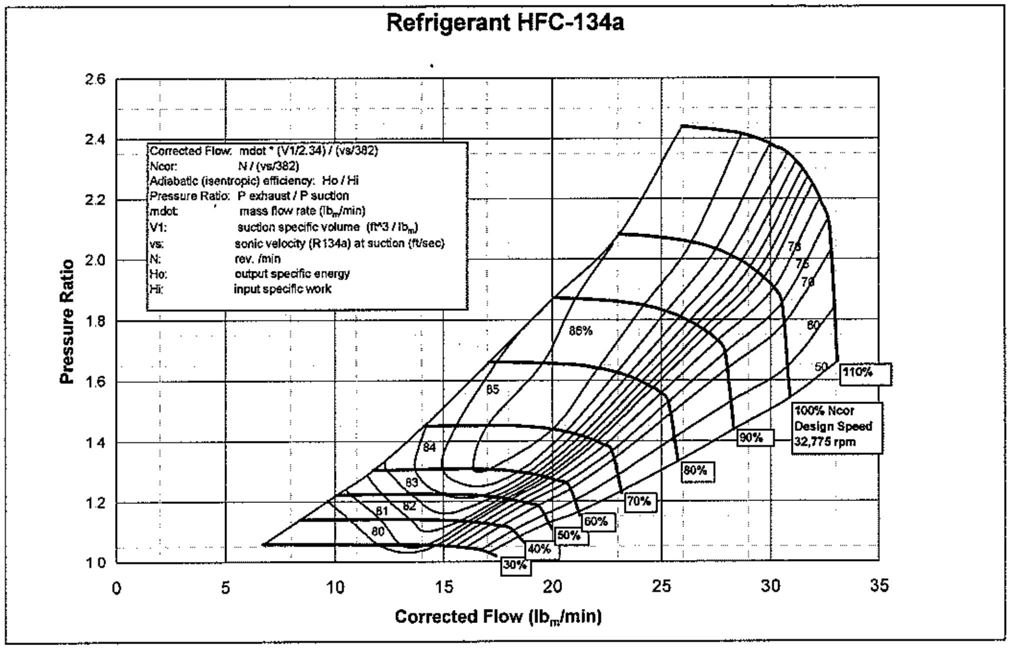

Chart 4: Stage 2 Compressor Map: Predicted Pressure Ratio Vs Corrected Flow

Impellers

The impellers were designed with closed shrouds: hub, blades and cover in one integrated part. The difference in efficiency between a shrouded and open impeller was about 6% due to elimination of leakage between rotor and shroud as well as across rotor blades. Leakage of open impellers was a function of clearance between impeller and fixed shroud, a clearance that had to be sufficient to allow for some axial movement of the rotating elements during operation.

Shrouded impellers were complex to produce and could only be made in a casting process as machining was not feasible because the gas passages were too small to allow access for cutting tools. Due to the low quantity and to avoid costly patterns, the impeller castings for both stages were made using rapid prototyping by stereolithography. The finish of the internal passages of the casting surfaces was improved using an extrude-hone process. The first stage impeller had 11 blades and the second stage had 13 blades (Figure 6).

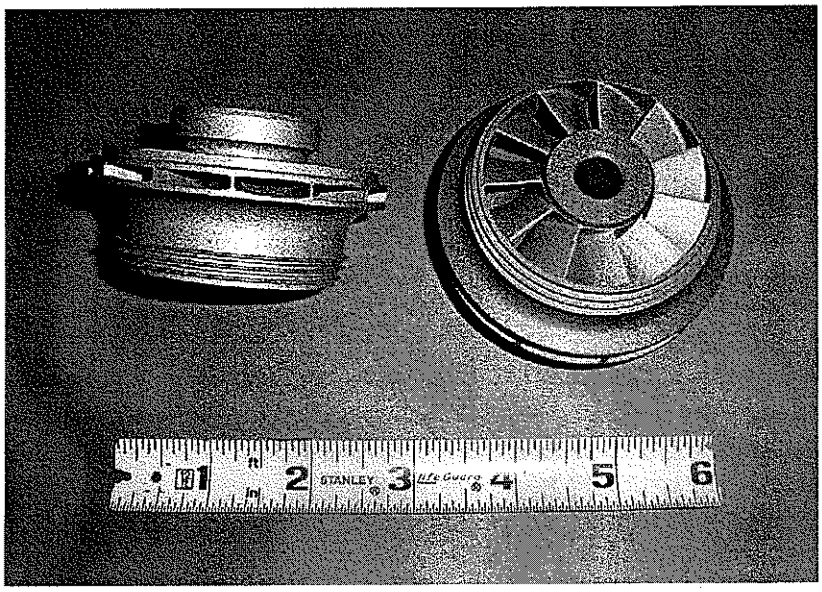

Figure 6: Shrouded Impellers Investment Cast from Rapid Prototype Models

Motor & Variable Frequency Drive (VFD) Controller

Ideally, the motor and controller should originate from the same manufacturer to assure complete compatibility. An extensive worldwide search of variable frequency drives to find an existing product and avoid going through a costly design development was not successful. This project required frequencies up to 1400 Hz, 45 kW power and ability to operate at speeds over 65,000 rpm.

Drives for DCPM motors with frequencies above 400 Hz and with power rating above 5 kW were not available. In a joint effort with a German manufacturer and a Korean consulting engineer a variable frequency drive controller was developed to control a permanent magnet motor without requiring an encoder (Figure 7). This combination of motor and controller generated substantially less heat thereby increased efficiency. Physically it was the size of a small filing cabinet but eventually the controller needs to be reduced in size and integrated into the compressor unit.

The new VFD sensorless controller was successfully operated at speeds in excess of 100,000 rpm.

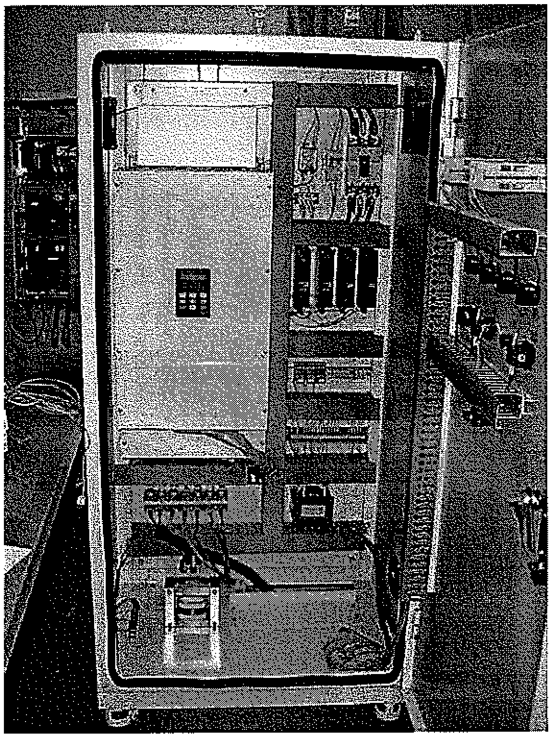

Figure 7: Sensorless VFD Controller

Motor

Permanent magnet, switched reluctance, flux switching and induction motors were investigated. Only induction motors were available as off-the-shelf standard products in the specified speed range. Other alternatives required time-consuming and costly development.

The motor selection for the prototype design A compressor was a four-pole induction motor rated for 45 kW power at 42,000 rpm. The rotor was 2 inches shorter and weighed 4.2 pounds less than the development compressor rotor. Both motors were manufactured by E+A of Switzerland, a supplier with an extensive motor program primarily used in high-speed machine tools.

The motor rotor was mounted on a single hollow steel shaft, which extended beyond both ends of the rotor providing journals for the air bearings. This was an improvement over the shaft of the development compressor, which was split into three sections making the shaft less rigid, increased machining cost and tolerance build up. A thrust runner was fitted to the end of the shaft. The entire rotating element was held together with a 7/16-inch tie rod, which was supported in the center of the hollow shaft.

The length of the rotating element was 18 inches, a reduction of 23% compared to the development compressor (Figure 8). The reduced length and weight of the rotating elements changed the first bending mode critical speed to 77,000 rpm compared to 57,000 rpm for the development compressor (Figure 9 and 10).

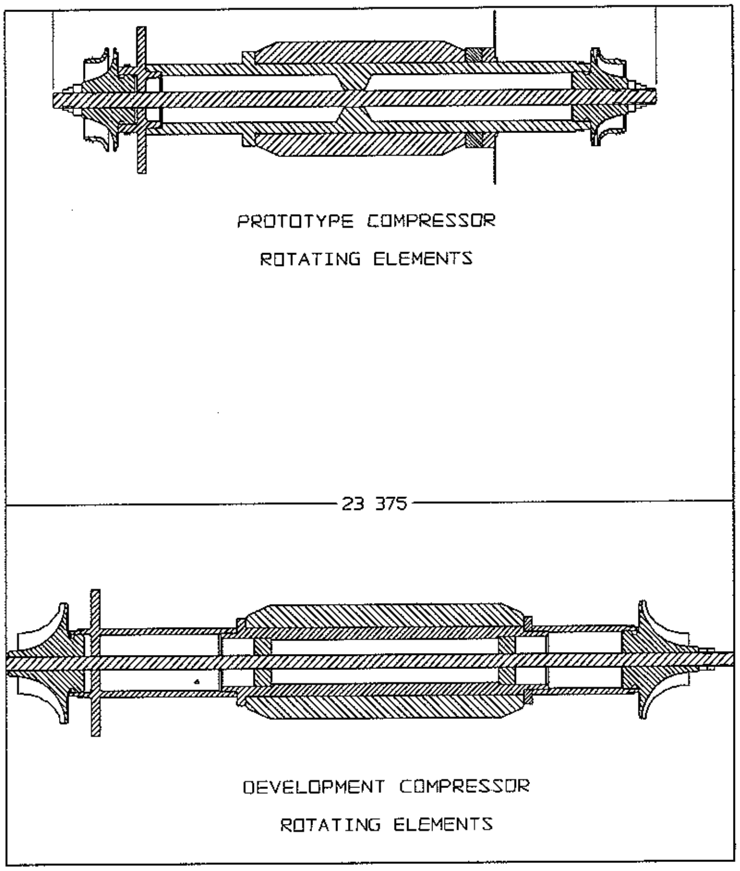

Figure 8: Comparison of Rotating Elements of Prototype and Development Compressor

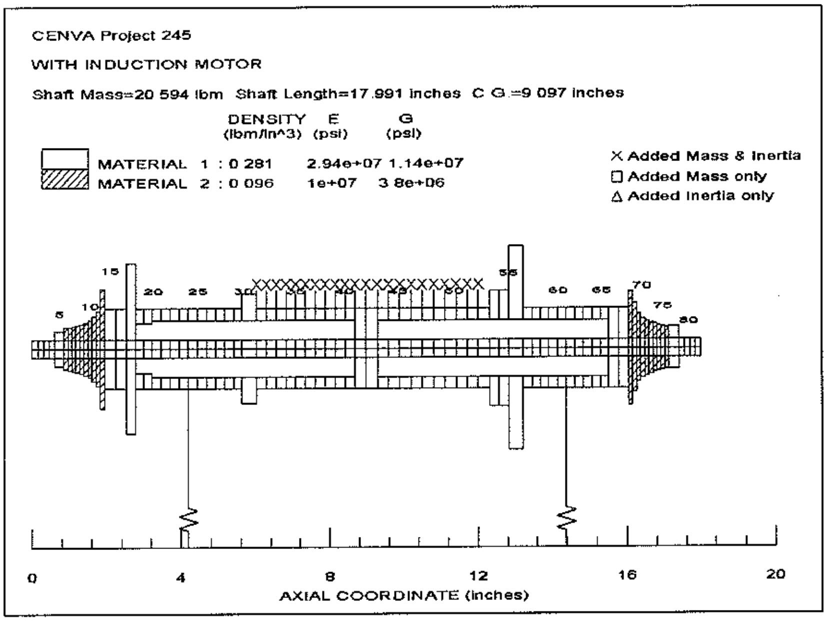

Figure 9: Shaft Geometry and Bearing Locations

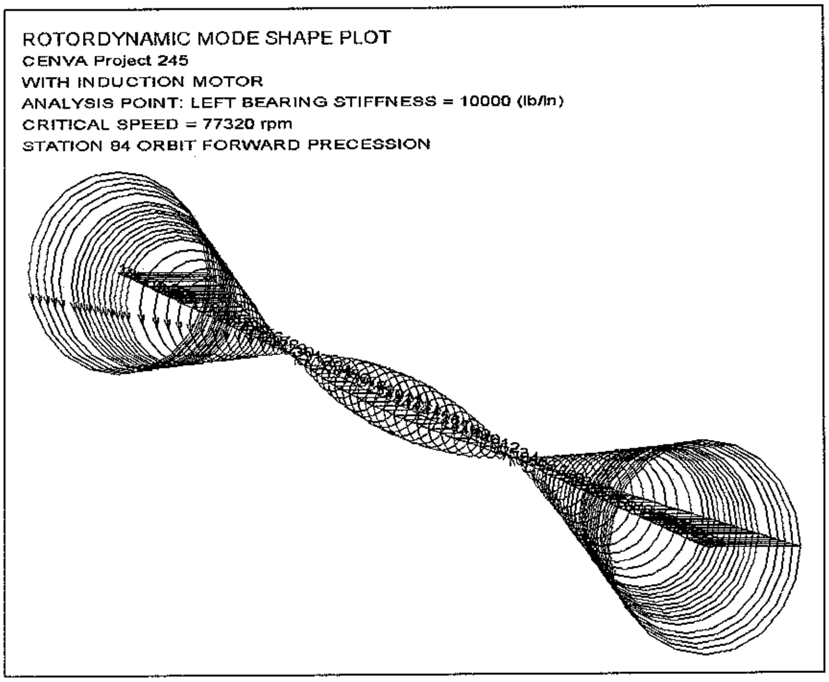

Figure 10: Rotordynamic Shape of Rotating Elements at First Bending Mode Critical Speed.

A total of 13 builds were made of this machine. Both breadboard and Prototype A designs had two stages mounted on either end of a motor, requiring a very substantial transition piping from the exit of stage 1 to the inlet of stage 2.

In tests the design had difficulty achieving expected efficiency and capacity. The problems manifested themselves both in how each stage performed and how they interacted. The physical separation of the two stages created flow problems at the inlet to the second stage and generated unacceptable pressure losses in the transition tubing from stage one to stage two resulting in low efficiencies.

Prototype Design B

Due to the performance deficiencies recognized in the extensive testing, a two-stage concept, design B, compressor with both stages cantilevered on the same side of the electric motor was developed (Figure 11 and 12). The design minimized pressure loss between stages as it eliminated the transition piping. After 3 builds and 15 test runs the concept was proven satisfactorily and was replaced by a prototype design rated for 15TR powered by a 30kW motor with a design speed of 65,000rpm. This machine was designed to apply an induction motor drive, which was later replaced by an in-house designed and manufactured permanent magnet motor (Figure 13) in order to alleviate dynamic stability problems at design speed, internal heat and aerodynamic losses. The prototype units totaled 15 builds and 146 test runs.

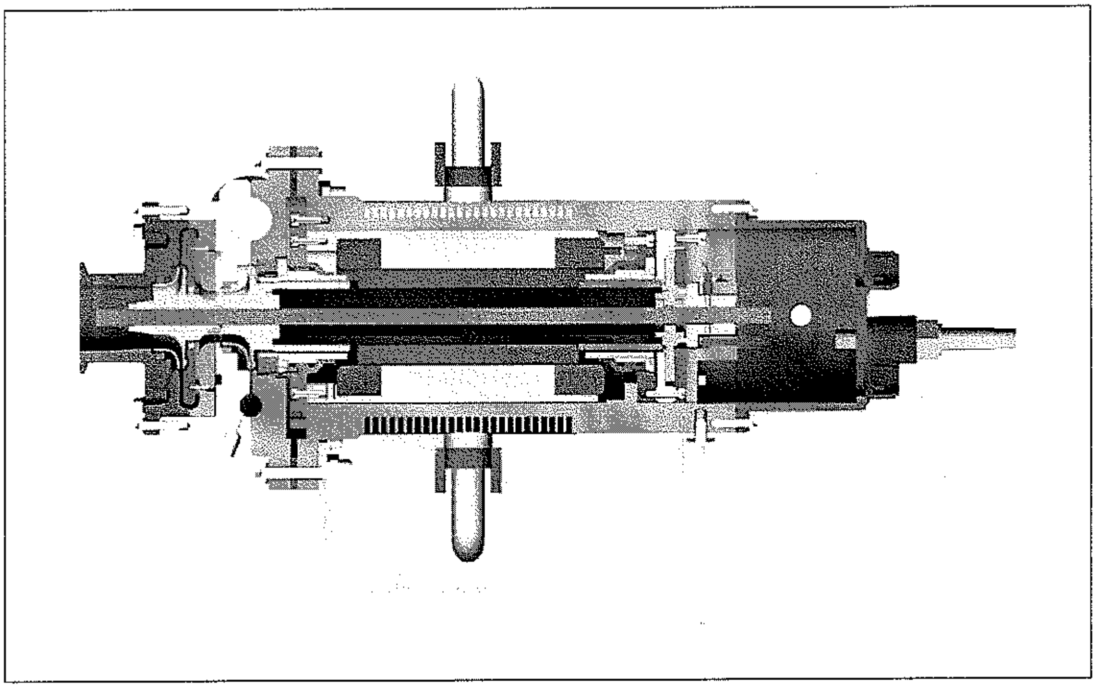

Figure 11: Schematic Cross Section of Design B Compressor

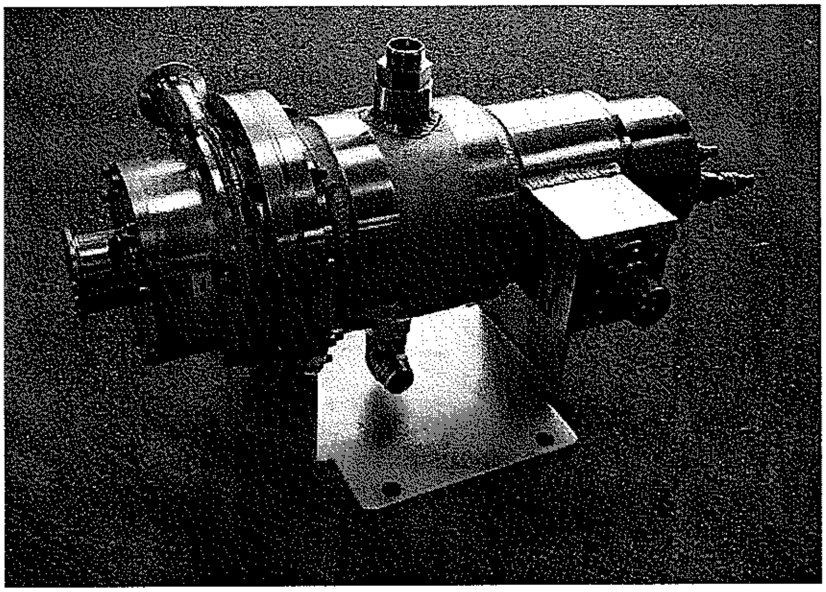

Figure 12: Prototype Design B Two-Stage Compressor

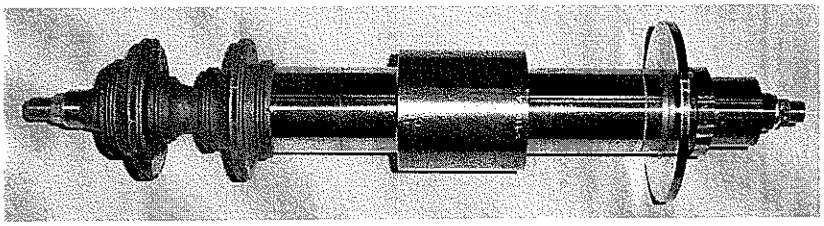

Figure 13: Design B Shaft with Two Impellers and Integral Permanent Magnet Motor Rotor

The new prototype design B had both impellers mounted on the same side of the drive motor, back to front. This compact prototype compressor had a design cooling capacity of 13 tons which could be modified into a 25-ton air conditioning compressor. The initial size of this compressor was chosen to expedite the new proof-of-concept design using existing in-house motor and controller.

The initial impeller, diffuser and return channel design would not allow speed to exceed 35,000 rpm as the operating conditions caused the compressor to go into surge. Based on mean line analysis, changes were made to the return channel and in further tests vanes were also removed from both diffusers. This allowed speeds of over 60,000 rpm, but flow and pressure ratios were insufficient to achieve the targeted performance.

Revised impeller designs with larger flow capacity were manufactured and tested. Speeds of over 65,000 rpm were achieved. Performance mapping was initiated to establish speed lines at mechanical speed from 50,000 to 65,000 rpm. However, the desired flow and pressure ratios were still not achieved. The machine experienced stability problems at speeds in excess of 65,000 rpm

Refrigerant Gas Cycle System

Testing of compressors was carried out on a custom-designed, in-house made refrigerant gas cycle where only a fraction of the discharge gas was condensed in a water-cooled condenser, sufficient to provide necessary control of inlet temperature and pressure. Liquid refrigerant from the condenser was mixed with discharge gas thereby allowing control of inlet temperature and pressure. The gas cycle was instrumented for continuous real time recording of temperatures, flows and pressures collected into a computerized data acquisition system (Labview) Data files from each test run allowed detail analysis and graphic display.

Testing

All compressors were initially tested in air to assure dynamic integrity of the rotating elements at relevant speed levels (Figure 14). Compressed air was forced through the compressors in order to provide internal cooling of motor rotor. Tests were completed with no-load impellers, followed by testing with actual impellers. A total of 62 test runs were conducted with the various compressors resulting in corrections and revisions of the critical elements.

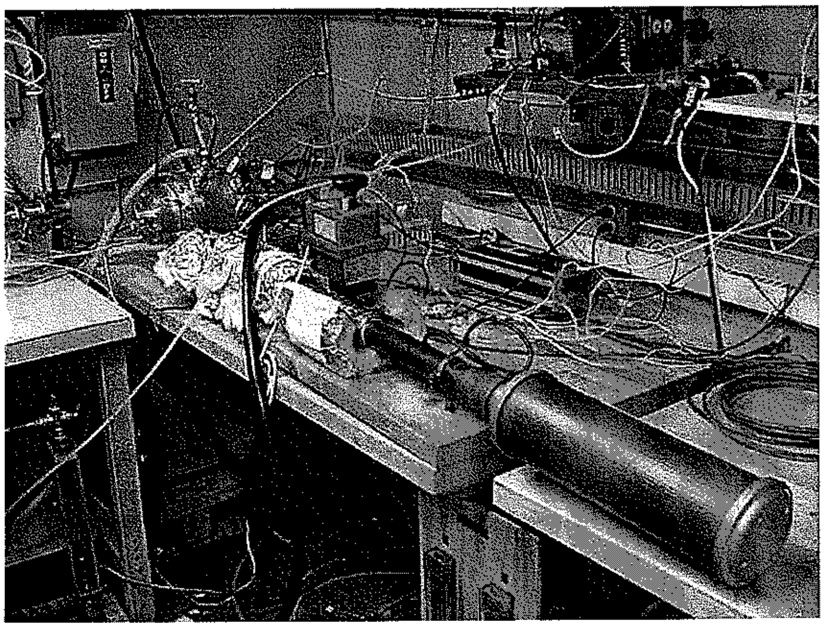

Figure 14: Compressor Being Tested Using Sensorless VFD Controller

The compressor was successfully operating at saturated suction temperature of 40°F and saturated condensing temperature of 105°F (the water cooled standard) at an overall efficiency of 40%.

Testing with R134a refrigerant was more difficult than testing in air due to the much higher density of refrigerant gas, the change in state from liquid to gas and back to liquid, the high pressures and the need for a completely leak tight system. A total of 135 test runs were conducted. Problems encountered were mainly related to high heat generation caused by windage, friction and motor losses. Dynamic stability issues were not overcome at speeds above 85% of design specification.

Technical Problems/Barriers

The compressor development encountered three technical problem areas, which prevented achieving the required efficiency goals:

1. Dynamic instability of the rotating elements at speeds above 80% of design specification
2. High internal losses in motor and rotating elements
3. Aerodynamic losses in the compressor

The dynamic problem was related to weight and length of the motor rotor and occurred when speed exceeded 55,000 rpm or 85% of design speed. Replacing the induction motor with a permanent magnet motor did significantly change the two physical dimensions. However, variable frequency drive controllers require encoders to indicate shaft position. High internal temperatures made use of encoders impractical.

The high internal losses were the reason that target efficiencies were not achieved. The losses related to windage, bearing friction, electrical motor and controller losses. Computer codes were developed for theoretical calculation of friction and windage losses as a function of operating conditions and component geometry. All loss categories grew exponentially with speed and were particularly obscured by motor/controller combination losses. Various cooling schemes were applied either by water cooling the main housing of the compressor or by flashing liquid refrigerant in the machine, but were largely unsuccessful. Further study and testing was in progress using a recently developed sensorless VFD controller.

The aerodynamic performance was evaluated through testing and using actual test data in computer models. The data suggested that the impellers were not perfectly matched. Stage one appeared to be efficient but stage two did not generate the required pressure ratio. A program to assess and map the individual stages separately was in progress but could not be completed by the end of the project.

System Testing

Testing with refrigerant R134a was successful up to almost 120°F saturated discharge temperature (SDT). See Chart 5. Excess heat generation prevented operating at the desired 125°F SDT required for air-cooled condensing.

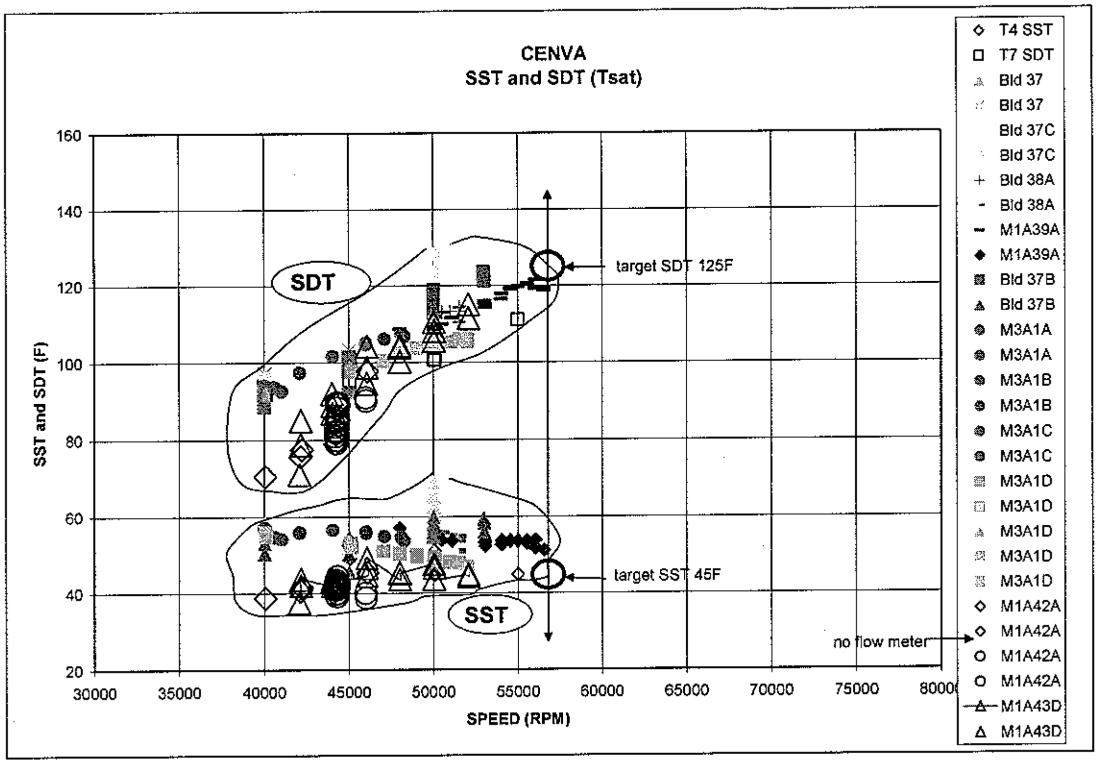

Chart 5: Test Results for Saturated Suction and Discharge Temperatures

Further testing was in progress with a revised compressor and VFD controller in order to meet the goal of 125°F SDT.

The original overall compressor efficiency target was 57% based on combination of efficiency of impellers, motor, controller, bearing friction and windage. Overall efficiency has improved from about 32% to an average of 40% at speeds of 50,000 to 56,000 RPM by a new manufacturing process for impellers. (See Chart 6)

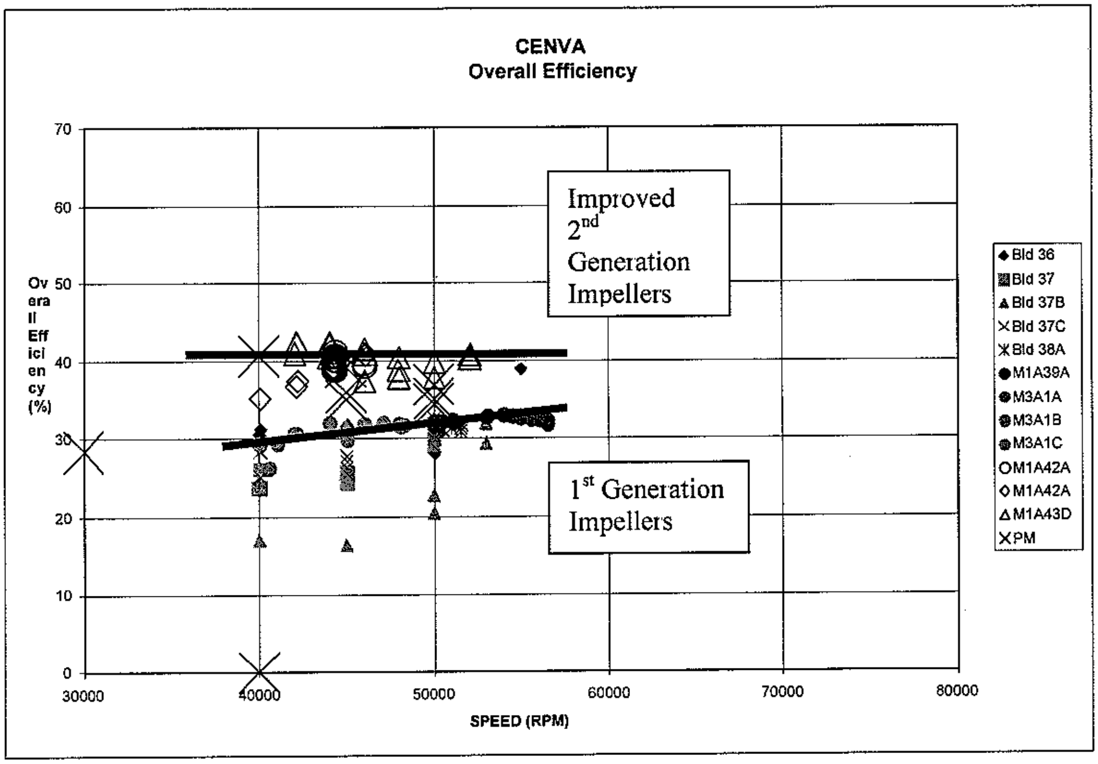

Chart 6: Test Results of Overall Compressor and Drive Efficiencies

Aerodynamic compatibility of the two compressor stages continued to cause stability problems at the 65,000 rpm design speed. Testing of the individual stages separately and study of the return channel design was in progress but was not concluded.

Field Testing

Field testing of the compressor is postponed until the system is fully developed.

CONCLUSION

A high-speed DC permanent magnet motor, essential to high efficiency compressors, was designed, manufactured and tested in R134a refrigerant. Other technologies, such as sensorless drives for DC permanent magnet motors and manufacturing of high quality shrouded impellers, were also developed. Additionally, the successful integration of the motor, impellers and bearings were demonstrated.

The overall compressor performance, through several design revisions, showed clear improvement towards the efficiency goals. The aerodynamic design of the two stages of impeller and diffuser were not perfectly matched preventing the machine from operate at design speed. The overall efficiency did not fully meet expectations due to high internal losses applying commercially available motors and controllers.

This phase of the project has clearly produced several important elements, which will be critical for delivering an efficient air conditioning compressor.

The following additional work is required to complete the development of a high-speed foil bearing supported refrigerant compressor:

- ❖ Redesign the impellers to improve matching
- ❖ Test sensorless drive with redesigned impellers
- ❖ Perform field testing

RECOMMENDATIONS

Since the oil-free high-speed foil bearing supported refrigerant compressor will save energy and improve the environment the work performed under this project should be continued.

Standard Form 269

FINANCIAL STATUS REPORT

(Long Form)

(Follow instructions on the back)

1 Federal Agency and Organizational Element to Which Report is Submitted U. S. Department of Energy	2 Federal Grant or Other Identifying Number Assigned By Federal Agency DE-FC26-02GO12014	OMB Approval No. 0348-0039	Page 1 of 1 pages

3 Recipient Organization (Name and complete address, including ZIP code)	State of Connecticut-Office of Policy and Management 450 Capitol Avenue Hartford CT 06106

4. Employer Identification Number 06-6000798	5 Recipient Account Number or Identifying Number 21549	6 Final Report ☒ Yes ☐ No	7 Basis ☒ Cash ☐ Accrual
8 Funding/Grant Period *(See instructions)* From: 09/15/02	To: (Month, Day, Year) 4/15/06	9 Period Covered by this Report From: ~~3/16/06~~ 9/15/02	To: (Month Day, Year) 6/30/06

10 Transactions:	I Previously Reported	II This Period	III Cumulative
a Total outlays	1 785,746 00	0 00	1,785,746 00
b Refunds, rebates, etc	0 00	0 00	0 00
c Program income used in accordance with the deduction alternative	0 00	0 00	0 00
d Net outlays *(Line a less the sum of lines b and c)*	1,785,746 00	0 00	1,785,746 00
Recipient's share of net outlays, consisting of: e. Third party (in-kind) contributions	785 755 00	0 00	785,755 00
f Other Federal awards authorized to be used to match this award	0 00	0 00	0 00
g Program income used in accordance with the matching or cost sharing alternative	0 00	0 00	0 00
h All other recipient outlays not shown on lines e, f or g	0 00	0 00	0 00
i Total recipient share of net outlays *(Sum of lines e, f, g and h)*	785,755 00	0 00	785,755 00
j Federal share of net outlays (line d less line i)	999 991 00	0 00	999 991 00
k Total unliquidated obligations			0 00
l Recipient's share of unliquidated obligations			0 00
m Federal share of unliquidated obligations			0 00
n Total federal share (sum of lines j and m)			999,991 00
o Total federal funds authorized for this funding period			999,991 00
p Unobligated balance of federal funds *(Line o minus line n)*			0 00
Program income, consisting of: q. Disbursed program income shown on lines c and/or g above			0 00
r Disbursed program income using the addition alternative			0 00
s Undisbursed program income			0 00
t Total program income realized *(Sum of lines q r and s)*			0 00

11 Indirect Expense	a Type of Rate *(Place "X" in appropriate box)* ☐ Provisional ☐ Predetermined ☐ Final ☒ Fixed			
	b Rate 55.36%	c Base 0.00	d Total Amount 0.00	e Federal Share 0.00

12 *Remarks: Attach any explanations deemed necessary or information required by Federal sponsoring agency in compliance with governing legislation.*

13 Certification: I certify to the best of my knowledge and belief that this report is correct and complete and that all outlays and unliquidated obligations are for the purpose set forth in the award documents.

Typed or Printed Name and Title M Joyal Gutis, Fiscal Administrative Manager	Telephone (Area code, number and extension) (860) 418-6285
Signature of Authorized Certifying Official M. Joyal Gutis	Date Report Submitted 7/21/2006

Standard Form 269
FINANCIAL STATUS REPORT
(Long Form)

Public reporting burden for this collection of information is estimated to average 30 minutes per response, including time for reviewing instructions, searching existing data sources, gathering and maintaining the data needed, and completing and reviewing the collection of information. Send comments regarding the burden estimate or any other aspect of this collection of information including suggestions for reducing this burden to the Office of Management and Budget Paperwork Reduction Project (0348-0039) Washington, DC 20503

PLEASE DO NOT RETURN YOUR COMPLETED FORM TO THE OFFICE OF MANAGEMENT AND BUDGET, SEND IT TO THE ADDRESS PROVIDED BY THE SPONSORING AGENCY.

Please type or print legibly. The following general instructions explain how to use the form itself. You may need additional information to complete certain items correctly, or to decide whether a specific item is applicable to this award. Usually, such information will be found in the Federal agency's grant regulations or in the terms and conditions of the award (e.g., how to calculate the Federal share, the permissible uses of program income, the value of in-kind contributions, etc) You may also contact the Federal agency directly

Item	Entry
1, 2 and 3	Self-explanatory.
4.	Enter the employer identification number assigned by the U S. Internal Revenue Service
5	Space reserved for an account number or other identifying number assigned by the recipient
6	Check yes only if this is the last report for the period shown in item 8
7.	Self-explanatory.
8.	Unless you have received other instructions from the awarding agency, enter the beginning and ending dates of the current funding period. If this is a multi-year program, the Federal agency might require cumulative reporting through consecutive funding periods. In that case, enter the beginning and ending dates of the grant period, and in the rest of these instructions, substitute the term "grant period" for "funding period "
9	Self-explanatory
10.	The purpose of columns, I, II and III is to show the effect of this reporting period's transactions on cumulative financial status. The amounts entered in column I will normally be the same as those in column III of the previous report *in the same funding period* If this is the first or only report of the funding period, leave columns I and II blank. If you need to adjust amounts entered on previous reports, footnote the column I entry on this report and attach an explanation.
10a	Enter total gross program outlays. Include disbursements of cash realized as program income in that income will also be shown on lines 10c or 10g. Do not include program income that will be shown on lines 10r or 10s. For reports prepared on a cash basis, outlays are the sum of actual cash disbursements for direct costs for goods and services, the amount of indirect expense charged, the value of in-kind contributions applied, and the amount of cash advances payments made to subrecipients For reports prepared on an accrual basis, outlays are the sum of actual cash disbursements for direct charges for goods and services, the amount of indirect expense incurred, the value of in-kind contributions applied, and the net increase or decrease in the amounts owed by the recipient for goods and other property received, for services performed by employees, contractors, subgrantees and other payees, and other amounts becoming owed under programs for which no current services or performances are required, such as annuities, insurance claims, and other benefit payments.

Item	Entry
10 b.	Enter any receipts related to outlays reported on the form that are being treated as a reduction of expenditure rather than income, and were not already netted out of the amount shown as outlays on line 10a
10 c	Enter the amount of program income that was used in accordance with the deduction alternative
Note:	Program income used in accordance with other alternatives is entered on lines q, r, and s Recipients reporting on a cash basis should enter the amount of cash income received; on an accrual basis, enter the program income earned. Program income may or may not have been included in an application budget and/or a budget on the award document. If actual income is from a different source or is significantly different in amount, attach an explanation or use the remarks section
10d, e, f, g, h, i, and j.	Self-explanatory
10k	Enter the total amount of unliquidated obligations, including unliquidated obligations to subgrantees and contractors Unliquidated obligations on a cash basis are obligations incurred, but not yet paid. On an accrual basis, they are obligations incurred, but for which an outlay has not yet been recorded Do not include any amounts on line 10k that have been included on lines 10a and 10j On the final report, line 10k must be zero
10l	Self-explanatory
10m	On the final report, line 10m must also be zero.
10n, o, p, q, r, s and t.	Self-explanatory.
11a	Self-explanatory.
11b	Enter the indirect cost rate in effect during the reporting period
11c	Enter the amount of the base against which the rate was applied
11d.	Enter the total amount of indirect costs charged during the report period
11e	Enter the Federal share of the amount in 11d
Note:	If more than one rate was in effect during the period shown in item 8, attach a schedule showing the bases against which the different rates were applied, the respective rates, the calendar periods they were in effect, amounts of indirect expense charged to the project, and the Federal share of indirect expense charged to the project to date