

Design of a 28 HP, 47,000 RPM Permanent Magnet Motor for Rooftop Air Conditioning

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Abstract

The transition to a new generation of air-conditioning and refrigeration products is fueled by a combination of environmental concern and increased world demand for improved performance at lower cost. Research and development programs worldwide are focussed toward bringing to market low-cost, better performing, advanced vapor compression refrigeration systems. Currently, direct-drive, high-speed compressors are not utilized in HVAC systems. This is because high-speed, refrigerant cooled motors do not exist for commercial use and efficient power electronic inverters, required to produce close-to-perfect sinusoidal waveforms, are not available at a reasonable price. Effective utilization of a high-speed, direct-drive compressor configuration in the cost and performance sensitive HVAC markets, requires very high motor and drive efficiencies at both part load and full load operation. Under funding provided by the U.S. Department of Commerce Technology Administration, SatCon Technology has joined a consortium of companies in the development of a 25-ton centrifugal compressor. Under this consortium, SatCon has designed, developed and is in the process of testing a 28hp, 47,000 rpm, permanent magnet motor drive. The potential benefits include an increase in the average energy efficiency of air-conditioning and refrigeration equipment. This increase in efficiency would simultaneously save industry users billions of dollars in energy costs and significantly reduce the emission of carbon dioxide and other pollutants. This paper summarizes the prototype motor design.

1. Introduction

The size of the global air conditioning market, which currently amounts to \$33 billion, exhibits a growth that has traditionally been driven by three exogenous factors: temperature, population, and income. Based on growth in U.S. housing initiations and increasing demand in emerging global markets (primarily in the warmest regions of the world), industry and government projections indicate that the market for

air-conditioning could increase by \$6 billion in sales by the year 2000.

In recent years, government regulations and environmental concern have also played an increasing role in the air-conditioning industry. In the late 1980s, concern over ozone depletion and global warming prompted the restriction of Chlorofluorocarbon (CFC) production. In addition, Hydrochlorofluorocarbon (HCFC) is expected to be discontinued as a refrigerant by the year 2030 in the U.S. (and even earlier in other countries). Moreover, the U.S. Energy Policy and Conservation Act of 1992 (EPACT) mandated the manufacture and use of energy- efficient motors. Improved motor efficiency will cut the amount of power generated by electric utilities, which in turn will reduce the use of fossil fuels that add to the “greenhouse” effect.

Another key trend has been the growth of the replacement market for air-conditioning, especially in North America. Seventy-two percent of US commercial buildings were built more than 20 years ago, with many still operating with their original HVAC equipment. Given that the average life of an air-conditioning unit is 20-25 years, the industry expects a major upgrade of systems due to a variety of reasons, including system failure, improved efficiency and cost savings, and environmental legislation.

Advances in compressor technology have led to a reduction in the number of moving parts, hence decreasing unit size, increasing system reliability, and decreasing energy consumption. In addition to becoming more sophisticated, system electronics are moving from proprietary protocols to standard protocols that permit greater levels of interconnectivity to other building systems, such as lighting, power and fire and safety systems. New building safety codes for smoke control require the interconnectivity of HVAC and fire and safety

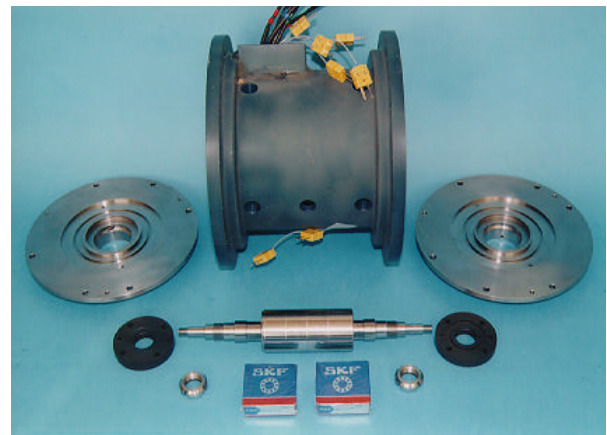
systems. Standard protocols also permit monitoring of building systems with a single personal computer, and they reduce the complexity of facility management. The integration of HVAC, lighting and power systems allow building supervisors to achieve better energy management with lower operating costs.

Through research and development programs, the US Department of Commerce is focussed toward helping US manufacturers develop low-cost, better performing, advanced vapor compression refrigeration systems. Besides enhancing the industrial competitiveness of US manufacturers in global markets, the government’s initiatives aim at increasing the average energy efficiency of air-conditioning and refrigeration equipment in the U.S. This would simultaneously save industry users billions of dollars in energy costs and significantly reduce the emission of carbon dioxide and other pollutants as a result of reduced fuel consumption at power plants. Under funding provided by the U.S. Department of Commerce Technology Administration, SatCon Technology has joined a consortium of companies (Carrier Corporation, Allied-Signal, General Electric, DuPont and Lockheed-Martin Control Systems) in the development of a 25-ton proof-of-concept centrifugal compressor. Under this consortium, SatCon has designed, developed and is in the process of testing a 28hp, 47,000rpm permanent magnet motor drive (Figure 1).

Figure 1. SatCon’s 28hp, 47,000 rpm compressor motor prototype.

The new air-conditioning system will offer a number of advantages:

- Increased system efficiency (approximately 15% over current commercial compressors).
- Weight and space savings (down to one fifth of the 500-pound traditional piston compressors).
- Fewer components and reduced system complexity. These benefits lower initial product cost, as well as labor, installation, and maintenance costs.
- Modularity (which positively impacts cost), standardization of equipment, and overall system simplification.



- Variable speed, which allows the supply to be matched to a variable demand for higher efficiency
- Quieter systems
- 10-15% annual operating cost savings

The following sections provide an overview of SatCon’s high-speed motor design.

2. Motor design specifications

The motor drive design point is based on a 25-ton air conditioning system. The drive requirements applicable to the design are provided in Table 1. The operating speed varies between 75% and 100% of maximum speed and the operating power varies from 20% to 100%.

Table 1: Compressor motor specifications.

Motor efficiency (%)	≥ 93-95 at full load < 5 drop off at part load
Shaft diameter	0.6 in (1.5 cm)
Cooling	HFC 134a
Rotor diameter	< 1.8 in (4.6 cm)
Reliability	30,000 hr lifetime
Cost (\$/hp)	10 for rotor/stator set only, assembly separate
Motor drive efficiency (%)	90
Supply voltage (V)	440, 3 <i>f</i> , 60 Hz

3. Stator and Rotor Magnetics Design

The permanent magnet motor topology is conventional, with the magnets located on the rotor shaft (Figure 2).

The motor is a two-pole pair machine with an envelope described by an outer diameter of 4.0 in. (10.2 cm) and an overall length of 4.8in. (12.2 cm)



Figure 2. Permanent Magnet Rotor and Stator Details.

Operation at high speeds allows for a considerable reduction in motor size. This, however, gives rise to a number of challenging design issues. Firstly, rotor losses are present in permanent magnet machines because a) the magnets are not perfect insulators and b) the rotor is exposed to time-varying magnetic fields from: temporal harmonics of the applied voltage and spatial harmonics of the stator current. Together with the core losses in the stator, these losses have to be accurately calculated to estimate drive efficiency. One way of reducing eddy current losses is to finely subdivide the magnets (in effect, “laminates” the magnets). Due to its high manufacturing cost, this process is not attractive. A more cost-effective way is to employ plastic bonded Neodymium-Iron-Boron magnets (which are already effectively subdivided). Although this material exhibits almost half the remanent flux density of other NdFeB materials (Table 2), its higher bulk resistivity results in lower rotor eddy-current losses for high-speed applications. Bonded magnets are easily fabricated into thin-walled ring magnets, their resin binder eliminates cracking and chipping, and dimensional precision can be achieved without secondary processes.

Table 2: Manufacturer data on BREMAG Type B PM material.

	BREMAG
Residual induction (kG)	6.85

Coercive Force (KOe)	5.6
Intrinsic Force (KOe)	9.1
Energy Product (MGOe)	10.0
Recoil Permeability (G/Oe)	1.1
Temp. Coeff. of Br (%/C)	-0.105
Density (gms/cc)	5.89
Ultimate Tensile Strength (MPa)	18
Young’s Modulus (MPa)	8100
Resistivity (ohm-m)	4.0E-5

There are two winding coils in each stator slot, and six turns per coil. Each turn consists of a bundle of 3 in-hand 26/30 Litz (16 AWG equivalent) wires. The resulting wire packing factor (calculated as the overall wire area in a slot divided by the slot area) is 57%. The slot area calculation takes into account the presence of slot liners and insulation.

Table 3 provides performance details at the 28hp, 47,000 rpm design point. This table also includes circuit parameters for the machine.

Table 3: PM motor performance predictions and circuit parameters.

Output power (hp)	28
Terminal voltage (L-N, Vrms)	200
Speed (rpm)	47,000
Frequency (Hz)	1567
Ambient temperature (°C)	88
Average gap flux (T)	0.276
Back emf (Vrms)	191.7
Phase current (A, rms)	38.5
Synchronous reactance (ohms)	3.81
Power factor (%)	91
Slot leakage inductance (H)	2.34E-4
Airgap inductance (H)	1.53E-4
Phase inductance (H)	3.87E-4
Phase DC resistance (ohms)	7.46E-2
Phase AC resistance (ohms)	9.7E-2
Rotor losses (W)	534
Core losses (W)	304
Ohmic losses (W)	432
Back emf constant (V rad/s)	0.039
Magnetic shear stress (N/m ²)	21,391
Motor efficiency (%)	94.3
Tip speed (m/s)	99

In the calculation of motor efficiency, the following loss components have been considered:

- Stator resistive losses
- Stator core losses (consisting of hysteresis and eddy current components)
- Rotor losses
- Harmonic losses on the rotor and stator

Windage has not been included. The rotor and stator harmonic losses have been estimated up to and including the 43rd harmonic. The harmonic spectrum assumed is the following (Table 4):

Table 4: Voltage spectrum input to the PM motor

N	V(rms)	N	V(rms)
1	146.9	23	60.5
5	9.6	25	56.0
7	13.3	29	13.9
11	28.4	31	3.3
13	41.7	37	15.4
17	13.4	41	37.0
19	18.7	43	34.3

The accuracy of loss calculations will only be verified through prototype testing. It is important to note that the retaining ring material is assumed to be Inconel 718.

The entire machine is cooled with R134a. Liquid Freon is preferable for stator cooling, although a water-cooled stator jacket can be used. The rotor and windage losses are removed by low-pressure, predominantly gaseous Freon.

4. Designing the Rotor Assembly

The three major issues that were addressed in attaching the magnetic material to the rotor shaft are:

- determining a method of repeatedly and accurately attaching the magnetic material to the rotor shaft,

- assuring that no part of the rotor fails under operation of the motor, and
- minimizing effects of rotor mechanical design on electromagnetic performance

Two methods of gluing the Bremag-10N to the rotor were investigated. One method involves sliding a cylinder made of Bremag-10N over the rotor, and the other involves gluing arc segments of the material directly onto the rotor. Next, the inclusion of a thin outer wrap around the assembly was examined because of the low elasticity and ultimate tensile strength of Bremag-10N. Finally, segmented retainer cups were selected as the most appropriate approach. Static finite element stress analyses were performed on different versions of the rotor assembly to obtain an understanding of the stress state and to design appropriate magnet assemblies that ensure successful operation without failure.

There are two simple methods of attaching the magnets to the rotor with a thin film adhesive. The magnets may be individual arc segments that can be directly applied to the adhesive on the rotor shaft or part of a single cylindrical “shell” that may be slid over the adhesive on the rotor shaft. Uniform film thickness and desired preloading for film curation are obtainable using individual magnet segments. Assuring the accurate positioning of the segments on the rotor shaft may be time consuming. The shell provides accurate positioning of the magnets with respect to each other, but sliding it onto the rotor shaft while maintaining uniform glue thickness and concentricity is difficult if done by hand. (This problem is virtually eliminated using individual magnet segments). Interference preloading may be obtained by freezing the rotor shaft, but the loading may not be uniform due to lack of concentricity. If an inexpensive shell-to-rotor shaft assembly process that ensures concentricity, uniform film distribution on the magnet surface, and appropriate curing is identified, the shell would most likely be the lesser expensive option.

A two-dimensional, axisymmetric finite element model of the shell magnet-rotor shaft assembly was created in ANSYS 5.3 with preliminary design dimensions. Static stress analyses

were performed under dynamic loading conditions to determine the principal stresses in the assembly. The critical area of stress in the assembly was determined by noting the allowable stresses in the materials in the assembly and by intuitively estimating locations of high stress. SatCon's in-house testing and experience indicates that the adhesive film has an allowable tensile stress of about 1 ksi. This is lower than the 2.611 to 2.901 ksi ultimate tensile strength that the vendor lists for the magnetic material. Also, since the elasticity of the magnetic material is much lower than that of the steel in the rotor shaft, it is expected that the radial and circumferential growth of the magnets under dynamic loading should be much higher than that of the rotor shaft. This effect should result in the highest tensile stresses being located at the interface between the magnets and the rotor shaft, i.e., at the adhesive film. The finite element analyses showed this to be true. They also show that the stresses in the steel rotor shaft are much lower than the allowable stresses in the rotor shaft. Therefore, the location of the adhesive film was the critical area of interest.

The first finite element model consisted of the magnet-rotor shaft assembly spinning freely and steadily at 52,000 RPM. The maximum stresses developed in the magnet were located at the interface to the rotor shaft, as expected. The radial stress there was about 4 ksi and the hoop stress was about 2 ksi. This state exceeds the 1 ksi allowable stress of the adhesive. The purpose of the second model was to view the limit of the effects of adding an outer wrap to the outer circumference of the magnets without a preload. In the limiting, best-case scenario, the wrap would offer infinite stiffness but have no loading effect. To accomplish these effects, the outer circumference of the magnets in the model was fixed. The radial stress at the magnet-rotor shaft interface was still about 2.9 ksi and the hoop stress was about 1.3 ksi. This stress state also exceeds the adhesive's allowable stress, suggesting that a compressive pre-stress is required. The next model included a 0.030 in thick outer carbon fiber wrap around the assembly. The carbon fiber used was M60J 60% (Toray). The results from this model showed a radial stress at the magnet-rotor shaft interface of 3.7 ksi and a hoop stress of 1.7 ksi. These stress results fall between those from the two previous models, lending

confidence to the expectation that an acceptable stress state cannot be obtained without utilizing the magnets' relatively high compressive strength. The maximum stress (20 ksi) in the wrap is far below the wrap's allowable stress, suggesting that it is not unreasonable to assume that the wrap can sustain the tensile stresses associated with pretension. However, it is difficult to vary or control the level of radial pre-stress during the application of a wound composite wrap. The resulting radial pre-stress is process-driven, and it is typically quite low. Therefore, there is not much confidence in obtaining a satisfactory compressive pre-stress to alleviate the tensile stresses in the adhesive. Options that do not include an adhesive layer between the magnets and the rotor shaft were then explored.

The next option included a kevlar composite wrapped over segmented magnets. With no adhesive between the magnets and the rotor shaft, the magnet load is born by radial compression in the magnets and hoop tension in the composite wrap. This approach takes advantage of the relatively high compressive strength in the magnets and the high tensile hoop strength in the composite material. However, stability of the magnets as they lift off the rotor shaft is a concern.

As a result, the approach illustrated in Figures 3 and 4 has been adopted for permanent magnet retention. Cup-shaped metallic retainers are machined to mount permanent magnets on the ID of a 'cup'. Magnets and Delrin spacers are mounted sequentially around the circumference of the retainer and epoxied in place. The retainer 'modules' are pressed onto a 4340 alloy steel shaft and electron-beam welded in place. The cup material was chosen to be Inconel 718 because of its high strength, electrical resistivity and ease of machining. This construction allows high rotor speeds while maintaining economical construction methods. It provides security for the magnets as they lift off the rotor shaft.

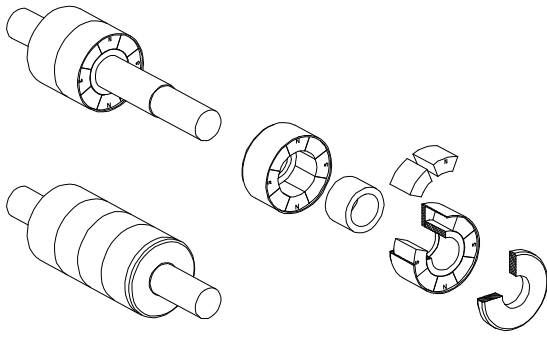


Figure 3. SatCon's approach to permanent magnet retention.

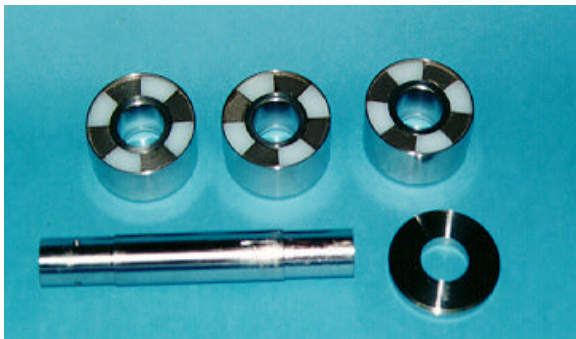


Figure 4. Rotor detail

5. Power Electronics Drive

For the proof-of-concept prototype development, an off-the-shelf inverter is being used (Figure 5).

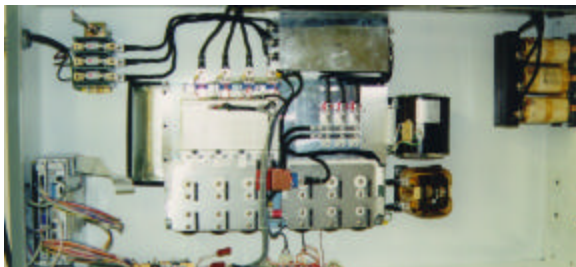


Figure 5. Test inverter for the PM motor drive

The inverter incorporates the following:

- floating point power supply,
- over temperature and over current protection
- SCR variable voltage front end
- Phase controlled
- Commercial DSP floating point controller
- Standard NEMA cabinet

Under different programs, SatCon is working towards achieving the industry cost goal of \$30/hp for the drive electronics.

6. Conclusions

SatCon is currently in the process of performing extensive testing on the motor drive integrated to a centrifugal compressor. Rotor spin tests have been performed confirming the displacement and critical speed estimates of the mechanical design phase. This tests were performed by positioning an eddy current displacement sensor was at the bottom of the shaft. This sensor was monitored using both a scope, and a frequency analyzer set up to plot displacement versus speed. The rotor was spun up to 55,000 rpm at a ramp rate of 200 rpm per second with minor increases in displacement at 20,000 rpm. Two risk areas are still being addressed in order to bring a feasible product to market. The first area is that of reliability under extensive and continuous product use. The other risk area is that of ensuring that the drive's cost meets the industry goal of \$40/hp.

8. About the Authors

Jerome Kiley is a senior mechanical engineer at SatCon. He has contributed innovative mechanical design, dynamic and vibration analysis, and thermo-fluid analysis to several projects involving Terfenol-D applications and high-speed, high power density motor design. He is also involved in thermo-fluid analysis of electric machines, and he has developed new mathematical thermo-fluid models for motor cooling applications. He holds an M.S. degree in Mechanical Engineering from Worcester Polytechnic Institute and a B.S. degree in Mechanical Engineering from Lafayette College.

Mary Tolikas is a senior magnetics engineer at SatCon. She is involved in the electromagnetic design and development of permanent magnet and variable reluctance motor drives for automotive, space, naval and consumer applications. She holds a Ph.D. and a M.S. degree in Electrical Engineering from the Massachusetts Institute of Technology, and a B.S. degree in Electrical Engineering from the University of Glasgow. She is currently pursuing an MBA degree in Strategic Management at the Sloan School of Management.