Topical Progress Report for the Variable Speed Integrated Intelligent HVAC Blower

Comprehensive Topical Report for BP1 (9/2000 - 11/2001)

Reporting Period Start Date: 9/29/00

Reporting Period End Date: 11/14/01

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Date of Issue: November 14, 2001

DOE Award: **DE-FC26-00NT40993**

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Abstract

This comprehensive topical report discusses the key findings in the development of a intelligent integrated blower for HVAC applications. The benefits of rearward inclined blades over that of traditional forward inclined blades is well documented and a prototype blower design is presented. A comparison of the proposed blower to that of three typical units from the industry is presented. The design of the blower housing is also addressed and the impact of size limitations on static efficiency is discussed. Issues of air flow controllability in the rearward inclined blower is addressed and a solution to this problem is proposed. Several motor design options are discussed including inside-out radial flux designs and novel axial flux designs, all are focused on the various blower needs. The control of the motor-blower and airflow through the use of a high density inverter stage and modern digital signal processor is presented. The key technical challenges of the approach are discussed. The use of the motor as a sensor in the larger heating/ventilating system is also discussed. Diagnostic results for both the motor itself and the blower system are presented.

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I. Introduction

Every year a considerable amount of energy is wasted due to the inefficient operation of commercial and residential heating, ventilation and air conditioning systems (HVAC). The Department of Energy (DOE) has estimated that HVAC, by far the largest consumer of energy in buildings, accounts for 6.7 Quads of energy consumed per year in the U.S. at a national cost of more than \$150B.

One of the best opportunities for reducing the energy consumption of HVAC is to improve the air handling and distribution system. Based on DOE data, more than 14% of the residential HVAC and 25% of the commercial energy consumption can be attributed to air handling equipment – primarily fans and motors which by themselves account for 1.23 quads and the equivalent production of 19.3 million metric tons of carbon emissions. The variable speed Electronically Commuted Motor (ECM) motor introduced several years ago by GE has been one of the few significant technical advancements in residential and commercial air handling systems. Today's most efficient two stage residential furnaces are dependent on the variable speed ECM that have a significant efficiency advantage over the single speed induction motor. The ability to vary fan speed depending upon HVAC usage cycles can significantly reduce the electrical energy consumption by 75-80% for a 3-5 ton residential 2-stage HVAC application since the blower will run at 60% of full rated speed for 90% of the time and the consumed power in an ECM varies with the third power of speed.

The number of commercial and residential HVAC vendors incorporating variable speed blowers into their products is still quite low; e.g. only ~7%. Several reasons for the lack of penetration have been the cost of a variable speed blower along with the more complex engineering and commissioning procedures associated with the technology. The overall efficiency of an ECM driven blower has also been compromised by poor fan and shroud designs.

To address these issues GE assembled a team that united several major vendors of HVAC subsystems with one of the US's leading industrial R&D laboratories. This team, consisting of GE Corporate Research and Development (CRD), GE Industrial Systems, Revcor, Trane, Carrier, and Nailor has the expertise and desire to substantially increase the efficiency of air handling systems. This team brought together not only a strong group of researchers and engineers capable of developing more efficient air handling and distribution technology, but also a group of leading HVAC vendors who are in a position to leverage the technology directly into products that will make a difference relative to U.S. energy consumption.

During the first budget period of the DOE-NETL award, the GE team has developed and demonstrated the basic elements of an intelligent integral motor-blower; advanced fan and blower systems, custom integral motor designs, repacked power electronics for increased reliability, self-diagnostic methodologies.

II. Executive Summary

The technical results to date of the first budget period (BP1) are summarized in this comprehensive topical report.

Programmatic:

The project is spending according to plan with no major diversions. The planned spending in BP1 will be the total \$1.95M, of which the DOE is billed 50.42%. The first budget period was extended from a finish date of Sept. 29, 2001 to Dec. 31, 2001. This recommendation was to coordinate the DOE project with standard project execution practices at GE Research and Development. The technology development team included members from GE motors in Fort Wayne, Indiana, and a multi-discipline team of research engineers at GE-CRD in Schenectady, NY.

A patent waiver was applied for granted.

Customer QFD's:

Several customer centric meetings were held with key HVAC OEM's. These sessions were instrumental in gauging the desires of the HVAC industry and learning of the constraints on them and is discussed in section III.B on page 9. These sessions increased the probability of producing a successful design via a close examination of the product requirements and characteristics. The greatest need was for increased blower efficiency, the second largest desire was for an integrated fan-motor solution which would greatly reduce the HVAC system development time. Other high impact concerns were reliability, cost, and repeatable/accurate air flow delivery.

Alliance with a Fan Manufacturer:

An alliance and sub-contract with a fan manufacturer was not fully acted upon during BP1. A few fan projects were initiated with the proposed manufacture ([Revor Inc](#)) and acted upon, but the high efficiency fan manufacturability issue was not ready for execution. The goals for the high efficiency fan were focused instead on understanding the blower technology and discovering the many trade-offs of the high efficiency blower system. Several key trade-offs in efficiency, noise, and air flow regulation have recently been documented, thus opening the door for the next state of productization. Budget period 2 (BP2) will bring a renewed focus on manufacturability of the GE wheel designs as we move from the laboratory to production prototypes.

Technical Deliverables:

The development of a high efficiency fan system is well underway with an initial result of 70% peak static efficiency with the GE-1 prototype rearward inclined wheel. This represents a 30-50% reduction in the mechanical power needed to satisfy the air flow requirements for a typical residential HVAC system as compared with today's forward inclined blowers. An issue of housing design is also presented in section IV.A.7 on page 21. The industry generally uses too small a housing due to total enclosure fitment and cost issues. The adoption of different aspect ratios and larger size blower housings can increase the static air moving efficiency beyond the 70% range. The cost of retooling for larger blowers is prohibitive, and will only be considered by the industry if there is sufficient SEER rating pressure or a deep need to investigate further system energy saving techniques.

The control of the air flow in traditional, low efficiency, forward inclined blowers is done via the motor's torque and speed information. The proposed high efficiency rearward inclined blowers do not feature a unique set of torque vs. speed curves and modifications to the blower wheel design are necessary to recoup some of this controllability. Alternatively, a method of airflow control via pressure rise information is presented in section IV.B on page 24, and equivalent air flow controllability can be achieved with the inclusion of a differential pressure sensor. Piezoresistive pressure transducers can be an enabling technology for high efficiency air flow control.

Several motor options were investigated for the integrated blower system. The ultimate objective was high system efficiency, and compact design. The design of several types of permanent magnet DC machines are presented, with a conventional radial flux, medium flux density rotor magnet being the overall best option. A detailed trade-off tool was used to evaluate the several fan-motor-controller options relative to performance, cost, reliability, and manufacturability. Axial

flux motor designs were also considered and a prototype unit will be compared in parallel with the radial flux design in order to gain more insight into this promising technology. See section IV.C on page 28 for details.

A high density electronics packaging method is presented which addresses a pareto of failed motor data. The reliability of the electronics can be traced back to failures in the power electronics stage due to high temperature operation. The use of more robust circuitry and derating the motor power as a function of over temperature will significantly improve the reliability of the blower system. The use of new circuits and packaging techniques also significantly reduced the footprint of the electronics by 50% which was needed due to an equivalent reduction in the motor cross sectional area which was driven by the requirement for a less restrictive intake path for the blower air. The proposed motor and electronics dimension is only 10 cm (4 in) in overall dimension, reduced from today's product which is 14 cm (5.5 in) in diameter.

Diagnostic measures for both the motor and for the blower system are demonstrated in section IV.G on page 39. Thus showing that a motor can be used as an effective sensor in the HVAC system. The inclusion of motor self diagnostics also helps the industry reduce the fear of electronics and technology in an otherwise low-technology industry. Often electronically controlled motors are improperly diagnosed and replaced unnecessarily when a system fault occurs. The self diagnostic function will give the installer and repair person a quick indication of the status of the blower unit with error codes to possible system faults.

III. HVAC Industry Perspective

The HVAC industry for commercial and residential systems was queried and several key observations were noted. The industry has only recently made air flow efficiency a high priority with the advent of stricter minimum SEER ratings under the Clinton administration. Normally the comfort and control features of the This section discusses GE's attempt to quantify the real requirements for the intelligent integrated blower, and to maximize the efficiency impact on residential and commercial air handling equipment.

A. Target Application

The target application for the proposed intelligent integrated motor-blower is a 3-ton Heat Pump with dual stage HVAC. The broad range of air moving equipment generally spans the 1 ton (400 CFM) to 6 ton (2400 CFM) range.* The target 3-ton application represents the middle of the market where there is significant sales but only a moderate presence of dual stage, high efficiency equipment with variable speed motor technology. The target HVAC requirements is summarized in Table III-1 and it shows both the SEER rating point and the maximum typical load for a blower in an actual installation. See section IV.A.1 on page 13 for detailed data on various HVAC blower ratings.

* for a definition of "ton" and "CFM" see section VIII Conversion Factors on page 44.

Table III-1 Operating condition for the target residential HVAC blower.

Operating Conditions	SEER rating	Maximum loading
Airflow	0.567 m ³ /sec (1200 CFM)	0.638 m ³ /sec (1350 CFM)
Blower pressure rise	186 Pa (0.75 inwg)	298 Pa (1.25 inwg)

B. HVAC Industry Requirements

In the early stages of the project (1Q01) three main OEM HVAC manufacturers (Carrier, Trane, and Nailor) were queried with GE Quality Function Deployment (QFD) sessions. Each session had various representatives from GE-CRD, GE-Industrial Systems and the customer. The end results of the three QFD sessions were averaged and are shown below in Figure III-1. The averaging of the results was done to smooth over the various interests of the sampled attendees. The "voice of the customer" (VOC) is measured through the number of issues in various product and service categories multiplied by a weighting factor of relative importance that the customer places on each issue. The net results from the three QFD sessions then give the product and technology development team a clear set of design goals and trade-off objectives.

A summary of the VOC results in Figure III-1

- Electrical and mechanical efficiency of the new blower *
- Reliability and diagnostic capabilities for increased quality and customer satisfaction
- Cost of the blower unit relative to a comparable low efficiency blower system
- HVAC performance relative to constant CFM control and creature comforts
- Environment, health and safety compliance of the new blower in HVAC systems
- Programming flexibility to adapt blower to any future communication or control methods
- Serviceability of the unit in the field

* the QFD's were held during the Clinton administration, which proposed a new minimum SEER rating of 13. This has since been relaxed by the Bush administration to a minimum of SEER 12 for heat pumps and AC units.

Figure III-1 Relative strength of customer concerns and GE response vs. category.

The response of GE to the VOC is documented in how we attack the features and technologies in the product. GE's response is summarized in Figure III-1 and Figure III-2. The main need was for a totally new fan design which would be attached to an integral motor with dedicated, pre-programmed digital controller. The use of plastics in the fan was also seen as a vital consideration for keeping costs low. Other top GE responses are focused on reliability, repeatability, and control / programming flexibility. The technology proposal outlined in the DOE-NETL award is in alignment with the objectives outlined during the QFD sessions.

Figure III-2 The top 13 blower design responses for residential and commercial HVAC .

The HVAC industry did have some constraints which effect the development of a truly novel and highly efficient product.

- 1) Using "suitable" plastics for the fan may be impossible in furnace applications due to regulations and limitations on plastic surface area. OEM's have tried unsuccessfully in the past to qualify products with plastic fans.

- 2) Redesigning the blower housing for increased efficiency is unfavorable from a legacy and retooling perspective. Overall enclosure size is a limitation which must be maintained. Retooling for a new blower dimension is a major investment by the OEM's.
- 3) A new 3 wire serial communication method may be unwelcome due to the existence of legacy systems which rely on the multi-wire thermostat interface to the motor. This also limits the benefit of digital communications and on-board diagnostics and intelligence. It takes approximately one year to develop a new furnace controller and longer to safety qualify it for use in a residence.

C. *Pareto of Faults*

The causes of blower motor failures were analyzed in order to produce a higher quality system. Three interesting results were uncovered in the area of motor faults. Most faults were easily attributed to issues with the integration of new components, or untested parts supplied by vendors. An alarming number of field rejects had no discernable failure mechanism, or the failure mode was not reproducible when the motor was being re-evaluated, thus indicating either an intermittent failure, or more probably the motor was replaced inappropriately along with other HVAC system components. The return of nominally good motors suggests that an on-board self diagnostic feature is needed. Section IV.G on page 39 discusses motor and HVAC system diagnostics in detail.

Figure III-3 Annual ECM failure percentage summary for selected Y2000 and Y2001 products.

The third most consistent failure mode was due to electrical component failure. These conditions can be improved by higher quality construction / design and component selection and limiting the thermal stresses in the motor. Section IV.F on page 38 discusses the use of a hybrid power module which can significantly impact the reliability of the power electronics. The module has a temperature sensitive thermistor, which can be used to dial-back the power flow during high ambient temperature operation.

D. *System Approach and Trade-off*

The proposed system solution for the intelligent integrated blower was arrived at via a method of system evaluation versus the weighted requirements from the customer. In the blower system, all of the components and technologies are integrated and a top-down method was used to evaluate the best candidates for a successful product.

A Trade-off study of potential motor-blower combinations with various magnetic materials and air flow control algorithms was initiated. Performance metrics were; efficiency, cost, air flow regulation repeatability, noise,

manufacturability, and reliability. The candidate solutions were evaluated during 2Q01-3Q01 when solid comparative data was accumulated for each of the evaluation indices. The overall numerical scores are shown in Table III-2.

The two best candidate topologies were the RI-RF-VAC design (rearward inclined blower - radial flux machine - moderate flux density rotor magnets) with or without pressure sensor for air flow control. The scoring was somewhat influenced by the experience of the GE team. The exclusion of the axial flux motor from the scoring was based on a determination that the GE team did not pose enough design and manufacturing experience to introduce the motor topology in the near term for HVAC applications. Further development of the axial flux motor in Y2002 will bring it into the technology portfolio. See section IV.C.2 on page 31 for a discussion of the axial flux machine. A detailed discussion of the trade-off tool evaluation methodology is given in the Appendix of this report.

Table III-2 List of best overall design candidates and scores

ECM 4.0 Design Candidate	Score
RI - RF - VAC no PS	11.08
RI - RF - VAC	10.62
RI - RF - Ferrite	10.08
RI - RF - NEO	9.85
FI - RF - NEO	9.31
FI - RF - Ferrite	9.23
RI - RF - VAC (inside out)	8.23
FI - RF - VAC	7.77
RI - RF - NEO (inside out)	7.46
FI - RF - NEO (inside out)	6.69
FI - RF - VAC (inside out)	5.92
FI - RF - Ferrite (inside out)	5.92

Key:

- Impeller type: RI = Rearward Inclined (high efficiency), FI = Forward Inclined (moderate efficiency)
- Motor type: RF = Radial flux, AF = Axial flux (eliminated due to near term doubt)
- Rotor Magnets: Ferrite = low flux, VAC = moderate flux, NEO = high flux (and high cost)
- Construction: inside out = exterior rotor (non conventional)
- Control: no PS = no pressure sensor (torque-speed control of air flow)

E. Conclusions

The HVAC industry is very concerned with the electrical power consumption of the blower due to the new SEER requirements. The industry can not instantly adopt new blower designs due to the time needed to safety qualify and retool. The GE team will concentrate on maximizing blower efficiency, air flow repeatability, diagnostic functions and system reliability through the development of compatible blower systems.

IV. Topical Technology Development

A. *Blower Development*

One of the most important objectives for the intelligent integrated blower project was the development of a high efficiency fan for the HVAC market. The project team studied three HVAC applications which were different in needs and approach; residential [furnaces](#), residential [air handlers](#), and commercial [constant volume terminal](#) units. Each of these applications has slightly different needs, but all could benefit from new blower and housing designs.

The end goal of the blower development addressed; static efficiency, audible noise, cost of manufacture, fan material, application, impact on motor design. The following sub-sections address first a characterization of the basic features of the three sampled applications. Then a discussion of the various forward and rearward inclined impellers is presented.

1. Residential HVAC Blowers

A residential 3-ton air handler and a 3-ton gas furnace blower assembly were evaluated and tested in GE's lab for baseline studies. A commercial constant volume termination unit was also studied and is discussed in the next sub-section.

Figure IV-1 Sample HVAC units; a) 3-ton air handler, b) gas furnace and c) variable volume terminal unit (yellow box marks blower location: not to scale)

Figure IV-1 shows the three units tested by GE. One will note that the blower is generally placed on the "cool" side of the heating or air conditioning elements. To specify the blower operating range, the internal airflow resistance for the three units at $0.567 \text{ m}^3/\text{sec}$ (1200 cfm) is shown in Table IV-1. One will note that the commercial terminal unit has significantly lower internal drop due to its minimalistic design and lack of any duct work.

Table IV-1 Typical internal airflow pressure rises for the targeted applications

UNIT	Nom. Air Flow	Pres. Rise	Max Air Flow	Pres. Rise
Air Handl.	0.567 m ³ /sec	151 Pa (0.60 inwg)	0.638 m ³ /sec	191 Pa (0.76 inwg)
Furnace	0.567 m ³ /sec	131 Pa (0.52 inwg)	0.638 m ³ /sec	166 Pa (0.66 inwg)
VAV	0.567 m ³ /sec	63 Pa (0.25 inwg)	0.638 m ³ /sec	71 Pa (0.28 inwg)
Remark	(1200cfm)		(1350 cfm)	

For residential applications, the external duct pressure drop varies from home to home and the air handler and furnace are designed to work with a wide external pressure range from 24.8 Pa (0.1 inwg) up to 149 Pa (0.6 inwg) at the 0.567 m³/sec airflow rate (1200 cfm). The typical external pressure drop is about 74~100 Pa (0.3~0.4 inwg). The resulting estimate of static pressure rise through across the blower is summarized in Table IV-2. One will note, that typical operation and the SEER rating test have significantly different operating conditions. The difference is seen in Table III-1 on page 9. The SEER test is without external duct work, hence the design of the blower must account for a wide range of operating conditions.

Table IV-2 Typical operating pressure rises for the units

UNIT	AIR FLOW	Min Pressure	Median Pressure	Max Pressure
Air Handler	0.567 m ³ /sec	176 Pa	238 Pa	300 Pa
Furnace	0.567 m ³ /sec	156 Pa	218 Pa	280 Pa

The next section evaluates the air flow performance and impeller characteristics of various blowers presently available in industry.

2. Blower Space Allocations

The various residential and commercial blower applications featured various amounts of space for the air handling equipment. The measured blower housing and total metal enclosure dimensions are recorded below in Table IV-3.

Table IV-3 Typical blower housing and total enclosure dimensions (cm)

Dimensions (cm)	Air Handler		Furnace		Commercial
	Housing	Enclosure	Housing	Enclosure	Hous/Encl
Length	45.7	50.8	54.6	62.2	49.8
Height	41.9	45.7	39.4	40.6	43.5
Width	28.6	48.3	24.4	44.5	40.6

Figure IV-2 Air handler unit photo; orange denotes maximum enclosure area, yellow denotes blower housing area, and blue denotes airflow path.

3. Blower Evaluation

The blowers used in the above three applications featured forward inclined (FI) blades and were of the "squirrel cage" type. This means that the impellers have an aggressive intake and exit angle on the blade, much like a spoon scooping ice cream. A non scientific study indicates that all the blowers in the residential air moving industry feature FI blower wheels.

Figure IV-3 Typical blower assemblies for ; a) air handler and b) furnace applications

a) Definitions of Blower Efficiencies

The evaluation of air moving equipment must consider both the potential and kinetic energy present in blower systems. The definition of dynamic and static efficiency is given below. *The evaluation of the blowers will always use the static efficiency at the SEER operating point*, which tends to be ~10% lower than the overall peak dynamic efficiency.

$$\text{Eff}_{\text{static}} = [(P_{\text{out}} - P_{\text{in}}) \cdot Q] / \text{Shaft Power} \quad (\text{accounts only for static pressure rise})$$

$$\text{Eff}_{\text{dynamic}} = [(P_{\text{out}} - P_{\text{in}}) \cdot Q + \frac{1}{2} \cdot (M/t) \cdot v^2] / \text{Shaft Power} \quad (\text{includes kinetic energy of exhaust air})$$

where P is pressure in units of Pascals, Q is flow in units of m³/sec, M is mass in units of kilograms, v is velocity in units of m/sec, t is time in units of seconds and Shaft Power in units of Watts.

The Seasonal Energy Efficiency Ratio (SEER) is how a heat pump or air conditioner is rated. The electrical consumption of the unit is monitored for a target cooling load. The SEER rating is then calculated by dividing the total BTU's of cooling by the electrical power consumption (Watts).

b) Forward vs. Rearward Inclined Blades

The FI blower wheels tend to require a relatively small blower housing and they operate at lower rotational speeds than rearward inclined (RI) blower wheels. RI blades are more like a butter knife smearing butter than the ice cream spoon analogy. The lower efficiency associated with the FI blower is due to the basic principle of the operation of this impeller:

- The airflow passing between the impellers reverses its direction as it travels from intake to exhaust.
- This reversing enables the FI impeller to generate large airflows at a lower speed.
- An abrupt change of the airflow direction causes poor flow patterns.
- Airflow separation is unavoidable in FI impellers and leads to significant airflow energy loss.
- The large amount of the energy gained by the airflow passing through the impeller is in the form of kinetic energy and then converted to static pressure in the housing. This conversion will also lead to extra energy losses.

The maximum static efficiency of a squirrel cage type FI blower is about ~70% and the typical achievable static efficiency is about ~60%, whereas the RI blower wheels can achieve 75% or better.

c) Definition of the Fan Curve

The "fan curve" is instrumental in characterizing blower wheel performance. The blower produces a certain pressure vs. flow curve when operated at a constant speed, and with variable flow resistance. The resulting curve can then be used to evaluate the operation of the blower.

Figure IV-4 shows the static efficiency performance and fan curves for five blower types; residential furnace, air handler, commercial terminal unit, industrial RI wheel and the proposed GE-1 RI wheel. The typical SEER rating point for heat pumps / air conditioning units is shown on the fan curve plot. The observed range of static blower efficiency is from 28% for the commercial terminal unit (which is not regulated by SEER ratings) to a potential 66% for the proposed GE-1 wheel.

Figure IV-4 Comparative performance data for five blowers; GE1, furnace, air handler, commercial, and industrial.

- all but the industrial RI blower were scaled to a typical SEER point for comparison

The given plots were for the given blower wheels operated in the housings in which they were installed. The GE-1 wheel was evaluated with a non-optimal housing so an additional 3~5% improvement in static efficiency is predicted.

The efficiency at the SEER rating point for the gas furnace application (which is not regulated by SEER ratings) shows that the blower is operating too "far down" on its fan curve, thus indicating that the blower is too small for the flow of 0.567 m³/sec (1200 CFM). Alternatively the airflow leaving the blower housing is too fast and that causes significant

kinetic energy loss. It is suggested the size of the blower be increased in order to achieve the optimal static efficiency. Figure IV-5 shows a CFD simulation of the FI furnace blower wheel with enclosure effects. There is actual reverse flow over a portion of the wheel, and generally poor air flow along the blades.

Oddly, an evaluation of the air handler blower revealed a similar problem (these are regulated by SEER ratings). The air handler blower wheel and housing was also undersized and it is predicted that the optimal FI blade design with larger housing could achieve ~55% static efficiency which would help with the SEER rating of the unit.

Figure IV-5 Residential FI blower CFD simulation showing poor air flow separation and reverse flow due to poor housing exit flow.

4. Industrial RI Blower

Rearward inclined impellers are generally more efficient than forward inclined impellers and an industrial RI blower was tested to help evaluate the state-of-the-art with respect to RI blade performance. A RI blower from an industrial manufacture, which was considered one of the best on the market, was obtained and evaluated. It was a single inlet wheel with dimension: 50 x 43.5 x 20.3 cm OD/ID/width (19 5/8", 17 1/8", 8") and the equivalent double inlet dimension is 40.6 cm (16") wide. It can be assumed that the airflow is doubled for a double inlet blower at the same pressure rise. The industrial blower was tested and found to feature a static efficiency of ~ 60% at the SEER rating point and a peak static efficiency of ~62%. This became the best case performance of any blower for the target 3-ton application. Figure IV-6 shows a drawing of the industrial eleven-blade RI wheel and also a photo, showing the single-inlet design with hub for a large shaft.

Figure IV-6 Industrial 11 blade RI blower wheel geometry and photo.

5. GE-1 Prototype Development

The 3-ton residential air handler unit became the test vehicle for the development of the target, high efficiency blower. The impeller design was chosen to be RI due to the superior static efficiency and the high scores in the trade-off study (see III.D System Approach and Trade-off). The major disadvantage to adopting a RI blower is the issue of airflow control via the motor's torque and speed information. See section IV.B Constant Air Flow Control for details on developing a low cost air flow control strategy for RI blowers.

The blower efficiency performance depends on the efficiency of the impeller and the losses due to the housing. Computational Fluid Dynamic (CFD) tools were used and models were developed to optimize airflow patterns through the careful design of the impeller blades and housing dimensions. The CFD modeling was done with two dimensional (2-D) analysis which ran more efficiently than three dimensional (3-D) models. The modeling tool was calibrated via a comparison to real test data on the industrial RI blower wheel in Figure IV-6. To see the modeling of the industrial RI blade, reference Figure IV-7 on page 19.

High efficiency blowers feature impellers which avoid serious airflow separation at the inlet and outlet of the blade. This is especially important in the target design case due to a relatively high air flow rate (1200 CFM) and relatively low static pressure < 250 Pa (< 1 inwg). Consequently, the dynamic head (kinetic energy density) is at the same level as the static pressure and significant loss of the dynamic head will lead to great loss of the efficiency. Therefore, special care must be taken at the place where flow velocity is high, especially, in the passage of the impeller where air flow is strongly interacted with the impeller blade and energy is transformed between them. An example case of poor air flow is demonstrated with the industrial RI blower wheel which up to now showed the best overall efficiency performances.

Figure IV-7 Comparison for the industrial and GE prototype RI blades; a) industrial 11 flat blade impeller, b) GE curved 20 blade impeller, c) air flow separation on the pressure side, d) smooth flow pattern.

Figure IV-7 shows how careful design of the impeller shape and number can significantly improve the airflow pattern and eliminate large scale air flow separation. Figure IV-7 c) reveals that the inside of the flow passage of the industrial RI impeller, airflow is separated and rolls up a big area of reverse flow. Some analytical modeling helped us understanding the basic phenomenon:

- The industrial RI wheel had too wide a gap between the blades that induce a over expansion of the airflow and the Coriolis force which directly cause the airflow stagnation near the pressure side.
- The wide blade separation also triggered the strong separation and induced large energy losses (6% AS estimated from the CFD simulation).

To eliminate the air flow separation, GE added more blades and made the blades somewhat curved. The calibrated 2-D CFD simulations were instrumental in evaluating several new designs which improved upon the static efficiency of the industrial RI blower. In the new GE designs, the blade gap has been narrowed made slightly curved, an example is shown in Figure IV-7.

Multiple RI blade geometries and separation gaps (number of impellers per wheel) were studied with the CFD tools. As the number of blades increased, the airflow separation could effectively be eliminated, but a second effect began to dominate the efficiency characteristic. The surface friction of the impeller has a negative effect on the efficiency and too many impeller blades will adversely effect the net static efficiency of the blower. See the Appendix for a plot of the net static efficiency of the RI wheel as a function of blade number per wheel for a certain shape impeller.

6. The design of prototype GE 1 impeller

The design of the GE-1 prototype RI blower wheel incorporated many design goals such as the tuning of the inlet shape, sizing the blower housing and adjusting the blower wheel's rotational speed.

Table IV-4 Key design parameters for the GE-1 prototype wheel.

Design Parameters	Dimensions (English units)
Impeller OD	29.2 cm
Impeller ID	18.8 cm
Passage Width	5.1 cm
Nominal speed	135 rad/sec
Blade exit angle	67.5 degree

Figure IV-8 The 16 blade GE-1 prototype impeller design and CFD airflow pattern.

The impeller blade was designed with a slight “S” shape to meet the entry and exit flow angles and to produce a smooth air flow expansion. It has been shown that air flow is smooth in the whole impeller passage and has a excellent flow pattern. The efficiency of the GE-1 impeller is estimated at ~92% through CFD simulation and experiment. An additional 8% of loss is associated with the intake region, and an additional 15% loss with the housing.

7. Housing Design

A major design restriction as discussed in the previous section is the size of the blower housing. To achieve optimal efficiency, the blower housing must be enlarged but that can increase the air handler size and cost and may result in an infeasible design. The converse argument is that the present size restrictions are constraining the static efficiency performance of the blower. Therefore, an optimal design is defined which balances the restrictions of the overall enclosure with that of housing size growth.

The housing allows the air flow to expand and converts the kinetic flow energy into static pressure energy. Ideally, the shape of the housing should be a “logarithmic spiral.” This is a direct consequence of the conservation of the circulation of air mass from the radially distributed impellers. Namely, the flow leaving the impeller will freely follow the logarithmic expansion passage. If the housing design follows this free stream line for a whole turn then the airflow is symmetric about the shaft axis and the blower wheel operates all impeller blades under the same condition. If this symmetry is broken then the impeller efficiency will be reduced. The housing width is usually twice the width of the impeller exit. Too wide a housing will increase the airflow expansion loss and too narrow a housing will make the expansion angle large and increase the size of the housing and the housing surface friction losses will increase. In the GE-1 blower case, the airflow leaving the impeller has an angle about 21 degrees, if the housing has double the width as the impeller exit, then the expansion angle is about half, or 10.5 degrees and that will lead to a large housing (about 76 cm in overall length) and is not acceptable. Therefore GE constrained the housing to ~3 times as wide as the impeller exit and the expansion angle was reduced to about 7 degrees.

Figure IV-9 Logarithmic housing expansion for optimal efficiency; a) spiral curve with expansion angle 7 degrees, b) 4-arc approximation for practical implementation.

The industry is reluctant to move away from the present array of housing and enclosure sizes. Much work has been done to (safety) qualify the present systems and the tooling costs for new housing designs is a few million dollars for some of our OEM customers. The efficiency disadvantages to not adopting new housing standards is great. Below is a summary of the penalties associated with not adopting the appropriate sized housing or wheels for the target 3-ton application. The least redesign and application of only one RI type wheel to a wide range of present blowers would only achieve modest efficiency gains. The best solution is to redesign the total air handling system and create many blower wheels for each air flow range.

Table IV-5 Static efficiency benefits to applying new housing size standards for air handler blowers

Proposed change to the system	Static Efficiency
Increase housing dimension to ideal size (larger unit enclosure)	75%
Increase housing dimension to the maximum allowed by present enclosure	65~70%
Use present housing size but with shape redesign	60~64%
Use conventional housing and add an adapter inlet ring for GE-1 type wheel	55~60%
Use only one RI blower wheel to fit a wide range of present day systems	mid 50's%

8. GE 1 Impeller Performance

The combination of the GE-1 prototype impeller wheel with the compromised logarithmic expansion housing (which was sized to fit a standard 3-ton air handler) resulted in a high efficiency blower with unprecedented performance. The resulting blower was targeted at the 3-ton application and enclosure size constraints. A summary of the efficiency performance and fan curve are shown in Figure IV-10. The target SEER rating point is shown on the graph and the static efficiency is ~64%. The blower's peak static efficiency was measured at 70% and the peak dynamic efficiency was ~80%. Table IV-6 shows the median static efficiency is around 68%, which is realizable in a typical residential home. The peak efficiency of 70% can be realized in a poorly designed residential application with high duct resistance and high static pressure. This first prototype is the starting point for a series of empirical prototype blower wheels which will be adjusted with the help of CFD tools.

Figure IV-10 Efficiencies and fan curve for the GE-1 prototype wheel at 135.1 rad/sec (1290 rpm) using a near-optimal housing which fits the target system enclosure.

Table IV-6 The GE-1 static efficiency performance for the target application flow of 0.567 m³/sec

GE-1 Performance	Min Pressure	Median Pressure	Max Pressure
Static Pressure	176 Pa	238 Pa	300 Pa
Static Efficiency	64%	68%	70%
Speed rad/sec	135.1	144.5	158.1

9. Optimal Housing Size

A study of the optimal blower wheel diameter and resulting enclosure size was under taken. The GE team discerned that the present air handler could achieve even better performance if the space inside the metal enclosure could be utilized. By using the fan-law scaling rules, and detailed information about the wheel and

enclosure the below efficiency vs. wheel diameter plot was generated. Figure IV-11 shows an estimate of the benefit which should be realizable if the industry could adopt larger enclosure sizes.

Figure IV-11 Static blower efficiency vs. overall wheel diameter for the target 3-ton application.

The commercial terminal unit features much lower static pressures than in residential blower applications. the typical airflow is about 0.520 m3/sec (1100 CFM) and pressure rise required is about 132 Pa (0.55 inwg) that includes the external pressure drop 62 Pa (0.25 inwg) and internal pressure drop 74 Pa (0.3 inwg). The baseline FI blower static efficiency at this operating condition is estimated at only about 28% and for the GE-1 type RI blower, only about 45% static efficiency. The restriction of the dimensions in the commercial terminal unit is much more severe than for the residential units, thus making high efficiency blowers less realizable. GE may attempt an axial fan design in place of the radial blower wheels for this challenging application.

10. Noise Reduction

The noise level of the blower is determined by the fluctuation of the air pressure, especially on the surface of the impeller. This is due to the more efficient radiation source of solid impeller than the airflow pattern (dipole source vs. quadruple source). It is true that the relative velocity in the impeller air flow passage of an RI blower is significantly higher than in a FI blower and higher velocity airflows tend to generate higher pressure fluctuation. Alternately the flow pattern in the RI impeller is much better than in a FI blower. The engineering world has commonly believed that FI blowers are more quiet than RI blowers, but a back-to-back evaluation of the air handler and furnace FI blower wheels and the GE-1 prototyped revealed the opposite. Table IV-7 shows the measured sound pressure levels for the three blowers at the SEER rating point.

Table IV-7 Sound pressure level at 1 meter and 0.567 m³/sec (1200 CFM)

Noise comparison	GE-1	FURNACE	INDUSTRIAL
Pressure rise	~170 Pa	176 Pa	176 Pa
SPL	60 dBA	62 dBA	63 dBA

The SPL only measures the RMS value of the full noise spectra. The full spectrum of noise was documented and the RI blower did not possess any significant tone noises which would cause problems. The clearances in the housing can be adjusted to avoid the excitation of blade tip frequency noises and GE is confident that the RI blower is not a “noisy” system.

11. Conclusion

The design of the GE-1 prototype wheel incorporated a novel "S" shaped RI curved impeller design to achieve excellent airflow patterns and impeller efficiencies. A near ideal housing design was proposed which tried to maximize the space available in the target air handler application. The restrictions of the blower enclosure did adversely effect the peak efficiency of the system, and the effect of the enclosure size on efficiency performance was documented.

B. Constant Air Flow Control

The control of air flow in a HVAC unit is critical to efficiency and comfort. Traditional forward inclined blower wheels feature a unique set of torque-speed relationships which makes it possible to control air flow from the estimated motor torque and measured speed. This method of control is not possible with rearward inclined blower wheels due to the arched efficiency characteristic which tends to give nearly overlapping torque-speed relations. Figure IV-12 shows how the residential FI and GE-1 prototype RI blades differ.

Figure IV-12 Sample speed-torque relationships for both forward and rearward inclined blade blowers.

An analysis of the relation between the airflow velocity and torque is instrumental in understanding the essential physics of the problem. The FI and RI blades impel the air differently and thus result in different torque-speed characteristics. The torque needed to sustain a certain airflow is related to the absolute velocity of the air as it leaves the impeller wheel. The absolute air velocity is the vector sum of the wheel's impeller tip velocity and the impeller air's exit velocity. The FI blade blower tends to produce an absolute velocity (torque) which is almost proportional to the air flow and rotational speed as is seen in Figure IV-12 and explained in Figure IV-13. The RI blower tends to have a set of nearly overlapping torque-speed relationships due to the poor alignment of the velocities which produce a near constant absolute air velocity.

Figure IV-13 Impeller exit air velocities for both RI and FI blades resulting in torque vs. speed variances.

Figure IV-13 shows the velocity triangle at the impeller tips of FI and RI blowers. In the RI case, we see, the contribution of the relative velocity to the absolute velocity is subtle: increasing the relative exit velocity will reduce the circumferential component of the absolute velocity and increase the radial component of the absolute velocity. In the FI blower, increase of the relative velocity will increase both circumferential component and radial component of the absolute velocity. *The torque applied to the shaft is roughly proportional to the product of the circumferential component (pressure) and radial component (flow) of the absolute velocity (under constant speed operation).* The following conclusions result from the above study:

1. FI blower: both radial and circumferential components of the absolute velocity increase with airflow, hence the torque will be an increasing function of the airflow.
2. RI blower: the behavior depends on the exit angle. If the exit angle is small, the reduction of the circumferential absolute velocity causes a decreasing torque as a function of airflow. If exit angle is close to 90 degrees, the circumferential absolute velocity is slightly reduced as airflow increases and the torque requirement increases linearly.

The turning point of the controllability of airflow with the motor's torque and speed information is a critical impeller exit angle. A model of the above paradigm shown in Figure IV-13 was made and the calculated blade torque's vs. airflow was plotted in Figure IV-14. Two actual sets of measurements for exits angles of 67 and 150 degrees are included to show the basic efficacy of the approach.

Figure IV-14 Torque vs. air flow relations for RI impellers with different exit angles.

In summary:

1. The blower airflow can not be resolved nor controlled by the motor's torque and speed information if the RI impeller exit angle is less than 50 degrees (for the target application).
2. The airflow controllability is significantly improved as the exit angle is increased to 90 degrees and the torque vs. airflow relationship is close to a linear function.
3. The current GE-1 wheel design features an exit angle of ~67 degrees and the airflow can be pseudo controlled by the torque-speed method but it requires high accuracy torque estimation in order to achieve the same accuracy of the airflow (as compared with the FI blower).

It should be noted that the actual torque-speed relationship follows the tendency of the above calculation qualitatively (or partially quantitatively) because the simplified analytic modeling assumes a constant slip factor. When the exit angle increases, the airflow tends to deflect more from the blade direction and that makes the above calculation over estimate the increase of the torque response to the airflow increase at large exit angles. This error is shown by the actual 150 degree exit angle curve is well below the calculated 130 degree curve.

The torque accuracy needed to regulate airflow to within 50CFM of target for the GE-1 blower wheel is presented. The plot of the torque-speed curves for various constant airflows (900 - 1200 cfm) are shown in Figure IV-15. The designed fan speed range is from 1290 rpm to 1700 rpm this covers the target operating range from the SEER rating point to the maximum loading point (see Table III-1). The motor speed was assumed to feature less than 1% error.

Table IV-8 Required torque estimation accuracy to meet 50 CFM error (or less) for the GE-1 wheel.

Operating Speed	Accuracy Required	% of Full Scale (2-Nm)
-----------------	-------------------	------------------------

1300 RPM	0.015 Nm	0.75 %
1500 RPM	0.025 Nm	1.125 %
1700 RPM	0.04 Nm	2 %

Figure IV-15 The GE-1 wheel's speed vs. torque relationship for various airflow levels.

The RI blower does have a unique set of pressure-speed curves. GE has developed a general approach of constant airflow control based on pressure information, and the sensitivity analysis is summarized below. Figure IV-16 shows the airflow of the GE-1 blower can be uniquely identified by the known motor speed and measured pressure rise across the blower. Table IV-9 gives the required accuracy of the pressure sensor for the desired air flow error of 50 CFM or less. A 4% error is acceptable for many commercially available piezoresistive pressure sensors.

Table IV-9 Required pressure sensor accuracy to meet 50 CFM error (or less) for the GE-1 wheel.

Operating Speed	Accuracy Required	% of Full Scale (1.2 inwg)
~1700 RPM	0.05 inwg	4%

Figure IV-16 The GE-1's wheel's speed vs. pressure rise relationship for various airflow levels.

C. Motor Development

The integrated intelligent blower required new motor designs which were tuned to the characteristics of the blower wheel, and to the restrictions of the blower housing and intake mouth. The thermodynamic cooling properties and temperature constraints also effected the proposed radial and axial flux motor designs. The designs for the two motor families (radial and axial) were both permanent magnet machines being driven by a controlled inverter. GE fine tuned the motor design tools so effective design comparisons for the blower application could be quickly developed. The design tools included mechanical dimensions, electro-magnetic model, equivalent electrical model, flux density modeling, and resulting torque performances.

Figure IV-17 Motor design tools used for the 3-phase PMDC motor iterations (radial flux).

Axial and radial flux motor designs were developed for both the rear-inclined blower and the forward-inclined blower. These designs are summarized in Table IV-10, Table IV-11, and Table IV-12. Designs were produced for three candidate rotor magnet materials: standard ferrite, a good performance compression bonded neodymium offered by the Arnold Corporation, and a low cost bonded neodymium offered by the VAC Corporation. For the radial flux machines; exterior rotor (inside out) designs were not considered because of the difficulty of cooling the interior windings. See thermal design section, Table IV-14.

1. Radial Flux Machine Design

All designs were produced for the same set of constraints. The outside diameter was set at 10.2 cm (4.0 inches), small enough not to impede the blower air flow. The current density and peak flux density, at the maximum load, were constrained to values consistent with current ECM design practice.

The FI blower is both less efficient and rotates at a slower speed than the RI blower, so the motor's torque requirement is considerably larger. Motor sizing is strongly dependent on this torque rating, so the FI designs are physically larger and heavier as is shown in Table IV-10 and Table IV-11.

The strongest rotor magnet material, the Arnold material, permits designs that are the most physically compact and the most efficient. The weakest magnet is the standard ferrite, so its designs are physically larger and less efficient. The VAC neodymium is weaker than the Arnold neodymium, not so much because of the lower residual flux density, B_r , but because of the greater recoil permeability. This greater recoil permeability requires a thicker magnet, and it reduces the margin against demagnetization.

Sizing studies indicate that better motor efficiency can be obtained by increasing the amount of magnetic material, although a situation of diminishing returns is observed, where designing with additional material gives an increase in iron loss that offsets a decrease in copper loss. The designs illustrated in the tables exhibit acceptable efficiencies.

The motor design for a rear-inclined blower, using the VAC bonded ring magnet material, was selected for prototyping. This design is only marginally more expensive ($< \$1$), from the material cost aspect, than the ferrite design, and the bonded ring magnet approach is preferred for ease and consistency of manufacture.

A magnetizing fixture has also been specially designed as shown in Figure IV-18. Obtaining full magnetization of neodymium has been found to be challenging, so the magnetizer design incorporates techniques to maximize performance. The magnetizer lamination material will be a special cobalt-iron alloy that permits the highest flux density. The magnetizer lamination material will also be very thin, .006", so to limit internal eddy currents that can diminish the magnetizing effort. The design of the magnetizer windings are coordinated to the magnetizer discharge circuit to permit the greatest magnetizing force with the smallest winding heating.

Table IV-10 Radial flux motor designs for the RI Blower

Motor Configuration	Rightside Out	Rightside Out	Rightside Out
Magnet Type	Arnold 3203	VAC low-cost	Ferrite
Magnet Br, kG	6.60	6.01	4.00
Magnet Recoil Permeability	1.18	2.20	1.06
Air gap Thickness, in.	0.010	0.010	0.025
Magnet Thickness, in.	0.100	0.125	0.250
Motor OD, in.	4.000	4.000	4.000
Motor Stack Length, in.	1.250	1.500	1.750
Total E.M. Weight, lbs.	3.54	4.14	4.68
Magnet Weight, lbs.	0.17	0.24	0.57
Copper Weight, lbs.	0.77	0.88	1.06
Iron Weight, lbs.	2.59	3.02	3.06
Iron Weight, incl. scrap, lbs.	4.26	5.12	5.97
Motor Efficiency at Max, %	86.1	84.7	83.6
Motor Efficiency at SEER, %	87.0	86.0	85.0
Copper Loss at Max, watts	33.9	37.8	46.1
Copper Loss at SEER, watts	17.6	19.2	23.6
Iron Loss at Max, watts	12.3	14.2	10.4
Iron Loss at SEER, watts	7.9	8.4	6.4
Current Density at Max, amps/in ²	4290	4250	4270
Current Density at SEER, amps/in ²	3090	3020	3060
Peak Flux Density at Max, Tesla	1.94	1.94	1.92
Peak Flux Density at SEER, Tesla	1.76	1.70	1.71
Power Input at Max, watts	359	371	378
Power Input at SEER, watts	208	212	216

Table IV-11 Radial flux motor designs for the FI Blower

Motor Configuration	Rightside Out	Rightside Out	Rightside Out
Magnet Type	Arnold 3203	VAC Low-Cost	Ferrite
Magnet Br, kG	6.60	6.01	4.00
Magnet Recoil Permeability	1.18	2.20	1.06
Airgap Thickness, in.	0.010	0.010	0.025
Magnet Thickness, in.	0.100	0.125	0.250
Motor O.D., in.	4.000	4.000	4.000
Motor Stack Length, in.	2.050	2.450	2.900
Total E.M. Weight, lbs	5.59	6.59	7.39
Magnet Weight, lbs	0.28	0.39	0.88
Copper Weight, lbs	1.07	1.24	1.65
Iron Weight, lbs	4.24	4.91	4.95
Iron Weight, incl. scrap, lbs	6.99	8.36	9.89
Motor Efficiency at Max, %	85.3	83.6	81.3
Motor Efficiency at SEER, %	86.0	84.6	82.4
Copper Loss at Max, watts	46.3	52.3	66.8
Copper Loss at SEER, watts	24.7	27.7	35.6
Iron Loss at Max, watts	10.5	11.8	8.8
Iron Loss at SEER, watts	6.7	7.2	5.5
Current Density at Max, amps/in ²	4270	4220	4240
Current Density at SEER, amps/in ²	3120	3070	3090
Peak Flux Density at Max, Tesla	1.94	1.93	1.94
Peak Flux Density at SEER, Tesla	1.78	1.73	1.75
Power Input at Max, watts	416	429	441
Power Input at SEER, watts	239	244	251

Figure IV-18 Magnetic outline of ECM 4.0 motor and magnetizing fixture (approx. to scale).

2. Axial Flux Machine Design

Emerging moldable soft magnetic materials are enabling new design topologies for electrical machines (e.g. axial flux machines). A high efficiency axial flux machine featuring moldable soft materials was designed and is currently being built employing commercially available moldable iron material from Mii Corporation. An exploded assembly view of this machine is shown in Figure IV-19.

Figure IV-19 Exploded View of Axial Flux Motor Technology Demonstrator

There are many axial flux topologies in the market place. Most computer disk drives, for example use axial flux motors. These motors typically lack an iron core to conduct flux and are therefore inefficient. A significant portion of the flux produced by the windings is leakage flux that is not utilized for torque production. Since the disk drive motors are small in terms of power, this is not important.

For motors in the fractional horsepower range for the HVAC application, efficiency is important and a ferromagnetic core that minimizes leakage flux is required. This core material must not only conduct magnetic flux, but also have low eddy current and hysteresis losses. Conventional radial flux machines use laminated steel to serve this purpose. In an axial flux configuration, a trapezoidal pole piece shape is required as shown in Figure IV-19. Building the

required structure from laminated sheets is awkward and costly. Moldable iron materials, however, provide an economical way of creating the required structure.

Axial flux motor topologies offer a radically different way to design and manufacture electrical machines. There are many potential advantages of an axial flux machine utilizing ferromagnetic pole pieces compared to more traditional motor topologies. These advantages include:

- Optimized tooth gap to reduce leakage flux
- No end turns and their associated losses
- More efficient mass production of winding elements on bobbins
- Copper fill factors improved by a factor of more than two, reducing motor size
- Top down assembly for throughput optimization
- Simplified integration with the fan blower

In a radial flux machine, the space between adjacent teeth is determined by the clearance required to insert the wires in the slots. This sacrifices the area available for flux to be transmitted and increases the reluctance of the magnetic circuit. In the axial flux machine under consideration, this constraint is removed. The gap between the adjacent trapezoidal pole pieces can be optimized to maximize the amount of flux produced by the coils that enters the air gap. There is a limit as to how close the teeth can be together, however, because eventually the flux will short from one pole to the next without entering the air gap. Figure IV-20 presents the normalized armature current flux in the air gap as a function of the spacing between adjacent teeth. Since torque is roughly proportional to air gap flux produced by the current, in excess of 10% more torque can be generated for the same current in an axial flux design with trapezoidal pole shoes compared with a radial design due to reduction in leakage flux.

Figure IV-20 Normalized armature current flux in air gap as a function of the spacing between adjacent teeth.

Axial flux motor technology made with moldable iron materials presents a totally different way to manufacture electrical machines. All iron components can be molded in a press. There are no laminations to cut, compress, stack and join. Coils with high copper fill factors are created on bobbins with simple shapes on automated precision winding equipment. Using precision winding copper fill factors in excess of 60% can be achieved on the bobbins. This is compared to a fill factor of approximately 36.5% for the ECM 2.3 product. The high copper fill factor reduces component size and increases the thermal conductivity of the coil. The coils can therefore run cooler with lower resistive losses. The wound bobbin coils are mounted on a circuit board containing the required interconnection of the coils, and electronics. The assembly of the stator occurs in a top down fashion starting with the pole pieces. After the stator is completed, the rotor is mated to the stator assembly. The characteristics of the axial flux motor under construction and those of the current product and the radial flux designs are shown in Table IV-12.

Table IV-12 Comparison of present product with the new RI blower motor designs.

	ECM 2.3	Axial flux	Radial Flux	Units
	(1/2 HP)	(3/8 HP)	(3/8 HP)	
# pole pairs	6	3	4	
Speed at 100hz	1000	2000	1500	RPM
Magnet material	ferrite	ferrite	bonded neo	
Magnet thickness	0.307	0.25	0.125	Inch
air gap	0.025	0.025	0.01	Inch
Rotor moment of inertia	1.086	4.615	0.64	lb*in^2
# turns per coil	128	108	80	
per phase inductance	45.7	14.8	15.3	mH
Rated power	375	288	288	Watts
Speed at max rated power	1050	1750	1750	Rpm
Maximum current draw	5	3.3	3.3	Amp
Copper losses (65C) max current	72	35	38	Watts
Stator iron losses max current	15	16	14	Watts
Length*	4.5	3.28	4.2	Inch
Stator diameter	5.48	4.56	4	Inch
air gap flux	3.8	3.8	3.7	kG
Peak iron flux	18	10	19	kG

*Note: the axial flux technology demonstrator was oversized by approximately 20% in this dimension to account for uncertainties associated with a new design.

Axial flux motor technology will be demonstrated with this prototype. This prototype will also be used to validate the axial flux design tool now under development.

3. High Performance Bonded Magnets

The current ECM product employs sintered ferrite magnet arcs that are glued to the rotor core. Recent developments in bonded Rare Earth (neodymium) – Iron - Boron magnetic materials have made these new materials an attractive economic alternative compared with ferrite materials. Many different companies are developing rare Earth-Iron-Boron based magnetic materials, and the cost of these materials continues to decrease. A survey of available alloys suitable for powders that would be used in a bonded magnet for ECM motors was conducted. These alloys include alloys from Vac Corporation, Magnequench, and even an alloy developed by GE-CRD for MRI components.

A table in the appendix presents a costs of different commercially available rotor magnet options for the ECM 2.3 product. While the bonded approach is still slightly more expensive, the bonded magnets have several advantages in the ECM motors as compared with glued on ferrite arcs. These advantages include:

Magnetic material can be injection molded directly onto the rotor iron, eliminating several assembly steps including a bake out after the arcs are glued onto the sleeve.

Pole to pole field uniformity and symmetry, both intra-rotor and rotor-to-rotor is improved

- No disturbance of magnet flux caused by magnet gaps.
- Rotor magnets cannot delaminate from rotor iron
- Rotor material is uniformly distributed; balance is not an issue.

To demonstrate this bonded magnet technology, bonded magnets were designed and fabricated for the ECM 2.3 stator. Because of the large amount of experience with ferrite magnet rotors in this product, the properties of the bonded “Lean Neo”® magnet rotors are compared to those of typical ferrite magnet rotors.

Figure IV-21 Flux density distribution of poles pairs in a) “Lean Neo” bonded magnet and b) ferrite magnet rotor with 0.062” air gap.

D. Thermal Design Considerations

The design of the integrated intelligent blower required a system approach and concurrent engineering between the many disciplines to achieve an effective, high performance product. The study of the heat transfer in the motor assembly and evaluating the requirement for air movement inside the motor ...

1. Thermal Design Evaluation

Problem statement and finite element analysis (FEA) consideration for radial flux motor design. The objective to conduct thermal analysis on motor design are:

- The design requirements: at max load and 60°C ambient temperature
- the copper winding temperature is <130°C, and the magnet temperature <100°C.
- Compare various motor designs
- Examine feasibility of exterior rotor (inside out) radial flux design
- Determine the amounts of internal motor cooling

Because of the geometry and materials symmetry, in the thermal finite element analysis, only a sliver of the whole motor needs to be modeled. The FEA was conducted by using ANSYS 5.7 software. An example of the modeling is shown in Figure IV-27. The modeling and simulations were calibrated with the use of a "dummy" motor and injected heat source.

Motor and material layers along the "x" direction are from in to out: rotor iron, magnet, air gap between rotor and stator, tooth and yoke of stator, coil windings and outside shell. Material properties are listed in Table IV-13. The FEA results of the present ECM 2.3 product were used to calibrate the testing results.

Table IV-13 Conductivity for different materials in motor (W/(m K))

	Kxx	Kyy	Kzz
Stator and rotor iron	25	25	0.5
Outside steel shell		50	
Magnet	4.5 (ECM2.3); 2.0 (ECM4.0, neo); 1.8 (ECM4.0, Vac); 4.0 (ECM4.0, ferrite)		
Air		0.03	
Coil winding (40% fill)	0.9	0.9	136.8

Subscripts x and y means radio and peripheral directions; z means axial direction.

Figure IV-22 1/24 sliver of an a) ECM 2.3 motor and b) temperature distribution results for the structure

2. Thermal Analysis for the Intelligent Blower Motor

The interior rotor radial flux ferrite motor was analyzed. First a CFD analysis of the airflow outside the motor body was administered. Figure IV-23 a) shows the air flow velocity magnitude outside the shell by assuming $0.567 \text{ m}^3/\text{sec}$ (1200 CFM) air flow rate in the proposed GE-1 blower and housing. Next Figure IV-23 b) shows the thermal boundary layer by a temperature contour plot when flowing air temperature is 60°C . Then the temperature on the shell was used as a boundary condition for the thermal conduction analysis within motor. The results are shown in Figure IV-23 c) and the interior rotor design was determined to be a reliable design from a thermal point of view.

Figure IV-23 a) air velocity , b) thermal boundary layer, c) temperature inside motor.

The comparisons of the interior and exterior motor designs are summarized in Table IV-14. In all cases the surface temperature was assumed to be 80°C for all cases at the largest load capacity. Thermal impedance (resistance), R , is defined as $\Delta T/q$, where ΔT is the temperature difference between the highest temperature within motor and the average temperature on the outside surface of the motor shell; q is the total heat generated inside motor in units of Watts.

Table IV-14 Thermal performance comparison of radial flux ECM motors

	Q (W)	ΔT (C)	R (C/W)	$T_{\max,coil}$	$T_{\max,magne}$
ECM 2.3*	136.0	13.8	0.1	93.8	91.4
Radial Flux (interior rotor)					
Ferrite rotor magnet	63.0	13.0	0.21	93	90
Vac rotor magnet	57.4	11.0	0.19	91	88
Neo rotor magnet	50.2	12.0	0.24	92	89
Radial Flux (exterior rotor)					
Neo rotor magnet	47.3	122.5	2.59	202.5	98

Results show that rotor magnet materials do not have significant effect on thermal performance, and the exterior rotor (inside out) design is not feasible, unless an internal air flow is forced to flow. At the largest load capacity (about 1200 CFM), the airflow is turbulent outside motor, the heat transfer coefficient between air and motor can be assumed to be 200 (W/m²°C). The required internal airflow rate for cooling the interior placed windings is shown in Figure IV-24. Achieving an internal air flow of 4 to 5 CFM is challenging in a low-cost fractional hp motor design.

Figure IV-24 Internal air flow requirement for stator cooling of an exterior rotor radial flux machine.

E. Motor Controls

There are many requirements for the successful demonstration of a viable motor control system that serves the HVAC blower application. These requirements are summarized below;

- Low-cost digital controller
- Digital communications -- to encouraging HVAC system diagnostics and ease of programmability
- Torque control (tight tolerance) -- repeatable motor torque regulation and estimation enables precise airflow control and permits predictable system efficiency and SEER ratings.
- Self-calibration and diagnostics – demonstrate fast motor torque control calibration techniques that do not require precise torque transducers or calibration on a factory dynamometer, and reduce number of false field rejects to help customer perception of technology as robust.

Precise and repeatable motor torque estimation is very important to the blower product which bases its air flow estimates on the motor's torque and speed conditions of the machine. Generally the speed information from the motor can be resolved to <1% error, but the ability to estimate torque has generally returned 5% errors.

1. DSP based Controls

The availability of low-cost digital signal processors (DSP) optimized for control of motor drives has enabled high performance cost effective ECM drive control solutions. The high level of on chip integration and powerful computational abilities of the DSP make it possible to combine the functions of the motor control ASIC and the application control micro-processor in the ECM 2.3 product into a single DSP chip while reducing the costs of the key control integrated circuits by a factor of 2¹. Since development of DSP based control firmware represents a significant investment and re-use of DSP control software is a key objective, a tradeoff analysis to compare the motor control DSP offerings from the leading manufactures was performed. The tradeoff analysis showed that the Texas Instruments (TI) TMS320LC2402A DSP was a clear choice². The TI C240xA family of DSPs are 3rd generation designs operate at 40 MIPS over a temperature range of – 40 to 85 degrees C and provide code compatibility among earlier as well as future planned devices. A block diagram of the flash memory version of the DSP is shown in Figure IV-31.

Details of the controller solution are given in the Appendix along with torque ripple minimization techniques and air flow control code implementation.

Figure IV-25 DSP architecture.

¹ TI DSP pitch with price quotations

² Tradeoff analysis of Motor control DSPs

F. Power Electronics

The redesign of the power processing stage for the intelligent integrated blower featured many focus areas. All of the focus areas were vital to the success of the final product.

Focus Areas:

- hybrid power module
- EMI tests
- thermal evaluation
- gate drive circuitry
- inverter topology study with integral power factor correction.

1. Hybrid Power Module

The current ECM product has six discrete IGBTs and six discrete gate drive circuits in the inverter stage. To modernize the circuit and to achieve higher levels of reliability, a hybrid power module and integrated gate drive circuits were studied. The end-goal was to dramatically improve the circuit reliability, reduce the assembly cost and to reduce the circuit footprint for the smaller motor dimensions. Although hybrid power modules have been on the market for many years, the cost of fractional hp units are finally approaching reasonable levels as compared with the cost of discrete components and insulated gate bi-polar transistors (IGBT's). Figure IV-26 shows the conventional approach with discrete switches and gate drive circuits, as well as a single board approach with bottom side mounted power hybrid module. The use of high voltage integrated circuits (HVIC's) in the gate drive circuit is also advantageous. These IC's can also impact the reliability and footprint of the circuits.

Figure IV-26 Packaging solutions for the a) ECM 2.3 discrete component approach (two boards); and b) single board with bottom side hybrid power module.

To identify the hybrid power module for ECM applications, several options have been listed and their specs are detailed in the [hybrid power module select specifications](#). Several approaches to hybridizing were considered.

- Three Phase IGBT & Gate Drive Power Hybrid Module. This is the most desired approach. With this approach, the IGBT bridge and gate drive units are integrated into one module.
- Three Phase IGBT Module + Gate Driver IC. This is also a desired approach. A three phase IGBT module and gate drive ICs are composed of a motor drive inverter. Compared to option 1, there are more parts. However it is still very easy to assemble.
- Discrete IGBTs + Gate Driver IC. Compared to the above approaches, this approach needs more assembly work.
- Discrete IGBTs + Discrete Gate Driver. This is the conventional approach and currently is used for ECM 2.3. The discrete gate drive consumes much board space as compared to the HVIC. In addition, discrete IGBTs have lower reliability and more assembly work.

All the options listed above are available on the commercial market from various manufactures. By working with several manufactures including Hitachi, IR, EUPEC ..., EUPEC hybrid module is considered to be the most appropriate component in ECM application. It includes a three phase IGBT bridge, a diode rectifier and a thermistor. The cost comparison is listed in table 4.1. It can be seen that the cost of EUPEC module is very close to the cost of discrete approach. Also EUPEC module [FB10R06K14](#) datasheet is linked.

G. HVAC Diagnostics

The use of an ECM motor and high performance fan, tightly integrated with DSP control capability, offers the potential for very effective on-line self diagnostics in residential and commercial HVAC applications. Detection of faults or degraded performance within the Motor, Fan, or Electronics as well as within the larger HVAC system (e.g. flow restrictions, mispositioned dampers) could play an important role in maintaining peak system efficiency, reliability and comfort. The benefits of a structured and modular DSP based diagnostic function include:

- Motor Health and Readiness Tests;
- Continual Fault Detection and Isolation for major faults;
- Detailed Fault Detection and Isolation Support Tool during servicing,
- Factory Test Tool.

The objectives of the diagnostics task were 1) determine the effects of HVAC faults and their observability within the Motor/Fan control, and 2) develop diagnostic concepts for detection of these faults. The approach taken in pursuit of these objectives involved the development of both a higher level System model and a more detailed Motor/Electronics/Fan Subsystem model which were used to simulate faults and to analyze the resulting system behavior.

Following this, the approach was to develop model based diagnostic concepts, employing simplified System and Motor/Electronics/Fan models suitable for on-line DSP implementation, which would accurately predict the system output when driven with the same control signals driving the actual Blower. Discrepancies between the predicted and actual sensed outputs would be used for the fault detection. In this way, the Motor/Electronics could potentially provide indication not only of internal Motor, Electronics, or Fan faults but also of System level faults, if the appropriate System/Blower interactions were included in the diagnostic model. It is envisioned that the diagnostic models would be constructed in modular form so that coverage could be added or removed as desired. The Motor/Electronics/Fan model would act as the core model with Load or System Models of varying detail being coupled in as required.

1. Results of Motor/Electronics/Fan Subsystem Model

A candidate HVAC diagnostics concept was developed based on a simplified averaged model of ECM Motor, Electronics and Fan behavior. A block diagram representation is shown in the figure below. The diagnostics routine is driven by three inputs from the DSP Motor Controller: the commutation switching event, the PWM duty cycle, and the speed estimate. For each Commutation interval, these inputs are averaged and used by the diagnostic model to predict the system state. The state vector consists of the individual motor phase currents, the rotor speed, and the rotor mechanical angle. The predicted state vector is transformed into equivalent measurement quantities that can be directly compared with actual measurements from the DSP Motor Controller. These measurements include the DC Link Current, the DC Link Voltage, the Phase Voltages, Motor Speed Estimate, and the Motor Torque Estimate. Differences between the actual measurements and the predicted measurements provide the information for system fault detection.

A detailed ECM Motor, Electronic and Fan simulation was also developed for the purpose of generating representative measurement data to the diagnostic routine. Simulation data representing normal and faulted systems was used as input to the diagnostic routine to assess the ability to detect particular faults. Actual data from a prototype ECM motor in a laboratory test setup (unfaulted operation only) was used to verify the simulation output, and was also used to drive the diagnostic model to verify performance of the averaged model technique. Details of the ECM simulation and the diagnostic model development are contained in the Addendum.

The Table below lists the various faults that were simulated and shows how these faults are observable in the diagnostic model measurement predictions. The Measurement Difference columns (DC Link Current, Motor Speed, and Motor Torque) show the magnitude of the discrepancy between the actual measurements and the predictions from the diagnostic model when the fault is present.

Table IV-15 Fault Detection Capability

Fault Type	Operational Fault Conditions	Fault Observability Level (Model Difference with Fault Present)		
		DC Link Current	Motor Speed	Motor Torque
Winding Failure	Open Circuit in Phase A Winding	25 A	> 1000 RPM	40 Nm
Demagnetized Rotor	20% Reduction in Back EMF Constant	1.1 A	100 RPM	1 Nm
Increased Bearing Friction	Friction Torque Increases to 10% of Commanded Torque	0 A	40 RPM	0.1 Nm
IGBT Failed Open	Open Circuit in Phase A Upper IGBT Switch	24 A	> 1000 RPM	40 Nm
Load Torque Anomaly	20% Increase in Fan Torque	0 A	60 RPM	0.15 Nm

The next figure shows the current, speed and torque response plots for the demagnetized rotor fault. At a time of 0.3 seconds, the back EMF amplitude is reduced by 20% to simulate the fault. As seen in the responses, the diagnostic model predictions for DC Link Current, Motor Speed, and Motor Torque track the measured quantities quite well prior to the fault, but deviate significantly after the fault occurs. These deviations provide the information for detecting a faulted condition. There are many approaches to processing the information, including simple threshold checking and Bayesian Likelihood Detection, and these will be addressed in the next phase of this work. The fact that the fault was simulated to occur suddenly is for convenience of presentation only; a slowly developing fault would be equally observable with the same model deviations, as soon as it reached the same magnitude. Also the motor is started from rest, and the diagnostic model tracks the startup, although the initial transient results in significant deviations. However, startup transients can be easily separated from true faults by either ignoring the startup period altogether, or by increasing the detection limits in this region.

Averaged Diagnostic Model

Fault: Back EMF Reduced 20% at $t = .3$

Detection: DC Link Current; Motor Speed; Motor Torque

Figure IV-27 Simulated Demagnetization Fault --- Diagnostic Model Response

2. Results of High Level System Model

In addition to the detailed ECM Motor and Electronics simulation, a high level system model simulation was developed in Matlab/Simulink for the purpose of simulating system level faults or disturbances and determining their effects at the Motor. A top level block diagram is shown in the figure below.

The major blocks modeled are a Flow Splitter, multiple Ducting and multiple Rooms. The model is driven by the Blower, which provides air flow and heat transfer into the system, and produces the load pressure acting on the Blower. The Blower supplies the Flow Splitter which contains adjustable dampers to balance the flow among the multiple Ducts. The Ducts deliver airflow to the rooms and contain adjustable registers at the entrance to each room. Air loss through leakage and heat conduction through the walls is also modeled. The Rooms contain an adjustable return register, a leakage path (normal leakage and also for disturbances such as a door opening), as well as modeling heat transfer through the walls. Flows, Pressures and Temperatures are computed throughout the system. The various dampers, leakage paths and flow restrictions can be manipulated to simulate the occurrence of system faults and allow determination of their diagnostic observability by the Blower. The following figure shows a sample output of Blower pressure disturbances resulting from a door opening, adjustment of a room inlet register, and the (near) closing of a balancing damper. The System Model will serve as the tool for analyzing the effects of system operation and fault conditions on the Blower and for assessing the level of observability

Balancing Dampers

Door

Room 1

Duct 1

Room 2

Duct 2

Return Registers

Room Registers

[illegible]

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Figure IV-30 Dynamic System Model Response to Various Disturbances

V. References

VI. Bibliography

VII. List of Acronyms and Abbreviations

CFD	computational fluid dynamics	OEM	original equipment manufacturer
CFM	cubic feet per minute	PFC	power factor correction
DSP	digital signal processor	QFD	quality function deployment
ECM	electronically commutated motors	RI	rearward inclined (blade)
EHS	environment, health and safety	SPL	sound pressure level
FI	forward inclined (blade)	VOC	voice of the customer
HVAC	heating ventilation air conditioning		
HVIC	high voltage integrated circuits		
IGBT	insulated gate bi-polar transistors		

VIII. Conversion Factors

Every attempt was made to use SI units in this report and research work. A few graphs do contain US conventional units which can be converted with the assistance of the below formulae.

Volumetric Flow:	$1 \text{ [CFM]} = 0.0004719 \text{ [m}^3\text{/sec]}$	or	$1200 \text{ [CFM]} = 0.566 \text{ [m}^3\text{/sec]}$
Pressure:	$1 \text{ [inH20]} = 248.4 \text{ [Pa]}$	or	$0.25 \text{ [inwg]} = 62.5 \text{ [Pa]}$
Rotational Speed:	$1 \text{ [RPM]} = 0.1047 \text{ [rad/sec]}$	or	$1150 \text{ [RPM]} = 120 \text{ [rad/sec]}$
Torque:	$1 \text{ [Oz-ft]} = 0.0847 \text{ [N-m]}$	or	$15 \text{ [Oz-ft]} = 1.27 \text{ [N-m]}$
Electrical Power	$1 \text{ [hp]} = 746 \text{ [Watts]}$		
Cooling Power	$1 \text{ [ton]} = 12000 \text{ [BTU-hour]} = 3520 \text{ [Watts]}$		
Air Flow Power	$\text{Power}_{\text{air}} = \text{Pressure} \cdot \text{Flow} = [\text{inH20}] \cdot [\text{CFM}] \cdot 0.1175 = [\text{Pa}] \cdot [\text{m}^3\text{/sec}] = [\text{Watts}]$		
Mechanical Power	$\text{Power}_{\text{motor}} = \text{Torque} \cdot \text{Speed} = [\text{Oz.-ft}] \cdot [\text{RPM}] \cdot 0.008868 = [\text{N-m}] \cdot [\text{rad/sec}] = [\text{Watts}]$		