Microturbine Performance Testing and Heat Transfer Analysis

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Project 1: Microturbine Performance Testing

1.1 Purpose and Introduction

Ingersoll Rand Energy Systems has recently redesigned some major subsystems of its 250 kW microturbine (model MT250). The objective of this effort was to make the unit easier to service, maintain and slightly improve the performance, and make manufacturing easier. This new program was identified as the Generation 3 microturbine or "G3". The highly-reliable rotor assembly, including the gearbox, shaft, compressor wheel, and turbine wheel remained unchanged in the G3 to minimize the risk of redesign. However, the static structure around the rotating equipment was redesigned. In addition, a whole new enclosure was designed, including the base, enclosure framing and paneling, and the cooling and fuel subsystem.

To validate the newly designed systems, an extensive qualification and endurance test campaign was defined. Risk identification meetings and design reviews were held in order to identify which component- and system-behavior had the largest risk of performance due to design modifications. Based on these risks, instrumentation and tests were identified to validate these components in full-scale engine testing. The test plan included steady state measurements, extensive engine thermal cycling to test cycle-sensitive hardware, and long full-power endurance testing to subject oxidation- and creep-sensitive hardware to validation testing.

1.2 Test Setup

Instrumentation was defined for both the new core engine modifications, as well as the microturbine enclosure. To limit the scope of this report, only the core engine testing will be reviewed. The redesigned static structure of the G3, along with some of the instrumentation is defined in Figure 1. Due to company proprietary restrictions, much of the figure has been obscured. The test plan included 42 pieces of instrumentation, including 36 thermocouples, 4 pressure taps, and emissions and fuel flow measurement devices.

Figure 1. G3 Core Engine Design

Testing initially commenced of the core alone on 3/31/2010. The testing included initial measurement of the performance and instrumentation readings of the G3 core without the new package. The core was then integrated into the package, which started testing on 7/24/2010. The engine was tracked with respect to cycles, hours, and the inlet temperature of the recuperator. A complete teardown and hardware inspection will occur in the fall of 2010.

1.3 Performance Test Results

Due to the extended test time of the new G3 core, the engine was run at full power over a variety of conditions. These included changes in ambient temperature, various ambient pressure changes, and different power conditions. A procedure was defined to essentially "correct" the performance data back to a standard condition: 59F, 14.696 psia, and 100% power. This would facilitate trending of the data to look for performance changes over time.

During testing, an assembly error was discovered that admitted some loose insulation into the engine flowpath downstream of the turbine, but upstream of the recuperator. This erroneous assembly issue can only be corrected with a teardown of the engine, and will be corrected at the next opportunity. Unfortunately, the impact of this insulation is that it collects on the face of the recuperator. This causes a higher-than-normal pressure drop across the recuperator, and "backpressures" the engine. This backpressure reduces overall performance (lower efficiency, higher flowpath temperatures). Therefore, a scheme to also correct for the pressure drop increase across the recuperator was needed.

Some test data was taken upstream of the recuperator at various times to determine how the recuperator was being back pressured over time, shown in Figure 2, so a correction model could be generated. After the curve fit was produced (not shown), the correction procedure involved entering the back pressure over time into a thermodynamic performance model to determine how the back pressure affected temperatures, pressures and efficiencies throughout the engine. The test data could then be set back to standard conditions using this model.

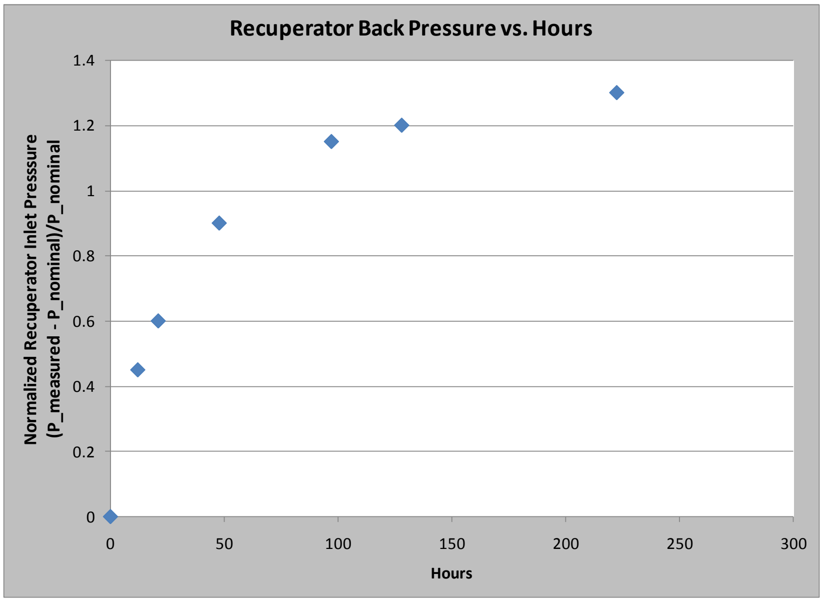

Figure 2. Normalized recuperator back pressure

One important parameter measured during testing was the recuperator inlet temperature (TA60) which indicates the health of the engine. The recuperator inlet temperature is often the limiting temperature parameter of the engine because the material is very thin and has a significant temperature and pressure gradient across it. Figure 3 shows how the recuperator inlet temperature varies over cycles (i.e. time). This value was measured with two different thermocouples for redundancy and accuracy, shown as blue and red on this figure. These values have been corrected to standard conditions and recuperator back pressure.

Figure 3 is divided into three different sections: PP2, PP1 indoor and PP1 outdoor. PP2 refers to a test cell that is simplified to test only engine hardware. It does not include many features of the current configuration, but is good for initial testing. PP1 refers to the full microturbine package, which has been adapted for indoor (ducting to outside) and outdoor (inlet louvers) use.

It is apparent that while in PP2, the engine reached an initial steady-state and then had a sudden rise which was the result of a decrease in ambient temperature. This indicates that there may be a problem with correcting for ambient temperature or there is a problem with the engine. Because the values began decreasing after more cycling lead to the belief that it is an issue with the ambient temperature correction. The results in PP1 indoor show no significant trend except for the large hump between 80 and 100 cycles. During this time, there were no irregularities in ambient temperature so the cause has yet to be determined. The engine has yet to be torn down for inspection so it is difficult to make any conclusions. Finally, the PP1 outdoor configuration shows no major trend other than a slight decrease in average temperature over the indoor unit which could be the result of reduced inlet pressure from the outdoor configuration.

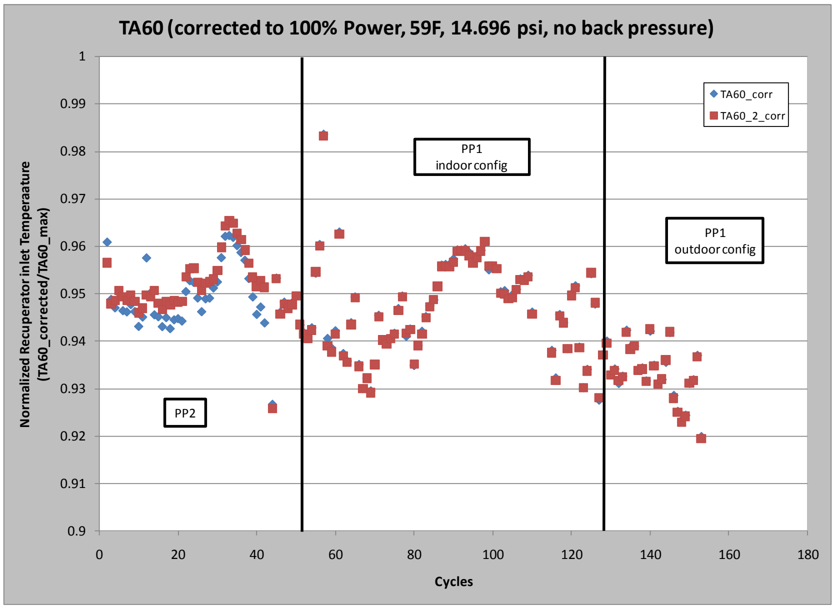

Figure 3. Normalized recuperator inlet temperature

Project 2: Heat Transfer Analysis

2.1 Purpose and Introduction

The purpose of this project is to develop an improved computational heat transfer model for the heat recovery unit on the microturbine to calculate heat transferred from the gas to water. This model will be integrated into a full engine cycle analysis tool to give accurate engine performance data and heat recovery unit performance curves for various conditions. A simplified model will also serve as a tool for the marketing and sales department to give customers more reliable heat recovery data.

The heat recovery unit, shown in Figure 4, is a cross-flow fin-tube heat exchanger where the hot exhaust gas flows between the fins and water (or any other liquid/gas) runs through the tubes. The idea of the heat recovery unit is to maximize the efficiency of the microturbine (combined heat and power). By extracting the un-used exhaust heat, the overall efficiency increases significantly; this reduces its impact on the environment and provides the customer with heated water.

Figure 4. Cogen Heat Recovery Unit

An old heat recovery model had been developed, but it was based on experimental data for a lower effectiveness heat exchanger, resulting in a very specific model that required extrapolation to reach certain conditions. The model created in this project is based off of empirical and theoretical calculations which allow the model to be robust and accepting of any inlet conditions, fluids and geometries. This model will accurately model the current higher-effectiveness heat exchanger.

2.2 Model

Figure 5 shows a schematic for the heat transfer model. The model takes into account the effect of air convection, water convection, tube material conduction and fin material conduction. The heat can either travel from the air through the fin through the tube and to the water or from the air through the tube to the water. The model takes into consideration the actual properties of exhaust gas and water (i.e. specific heat, viscosity, density, thermal conductivity, etc.) depending on the fuel used, temperature, and pressure to calculate the Reynolds numbers, Prandtl numbers, Nusselt numbers and eventually the thermal convection. The conduction of the fin and tube material is calculated by knowing the fin material properties and temperatures. With the thermal convections and conductions calculated, the thermal resistivity of the fin-tube model can be determined which ultimately yields the heat transfer given the inlet water and air temperatures and flow rates.

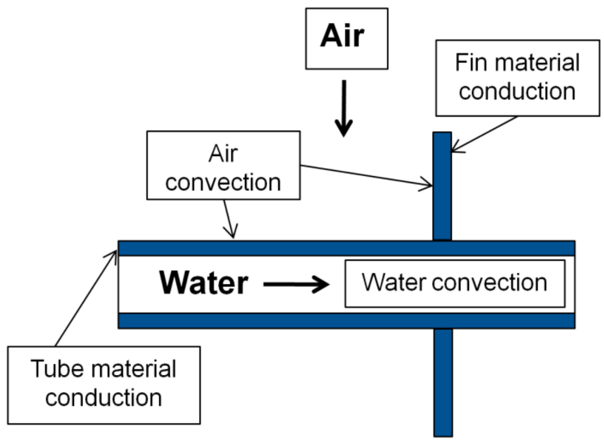

Figure 5. Heat Transfer Model

2.3 Results and Discussion

Extensive analysis was performed with the heat transfer model to validate it with test data. The average error in heat transfer between the model and the production testing was 1.17%. The water temperature out was also predicted through the model to be 1.91% higher than the test data. These differences are likely due to the unaccounted imperfections in the production test heat recovery unit. However, this data is very promising, indicating that the model is indeed accurate at standard conditions. The model should also be accurate at extreme conditions since the model analyzes the correct physics. Unfortunately, there were no test resources available to do additional testing for confirmation.

The model was then used to create data and curve fits for the marketing department. Figure 6 shows the iso-heat recovery curves with respect to inlet water temperature and inlet water flow. This figure is useful because it can quickly tell a customer how much heat they will recover given their inlet conditions. The figure also indicates a grayed out area where the outlet water temperature is near the boiling point of water under ISO conditions (with a safety factor), which the customer should stay away from. The major conclusion drawn from Figure 6 is that at low inlet water temperatures, the heat recovery is very sensitive to inlet water flows rates and not water temperatures. The opposite is also true, where at high inlet water flow rates the heat recovery is sensitive to inlet water temperatures, but not water flow rates. The variables of inlet gas flow rate and temperature are not included in the figure because they would complicate the figure and are not variables the customer can adjust.

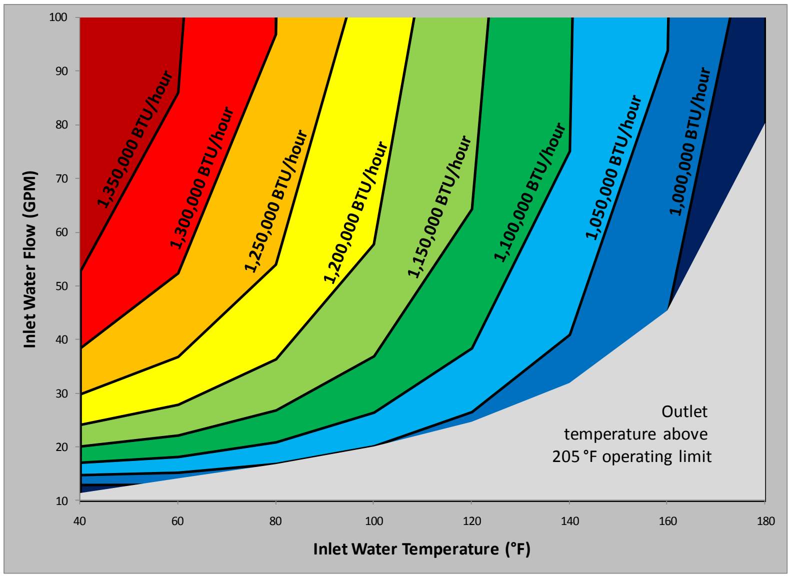

Figure 6. Heat recovery unit performance curves

Finally, quick parametric studies were performed to obtain the basic performance change with respect to single variable change. For this investigation, the two parameters studied were heat recovery and microturbine higher heating value efficiency, and the three parameters varied were percent power of the engine, altitude of operation, and fuel type.

The percent power study provided results as expected, shown in Figure 7. As power decreases, the heat recovery decreases because the gas temperature goes down because the combustor is burning leaner. The higher heating value efficiency decreases two-fold because both the power provided by the generator and the heat recovery go down.

As the altitude increases, the results are a little more interesting, which are displayed in Figure 8. The heat recovery goes down because the ambient pressure goes down, which requires less fuel, therefore a lower gas temperature. However, because the power provided and heat recovery only decrease slightly with a significant drop in fuel flow rate, the higher heating value efficiency sees a mild increase.

The last parametric study, fuel type, has its results provided in tabular form in Table 1. The difference in fuel types is the percent of combustible content, which relates directly to the fuel flow rate required. Natural gas was assumed to have 96% methane, where anaerobic digester gas only has about 60% methane and landfill gas only has about 40% methane, the rest being filled with mostly carbon dioxide. As seen in Table 1, with the increasingly diluted fuels the heat recovery and higher heating value efficiency increase. This is because the specific heat and temperature of the diluted gas are slightly higher. Also, although the fuel flow rate must increase significantly, the higher heating value decreases significantly giving the slight rise in efficiency.

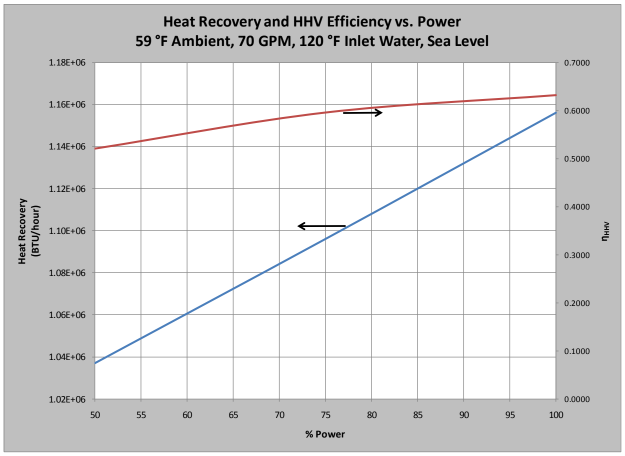

Figure 7. Percent power parametric study

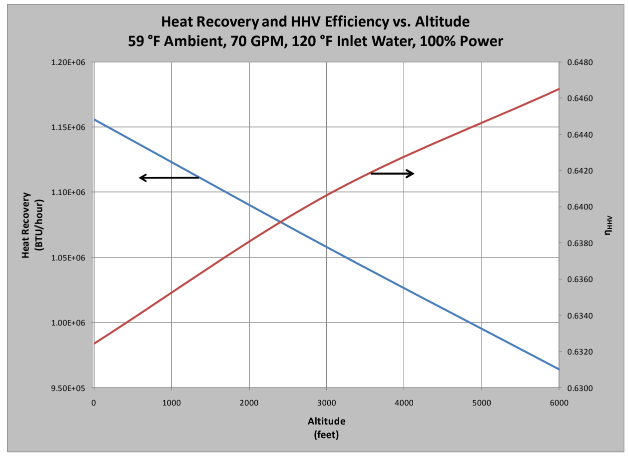

Figure 8. Altitude parametric study

Table 1. Fuel type study

Test	Gas Type	q_calc	η_HHV
#		BTU/hour	
1	Natural	1.16E+06	0.6324
2	Anaerobic	1.17E+06	0.6333
3	Landfill	1.19E+06	0.6340

Acknowledgements

My experience at Ingersoll Rand provided me with an immeasurable amount of knowledge in electrical, mechanical and heat transfer systems, as well as the integration of all the systems and the impact on the environment. I also learned how to work in an industry environment and how to work effectively with a team of many people with vastly different backgrounds. All of this experience has made me well-rounded and has given me the confidence to be a great engineer.

Without the guidance, patience, and work ethic of all the engineers, technicians, managers, and support staff, my experience would not have been possible. Thank you to everyone at Ingersoll Rand, especially my mentors Jeff Armstrong and Brian Finstad, who helped me grow and gave me some fascinating projects to work on. Thank you also to Gary Manter, who performed a large portion of the specialized mechanical testing (not detailed in this report).

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I am thankful for the experience and I highly recommend the UTSR program to any engineering student that has an interest in turbomachinery and wants to gain invaluable knowledge in the field.