

A CFD study of Screw Compressor Motor Cooling Analysis

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Abstract. Screw compressors use electric motors to drive the male screw rotor. They are cooled by the suction refrigerant vapor that flows around the motor. The thermal conditions of the motor can dramatically influence the performance and reliability of the compressor. The more optimized this flow path is, the better the motor performance. For that reason it is important to understand the flow characteristics around the motor and the motor temperatures. Computational fluid dynamics (CFD) can be used to provide a detailed analysis of the refrigerant's flow behavior and motor temperatures to identify the undesirable hot spots in the motor. CFD analysis can be used further to optimize the flow path and determine the reduction of hot spots and cooling effect. This study compares the CFD solutions of a motor cooling model to a motor installed with thermocouples measured in the lab. The compressor considered for this study is an R134a screw compressor. The CFD simulation of the motor consists of a detailed breakdown of the stator and rotor components. Orthotropic thermal conductivity material properties are used to represent the simplified motor geometry. In addition, the analysis includes the motor casings of the compressor to draw heat away from the motor by conduction. The study will look at different operating conditions and motor speeds. Finally, the CFD study will investigate the predicted motor temperature change by varying the vapor mass flow rates and motor speed. Recommendations for CFD modeling of such intricate heat transfer phenomenon have thus been proposed.

1. Introduction

There are two methods to consider for hermetic motor cooling. The first is liquid using either gravity, spray, or pool distribution. The second is vapor. A third method using an external air cooled motor isn't practical or cost effective compared to hermetic motor cooling methods. There are several benefits to vapor cooling in refrigeration screw compressors. The vapor temperature is controlled by the evaporator in a temperature range that is easily applicable to cooling the motor. The vapor is readily available as it travels into the compression process. Using the relatively cool suction gas allows for a straight forward cooling flow path design, a cool running motor, and increases the motor power density for a given motor size.

Temperature distribution of the motor can influence the performance of a screw compressor. If the insulation gets too hot, it breaks down and reduces the life of the motor. To understand the temperature distribution of the screw compressor motor, thermocouples are added to the stator and then evaluated in the lab on a test stand. In Figure 1, the large dots illustrate the locations where these thermocouples are placed. Even though many thermocouples are added to the stator portion of the motor, only a small percentage of the motor is being monitored.



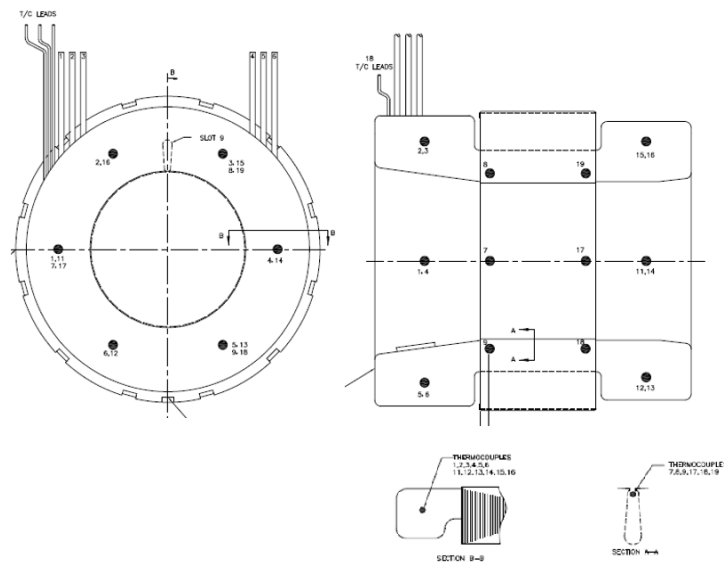


Figure 1. Thermocouple Locations for the Stator

By using CFD to evaluate the thermal profile of the motor, it can be done without thermocoupling. CFD can provide a more detailed distribution of the temperature profile than a thermocoupled motor could provide and more easily discover hot spots in the motor that can cause the motor to fatigue and shorten its life. CFD can also look at the flow path of the fluid around and through the motor which can be used to reduce the thermal hot spots in the motor, increase its life, and improve compressor efficiency. Literature review found two papers that looked at modelling a compressor motor; one from Chin (1) in 2012, and one from Tang (2) in 2016. Chin (1) used the lump parameter method by describing the heat transfer characteristics on a screw compressor by modelling a thermal equivalent circuit. Tang (2) performed a CFD analysis on a hermetic motor for a scroll compressor.

Motors consist of many parts, some of which consist of composite materials. Orthotropic thermal conductivity material properties were used during this study to simplify the composite materials into one volume. A more accurate representation of the material properties could then be used without having to create multiple volumes to represent the composite materials. A journal from Pietrak (3) in 2015 wrote about a Rayleigh model that addresses a process to calculate composite materials. A process described by Williams (4) in 2003 describes how to use orthotropic thermal conductivity in ANSYS CFX.

2. Motor Power Distribution

In the paper by Tang (2), motor efficiency FEA simulations were done to help determine the motor power. For this study, the motor power values were provided by the motor vendor. A simple motor power formula is **Watt = Volts x Amps x $\sqrt{3}$ x Motor Power Factor**. Motor power factor, is a ratio of the active power to apparent power. Active power is the real power doing the work. Apparent power is the vector sum of active and reactive powers. Reactive power is the power required for magnetization of the motor; it has no action and is influenced by the inductance of the motor.

These simulations used a uniform power distribution that did not change with temperature, and were applied to the following motor volumes:

- Rotor Aluminium
- Stator Laminate
- Inside and End Windings.

3. Review of Modelling Approach

CFD simulations were conducted using ANSYS CFX version 16.2 and were built with several domains. The laminates and windings used orthotropic thermal conductivity material properties. Most domains were stationary except for the rotor and the shaft. Figure 2 shows a cross section of the simulation setup. The motor rotor consisted of the aluminium rotor and the rotor laminate. The stator was created with many parts, including the liner, windings, wedges, and the laminate. The liner goes in between the outside of the copper windings and wedges on the inside of the stator laminate slots. The liner was simulated using a 2-D feature which was modelled as a thin material. The stator laminate, air gap, and inside copper windings are shown in Figure 3. Figure 4 highlights the inside copper windings and copper end windings.

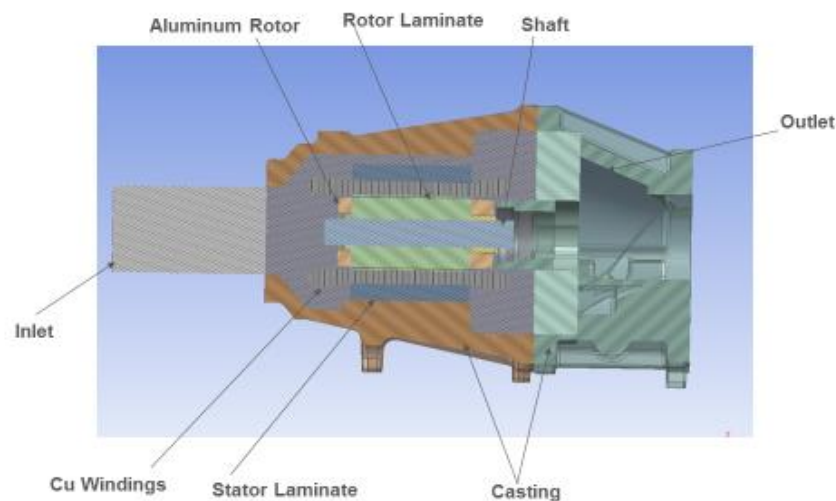


Figure 2. Simulation Setup

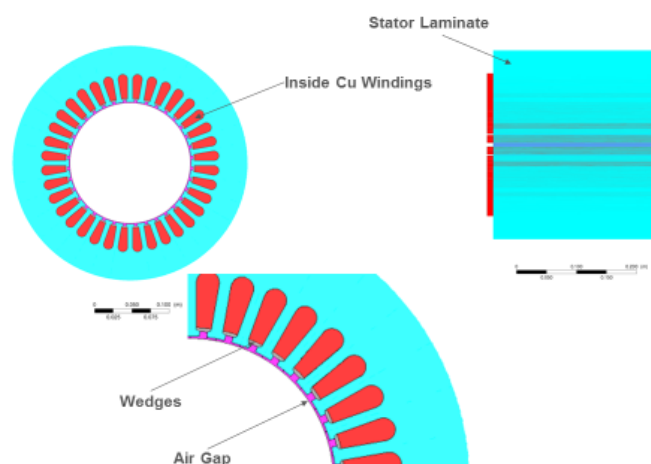


Figure 3. Simulation Setup

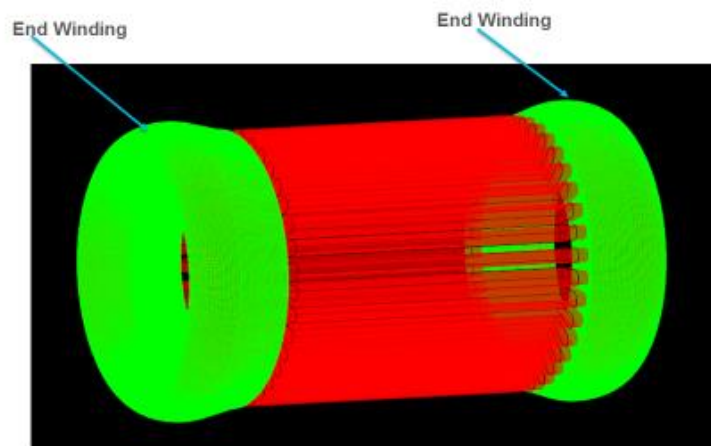


Figure 4. Simulation Setup

The flow path for the simulation went from the inlet to the radial and axial entrances to the rotors. This is shown in Figure 5, which includes the air gap between the stator and the rotor.

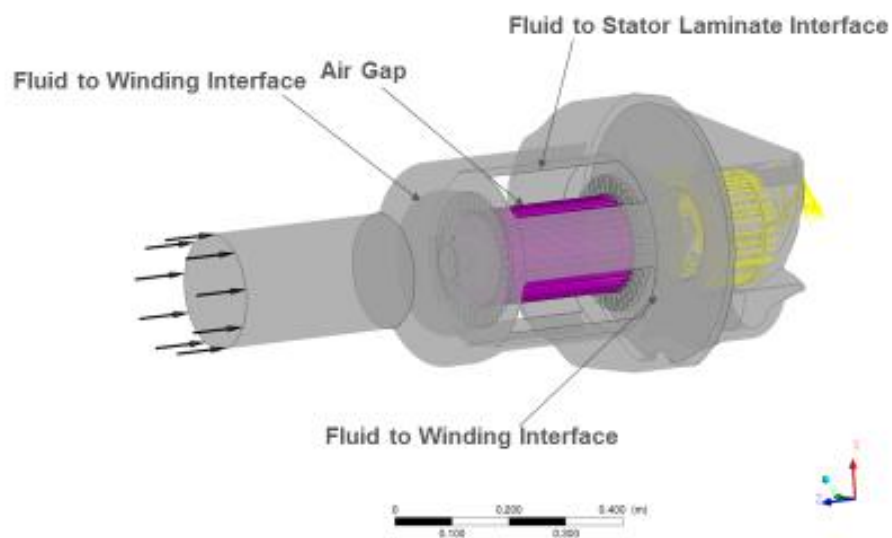


Figure 5. Simulation Setup

A rough heat transfer coefficient was applied to the outside of the casting using a flat plate heat transfer coefficient equation. The CFD setup assumptions are shown below.

- ANSYS CFX Version 16.2 was used for the CFD Analysis
- Steady State Simulation
- Heat Transfer = Total Energy w/ Vis Work Term.
- Turbulence Model = SST
- The outside walls had a heat transfer coefficient applied.
- RPM varied (3532 RPM, 864 RPM, 1679 RPM)
- Fluid = R134a Gas
- No oil
- Radiation is not considered.
- Orthotropic thermal conductivity applied to
 - rotor laminate
 - stator laminate
 - windings
- Mesh size = 77.9 million nodes
- Interfaces = 449

The simulations were run on a cluster using 162 cores with several E5-2697 v3 CPU's. Simulation times ranged from one to six days depending on the differences in the operating conditions from the initial conditions.

The orthotropic material properties were calculated using the Rayleigh models which came from the Journal "A review of models for effective thermal conductivity of composite materials", on page 16 from Pietrak (3). The equations used are shown below where k_1 is treated as a cylinder of the metal and k_m = the lamination or varnish in the windings, and ϕ = volume fraction of the materials.

Assuming z is the axis of the cylinder, the formulas are

$$\begin{aligned}
 & \bullet \frac{k_{eff,ZZ}}{k_m} = 1 + \left(\frac{k_1 - k_m}{k_m} \right) \phi \\
 & \bullet \frac{k_{eff,XX}}{k_m} = \frac{k_{eff,YY}}{k_m} = 1 + \frac{2\phi}{C_1 - \phi + C_2(0.30584\phi^4 + 0.013363\phi^8 + \dots)} \\
 & \bullet C_1 = \frac{k_1 + k_m}{k_1 - k_m}, C_2 = \frac{k_1 - k_m}{k_1 + k_m}
 \end{aligned}$$

Williams (4) describes the setup and how to incorporate an orthotropic material property in cylindrical or Cartesian coordinates for ANSYS CFX. He also performs a test case that shows how well it works. In CFX, this is a beta feature. To be able to use beta features, that option needs to be turned on. The CEL expression in cylindrical coordinates for how to set this up in CFX are shown in Figure 6.

```

MATERIAL: Material
Material Group = User
Option = Pure Substance
Thermodynamic State = Solid
PROPERTIES:
Option = General Material
EQUATION OF STATE:
Density = XX.X [kg m^-3]
Molar Mass = X [kg kmol^-1]
Option = Value
END
REFERENCE STATE:
Option = Specified Point
Reference Specific Enthalpy = X [J kg^-1]
Reference Temperature = XX [C]
END
SPECIFIC HEAT CAPACITY:
Option = Value
Specific Heat Capacity = X.XX [J kg^-1 K^-1]
END
THERMAL CONDUCTIVITY:
Option = Orthotropic Cylindrical Components
Thermal Conductivity Axial Component = XX.X [W m^-1 K^-1]
Thermal Conductivity Theta Component = XX.X [W m^-1 K^-1]
Thermal Conductivity r Component = XX.X [W m^-1 K^-1]
AXIS DEFINITION:
Option = Coordinate Axis
Rotation Axis = Coord X.X
END
END
END
END

```

Figure 6. Simulation Setup

4. Test and Analysis

Before the CFD test cases were conducted, a thermal boundary layer study was performed to determine the correct inflation layer for the simulations. It is important to note that a viscous boundary layer is not the same as a thermal boundary layer, but the relationship is the Prandtl number. After archiving mesh independence, the simulations were started. All simulations were run until the RMS residuals converged to at least $1e-4$ and all of the imbalances were converged to at least one percent.

The results from the first three cases had a motor installed with several thermocouples on the stator as shown in Figure 1, which were monitored and measured in the lab. Results from these cases have been normalized by setting the lowest temperature value in each experiment to zero and setting the largest temperature value in each experiment to one. In all cases, the inlet was a pressure and temperature boundary condition, and the outlet was a mass flow rate boundary condition. For cases 1 through 3 the volumetric average, minimum and maximum temperature values came from the CFD simulations. The thermocouple values came from the average values of the thermocouples on the motor from lab testing.

For case 4, the motor speed and mass flow rates were doubled from case 3, and the motor temperature drop was predicted. Case 4 was not tested in the lab, so no thermocouple data was available. The operating conditions are provided for all cases in the format of saturated suction temperature, saturated discharge temperature, and super heat.

For all cases, the flow path of R134a fluid goes towards the negative Z direction. The rotor side end windings are downstream from the suction side end windings. The temperature profile of one of the simulations is seen in Figure 7.

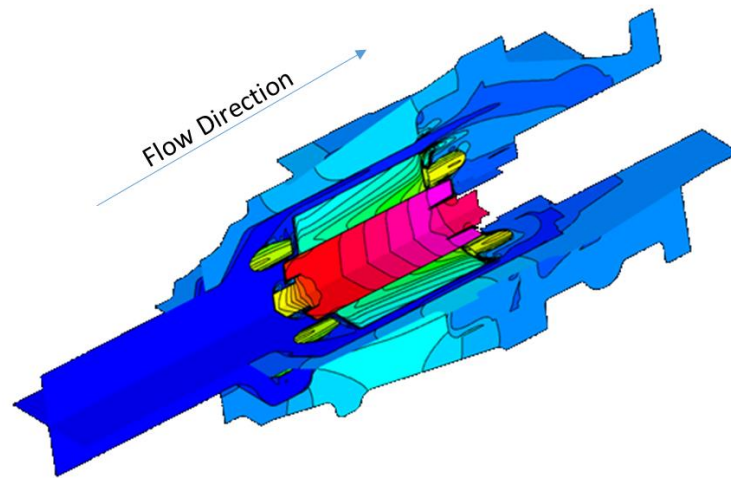


Figure 7. Temperature Profile of the Simulation

Even though the stator geometry is symmetrical, the stator contour temperature profiles is not symmetrical because the flow path around the motor is not symmetrical. For Case 1, the compressor was operated at a (40/100/0) °F operating condition. The motor was rotating at 3532 rev/min. The inlet density of the fluid is 0.8247 lbm/ft³. The CFD contour plot of the motor is shown in Figure 8. The normalized numerical results are in Table 1. From the data in Table 1 the lowest temperature is the suction side winding and the largest temperature is the rotor.

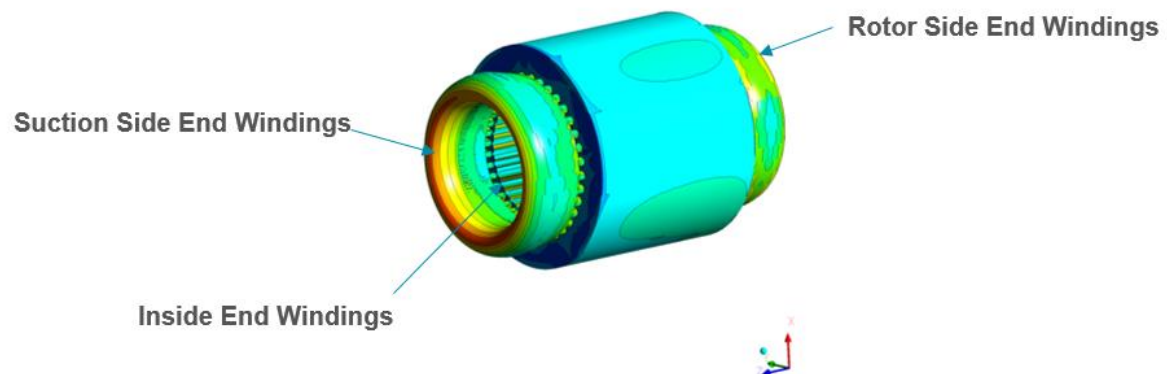


Figure 8. Case 1 Simulation Results

Table 1. Normalized Case 1 Temperature Results

	Volumetric			
	Average	Min.	Max.	Ave. Thermocouple
Suction Side End Windings	0.57	0.00	0.89	0.37
Inside End Windings	0.60	0.05	0.91	0.44
Rotor Side End Windings	0.66	0.03	0.84	0.58
Rotor Aluminum	0.73	0.51	1.00	

For Case 2, the compressor was operated at a lower pressure ratio than Case 1, and a much lower speed. The operating condition was (40/75/0) °F, and the motor speed was 864 rev/min. The inlet density of this simulation is the same as in Case 1. The lower speed reduced the mass flow rates of the R134a by 4.5x. Along with lower mass flow rates, the power distribution in the motor is 3.5x lower than in Case 1. The CFD contour plot of the motor is shown in Figure 9. The normalized numerical results are in Table 2.

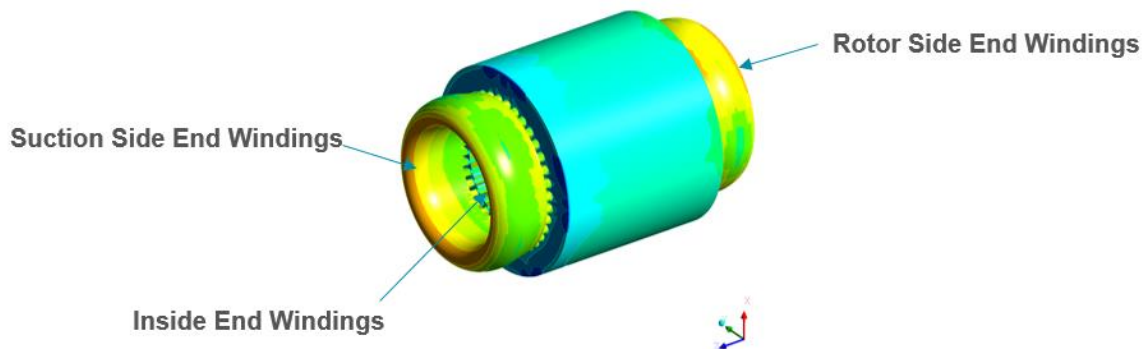
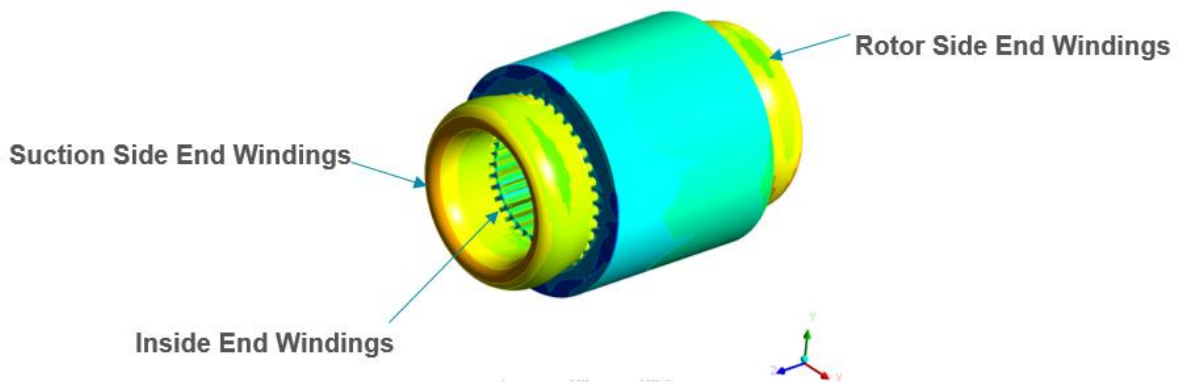


Figure 9. Case 2 Simulation Results

Table 2. Normalized Case 2 Results

	Volumetric Average	Min.	Max.	Ave. Thermocouple
Suction Side End Windings	0.30	0.00	0.42	0.29
Inside End Windings	0.33	0.01	0.50	0.24
Rotor Side End Windings	0.40	0.02	0.53	0.43
Rotor Aluminum	0.85	0.70	1.00	

For Case 3, the compressor was operated at the largest pressure ratio of the three cases and at a medium speed. An operating condition of (17.3/116/0) °F was tested. The motor speed was 1679 rev/min and the inlet density of the fluid is 0.3586 lbm/ft³. The mass flow rates of this case is 4.53x slower than Case 1. The main differences in Case 3 over the other two is that the mass flow rates are lower like Case 2 but the power is increased by 1.4x over Case 1. The increase in power is due to the higher pressure ratio, which caused the increase in torque. The CFD contour plot of the motor is shown in Figure 10. Normalized numerical results are listed in Table 3.

**Figure 10.** Case 3 Simulation Results**Table 3.** Normalized Case 3 Temperature Results

	Volumetric Average	Min.	Max.	Ave. Thermocouple
Suction Side End Windings	0.13	0.00	0.23	0.04
Inside End Windings	0.14	0.01	0.29	0.08
Rotor Side End Windings	0.22	0.03	0.68	0.25
Rotor Aluminum	0.62	0.39	1.00	

Table 4 was created to compare Cases 1-3. Only the volumetric average values were used in this comparison. The numbers were normalized with the smallest values being set to 0 and the largest values being set to one.

Table 4. Normalized Cases 1,2,3 Volumetric Averaged Temperature Results

	Volumetric Average		
	Case 1	Case 2	Case 3
Suction Side End Windings	0.06	0.00	0.41
Inside End Windings	0.06	0.00	0.42
Rotor Side End Windings	0.07	0.01	0.52
Rotor Aluminum	0.09	0.08	1.00

Case 4 was not tested in the lab, but was simulated to predict the differences in motor temperature drop by doubling the motor speed and mass flow rates from Case 3. The motor speed for this simulation was 3358 rev/min. The operating condition was the same as Case 3 at (17.3/116/0) °F. Table 5 shows the percent reduction in motor temperature.

Table 5. Average Volumetric Percent Reduction

	Average Volume % Reduction
Suction Side End Winding	15.7
Inside End Winding	19.8
Rotor Side End Winding	20.4
Rotor Aluminum	38.8

5. Conclusion

- The CFD study from the first three test cases looked at various operating conditions at different speeds. These CFD solutions were compared with the lab results where the motor was installed with thermocouples and measured in the lab. The CFD results were consistent with the thermocoupled measured data.
 - The coolest parts of the motor were the end windings and the inside windings. The hottest part of the motor was the rotor.
- When all three cases were compared to each other, Cases 1 and 2 were very similar. With the decreased mass flow rates along with the increased power in Case 3, the temperature of the motor was increased considerably.
- Case 4 looked at motor response when the mass flow rates were doubled along with the rotational speed from Case 3. The largest reduction was the rotor at 38.8%. The motor temperature drop improved as the gas moved through the motor from the suction side to the rotor side.
- In time, additional improvement could be gained by incorporating the motor efficiency FEA simulations methods done from Tang (2).

Acknowledgement

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References

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