DEVELOPMENT OF AN AIR-CYCLE ENVIRONMENTAL CONTROL SYSTEM FOR AUTOMOTIVE
APPLICATIONS

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TITLE: DEVELOPMENT OF AN AIR-CYCLE ENVIRONMENTAL CONTROL SYSTEM FOR AUTOMOTIVE APPLICATIONS

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Abstract

Development of an Air-Cycle Environmental Control System for Automotive Applications

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An air-cycle air conditioning system, using a typical automotive turbocharger as the core of the system, was designed and tested. Effects on engine performance were kept to a minimum while providing the maximum amount of cooling possible and minimizing weight and space requirements. A test stand utilizing shop compressed air was developed to measure component performance. An unmodified automotive turbocharger was tested initially as a baseline in a Reversed-Brayton Cycle air cooling system. Once the baseline was established, another air-cycle machine, assembled from commercial turbocharger components chosen individually to optimize their performance for cooling purposes, was tested to improve the overall cycle efficiency. Finally, once the air-cycle air conditioning system was optimized, it was tested on an engine to simulate more realistic operating conditions and performance. The shop-air test stand experiments showed a peak dry-air-rated (DAR) coefficient of performance (COP) of 0.38 and a DAR cooling capacity of 0.45 tons for the baseline turbocharger, and a peak DAR COP of 0.73 and DAR cooling capacity of 1.5 tons for the optimized system with a modified turbocharger. The on-engine testing was limited due to a thrust bearing failure in the ACM. However, the data collected at lower engine load and speed indicates a DAR COP of 0.56 and a DAR cooling capacity of 0.72 tons. On-engine testing was planned to include operating points where the stock turbocharger was utilizing turbine-bypass to limit boost pressure. While it wasn't possible to continue testing, it is expected that DAR COP and cooling capacity would have increased at higher engine load and speed, where turbine-bypass operation typically occurs.

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Nomenclature

P_{1C}	Compressor inlet pressure
P_{2C}	Compressor discharge pressure
P_{1T}	Turbine inlet pressure
P_{2T}	Turbine discharge pressure
T_{1C}	Compressor inlet temperature
T_{2C}	Compressor discharge temperature
T_{1T}	Turbine inlet temperature
T_{2T}	Turbine discharge temperature
PR_C	Compressor pressure ratio
PR_T	Turbine pressure ratio
c_p	Specific heat capacity
N_C	Physical compressor speed
N_C^*	Corrected compressor speed
N_T	Physical turbine speed
N_T^*	Corrected turbine speed
W_C	Physical compressor mass flow rate
W_C^*	Corrected compressor mass flow rate
W_T	Physical turbine mass flow rate
W_T^*	Corrected turbine mass flow rate
$\dot{W}_C$	Compressor Power
$\dot{W}_T$	Turbine Power
R	Gas constant
$\bar{R}$	Universal gas constant
M	Molecular weight
h	Specific enthalpy

Greek:

γ	Ratio of specific heats
----------	-------------------------

Acronyms:

ACM	Air-cycle machine
BSFC	Brake specific fuel consumption
CFM	Cubic feet per minute
DAR	Dry air rated
SCFM	Standard cubic feet per minute
LFE	Laminar flow element
CHRA	Center housing rotating assembly
CI	Compression ignition

SI	Spark ignition
COP	Coefficient of performance
VE	Volumetric efficiency
EGT	Exhaust gas temperature

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Chapter 1: Introduction

Automotive air conditioning systems are in the process of undergoing some major changes due to environmental concerns. The new changes are to prevent global warming, in addition to ozone depletion. These changes tend to have negative impacts on air conditioning performance due to a shift in refrigerant selection criteria, from maximum coefficient of performance (COP) to lifecycle environmental impact of the system. The new criteria considers designs that have lower COP but may still have less overall negative impact on the environment throughout the lifecycle of the system. Some of the other considerations in selecting a refrigerant, are the system weight, manufacturing costs, refrigerant toxicity, and effects of potential leaks. Looking into the future, additional changes to automotive air conditioning systems are likely, and use of a refrigerant that is naturally available and does not need to be retrofitted is convenient.

Air-cycle machines (ACM) conveniently use air as the refrigerant and have the potential to provide convenient air conditioning for high performance vehicles, where weight is at a premium. A typical ACM can consist of a compressor, heat exchanger, and cooling turbine, which is supplied with pressurized air. The main components of an air-cycle cooling system are the compression process, heat removal, and controlled expansion. The cool air from the turbine outlet can be routed directly into the passenger cabin.

In racing applications air conditioning is a secondary consideration in vehicle design, and often not used due to weight and power requirements. Any amount of cooling that can be provided without loss in engine performance and minimal weight increase is considered to be an improvement. A bleed-air control configuration is best-suited to racing applications because the amount of cooling required is not set, any cooling provided is beneficial. The bleed-air

configuration requires a forced-induction system as the primary source of compressed air for the Reverse Brayton Cycle to operate. This configuration will work with either a supercharged or turbocharged system. The latter is preferred to minimize the parasitic power loss on the engine caused by the ACM operation.

Instead of "waste-gating", a process in which a fraction of the exhaust to bypass the turbine stage to limit compressor outlet pressure, the compressor outlet pressure can be limited by bleeding air through the ACM. This has a reduced impact on power consumption from the engine because the power would have gone to waste if not used for cooling.

A variable nozzle turbine (VNT) can be used to achieve a desired cooling capacity while maintaining boost levels with increased primary compressor power requirements during full ACM operation by varying turbine power. This configuration can be designed to increase overall system efficiency by minimizing lower-loop losses, which is important for minimizing fuel consumption. The lower-loop losses can be seen on the P-v diagram of the Otto Cycle, in Figure 1.1. The lower-loop on either the Otto or Diesel Cycles represents the net pumping losses induced on the engine by the exhaust and intake systems. The net power can be determined by integrating the P-v curve forward through the intake and exhaust strokes. The sum of the signed areas results in the net power consumed. An intake pressure greater than the exhaust backpressure will result in a negative pumping loss, or increased engine power.

Figure 1.1 P-v diagram for an Otto Cycle.

The VNT control method was found to be the most practical to implement for this testing, and it demonstrates that the primary turbocharger can supply the necessary airflow to operate the ACM effectively in a bleed-air configuration. There are two objectives in this project; 1) demonstrate the feasibility of using commercially available automotive turbocharger components to design an ACM unit; and, 2) demonstrate that an engine-turbocharger system can be re-designed to support the additional airflow required by the ACM.

Chapter 2: Background

Air-cycle technology has been in use since the 1940's in aircraft environmental control systems (ECS) [1]. Until then it was considered commercially unfeasible due to its relatively low coefficient of performance (COP) compared to other refrigeration methods available, namely vapor-cycle systems and dry ice. The appeal that made it a consideration for use in aircraft is the size and weight of a typical ACM. In addition to this, there is a convenient source of compressed air on-board already, the jet engine's compressor stages. For comparable systems, in terms of cooling capacity, air-cycle systems can reduce space requirements by approximately 25% and weight by 50% [1]. This is important in racing and high-performance applications because engine bays are typically very tight on space, and weight can diminish acceleration and handling. The increased weight can be especially detrimental in vehicles with down-force since the increase in weight does not necessarily increase the lateral load capability by a proportional amount.

The air-cycle machine is based around the concept of a cooling turbine. This is the common component among all of the various ACM configurations, such as the simple, two-wheel, or three-wheel bootstrap cycles. The cooling turbine operates by providing a controlled expansion of air; an adiabatic, controlled expansion requires the air to do work on its boundaries to provide any cooling effect. The work done on the turbine wheel is transmitted by the shaft to a compressor or fan to utilize this power. The compressor and fan are methods of dissipating turbine power to do useful work in the system.

2.1 Simple Cycle

Figure 2.1 Simple air cycle machine schematic.

The main advantage to using the simple air-cycle system, as the name implies, is simplicity.

Decreased component numbers reduce weight and space requirements. The air coming from the main jet engine is typically in the range of 400-600°F. This air is then passed through a heat exchanger with high effectiveness to get the air cooled down to as close to the ram air temperature as possible. Once the air is cooled by the heat exchanger, it is introduced into the turbine for expansion. The power produced by the turbine is transmitted, by direct drive, to the fan. The turbine power corresponds to the enthalpy change in the air passing through it. There are two components to cooling the air; 1) Heat rejection from the ACM intercooler; 2) Turbine power derived from the airflow reduces enthalpy and air temperature. The fan helps improve heat exchanger effectiveness by increasing the airflow over the cold side of the heat exchanger. The fact that the simple cycle utilizes a fan and does not rely only on ram air for cooling the heat

Cooling Air Exi

instead of a fan, and a second heat exchanger. While this system relies on ram air for cooling and does not provide cooling while the aircraft is stationary, it does provide significantly lower turbine discharge temperatures. The turbine discharge temperature is low enough to have concerns about freezing the moisture out of the airflow. Although the ice does not form on the turbine wheel itself, it can build up to an amount that restricts airflow to the cabin, reducing the pressure ratio across the turbine and decreasing performance.

The air is drawn from one of the main engine compressor stages and cooled through a primary heat exchanger, just as the simple cycle does. Next, the cooler air enters the secondary compressor stage. This provides another pressure rise to further compress and heat the air. The power required by the secondary compressor to accomplish this task is provided by the cooling turbine. The air exiting the secondary compressor stage is cooled by another heat exchanger before entering the turbine. The turbine in the bootstrap configuration receives air at approximately the same temperature as the simple cycle, but at a significantly higher pressure ratio across the turbine. This is how the bootstrap air cycle machine is able to provide lower turbine discharge temperatures than the simple cycle.

2.3 Three-wheel Bootstrap Cycle

The three-wheeled bootstrap cycle is essentially the same as the two-wheeled assembly with the addition of a fan mounted on the same shaft as the ACM compressor and turbine or gear-driven from the shaft. The fan ensures adequate airflow on the cold side of the intercooler, but it will consume power from the turbine and reduce the pressure ratio across the compressor. This configuration requires one-off design and fabrication of ACM components that are not readily available with automotive parts.

2.4 Project Goals

The goal of this project is to demonstrate that an ACM air conditioning unit can be manufactured from existing automotive turbocharger components and provide acceptable cooling with a minimal performance penalty. There are many choices for ACM configuration, but some of them lend themselves more easily to automotive applications than others.

The simple cycle has few components, but the turbocharger compressor would have to be replaced by a fan. This requires a significant amount of fabrication and does not meet the goal

of using existing automotive turbocharger components. Using a fan mounted on the turbine shaft requires ducting that may be difficult to package in a car engine bay.

The three-wheel bootstrap configuration makes for a nicely-packaged rotating assembly. The main concern with this setup is the manufacturing and balancing of a complicated rotating assembly. This requires extensive modification to a turbocharger and is not appropriate for meeting the goals of this project.

The two-wheel bootstrap configuration requires more components, but there is a compressor-turbine assembly that allows the use of a turbocharger to package those components. An electric fan is needed to provide airflow over the intercooler at low vehicle speeds. This allows for simpler packaging in a small engine bay than ducting air from a remotely located fan. The two-wheel configuration provides the lowest turbine discharge temperature of all the setups. This is important in an automotive application because the pressure ratios available are limited due to engine constraints. A lower pressure ratio will reduce the temperature drop across the turbine, assuming everything else remains constant. The two-wheel bootstrap configuration is used for this project because it meets the goals that have been set. This configuration, in the context of the engine-mounted system to be tested in this project, can be seen below.

Figure 2.3 On-engine two-wheel bootstrap schematic.

The on-engine ACM configuration operates by bleeding air from the engine's forced-induction pressurized intake tubing to the intake manifold. The bleed-air configuration allows for control of intake manifold pressure or boost, a common name referring to intake manifold gauge pressure. Boost control is typically performed by using a pneumatic actuator that senses compressor outlet pressure and closes or opens a flapper valve located next to the turbine. This allows for a fraction of the exhaust gas to bypass the turbine, limiting turbine power and boost. The turbine bypass valve is commonly referred to as a waste-gate. It is referred to as "waste-gating" because any fraction of the exhaust that is allowed to bypass the turbine is wasted energy. The compressor bleed configuration uses this wasted energy to power the ACM while preventing excessive boost.

While the on-engine, two-wheel bootstrap configuration is the primary goal of this project, the turbine-compressor match must first be validated. This is performed on a stand-alone test stand with compressed air supplied to the ACM by the shop air compressor system. This configuration can be seen in Figure 3.1.

2.5 COP Defined

Traditionally COP is defined as the ratio of cooling load compared to the power required to drive the system. This is a practical definition for most air conditioning systems where the compressor, dynamic or fixed-displacement, is driven from a motor or engine. The power, with an associated cost, going to the air conditioning system is easier to determine than that of the turbocharger-based system described in this project since there is a physical connection between the motor and compressor.

$$COP_{traditional} = \frac{\dot{Q}_{cooling}}{W_{compressor,fans}} \text{ (Eqn. 2.1)}$$

This is not as clear when considering an ACM powered by the engine's turbocharger; not all of the power delivered to the ACM has a cost associated with it. A portion of the power that went to drive the primary compressor was from heat energy that would have been wasted through the exhaust anyways or lost through waste-gating.

A new method for determining power consumption specifically by the ACM is proposed. This can be done by finding the change in fuel flow rate with and without the ACM operating and using brake-specific fuel consumption (bsfc) to calculate power used by the ACM. This represents the effective power to drive the ACM, in other words, only power that has an associated increase in fuel cost. This applies to ACM operation outside of normal waste-gate

operation to limit boost. ACM operation to limit boost levels instead of using the waste-gate has minimal impact on engine fuel consumption, lending to a significantly increased COP.

$$COP_{effective} = \frac{\dot{Q}_{cooling}}{\dot{W}_{Engine-ACM}} \quad (\text{Eqn. 2.2})$$

$$\dot{W}_{Engine-ACM} = \frac{\Delta \dot{m}_{fuel, ACM}}{bsfc} \quad (\text{Eqn. 2.3})$$

$COP_{effective}$ is not the only concern in ACM design; the cooling capacity is the driving requirement in the design process. In optimizing the system, while meeting cooling capacity requirements, COP is maximized for a given set of ambient and ACM inlet conditions. The effective COP and cooling capacity together are referred to as ACM performance.

2.6 Previous work performed at Queen's University, Belfast

To the author's knowledge, there has only been one other recent group to perform air-cycle research for automotive applications. The idea of using air-cycle air conditioning for this type of application is well-founded by theory, but practical limitations in implementing the system can considerably decrease the performance of an ACM. This can be observed in the previous work described below.

A research group at Queen's University, Belfast, has designed and implemented a supercharger-based ACM for refrigerated trailers in road transport applications [2,3]. The group implemented a two-wheel bootstrap cycle similar to the one in this project. The primary compressor is gear-driven from the crankshaft, and it feeds compressed air to the typical bootstrap ACM. Instead of bleeding air from the engine intake manifold as done in this project, the supercharger is dedicated to supplying air to the ACM. This configuration somewhat decouples engine and ACM operation, making it easier to control and quantify ACM performance. It has a severe

shortcoming in that it will always be a parasitic power loss from the engine crankshaft, rather than “free” power from a turbocharger-based system that is demonstrated in this system. The theoretical COP of their ACM was 0.294, corresponding to the $COP_{\text{traditional}}$ defined above, while an optimized model showed a possibility of 0.62. Their performance goal was not achieved due to excessively low efficiency of the gearbox to drive the supercharger, heat exchanger performance, and excessively large turbocharger bearing losses. Overall, the fuel consumption of the engine was approximately three times greater with the ACM operating, and no actual COP figures were provided from this testing. While their testing did not show promising results, the lower-than-expected performance likely came from limitations in components selected rather than the air-cycle concept itself.

The turbocharger-based ACM differs from the supercharger-based unit because there is energy available that would have been wasted during the turbine bypass process, waste-gating, without the presence of the ACM. Instead of wasting this energy, it can be utilized for powering the ACM without any additional cost to the engine. This is essentially free cooling, or infinite COP. Since the use of a turbocharger as the air source allows increased upper limits of performance than the supercharger-based system, there is an improved likelihood of success.

2.7 Theoretical ACM Performance

Air-cycle analysis can be performed assuming dry or moist air. The analysis for this project assumes dry air, except for using the dry-air rated (DAR) analysis for cooling capacity. DAR temperature is the equivalent temperature if the entrained water or ice in the air exiting the turbine is adiabatically evaporated [4]. Dry air is assumed because, for most of the system, the difference in enthalpy change across components is minimal between the two methods. This, however, is not true for the turbine because the air temperature falls below that of the ambient

conditions and moisture can drop out as a liquid, or possibly ice. This can have a significant impact in the enthalpy change across the turbine, affecting turbine power and discharge temperature. Dry air is assumed for simplicity in designing the first prototype. Detailed derivations of the dry-air, perfect gas equations can be found in Appendix B.

The turbine-compressor matching process for an ACM is different than that of a match performed for an engine application. This is primarily due to the difference in turbine inlet conditions. Corrected parameters, such as flow and speed, for the turbine are dependent upon temperature. The turbine inlet temperature is significantly lower for an ACM than typical exhaust gas temperatures of either diesel or gasoline engines. This presents a challenge because commercial turbochargers are designed for engine applications, and it requires a new compressor to be matched to the turbine to maximize performance in the ACM application. The compressor-turbine matching process can be found in Appendix C. The differences in the typical on-engine application and the ACM application become apparent when viewing a T-s diagram with both the Brayton and Reverse-Brayton cycles. The Brayton Cycle is modeling the engine as a heat input to the turbocharger system. Figure 2.4 shows the relative temperature and entropy changes through each cycle.

The process starts at the compressor inlet and the volume decreases as the pressure increases. The Brayton Air-Standard Cycle is modeled with the engine as a continuous thermodynamic machine and with a mass-fraction bypassing the turbine to prevent over-pressurizing the intake. The intake air is then drawn into the engine where it goes through the combustion process where the burned fuel adds heat to the flow before it is discharged into the exhaust manifold. The exhaust flow increase volume and decreases in pressure as it expands through the turbine. A line connecting the turbine outlet and the compressor outlet represents the air possibly being

recycled at some point and closes the cycle. The Reverse-Brayton Cycle involves much less energy than compared to the Brayton Cycle. This results in changes in corrected conditions for the compressor and turbine, lower compressor pressure ratios, and requires a somewhat different approach to the compressor-turbine matching process. A relatively larger compressor wheel proves to be a better match for reducing excessive corrected turbine speeds for maximizing turbine efficiency, while still keeping compressor efficiency in mind.

Figure 2.4 Comparison of the Reverse-Brayton Air Cooling Cycle and the Brayton Cycle.

There are a few things known about the system, as with many systems, such as mass balance, energy balance, and operating speeds of the compressor and turbine wheels. Assuming there are no leaks in the system, the mass flow rate through the compressor and turbine will be the same. With the assumption that the bearing losses are negligible, which is the case in most

engine applications where the uncertainty of many other parameters outweigh this, the power output of the turbine will be matched by the compressor power requirement in steady-state operation. Considering the compressor and turbine wheels are directly mounted on the same shaft, the shaft speeds will be equal. Even with as many known parameters as there are, it is an iterative process to find a mass, power, and speed balance because of the graphical nature of the compressor and turbine performance maps.

It is important to investigate the effects of various component performances on overall ACM performance. A parametric study of ACM component efficiencies was conducted and the results are in Figure 2.5-2.6 and Table 2.2. With the results from the parametric study, attention can be given to the most critical components first. The analysis presented next shows that heat exchanger effectiveness (E_{htxr2} , Figure 2.5-2.6) and turbine efficiency (η_{t2} , Figure 2.5-2.6) are more important to overall system performance than compressor efficiency. The data in the following figures are calculated assuming a primary compressor efficiency of 70% and ambient conditions similar to those found at the testing location. The compressor efficiency chosen is representative of attainable performance over a wide range in air flow in modern automotive turbocharger systems [6]. The code used to generate the following plots can be found in Appendix K.

Table 2.1 Ambient conditions for theoretical performance model.

Ambient Condition	
Temperature [Deg. F]	70
Pressure [psia]	14.69
Relative Humidity [-]	0.50

Figure 2.5 Parametric study of ACM performance for a compressor with 60% efficiency and varying levels of turbine efficiency and heat exchanger effectiveness.

Figure 2.6 Parametric study of ACM performance for a compressor with 80% efficiency and varying levels of turbine efficiency and heat exchanger effectiveness.

This analysis indicates that minimum acceptable intercooler effectiveness is approximately 0.70 before rapidly decreasing overall ACM performance. While compressor and turbine efficiencies remain constant, the curves for ACM COP decrease by larger amounts as heat exchanger effectiveness decreases.

Using representative values from the previous analysis, attainable performance with automotive turbocharger components, for intercooler effectiveness, turbine efficiency, and compressor efficiency, the following table was constructed:

Table 2.2 Tabulated data from parametric study of performance analysis for a relative humidity of 50%.

Pressure Ratio [-]	Intercooler Effectiveness [-]	Turbine Efficiency [-]	Compressor Efficiency [-]	COP_{DAR} [-]
1.5	0.80	0.80	0.60 → 0.80	1.05 → 1.25
		0.60 → 0.80	0.80	0.82 → 1.25
	0.60	0.80	0.60 → 0.80	0.86 → 1.07
		0.60 → 0.80	0.80	0.61 → 1.07

It can be observed that the turbine performance is more critical to the overall cycle performance. This is to be expected because the more efficient the turbine is, the more heat will be removed from the air expanding through the turbine, and the more power will be extracted to drive higher pressure ratios across the compressor. The higher pressure and temperature entering the heat exchanger allows more heat to be removed before entering the turbine. Obviously, there is a limit to this cycle and a steady-state operating point because of the decreasing energy content in the air flow to the turbine as more heat is removed by the intercooler.

It should be noted that the ACM in this testing is operated at the lower pressure ratio range due to boost limitations on the test engine and typical boost pressures for gasoline engines. The theoretical analysis is extended to the upper limits of pressure ratios that can be achieved by

single stage compression. This is representative of what can be seen in operating intake pressures in diesel engines.

Chapter 3 Experimental Apparatus, Procedures, and Conditions

There are two testing configurations needed to complete testing for this project. The first is designed to verify the compressor-turbine match for the ACM itself and to demonstrate that the desired performance can be achieved. The second test is designed to demonstrate on-engine performance.

3.1 Shop-air Test Configuration

Figure 3.1 Shop-air testing configuration

The first set of tests consists of the ACM turbocharger, intercooler, and required ducting between components powered by shop air. Figure 3.1 shows the connection to the compressed air source and the locations of the sensors. The test facility has a 75-hp reciprocating compressor and large air storage tanks. A relatively large shop air supply is needed because the compressor has to supply the tanks with 125psig, and the ACM air supply is regulated down to a

nominal 11psig. This pressure represents the ACM inlet conditions that will be similar to operation with the diesel engine. With this in mind, there is not an exact method for finding primary compressor power for COP calculations. To represent realistic primary compressor performance in a typical turbocharger application, an isentropic compressor efficiency of 70% is assumed. This assumption applies to the following equation to calculate the air source compressor power requirements, solely for the purpose of COP estimation:

$$\dot{W}_c = \dot{m} c_p \frac{T_1}{\eta_c} \left[P R_c^{\gamma-1/\gamma} - 1 \right] \quad (\text{Eqn. 3.1})$$

This testing configuration allows for a baseline to be established with the unmodified, Garrett GT1241, turbocharger. Garrett Turbochargers manufactures a large range of turbocharger sizes and many combinations of compressors and turbines. The turbocharger base model selected has the smallest turbine in the product lineup, but it closely matches the requirements for this project. The next step is to optimize performance for an ACM application, rather than a gasoline engine application. This involves sizing another compressor for the GT12 turbine. The compressor-turbine matching process is discussed in greater detail in Appendix C.

Figure 3.2 Physical ACM test stand using shop air.

Type-T thermocouples were used because they work well with the lower temperature range that the ACM will be operating in. In addition to this, the standard limit of error for type-T thermocouples is 1°C , compared to 2.2°C for type-K. Each thermocouple was calibrated within the standard limit of error. All sensors were checked periodically between sets of experiments to ensure quality data. The calibration data are in Appendix E.

This testing configuration gathered data for a baseline using an unmodified GT12 turbine-41mm compressor wheel assembly for the ACM and a GT12 turbine-44mm compressor wheel assembly for improved efficiency. The larger compressor reduced corrected turbine speeds to improve performance. The 41mm compressor wheel placed corrected turbine speed operating point off the high end of the turbine performance map. The compressor provided similar efficiency contours to the smaller compressor wheel, so there wasn't a significant decrease in performance on that end. The differences in the unmodified GT1241 and modified GT1244

compressor-turbine matches are due to the differences in the component inlet operating conditions for the intended applications, as discussed in Section 2.7.

The installation of the new compressor required manufacturing a shaft adapter due to a larger bore size in the larger compressor wheel. The specifications from Garrett Turbochargers indicated that bore inside-to- outside diameter runout and perpendicularity to the end faces need to be within 0.0001 inch. A shaft adapter was produced that met the acceptable vibration limits at peak operating speeds. This was verified before continuing with testing by slowly increasing the turbocharger speed, while directly measuring shaft speed and bearing housing vibration levels.

The procedure for performing the tests on both the unmodified GT1241 and modified GT1244 assemblies included: blowing out liquid water from the shop air lines and tanks, gathering local ambient conditions, and manually regulating the ACM inlet pressure to achieve desired operating points and steady-state conditions before collecting each data set. Steady-state operating conditions can be detected and verified several ways, such as monitoring shaft speed, temperatures, and pressures.

The component that contributed most to the length of time required to reach steady state was the intercooler. It has a relatively large mass and took some time to reach a steady temperature. It is very important to wait for steady-state conditions before collecting data because fictitiously high performance numbers can be observed. This is because of the thermal capacitance of the aluminum comprising the intercooler core and end caps; it provides a second means of heat removal from the airflow until the intercooler is saturated with heat. Once the intercooler is "heat-soaked", the only means of dissipating heat is to the cold air flow stream, instead of transferring heat from the hot air stream to the thermal-capacitance of the aluminum

and to the cold air flow stream. Initial testing showed large apparent intercooler effectiveness during the transient period. All data was collected at steady-state conditions for this project, eliminating this effect.

The shop air stand testing demonstrated that the baseline GT1241 turbocharger provided an adequate compressor-turbine match, while the modified GT1244 turbocharger provided a larger useable ACM operating range by increasing the efficiency over that range closer to the peak efficiency. The results of the shop air test stand experiments are discussed in Section 4.1 and Section 4.2. The uncertainty calculations for these measurements can be found in Appendix A. With the compressor-turbine match verified and the performance of the ACM optimized, the next step is on-engine testing.

3.2 On-engine Testing Configuration

Figure 3.3 On-engine testing configuration schematic

The on-engine configuration uses the same ACM test stand configuration with the exception of the compressor inlet being fed compressed air from the intake manifold of the diesel engine. This was achieved using a y-pipe after the primary, or engine, intercooler. The flow control valve is located before the ACM compressor inlet and is capable of turning the flow to the ACM on or off and throttling flow. The connection from the engine to the ACM is illustrated in Figure 3.3.

The engine used in this experiment is a six liter Perkins diesel engine. It is a four-stroke, inline six-cylinder engine. This choice of engine platform is relevant to sports car and racing applications because the engine displacement is similar, even though the speed range is lower. This is not necessarily a problem because the engine turbocharger's turbine is sized for the engine in consideration, and in the worst case, it will provide a lower limit for load and speed that the ACM can be effectively operated. Since, in racing applications, the engine is typically at higher load and speed, this is not a concern.

The engine is fully-instrumented and is connected to a dynamometer. The engine-dynamometer setup is part of an engine's class, and the sensors are calibrated. A calibration log is maintained, but even so, the sensors were verified to match their previous calibrations.

The Perkins diesel engine was first tested with the stock turbocharger to set a baseline for comparison of the new compressor for the primary, or engine, turbocharger. The second turbocharger is sized to efficiently accommodate the additional air flow required for the operation of the engine and ACM. The new turbocharger has a larger compressor and a variable nozzle turbine.

The experiments performed to assess the effects of ACM operation on engine performance were conducted by running the engine at an operating point allowing maximum boost levels to

be reached before opening the diverter valve to the ACM. This demonstrates the effect of ACM system air flow requirements on boost level available to the engine and change in fuel consumption, the two primary factors considered in a racing application.

3.3 ACM Turbine-Compressor Matching

The matching process for the ACM rotating assembly follows the method presented in Appendix C, but there are some new considerations regarding the ACM application. The turbine and compressors that are paired on production turbochargers are well-matched for gasoline or diesel engine applications, where there is hot exhaust gas driving the turbine. This temperature difference in the engine and ACM applications causes a large change in corrected turbine speed and corrected mass flow rate, two parameters used to map turbine performance. Since the ACM turbine inlet conditions will always be cooler than the original engine application, a larger compressor will be needed to reduce corrected turbine speeds. The compressor matching technique is essentially unchanged, since the compressor conditions are similar in both applications.

3.4 Primary Compressor Resizing for ACM operation

Resizing the primary engine turbocharger compressor was required to accommodate the additional airflow required during ACM operation but still has a sufficient surge margin for safe engine operation. The method employed in sizing a new compressor for this project split the operating points symmetrically around the imaginary line that passes through the center of the efficiency contours up the map. This is graphically demonstrated below.

Figure 3.4 Primary engine compressor matching process with an engine operating point of 1600rpm and full load.

Splitting the operating points around the peak efficiency line provides the best average efficiency if the ACM was operated the same amount of time as it was off. This is not likely to be the case. However, since this project does not involve a specific application, the actual duty cycle is unknown and a 50% duty cycle is assumed. The split on the actual compressor match was not 50% due to available commercial compressor assemblies. Therefore, the nearest available compressor size was chosen. The primary compressor efficiency remains nearly constant whether or not the ACM is operating by placing the operating point on each side of the nearest efficiency contour. There is opportunity for optimizing the compressor selection process, but detailed, application-specific data is required. The compressor operating boost curves for this project are shown in the figure below.

Figure 3.5 Primary engine compressor match with the Perkins diesel boost curve and the target boost curve with the ACM operating.

The compressor selected for the Perkins diesel engine allows for full engine operation with and without the ACM operating. This match provides a sufficient surge margin and low risk of compressor overspeed, while optimizing efficiency as much as possible for this project.

Chapter 4: Results, Conclusions, and Future Work

4.1 Results and Discussion from Unmodified GT1241 Shop Air Test Stand

The unmodified GT1241 turbocharger was used as a performance baseline for comparison of modifications to the turbocharger, and it performed better than expected. The ACM performance is shown in Figures 4.1 and 4.2. An accurate prediction of performance was hard to obtain due to limited compressor and turbine performance data from the manufacturer. This is because the ACM operating conditions, specifically turbine inlet temperature and pressure, caused an increase in corrected turbine speed, compared to an on-engine application. The ACM operating conditions put the target corrected turbine speed outside of the known turbine performance map. However, the increased corrected turbine speed did not cause a sharp turbine efficiency drop, which was likely to occur due to excessive speed from the relatively small compressor. The limiting factor in this configuration was the intercooler, which can be seen in Figure 4.3. The relatively low intercooler effectiveness limited heat removal from the air entering the turbine. This is detrimental to performance as an increase in turbine inlet temperature will generally cause an increase in turbine outlet temperature, decreasing the cooling capacity of the ACM. The data points in Figures 4.1-4.3 were collected simultaneously, so the mass flow rate can be related to the ACM pressure ratio, DAR COP, and DAR cooling capacity. For example, the third point from the right in Figure 4.1 corresponds to the third point from the right in Figure 4.2-4.3.

Figure 4.1 Dry air rated COP for the unmodified GT1241 turbocharger. Data points in Figures 4.1-4.3 have a one-to-one correspondence.

Figure 4.2 Dry air rated ACM cooling capacity. Data points Figures 4.1-4.3 have a one-to-one correspondence.

The cooling capacity required in a performance automotive application is approximately one ton of cooling. This was determined from reviewing R134a compressor performance maps from Sanden [Appendix H]. This is discussed in greater detail in Section 4.4.

Figure 4.3 Intercooler effectiveness variation with mass flow rate in the GT1241 ACM with the electric fan.

Data points Figures 4.1-4.3 have a one-to-one correspondence.

The ACM intercooler effectiveness was limited because of poor electric fan performance. The heat exchanger performance was shown to have a significant impact on overall ACM performance in Section 2.7, so this was remedied in later tests by manually regulating compressed air to pass over the cold side of the heat exchanger. However, to continue with testing on schedule, the next compressor was installed before the heat exchanger performance was improved. The DAR COP has a peak because as the pressure ratio across the ACM starts at unity and increases, both the ACM compressor and turbine will pass through their peak efficiencies. The DAR cooling capacity will increase until both the compressor and turbine have

passed their peak efficiencies and begin to decline in performance as the pressure ratio across the ACM is further increased.

The peak dry air rated COP, with the 70% primary compressor efficiency assumption, is on par with a COP of the theoretical optimization of the ACM from Queen's University, Belfast [3], a COP of 0.62. However, the maximum cooling capacity does not occur at the maximum COP; the dry air rated COP falls to approximately 0.48 at that point.

4.2 Results and Discussion from Modified GT1244 Shop Air Test Stand

The initial testing of the GT1244 turbocharger used the same intercooler-fan assembly as the previous test to have a direct comparison of a single component change in the system. The change in the compressor wheel and housing did not change the peak dry air rated COP of the ACM, but it did greatly expand the useful operating range of the ACM and increase the system cooling capacity, pushing it closer to one ton of cabin cooling. This can be seen in Figures 4.4-4.5. The performance improvements from the compressor wheel and housing change from the unmodified GT1241 configuration are shown below. The intercooler effectiveness is approximately the same as in the unmodified GT1241 testing, Figure 4.6. The intercooler is still a limiting factor. However, with its performance approximately the same as in the GT1241 ACM testing, the results still show a relative improvement of the 44mm compressor wheel over the previous 41mm compressor wheel. The data points were collected simultaneously during this testing, and each point corresponds to the same data point number in the other plots in Figure 4.4-4.6.

Figure 4.4 Dry air rated COP for the modified GT1244 turbocharger. Data points Figures 4.4-4.5 have a one-to-one correspondence.

Figure 4.5 Dry air rated ACM cooling capacity. Data points Figures 4.4-4.5 have a one-to-one correspondence.

Figure 4.6 Intercooler effectiveness variation with ACM mass flow rate using the electric fan. Data points

Figures 4.4-4.5 have a one-to-one correspondence.

Next, the intercooler cold-side airflow problem was fixed. This was achieved by routing manually-regulated compressed air over the cold side. The airflow over the cold side of the intercooler was kept within reasonable levels that could be obtained with a typical configuration seen in automotive applications. This was determined to be approximately 350 CFM, and this agrees with volumetric flow rates observed in R134a condensers [9]. Based on intercooler cold flow entrance area, this results in an average air velocity of approximately 8.5 ft/s entering the face of the intercooler core. This is even easier to achieve when the vehicle would be moving. The results from the improved intercooler configuration can be seen below. The intercooler performance increase improved both ACM COP and DAR cooling capacity, Figure 4.7-4.8. The increase in intercooler effectiveness can be seen in Figure 4.9. The data points were collected

simultaneously during this testing, and each point corresponds to the same data point number in the other plots in Figure 4.7-4.9.

Figure 4.7 Dry air rated COP for the modified GT1244 turbocharger. Data points Figures 4.7-4.9 have a one-to-one correspondence.

Figure 4.8 Dry air rated ACM cooling capacity. Data points Figures 4.7-4.9 have a one-to-one correspondence.

Figure 4.9 Intercooler effectiveness variation with ACM mass flow rate. Data points Figures 4.7-4.9 have a one-to-one correspondence.

The ACM intercooler proved to be a critical component in overall ACM performance. The improved intercooler effectiveness approximately doubled the dry air rated COP and cooling capacity.

The theoretical performance analysis was shown to be a good approximation of the actual performance of the ACM. The flow prediction from the compressor-turbine match and the turbine pressure ratio-flow profile and actual flow conditions are over-plotted on the compressor performance map for the 44mm compressor wheel. At lower pressure ratios, all measurements and corresponding predicted values agree within 7% difference or better. At higher pressure ratios across the ACM the dry-air rated COP and DAR cooling capacity calculations both matched actual performance within approximately 13%, but the mass flow rate deviates more. The mass flow rate deviates up to 12% difference from the measured mass flow rate. At this point, the pressure ratio across the ACM is approximately 1.84. The results of the compressor operating points comparison can be seen in Figure 4.10. The DAR cooling capacity and DAR COP comparisons agree well and are shown in Table 4.2.

Figure 4.10 Locus of theoretical and actual operating points over-plotted on GT1244 compressor map.

Since the operation of the ACM placed the operation of the ACM at the outer limits of performance data collected by the turbocharger manufacturer, limited speed, power, and mass balance points could be obtained. If more component performance data was available, a more complete locus of operating points could be obtained with the computer simulation. Even with limited data, the part of the locus of operating points that was obtained from the computer simulation matches well with the experimental data. The computer simulations, in both Matlab and Engineering Equation Solver (EES), use the equations derived in Appendix B and the matching process described in Appendix C. A comparison of the theoretical and actual

operation using the conditions specified in Table 4.1 results in the performance presented in Table 4.2.

Table 4.1 Operating Conditions for single point comparison of theoretical and actual performance.

<u>Ambient and Operating Conditions</u>	<u>Value</u>	<u>Units</u>	<u>Uncertainty [+/-]</u>
Ambient Temperature	531	[Deg. R]	1.7
Ambient Pressure	14.69	[psia]	0.02
Ambient Relative Humidity	0.4	[-]	0.08
ACM Compressor Inlet Temperature	526	[Deg. R]	1.7
ACM Compressor Inlet Pressure	26.95	[psia]	0.02
ACM Compressor Efficiency	0.72	[-]	0.02
ACM Compressor Outlet Pressure	37.08	[psia]	0.02
ACM Intercooler Effectiveness	0.83	[-]	0.04
ACM Intercooler Pressure Drop	0.3	[psi]	0.03
ACM Mass Flow Rate	13.73	[lbm/min]	0.6
ACM Turbine Inlet Temperature	536	[Deg. R]	1.7
ACM Turbine Efficiency	0.56	[-]	0.02

Table 4.2 Results of single point comparison of theoretical and actual performance.

Performance Parameter	Theoretical Operation	Actual Operation	Uncertainty [+/-]
COP, Dry Air Rated [-]	0.61	0.64	0.04
Cooling Capacity, Dry Air Rated [ton]	1.7	1.5	0.06

4.3 Results and Discussion from GT1244 On-Engine Testing

The on-engine testing was limited because of a bearing failure in the ACM turbocharger.

However, one operating point was tested before the failure occurred. The testing that was completed before failure was still a success with good performance. The COP based on change in bsfc and fuel flow rates indicated a COP of 0.56, with a corresponding 0.72 tons of cooling capacity. The theoretical optimization of the ACM from Queen's University, Belfast had a COP of 0.62, while actual performance being much less [2,3]. Their actual performance number was not stated, other than being less than expected. The COP for the on-engine testing for this

project is assuming the ACM is running at a time the engine's turbocharger would not have been waste-gating, otherwise the COP is essentially infinite because the potential power would have been wasted in the turbine bypass free expansion process.

The testing conducted with the ACM being powered by the Perkins diesel's turbocharger had reduced performance from the shop air test stand. The DAR cooling capacity decreased from approximately 0.90 to 0.72 tons, and the DAR COP decreased from approximately 0.73 to 0.56 for corresponding operating points of the shop-air testing of the GT1244 and the on-engine testing. This is due to much higher ACM inlet temperatures coming from the intake manifold of the engine. The on-engine testing conditions represent real world operation more accurately than the shop air test stand.

4.4 Comparison of ACM to R134a Systems

The target cooling capacity for the ACM was determined by observing the average cooling capacity based on the Sanden performance charts of the SD5H09 R134a system compressors [Appendix H]. This performance chart indicates the cooling capacity of a typical R134a system using the Sanden compressor. The cooling capacity will vary with suction superheat and operating pressures. The average range was selected from the Sanden chart to represent the target cooling capacity. This is approximately one ton of cooling capacity (12000 BTU/hr). The Sanden performance chart was used because Sanden is a worldwide manufacturer of automotive air conditioning compressors, used in all types of vehicles [10] and would create a good approximation of the cooling requirements for a high-performance vehicle. While the GT1244 ACM fell a little bit short of the cooling capacity goal, the R134a systems are pre-charge pressure and engine speed sensitive. Their actual capacity varies somewhat and are seldom used at full duty cycle at all times (i.e. cabin fan speed and temperature shutoff switch).

The system weight for a typical R134a system varies between 60-75 lbf¹. The ACM system weight and estimated bracket weights totals at 40lbf. This is approximately a 40% weight reduction. The physical size and number of components of the system is smaller than compared to R134a or Freon systems [1].

Table 4.3 ACM component weight breakdown.

<u>Component</u>	<u>Weight [lbf]</u>
Turbocharger Assy.	10
Intercooler Core	10
IC end caps	4
electric fan	5.6
ducting	2.5
Clamps	1
silicone connectors	2
Brackets (for vehicle)	5
Total	40

Another aspect of the ACM that differs from typical R134a air conditioning systems is that the ACM does not always induce a parasitic loss on the engine crankshaft, as opposed to a belt-driven R134a compressor. This can be seen on a P-v diagram for a four-stroke engine.

¹ 2008 e-mail to Vintage Air; unreferenced

Figure 4.11 P-v diagram for four-stroke gasoline engine.

The lower loop on the P-v diagram is an indicator of the pumping losses on the engine. The pumping losses are the area under the exhaust line minus the area under the intake line in Figure 4.11 or Figure 1.1. It should be noted that in Figure 1.1, the exhaust pressure is greater than the intake pressure. A positive lower-loop loss is typical of naturally-aspirated engines. In Figure 4.11 the net pumping losses are negative, indicating that a forced-induction system is creating a higher engine intake manifold pressure than exhaust backpressure. Turning the R134a system compressor on will induce a load on the engine and always hurt engine performance. On the other hand, a turbocharger can actually create a higher intake pressure than exhaust back pressure. This is due to the turbine recovering wasted exhaust heat, with some exhaust backpressure, providing enough work to the compressor to deliver higher than exhaust pressures to the engine's intake manifold. This helps to minimize engine bsfc.

4.5 Conclusion

The shop-air stand testing successfully demonstrated that an ACM can be designed around automotive turbocharger components. The use of automotive turbocharger components is critical to reducing cost in development of commercial systems for automotive use. An off-the-shelf turbocharger can be used in certain applications without modification and provide adequate performance. The testing on the unmodified GT1241 turbocharger indicated a peak DAR COP of approximately 0.38, and a DAR cooling capacity of 0.45 tons. The performance was somewhat limited due to the intercooler effectiveness initially, in the first two experiments of the GT1241 and GT1244, before the heat exchanger issue was remedied. It is expected that if the intercooler effectiveness was increased with this turbocharger configuration, the performance would be significantly increased, as seen in the modified GT1244 testing.

The GT1244 was optimized over the baseline GT1241 turbocharger. The ACM turbine and compressor were selected for the ACM operating conditions, as opposed to the typical on-engine application these turbochargers are normally used for. Initially, the peak DAR COP and DAR cooling capacity was approximately the same as the baseline GT1241 configuration. However, even with the intercooler performance being limited, the operating range was significantly increased (Figure 4.4-4.5). Once the intercooler effectiveness was increased by providing adequate airflow across the cold side of the heat exchanger core, the peak DAR COP reached 0.73 with a peak DAR cooling capacity of 1.5 tons (Figure 4.7-4.9). The DAR COP figures for the shop-air testing were determined by assuming a primary compressor efficiency of 70%. This primary compressor efficiency is typical of modern turbochargers in application. For reference, the compressor used for the on-engine testing in this project has an isentropic efficiency of 70%, or greater at the target boost pressure of 11psig, over a mass flow range of approximately 16-32 lbm/min.

The on-engine testing has shown that an air-cycle air conditioning system can be effective for racing applications, where there is plenty of opportunity to take advantage of time that would have otherwise been spent waste-gating. The testing performed prior to the ACM thrust bearing failure indicated a DAR COP of 0.56 and DAR cooling capacity of 0.72 tons. This operating point was not at full boost pressure due to the lower engine load and speed operating point initially tested, and the full benefit of the setup, operation during typical waste-gating conditions, could not be tested.

In comparison to previous work at Queen's University, Belfast, the ACM in this project performed well. The theoretical COP of the previous work was 0.294 [1,2]. They performed a theoretical optimization of the same system with an expected COP of 0.62. The theoretical optimization used compressor and turbine efficiencies of state-of-the-art components. However, their final results from actual testing were not presented other than a short statement saying that fuel consumption of the engine increased over three times the amount for the vapor-compression cycle they implemented as well. The group attributed the less-than-expected performance to excess losses in components such as the ACM bearings and primary compressor on the engine.

If the ACM bearing failure had not occurred, the next operating point would have been at higher load and speed to provide a better comparison to the waste-gated turbocharger operation. The operating point tested corresponds to a point where the stock turbocharger was not yet waste-gating. This means that the boost pressure can be increased to the original limit, resulting in increased cooling capacity due to increased pressure ratio across the ACM. Also, the effects on engine performance can be decreased because the ACM will be consuming a smaller fraction of the overall engine airflow. At larger engine loads and speeds, there is significant excess turbine

power that would normally be limited by waste-gate operation. Factors that influence turbine power are turbine mass flow rate, exhaust gas temperature, and pressure ratio across the turbine. Both turbine mass flow rate and exhaust gas temperature typically increase with engine load and speed. With this in mind, engine operating points corresponding to waste-gate operation are expected to have increased DAR COP following the proposed definition of $COP_{\text{Effective}}$ in Eqn. 2.2.

The engine turbocharger matching criteria has potential to be further optimized for specific applications. This will minimize effects of ACM operation on engine performance. The nominal ACM mass flow rate relative to the engine mass flow rate at typical engine operating conditions is an important consideration because too large of an ACM and the primary compressor will not be able to maintain desired boost pressures at higher combined system mass flow rates. The air flow to the ACM can be throttled, but this will reduce the potential cooling capacity of an ACM being designed for the vehicle. While throttling the ACM may be necessary in some engine operating conditions, it is not optimal for performance and should be avoided as much as possible.

4.6 Future Work

While there is a considerable amount of work to still be done in designing an ACM for production, most of it can be easily handled by a turbocharger manufacturer. The conditions the ACM turbocharger compressor is operating in are similar to a second turbo in a series turbocharger setup used on modern diesel engines to more efficiently handle increased boost pressures. The main consideration is the bearing system for the ACM. The bearing system experiences higher thrust loading because it needs a larger compressor wheel relative to the turbine wheel. This is because the compressor inlet sees higher than atmospheric pressures

while the turbine still exits to ambient pressure, and with a larger compressor wheel, there is increased area for the inlet pressure to act upon. Stability of the rotating assembly is the first priority. However, the bearing system efficiency is increasingly critical in low-temperature applications because there is less power being transmitted by the shaft, and the same amount of bearing drag will now be a larger fraction of the power transmitted. To increase stability and decrease bearing losses, magnetic or air bearings may be considered [5,7].

The ACM turbocharger can operate in a smaller range than a typical on-engine application, and this allows for more optimization of the compressor and turbine designs for higher peak efficiency at the cost of usable mass flow rate range [8]. Once a particular application is well-defined, the compressor and turbine can be designed to operate in a narrower range that is adequate for the application. This requires a considerable amount of research and development and would only be practical for larger production quantities.

To completely prove out the concept with specific packaging requirements, an in-vehicle demonstration would be ideal. The type of vehicle could be anything performance-oriented with a medium to large engine, due to the minimize size of turbochargers available for ACM design. The turbocharger used for the ACM in this project is the smallest commercially available from Garrett Turbochargers. The in-vehicle demonstration would allow for an operation duty-cycle to be recorded and facilitate in optimizing the engine turbocharger's compressor matching process.

References

1. Scofield, Paul C. (1949). Air Cycle Refrigeration. *Refrigerating Engineers*, 57, [558-563, 611-612].
2. Spence, Stephen W.T., Doran, W. John, & Artt, David W. (2004). Design, construction and testing of an air-cycle refrigeration system for road transport. *International Journal of Refrigeration*, 27, 503-510.
3. Spence, Stephen W.T., Doran, W. John, Artt, David W., & McCullough, G. (2005). Performance analysis of a feasible air-cycle refrigeration system for road transport. *International Journal of Refrigeration*, 28, 381-388.
4. Arora, C P. (2000). Refrigeration and air conditioning. West Patel Nagar: Tata McGraw-Hill Publishing Company Limited.
5. Murray, Charles J. (1994). Magnetic Bearing Improves Air Cycle Cooling Reliability. *Design News*, 49, 85-86.
6. *Turbo Tech 101*. Retrieved from http://www.turbobygarrett.com/turbobygarrett/tech_center/turbo_tech101.html
7. Burgmeler, Lyman, & Poursaba, Matt (1994). Ceramic Hybrid Bearings in Air-Cycle Machines. *International Gas Turbine and Aeroengine Congress and Exposition*, 94-GT-393, 1-9.
8. Dresser-Rand, Olean & Wellsville Operations. (2006). *Range Versus Efficiency - A Dilemma for Compressor Designers and Users* [Brochure]. Olean, NY: James M. Sorokes.
9. Hosnoz, M., & Direk, M. (2006). Performance evaluation of an integrated automotive air conditioning and heat pump system. *Energy Conversion & Management*, 47, 545-559.
10. Retrieved from www.sanden.com

Appendix A – Uncertainty Analysis

Uncertainty of a dependent variable $F(x,y,z)$, in general can be calculated as:

$$u_F = \left[\left(\frac{\partial F}{\partial x} u_x \right)^2 + \left(\frac{\partial F}{\partial y} u_y \right)^2 + \left(\frac{\partial F}{\partial z} u_z \right)^2 \right]^{1/2} \quad (\text{Eqn. A.1})$$

where u_F is the uncertainty in the parameter of interest, such as compressor efficiency, and u_x , u_y , and u_z are the measurement uncertainties.

Table A.1 Measurement Uncertainty Based on Calibration Instrumentation

Measurement	Measurement Uncertainty
Pressure	0.02 [psi]
Temperature	1.7 [Deg. F]
Volumetric Flow Rate	approx. 5 [CFM]

A.1 Compressor Efficiency

The compressor efficiency uncertainty was calculated using conditions representative of ACM operating conditions. In doing so some assumptions are required. It is assumed the compressor and turbine efficiency are 70% and the intercooler effectiveness is 70%.

The compressor efficiency can be calculated as:

$$\eta_C = \frac{T_{1c} \left(PR_c^{\gamma_c - 1 / \gamma_c} \right)}{T_{2c} - T_{1c}} \quad (\text{Eqn. A.2})$$

The partial derivatives are calculated:

$$\frac{\partial \eta_C}{\partial T_{1c}} = \frac{\left[\left(\frac{P_{2c}}{P_{1c}} \right)^{\gamma_c - 1 / \gamma_c} \right]}{(T_{2c} - T_{1c})} + \frac{T_{1c} \left[\left(\frac{P_{2c}}{P_{1c}} \right)^{\gamma_c - 1 / \gamma_c} \right]}{(T_{2c} - T_{1c})^2} \quad (\text{Eqn. A.3})$$

$$\frac{\partial \eta_C}{\partial T_{2c}} = - \frac{T_{1c} \left[\left(\frac{P_{2c}}{P_{1c}} \right)^{\gamma_c - 1 / \gamma_c} \right]}{(T_{2c} - T_{1c})^2} \quad (\text{Eqn. A.4})$$

$$\frac{\partial \eta_C}{\partial P_{1c}} = - \frac{T_{1c} \left(\frac{P_{2c}}{P_{1c}} \right)^{\gamma_c - 1 / \gamma_c}}{P_{1c} (T_{2c} - T_{1c})} \frac{\gamma_c - 1}{\gamma_c} \quad (\text{Eqn. A.5})$$

$$\frac{\partial \eta_c}{\partial P_{2c}} = \frac{T_{1c} \left(\frac{P_{2c}}{P_{1c}} \right)^{\gamma_c - 1 / \gamma_c}}{P_{2c} (T_{2c} - T_{1c})} \frac{\gamma_c - 1}{\gamma_c} \quad (\text{Eqn. A.6})$$

Equations A.3-6 can be used to find the total uncertainty in efficiency by,

$$u_{\eta_c} = \sqrt{\left(\left(\frac{\left(\frac{P_{2c}}{P_{1c}} \right)^{\gamma_c - 1 / \gamma_c}}{(T_{2c} - T_{1c})} + \frac{T_{1c} \left(\frac{P_{2c}}{P_{1c}} \right)^{\gamma_c - 1 / \gamma_c}}{(T_{2c} - T_{1c})^2} \right)^2 u_{T_{1c}}^2 + \left(- \frac{T_{1c} \left(\frac{P_{2c}}{P_{1c}} \right)^{\gamma_c - 1 / \gamma_c}}{(T_{2c} - T_{1c})^2} \right)^2 u_{T_{2c}}^2 + \left(- \frac{T_{1c} \left(\frac{P_{2c}}{P_{1c}} \right)^{\gamma_c - 1 / \gamma_c}}{P_{1c} (T_{2c} - T_{1c})} \frac{\gamma_c - 1}{\gamma_c} \right)^2 u_{P_{1c}}^2 + \left(\frac{T_{1c} \left(\frac{P_{2c}}{P_{1c}} \right)^{\gamma_c - 1 / \gamma_c}}{P_{2c} (T_{2c} - T_{1c})} \frac{\gamma_c - 1}{\gamma_c} \right)^2 u_{P_{2c}}^2 } \right)}$$

(Eqn. A.7)

The results for uncertainty in compressor efficiency are:

Figure A.1 Uncertainty in ACM compressor efficiency during typical operating conditions.

The uncertainty grows large at low compressor pressure ratios. The calculations are mainly limited by the measurement uncertainty associated with the thermocouples. With low pressure ratios across the compressor, there will be small changes in temperature, and the uncertainty in temperature measurements will be a larger fraction of the temperature difference across the compressor.

Table A.2 Values used in calculating uncertainty in ACM compressor efficiency calculations.

Compressor Inlet Pressure [psia]	Compressor Outlet Pressure [psia]	Compressor Inlet Temperature [Deg. R]	Compressor Outlet Temperature [Deg. R]	Compressor Efficiency Uncertainty [-]	Compressor Pressure Ratio [-]
16.69	18.71	530	555.1	0.0692	1.121
18.69	22.92	530	575.3	0.0391	1.226
20.69	27.29	530	592.1	0.0289	1.319
22.69	31.8	530	606.5	0.0237	1.401
24.69	36.45	530	618.9	0.0206	1.476
26.69	41.22	530	629.9	0.0185	1.544
28.69	46.1	530	639.6	0.017	1.607

A.2 ACM Mass Flow Rate

Mass flow rate can be calculated using the air density and volumetric flow rate as follows.

$$\dot{m} = \rho \dot{V} \quad (\text{Eqn. A.8})$$

Where ρ is the air density, and $\dot{V}$ is the volumetric flow rate. The uncertainty is calculated in a similar way to the compressor efficiency uncertainty. First, the mass flow rate should be expressed in terms of measured quantities.

$$\dot{m} = \left(\frac{RT}{P} \right) \dot{V} \quad (\text{Eqn. A.9})$$

The resulting uncertainty in mass flow rate for the following conditions is approximately 0.6 lbm/min, or 6.7%.

Table A.3 Conditions for evaluating uncertainty in mass flow rate.

Compressor Inlet Condition	Value
Temperature [Deg. R]	560
Pressure [psia]	25
Volumetric Flow Rate [CFM]	75

A.3 Compressor Power

Compressor power can be calculated by the following equation:

$$\dot{W}_c = \dot{m}c_p(T_{2c} - T_{1c}) \quad (\text{Eqn. A.10})$$

Using units of pound-mass, BTU/lbm/°R, and °R, the result can be converted to hp by applying a conversion factor of 0.02358 to the right-hand side of Eqn. A.10.

The result of the uncertainty analysis for the following conditions is approximately 0.21 hp, or 7.5%.

Table A.4 Conditions for evaluating uncertainty in ACM compressor power.

Compressor Inlet and Outlet Conditions	Value
Specific Heat, Constant Pressure [BTU/lbm/R]	0.241
Inlet Temperature [Deg. R]	530
Outlet Temperature [Deg. R]	575
Inlet Pressure [psia]	22.5
Volumetric Flow Rate [CFM]	100

A.4 Turbine Power

Turbine power can be expressed using the following equation.

$$\dot{W}_t = \dot{m}c_p(T_{1t} - T_{2t}) \quad (\text{Eqn. A.11})$$

Alternatively, it can be expressed in terms of measured quantities.

$$\dot{W}_t = \left(\frac{RT_{1c}}{P_{1c}}\right) \dot{V}c_p(T_{1t} - T_{2t}) \quad (\text{Eqn. A.12})$$

The resulting uncertainty is 0.16 hp, or 4.5%.

A.5 Cooling Capacity

The cooling capacity can be expressed using the following equation.

$$\dot{Q} = \dot{m}c_p(T_{ambient} - T_{2t}) \quad (\text{Eqn. A.13})$$

The cooling capacity can be expressed in terms of measured quantities.

$$\dot{Q} = \left(\frac{RT_{1c}}{P_{1c}}\right) \dot{V}c_p(T_{1t} - T_{2t}) \quad (\text{Eqn. A.14})$$

The resulting uncertainty is 11.7 BTU/min, or 6.1%.

A.6 Turbine Efficiency

The turbine efficiency can be expressed as

$$\eta = \frac{\dot{m}c_p(T_{1t}-T_{2t})}{\dot{m}_t c_{p_t} T_{1T} \left[\frac{PR_t^{\frac{\gamma_t-1}{\gamma_t-1}}}{PR_t^{\frac{\gamma_t-1}{\gamma_t}}} \right]} \quad (\text{Eqn. A.15})$$

Alternatively, it can be expressed in terms of measured quantities.

$$\eta = \frac{(T_{1t}-T_{2t})}{T_{1T} \left[\frac{P_{1t}/P_{2t}^{\frac{\gamma_t-1}{\gamma_t-1}}}{P_{1t}/P_{2t}^{\frac{\gamma_t-1}{\gamma_t}}} \right]} \quad (\text{Eqn. A.16})$$

The resulting uncertainty is 0.02 [-].

A.7 Intercooler Effectiveness

The intercooler effectiveness can be expressed as

$$E_{htxr} = \frac{(T_1-T_2)}{(T_1-T_{ambient})} \quad (\text{Eqn. A.17})$$

The resulting uncertainty is 0.04 [-].

A.8 Intercooler Pressure Drop

The intercooler effectiveness can be expressed as

$$\Delta P_{htxr} = (P_1 - P_2) \quad (\text{Eqn. A.18})$$

The resulting uncertainty is 0.03 [-].

A.9 Numerical Uncertainty Propagation Analysis

Numerical uncertainty analysis was required for the dry-air rated parameters and relative humidity calculations. This was required because properties of moist air were found using the psychrometric lookup functions in Engineering Equation Solver (EES), and an analytical solution for finding uncertainty was not possible. The calculated values that required numerical treatment are DAR COP, DAR Cooling Capacity, and the calculation of the relative humidity from wet-bulb, dry-bulb temperature, and ambient pressure measurements. More information about the method used in EES can be found at:

http://www.fchart.com/download/ees_manual.exe

Alternatively, it can be found in PDF format at:

<http://www.unh.edu/p2/ihf/ees.pdf>

Appendix B – Derivation of Equations for Air-cycle Analysis

B.1 Isentropic Compressor Power Requirement

Figure B.1 Compressor control volume analysis.

Assumptions

- Isentropic
 - Adiabatic
 - Reversible
- Perfect Gas
 - Ideal gas behavior
 - Constant properties

Governing Equations

$$Pv = RT \quad (\text{Eqn. B.1})$$

$$Pv^\gamma = \text{constant} \quad (\text{Eqn. B.2})$$

Continuity

$$\dot{m}_1 = \dot{m}_2 \quad (\text{Eqn. B.3})$$

Energy

$$\dot{W}_{in} = \dot{m}(h_2 - h_1) \quad (\text{Eqn. B.4})$$

For a perfect gas, enthalpy can be found with the following equation

$$h = c_p(T - T_{ref}) \quad (\text{Eqn. B.5})$$

Substituting Eqn. B.5 into Eqn. B.4 results in the following energy equation

$$\dot{W}_{in} = \dot{m}c_p(T_2 - T_1) \quad (\text{Eqn. B.6})$$

Using the equation of state for an ideal gas (Eqn. 3.1.1), we get

$$\dot{W}_{in} = \dot{m} \frac{c_p}{R} (P_2 v_2 - P_1 v_1) \quad (\text{Eqn. B.7})$$

Rearranging Eqn. B.7 results in

$$\dot{W}_{in} = \dot{m} \frac{c_p}{R} P_2 v_1 \left[\frac{v_2}{v_1} - \frac{P_1}{P_2} \right] \quad (\text{Eqn. B.8})$$

Applying the polytropic equation for an adiabatic boundary condition to Eqn. B.8 gives us

$$\dot{W}_{in} = \dot{m} \frac{c_p}{R} P_2 v_1 \left[\frac{P_1^{1/\gamma}}{P_2} - \frac{P_1}{P_2} \right] \quad (\text{Eqn. B.9})$$

Rearranging and defining pressure ratio as $PR_c \stackrel{\text{def}}{=} P_{2c}/P_{1c}$ we get,

$$\dot{W}_{in} = \dot{m} c_p P_1 v_1 \left[PR_c^{\gamma-1/\gamma} - 1 \right] \quad (\text{Eqn. B.10})$$

Finally, substituting the ideal gas equation of state, we get

$$\dot{W}_C = \dot{m} c_p T_{1C} \left[PR_C^{\gamma-1/\gamma} - 1 \right] \quad (\text{Eqn. B.11})$$

If we define an isentropic efficiency as

$$\eta \stackrel{\text{def}}{=} \frac{\dot{W}_{\text{isentropic}}}{\dot{W}_{\text{actual}}} \quad (\text{Eqn. B.12})$$

Eqn. B.11 becomes

$$\dot{W}_C = \dot{m} c_p \frac{T_1}{\eta_c} \left[PR_C^{\gamma-1/\gamma} - 1 \right] \quad (\text{Eqn. B.13})$$

B.2 Isentropic Turbine Power

Figure B.2 Turbine control volume analysis.

Assumptions

- Isentropic
 - Adiabatic
 - Reversible
- Perfect Gas
 - Ideal gas behavior
 - Constant properties

Governing Equations

$$Pv = RT \quad (\text{Eqn. B.14})$$

$$Pv^\gamma = \text{constant} \quad (\text{Eqn. B.15})$$

Continuity

$$\dot{m}_1 = \dot{m}_2 \quad (\text{Eqn. B.16})$$

Energy

$$\dot{W}_{out} = \dot{m}(h_1 - h_2) \quad (\text{Eqn. B.17})$$

For a perfect gas, enthalpy can be found with the following equation

$$h = c_p(T - T_{ref}) \quad (\text{Eqn. B.18})$$

Substituting Eqn. B.18 into Eqn. B.17 results in the following energy equation

$$\dot{W}_{out} = \dot{m}c_p(T_1 - T_2) \quad (\text{Eqn. B.19})$$

Using the equation of state for an ideal gas (Eqn. B.14), we get

$$\dot{W}_{out} = \dot{m} \frac{c_p}{R} (P_1 v_1 - P_2 v_2) \quad (\text{Eqn. B.20})$$

Rearranging Eqn. B.20 results in

$$\dot{W}_{out} = \dot{m} \frac{c_p}{R} P_2 v_1 \left[\frac{P_1}{P_2} - \frac{v_2}{v_1} \right] \quad (\text{Eqn. B.21})$$

Applying the polytropic equation for an adiabatic boundary condition to Eqn. B.21 gives us

$$\dot{W}_{out} = \dot{m} \frac{c_p}{R} P_2 v_1 \left[\frac{P_1}{P_2} - \frac{P_1}{P_2}^{1/\gamma} \right] \quad (\text{Eqn. B.22})$$

Rearranging and defining pressure ratio as $PR_c \stackrel{\text{def}}{=} P_2/P_1$ we get,

$$\dot{W}_{out} = \dot{m} c_p P_1 v_1 \left[1 - PR_c^{\gamma-1/\gamma} \right] \quad (\text{Eqn. B.23})$$

Making a substitution with the ideal gas equation of state,

$$\dot{W}_{out} = \dot{m} c_p T_1 \left[1 - PR_c^{\gamma-1/\gamma} \right] \quad (\text{Eqn. B.24})$$

Dividing Eqn. B.24 by $PR_C^{\gamma-1/\gamma}$ results in an equation of similar form to the compressor power equation. It is not necessary, but it is a more common form and makes it easier to remember for many people.

$$\dot{W}_{out} = \dot{m}c_p T_1 \left[\frac{PR_T^{\gamma-1/\gamma-1}}{PR_T^{\gamma-1/\gamma}} \right] \quad (\text{Eqn. B.25})$$

If we define an isentropic efficiency as

$$\eta \stackrel{\text{def}}{=} \frac{\dot{W}_{actual}}{\dot{W}_{isentropic}} \quad (\text{Eqn. B.26})$$

Eqn. B.25 becomes

$$\dot{W}_{out} = \dot{m}c_p T_1 \eta_T \left[\frac{PR_T^{\gamma-1/\gamma-1}}{PR_T^{\gamma-1/\gamma}} \right] \quad (\text{Eqn. B.27})$$

B.3 Bearing Losses

Figure B.3 Bearing system control volume analysis.

Energy

$$\dot{W}_{Bearing\ Drag} = \dot{W}_T - \dot{W}_C \quad (\text{Eqn. B.28})$$

Substituting Eqn. B.27 and Eqn. B.13

$$\dot{W}_{Bearing\ Drag} = \dot{m}_c c_{p_c} \frac{T_{1c}}{\eta_c} \left[PR_C^{\gamma_c^{-1}/\gamma_c} - 1 \right] - \dot{m}_t c_{p_t} T_{1T} \eta_t \left[\frac{PR_t^{\gamma_t^{-1}/\gamma_t - 1}}{PR_t^{\gamma_t^{-1}/\gamma_t}} \right] \quad (\text{Eqn. B.29})$$

B.4 Isentropic Efficiency

B.4.1 Compressor Isentropic Efficiency Measurement

Referring to Figure 3.1 for sensor layout, there are pressure and temperature measurements at the compressor inlet and outlet.

Starting with Eqn. B.12, and substituting Eqn. B.6 and Eqn. B.13

$$\eta = \frac{\dot{m}_c c_{p_c} T_{1c} \left[PR_C^{\gamma_c - 1 / \gamma_c - 1} \right]}{\dot{m}_c p (T_{2_{measured}} - T_{1_{measured}})} \quad (\text{Eqn. B.30})$$

This can be reduced to

$$\eta_c = \frac{T_{1c} \left[PR_C^{\gamma_c - 1 / \gamma_c - 1} \right]}{(T_{2c} - T_{1c})} \quad (\text{Eqn. B.31})$$

B.4.2 Turbine Isentropic Efficiency Measurement with Bearing Losses Combined

Rearranging Eqn. B.28

$$\dot{W}_C = \dot{W}_T - \dot{W}_{\text{Bearing Drag}} \quad (\text{Eqn. B.32})$$

In words, the next step is equating compressor power with effective turbine power. The right-hand side is the lumped parameter containing the turbine power being reduced by the bearing losses. This reduces the apparent turbine isentropic efficiency. Turbine isentropic efficiency can be defined as

$$\eta \stackrel{\text{def}}{=} \frac{\dot{W}_{\text{actual}}}{\dot{W}_{\text{isentropic}}}, \quad (\text{Eqn. B.33})$$

where the actual work represents the power consumed by the compressor, not the power produced directly by the turbine. This is typically the efficiency presented commercial turbocharger datasheets.

Substituting Eqn. B.13 and Eqn. B.27 into Eqn. B.33

$$\eta = \frac{\dot{m}_c c_{p_c} \frac{T_{1C}}{\eta_c} \left[\frac{PR_C^{\gamma_c-1/\gamma_c-1}}{PR_t^{\gamma_t-1/\gamma_t-1}} \right]}{\dot{m}_t c_{p_t} T_{1T} \eta_t \left[\frac{PR_t^{\gamma_t-1/\gamma_t-1}}{PR_t^{\gamma_t-1/\gamma_t}} \right]} \quad (\text{Eqn. B.34})$$

For most systems, the mass flow rate of the turbine is the mass flow rate of the compressor with the addition of a mass flow rate of fuel. For the ACM, the turbine mass flow rate is equal to the compressor mass flow rate, and, since the assumption of a perfect gas has been made, the gas properties remain constant throughout the cycle calculations. In this case, the turbine isentropic efficiency reduces to

$$\eta = \frac{\frac{T_{1C}}{\eta_c} \left[\frac{PR_C^{\gamma-1/\gamma-1}}{PR_t^{\gamma-1/\gamma-1}} \right]}{T_{1T} \left[\frac{PR_t^{\gamma-1/\gamma-1}}{PR_t^{\gamma-1/\gamma}} \right]} \quad (\text{Eqn. B.35})$$

B.4.3 Turbine Isentropic Efficiency Measurement without Bearing Losses Combined

Unlike the previous turbine isentropic efficiency calculations, the physical work will be represented as

$$\dot{W}_{actual} = \dot{m}_{measured} c_p (T_{1,measured} - T_{2,measured}) \quad (\text{Eqn. B.36})$$

This is the physical power produced by an adiabatic turbine before bearing losses. Eqn. B.36 and

Eqn. B.25 can be substituted into Eqn. B.33 to form

$$\eta = \frac{\dot{m}_{measured} c_p (T_{1,measured} - T_{2,measured})}{\dot{m}_t c_{p_t} T_{1T} \left[\frac{PR_t^{\gamma_t-1/\gamma_t-1}}{PR_t^{\gamma_t-1/\gamma_t}} \right]} \quad (\text{Eqn. B.37})$$

This can be reduced to

$$\eta = \frac{(T_{1,measured} - T_{2,measured})}{T_{1T} \left[\frac{PR_t \gamma_t^{-1} / \gamma_{t-1}}{PR_t \gamma_t^{-1} / \gamma_t} \right]} \quad (\text{Eqn. B.38})$$

B.5 Compressor Discharge Temperature

Compressor discharge temperature can be calculated using the compressor work equation. In real cases, there will be a temperature rise due to isentropic, adiabatic compression and a component due to compressor inefficiency. The latter part is accounted for by the isentropic efficiency. Equating Eqn. B.6 and Eqn. B.13 results in the following equation.

$$\dot{m}c_p(T_2 - T_1) = \dot{m}c_p \frac{T_1}{\eta_c} \left[PR_C^{\gamma-1/\gamma} - 1 \right] \quad (\text{Eqn. B.39})$$

This results in an equation relating the compressor discharge temperature to a function of the compressor inlet temperature, pressure ratio, and compressor isentropic efficiency, three pieces of information that are typically known. By simplifying and rearranging Eqn. B.39, we get

$$T_2 = T_1 + \frac{T_1}{\eta_c} \left[PR_C^{\gamma-1/\gamma} - 1 \right] \quad (\text{Eqn. B.40})$$

B.6 Corrected Compressor Flow Rate

Corrected compressor flow rate maps the physical flow rate to a standard reference condition.

The compressor test conditions must be known to predict performance at conditions other than those identical to the testing conditions. Having one defined standard temperature and pressure condition for all compressor maps reduces chances of error by the engineer.

Honeywell Turbo Technologies (Garrett Turbo) uses the compressor reference temperature and pressure as $545^{\circ}R$ and 28.4 in Hg, respectively [5].

$$W_C^* = \frac{W_C \sqrt{T_{1C}/545}}{P_{1C}/28.4} \quad (\text{Eqn. B.41})$$

B.7 Corrected Compressor Speed

The corrected compressor speed needs to be corrected in a similar fashion to compressor flow rate, except for dependence on pressure [5].

$$N_C^* = \frac{N_C}{\sqrt{T_{1C}/545}} \quad (\text{Eqn. B.42})$$

B.8 Turbine Discharge Temperature

The turbine outlet temperature cannot be predicted exactly from typical turbine maps. Most turbine maps provide the turbine efficiency including bearing losses, so this will indicate less than the actual power the turbine is producing. This will tend to underestimate the temperature drop across the turbine. For most applications of turbochargers, the turbine discharge temperature is not of much concern; the power provided to the compressor is of more interest.

Starting with a compressor-turbine power balance (Eqn. B.13 and Eqn. B.27), the turbine discharge temperature can be found.

$$\dot{m}c_p \frac{T_{1c}}{\eta_c} \left[PR_C^{\gamma-1/\gamma} - 1 \right] = \dot{m}c_p (T_{1T} - T_{2T}) \quad (\text{Eqn. B.43})$$

The compressor and turbine flow rates are the same and, it is being modeled as a perfect gas, so it reduces to

$$\frac{T_{1c}}{\eta_c} \left[PR_C^{\gamma-1/\gamma} - 1 \right] = (T_{1T} - T_{2T}) \quad (\text{Eqn. B.44})$$

This can be arranged in the form

$$T_{2T} = T_{1T} - \frac{T_{1c}}{\eta_c} \left[PR_C^{\gamma-1/\gamma} - 1 \right] \quad (\text{Eqn. B.45})$$

B.9 Corrected Turbine Flow Rate

Corrected turbine flow rate calculation is the same as corrected to corrected compressor flow rate, except for reference conditions. Honeywell Turbo Technologies (Garrett Turbo) uses a turbine reference temperature and pressure of 519^{OR} and 29.92 in Hg, respectively [5].

$$W_T^* = \frac{W_T \sqrt{T_{1T}/519}}{P_{1T}/29.92} \quad (\text{Eqn. B.46})$$

B.10 Corrected Turbine Speed

Corrected turbine speed calculation is the same as for corrected compressor speed, with the exception of reference temperature [5].

$$N_T^* = \frac{N_T}{\sqrt{T_{1T}/519}} \quad (\text{Eqn. B.47})$$

B.11 Dry Air-Rated Temperature

Treating systems with multi-component flow can be difficult and cumbersome. The dry air-rated method allows the engineer to treat moist air systems as a single-component gas. The dry air rated temperature is a pseudo temperature of dry air that has equivalent enthalpy to the original air-entrained water mixture if the entrained water were to adiabatically vaporize. This can greatly simplify calculations by allowing treatment as an ideal gas or simplified further to a perfect gas.

Appendix C - Compressor-Turbine Matching Process

Before beginning the cycle analysis, compressor and turbine performance maps are needed from the manufacturer. These maps provide the isentropic efficiency of the components over their operating range. Obtaining a good turbine-compressor match is a critical step in achieving minimum turbine discharge temperatures.

Figure C.1 GT1244 compressor performance map.

Figure C.2 GT1244 turbine performance map.

To start the compressor-turbine analysis, it is important to start with the compressor first because the compressor performance map will define a corrected compressor speed based on the pressure ratio and corrected compressor flow rate. One can assume values for corrected compressor flow rate and pressure ratio. Using these assumed values the compressor map will provide compressor speed and isentropic efficiency. Using Eqn. C.1, the required compressor power can be calculated.

$$\dot{W}_c = \dot{m} c_p \frac{T_1}{\eta_c} \left[PR_c^{\gamma-1/\gamma} - 1 \right] \quad (\text{Eqn. C.1})$$

The next step is to convert corrected quantities to physical quantities, such as compressor mass flow rate and speed. These can be modified to the actual operating conditions by the following equations.

$$W_c = W_c^* \frac{\sqrt{T_{1c}/545}}{P_{1c}/28.4} \quad (\text{Eqn. C.2})$$

$$N_c = N_c^* \sqrt{T_{1c}/545} \quad (\text{Eqn. C.3})$$

The compressor outlet temperature can be determined using the known or assumed compressor inlet temperature, compressor pressure ratio, and compressor efficiency.

$$T_2 = T_1 + \frac{T_1}{\eta_c} \left[PR_c^{\gamma-1/\gamma} - 1 \right] \quad (\text{Eqn. C.4})$$

The next component after the compressor will be the heat exchanger, followed by the turbine. There will be a pressure drop across the heat exchanger. Typical values can be assumed for the pressure drop and effectiveness, unless actual heat exchanger data is available. In this project, engineering data on the heat exchanger was not available, so the pressure drop was assumed to

be 1psi and effectiveness to be 0.80. From this, turbine inlet pressure and temperature can be found.

$$P_{1T} = P_{2c} - \Delta P_{HTXR} \text{ (Eqn. C.5)}$$

$$T_{1T} = T_{2c} - \varepsilon(T_{2c} - T_{amb}) \text{ (Eqn. C.6)}$$

Typically the turbine outlet pressure can be assumed to be the ambient pressure. The turbine pressure ratio can be found from the following equation.

$$PR_T = P_{1T}/P_{2T} \text{ (Eqn. C.7)}$$

To acquire data from the turbine performance map, corrected turbine mass flow rate and speed are required. These can be found from the following equations.

$$W_T^* = \frac{W_T \sqrt{T_{1T}/519}}{P_{1T}/29.92} \text{ (Eqn. C.8)}$$

$$N_T^* = \frac{N_T}{\sqrt{T_{1T}/519}} \text{ (Eqn. C.9)}$$

Turbine efficiency can be found from the turbine performance map at this point, and turbine power can be found using the following equation. The corrected turbine mass flow rate must fall on the flow lines on the turbine performance map, and if they do not, conservation of mass is not satisfied. Iteration with new assumed values is required.

$$\dot{W}_{out} = \dot{m} c_p T_1 \eta_T \left[\frac{PR_T^{\gamma-1/\gamma-1}}{PR_T^{\gamma-1/\gamma}} \right] \text{ (Eqn. C.10)}$$

Once this is complete the turbine power generation can be compared to the compressor power consumption. They should be relatively close, within a tolerance determined by the engineer. If

not, a new corrected compressor mass flow rate and pressure ratio can be selected and repeat this process until mass, power, and speed are balanced.

Appendix D - Detailed Calibration Procedure

The experimental data is collected using a National Instruments DAQPad 6015 data acquisition (DAQ) system and a NI SCB-68 breakout box with cold-junction compensation (CJC). The type-T thermocouple inputs are configured as a differential input for noise cancellation purposes.

Noise is an issue in the thermocouple readings because the span of the wire reached up to 15 feet. Shielding with single-ended grounding provides enough protection from line noise to obtain standard accuracy of $\pm 1^\circ\text{C}$ for type-T thermocouples. Each thermocouple channel is calibrated using a Fluke 714 Thermocouple Calibrator unit. The pressure transducers are automotive 3-bar MAP (Manifold Absolute Pressure) sensors. The pressure transducers are powered from a +5V source available from the NI equipment, as to provide a stable input to improve the quality of the measurements. The pressure transducers are calibrated using a Fluke 718 Series Pressure Calibrator unit.

The laminar flow element (LFE) is a Meriam 50MC2-2. The calibration sheet that relates the volumetric flow rate to pressure differential correlation is provided by the manufacturer. The pressure differential across the LFE is measured by a Meriam 2110P Smart Pressure Gauge differential pressure transducer. The differential pressure transducer is calibrated using a Fluke 718 Series Pressure Calibrator unit. The two calibration curves, from the LFE and differential pressure transducer, are combined to relate the volumetric flow rate to a voltage output.

All measurements were oversampled and averaged to provide the most stable measurements. The noise measured was nearly normally distributed, so averaging does not generate a significant offset in the readings. In any case, the oversampling and averaging method used was employed at the time of calibration, so any offset is captured in the calibrations obtained. The largest components of the noise were at 60Hz and 120Hz. These two frequencies correlate to

noise that is generated by single-phase AC and full-bridge rectified single-phase AC. The sensor calibration was checked periodically to ensure proper sensor operation. The severity of the electromagnetic interference (EMI) in the testing facility can be seen in the next plots.

Figure D.1 Thermocouple measurement noise (pre-filtered) before and after shielding the wires.

Figure D.2 Temperature measurement output after oversampling and averaging.

The temperature output from the DAQ system always agreed with the Fluke calibrator unit within 1.7 Deg. F, after implementing the shielding, oversampling, and averaging. This is approximately the standard limit of error for the thermocouple wire used.

Appendix E - Sensor Calibration Charts

E.1 Pressure Transducer Calibration

Figure E.1 MAP sensor calibration chart.

E.2 Laminar Flow Element and Differential Pressure Transducer Calibration

The laminar flow element and differential pressure transducer setup requires multiple calibrations. The first calibration comes with the LFE manufacturer, and this relates volumetric flow rate to a pressure differential across the element. It is important to note that in order to make accurate measurements with a LFE, using the provided calibration sheet, the required straight, smooth pipe entrance and exit lengths must be at an approximate minimum of 10 and 5 pipe diameters, respectively.

Figure E.2 This schematic shows the configuration of the LFE, differential pressure transducer, and resistor.

Figure E.3 Laminar flow element calibration curve. Data is provided by Meriam Process Technologies.

The second calibration required is for the differential pressure transducer to relate output voltage to the pressure differential. The sensor provides a standard 4-20ma current loop output. This is an excellent choice for transmitting a signal accurately over larger distances. While there is a voltage drop associated with a length of wire, there is continuity in the current through the wire. One drawback is that the current must be converted to a voltage by placing a resistor in the loop and measuring the voltage differential across the resistor. There is a tolerance on that resistor that is often 5-10%. There are closer tolerances available at additional cost, but a pressure transducer calibration with the implemented resistor in place will provide the most accurate measurements.

Figure E.4 Differential pressure transducer and resistor combined calibration.

Once both calibrations are complete, you have a relationship between volumetric flow rate and pressure drop and another relationship between pressure drop and voltage. The two can be related by doing a linear curve-fit on each calibration, and using the resulting equations to make a new calibration chart relating the voltage output to volumetric flow rate.

Figure E.5 Laminar flow element setup calibration.

E.3 Thermocouple Calibration

The thermocouples were calibrated using a Fluke 714 Calibrator Unit, as described in Appendix D. With sufficient electro-magnetic interference (EMI) protection, the standard type-T calibration, as implemented by default in Labview, was sufficient to match the calibration unit within 1.7 [Deg. F] over the expected temperature range of ACM operation. This range was expected to be approximately 32-180 [Deg. F].

Appendix F - Perkins Diesel Performance, Stock Turbocharger

Figure F.1 BSFC contour map using the stock turbocharger.

Figure F.2 VE contour map using the stock turbocharger.

Figure F.3 Intercooler effectiveness contour map using the stock turbocharger.

Figure F.4 Pressure differential between the exhaust and intake manifolds. ($P_{int} - P_{exh}$)

Figure F.5 Engine turbocharger compressor efficiency.

F.6 Engine turbocharger turbine efficiency.

F.7 Engine fuel mass flow rate.

Appendix G - Perkins Diesel Performance, VNT Turbocharger

The VNT turbocharger was tested over a smaller torque and speed range due to fuel measurement time limitations and the extra time necessary for turbine vane position adjustments.

Figure G.1 BSFC contour map using the VNT turbocharger

Figure G.2 VE contour map using the VNT turbocharger.

Figure G.3 Intercooler effectiveness contour map using the VNT turbocharger.

Figure G.4 differential between the exhaust and intake manifolds. ($P_{int} - P_{exh}$)

Figure G.5 Engine turbocharger compressor efficiency.

Figure G.6 Engine turbocharger turbine efficiency.

Figure G.7 Engine fuel mass flow rate.

Appendix H - Sanden R134a Compressor Performance Chart

Appendix I - ACM Performance Prediction Code in Matlab

```

% Chris Forster

% This is the main ACM modeling program.

% 1) Load compressor and turbine data into tab-delimited text files.
% 2) Set system parameters (Main_Parameters.m)
% 3) Execute this file.
% 4) Everything else is automatic.

%␣
-----␣
----
% Program starts here.
%␣
-----␣
----

% Clear the workspace, command window, close open plots/windows,␣
etc...
    clear;
    close all;
    clc;

    %matlabpool(2)

% Load/preprocess compressor data
    comp_data_process;

% Load/preprocess turbine data
    turb_data_process;

% Load/preprocess heat exchanger data
    %HTXR1
        % n/a for now
    %HTXR2
        htxr2_data_process;

% Load system parameters
    Main_Parameters;

% Load psychometric table data
    PsychometricTables = PsychometricTables(0);

% Find humidity stuff...
    water_mass_init = PsychometricLookup(ambient.P,ambient.T,ambient.␣
rel_hum,PsychometricTables)*7000;    %grains

% T-C Operating Point Matching

```

```

% Set mass flow rate tolerances and limits
    rel_tol_mass = 1e-2;
    w_mass = 0.8;    %relaxation factor
    n = 0;

% Set Initial Guesses
    m_dot_c_corr = 6:0.05:8;
    PR_c = 1.25:0.01:1.55;

    n_calcs = length(m_dot_c_corr)*length(PR_c);

for index = 1:length(m_dot_c_corr)
    percent_complete = index/length(m_dot_c_corr)*100;
    percent_display = ['Calculations are ' num2str(percent_complete)
'% complete.'];
    display(percent_display)
    for index2 = 1:length(PR_c)

multi_acm(index,index2) = TC_match(acm,engine,ambient,water_mass_init,
m_dot_c_corr(index),PR_c(index2));

% Cleanup time...
%clear index;
%clear index2;
%clear counter;
    end
end

% Keep the points that meet power, speed, and mass balance criteria
k_index = 1;

for index = 1:length(m_dot_c_corr)
    for index2 = 1:length(PR_c)

        if (isnan(multi_acm(index,index2).power_t) == 0) & (isnan(
(multi_acm(index,index2).power_c) == 0) & (isnan(multi_acm(index,
index2).m_dot_t_phy) == 0) & (isnan(multi_acm(index,index2).
m_dot_c_phy) == 0)
            keepers_nan(k_index,:) = [index, index2];
            k_index = k_index+1;
        end

    end
end

k_index = 1;

for index = 1:length(keepers_nan(:,1))

```

```

        if (abs(multi_acm(keepers_nan(index,1),keepers_nan(index,2)). $\sqrt{\text{power\_t}-\text{multi\_acm(keepers\_nan(index,1),keepers\_nan(index,2)).power\_c}}$ 
<= 0.2)
            keepers_power(k_index,:) = keepers_nan(index,:);
            k_index = k_index+1;
        end
    end

    k_index = 1;

    for index = 1:length(keepers_power(:,1))

        if (abs(multi_acm(keepers_power(index,1),keepers_power(index,2)). $\sqrt{\text{m\_dot\_t\_phy}-\text{multi\_acm(keepers\_power(index,1),keepers\_power(index,2)).m\_dot\_c\_phy}}$ 
<= 0.1)
            keepers_mass(k_index,:) = keepers_power(index,:);
            k_index = k_index+1;
        end
    end

    figure(2)
    subplot(2,1,2)

    hold on

    for index = 1:length(keepers_mass(:,1))
        scatter(multi_acm(keepers_mass(index,1),keepers_mass(index,2)). $\sqrt{\text{Pl\_t}/\text{ambient.P},\text{multi\_acm(keepers\_power(index,1),keepers\_power(index,2)).m\_dot\_t\_corrected}}$ 
,Pl_t/ambient.P,multi_acm(keepers_power(index,1),keepers_power(index,2)).eta_t)
    end

    figure(2)
    subplot(2,1,2)

    hold on

    for index = 1:length(keepers_mass(:,1))
        scatter(multi_acm(keepers_mass(index,1),keepers_mass(index,2)). $\sqrt{\text{Pl\_t}/\text{ambient.P},\text{multi\_acm(keepers\_power(index,1),keepers\_power(index,2)).eta\_t}}$ 
,Pl_t/ambient.P,multi_acm(keepers_power(index,1),keepers_power(index,2)).eta_t)
    end

    % Plots
    plots

    figure(1)
    subplot(1,2,1)

    hold on

```



```

for index = 1:length(keepers_mass(:,1))
    scatter(multi_acm(keepers_mass(index,1),keepers_mass(index,2)).\
m_dot_c_corr,multi_acm(keepers_power(index,1),keepers_power(index,2)).\
PR_c)
    pts_scatter(:,index) = [multi_acm(keepers_mass(index,1),\
keepers_mass(index,2)).m_dot_c_corr; multi_acm(keepers_power(index,1),\
keepers_power(index,2)).PR_c; multi_acm(keepers_power(index,1),\
keepers_power(index,2)).Q_dot_DAR; multi_acm(keepers_power(index,1),\
keepers_power(index,2)).eta_t; multi_acm(keepers_power(index,1),\
keepers_power(index,2)).Pl_c/ambient.P; multi_acm(keepers_power(index,\
1),keepers_power(index,2)).T2_t;multi_acm(keepers_mass(index,1),\
keepers_mass(index,2)).m_dot_c_phy];
end

figure(3)      % DAR Cooling Capacity vs. ACM Pressure Ratio
scatter(pts_scatter(5,:),pts_scatter(3,:))

figure(4)      % Turbine efficiency vs. Corrected flow rate
scatter(pts_scatter(1,:),pts_scatter(4,:))

figure(6)      % Turbine efficiency vs. Physical flow rate
scatter(pts_scatter(7,:),pts_scatter(4,:))

figure(5)      % Turbine Outlet Temperature vs. ACM Pressure Ratio
scatter(pts_scatter(5,:),pts_scatter(6,:))

% Optional windows closing...
%close all;

%matlabpool close

```

```

%Program by Chris Forster

%Ambient Conditions
ambient.T = 526;      %Rankine
ambient.P = 14.69;    %psia
ambient.rel_hum = 0.40; %unitless

%Air Properties
%gamma = 1.395;

%Operating Parameters
%Valve_opening = 0.5;      %Fraction, unitless
%M_dot_s = 15;

%Valve Data
%Valve_area = 0.5;        %in^2
%Valve_C = 1;
%Chester_constant = 31.82;

%Engine TC Data
engine.eta_c = 0.7;
engine.eta_t = 0.65;
engine.boost = 11;      %psig
engine.PR = (ambient.P+engine.boost)/ambient.P;

%Primary HTXR Data
engine.htxr = 0.8;
engine.dP_htxr = 1;

%Secondary HTXR Data
%E_htxr2 = 0.9;
acm.dP_htxr = 0.3;

%HTXR2 Fan Operating Parameters
%acm.m_dot_fan = 12;      %lbm/min

%Initialize My Functions
N_table = N_air_gen(1,1.876,0.001);

```

```

% Chris Forster
% Turbine Power

function[W_t] = turbine_power(P1,P2,T1,m_dot,eta_t)

if (isnan(P1) == 1) | (isnan(P2) == 1) | (isnan(T1) == 1) | (isnan(m_dot) == 1) | (isnan(eta_t) == 1)
    W_t = 0;
    %display('turbine_power NaN input warning, W_t set to zero!')
else

PR = P1/P2;

C_p = 0.241;

W_t = m_dot*C_p*T1*eta_t*((PR^0.285-1)/PR^0.285)*0.02358;    %Units in HP

end

end %End of function

```

```

% Chris Forster
% Turbine Mass Flow Rate

%turb_data(:,1) = Corrected Speed
%turb_data(:,2) = Pressure Ratio
%turb_data(:,3) = Corrected mass flow rate data
%turb_data(:,4) = Turbine Efficiency (0-1 range)

%This function returns actual mass flow rate (not corrected).

function[m_dot] = turbine_massflow(P1,P2,T1,N_phy,turb_data)

if (isnan(P1) == 1) | (isnan(P2) == 1) | (isnan(T1) == 1) | (isnan(N_phy) == 1)
    m_dot = 0;
    %display('turbine_massflow NaN input warning, m_dot set to zero!')
else

PR = P1/P2;

N_corrected_t = N_phy/sqrt(T1/519);

% Lookup speeds, then calc two flowrates based on PR
for index = 1:length(turb_data.speeds)
    if N_corrected_t < turb_data.speeds(1)
        m_dot_corrected_t = 0;
        %display('Corrected Turbine Speed Too Low, NO DATA')
        break
    end

    if N_corrected_t > turb_data.speeds(length(turb_data.speeds))
        m_dot_corrected_t = 0;
        %display('Corrected Turbine Speed Too High, NO DATA')
        break
    end

    if N_corrected_t == turb_data.speeds(index)
        m_dot_corrected_t = turb_data.fits.flow(index,1)
        *PR^2+turb_data.fits.flow(index,2)*PR+turb_data.fits.flow(index,3);
        break
    end

    if N_corrected_t < turb_data.speeds(index)
        m_dot1 = turb_data.fits.flow(index-1,1)*PR^2+turb_data.fits.
flow(index-1,2)*PR+turb_data.fits.flow(index-1,3);
        m_dot2 = turb_data.fits.flow(index,1)*PR^2+turb_data.fits.flow
(index,2)*PR+turb_data.fits.flow(index,3);

        m_dot_corrected_t = m_dot1 + (m_dot2-m_dot1)/(turb_data.speeds

```

```

(index)-turb_data.speeds(index-1))*(N_corrected_t - turb_data.speeds
(index-1));
        break
    end
end

m_dot = m_dot_corrected_t*(P1*2.03625437/29.92)/sqrt(T1/519);

end

end %End of function

```

```

% Chris Forster

% This function finds the turbine efficiency from the turbine data
file.

%turb_data(:,1) = Corrected Speed
%turb_data(:,2) = Pressure Ratio
%turb_data(:,3) = Correct Turbine Mass Flow Rate
%turb_data(:,4) = Turbine Efficiency

function[eta_turb] = turbine_efficiency(P1,P2,T1,N_phy,turb_data)

if (isnan(P1) == 1) | (isnan(P2) == 1) | (isnan(T1) == 1) | (isnan(
(N_phy) == 1)
    eta_turb = 0;
    %display('turbine_efficiency NaN input warning, m_dot set to
zero!')
else

PR = P1/P2;

N_corrected = N_phy/sqrt(T1/519);

% Lookup speeds, then calc two flowrates based on PR
for index = 1:length(turb_data.speeds)
    if N_corrected < turb_data.speeds(1)
        eta_turb = 0;
        %display('Corrected Turbine Speed Too Low, NO DATA')
        break
    end

    if N_corrected > turb_data.speeds(length(turb_data.speeds))
        eta_turb = 0;
        %display('Corrected Turbine Speed Too High, NO DATA')
        break
    end

    if N_corrected == turb_data.speeds(index)
        eta_turb = turb_data.fits. eff(index,1)*PR^2+turb_data.fits. eff
(index,2)*PR+turb_data.fits. eff(index,3);
        break
    end

    if N_corrected < turb_data.speeds(index)
        eta1 = turb_data.fits. eff(index-1,1)*PR^2+turb_data.fits. eff
(index-1,2)*PR+turb_data.fits. eff(index-1,3);
        eta2 = turb_data.fits. eff(index,1)*PR^2+turb_data.fits. eff
(index,2)*PR+turb_data.fits. eff(index,3);

```

```

        eta_turb = eta1 + (eta2-eta1)/(turb_data.speeds(index)-
turb_data.speeds(index-1))*(N_corrected - turb_data.speeds(index-1));
        break
    end
end

end

end %End of function

```

```

turb_data_loader

%turb_data(:,1) = Corrected Speed
%turb_data(:,2) = Pressure Ratio
%turb_data(:,3) = Correct Turbine Mass Flow Rate
%turb_data(:,4) = Turbine Efficiency

% Using structures...
for index = 1:length(acm.turb_data.speeds)
    counter = 1;
    for index2 = 1:length(acm.turb_data.file(:,1))
        if (acm.turb_data.file(index2,1) == acm.turb_data.speeds(
(index))
            acm.turb_data.data(counter,1,index) = acm.turb_data.speeds(
(index);
            acm.turb_data.data(counter,2,index) = acm.turb_data.file(
(index2,2);
            acm.turb_data.data(counter,3,index) = acm.turb_data.file(
(index2,3);
            acm.turb_data.data(counter,4,index) = acm.turb_data.file(
(index2,4);
            counter = counter + 1;
        end
    end
end

acm.turb_data.forcedpoint = [1 0 0];

for index = 1:length(acm.turb_data.speeds)
    tempvar1 = cat(2,acm.turb_data.speeds(index),acm.turb_data.(
forcedpoint);
    tempvar2(:, :, index) = cat(1,acm.turb_data.data(:, :, index),
tempvar1);
    clear tempvar1;
end

clear acm.turb_data_spds.data;
acm.turb_data.data = tempvar2;
clear tempvar2;

for index = 1:length(acm.turb_data.speeds)
    acm.turb_data.fits.flow(index, :) = polyfit(acm.turb_data.data(:, 2,
index), acm.turb_data.data(:, 3, index), 2);
    acm.turb_data.fits.eff(index, :) = polyfit(acm.turb_data.data(:, 2,
index), acm.turb_data.data(:, 4, index), 2);
end

```



```
%Chris Forster

%The file format is a tab-delimited text file

    %turb_data(:,1) = Corrected Speed
    %turb_data(:,2) = Pressure Ratio
    %turb_data(:,3) = Correct Turbine Mass Flow Rate
    %turb_data(:,4) = Turbine Efficiency

%This file will be run from within the simulation (automatically...)

acm.turb_data.file = load('turb_data.txt');
acm.turb_data.speeds = load('turb_speeds.txt');
```

```

% Chris Forster

function[acm2] = TC_match(acm,engine,ambient,water_mass_init,
m_dot_c_corr,PR_c)

    acm.m_dot_c_corr = m_dot_c_corr;
    acm.PR_c = PR_c;

    % Find Compressor Speed (Physical Speed)
    acm.P1_c = engine.boost+14.7;
    acm.P2_c = acm.P1_c*acm.PR_c;
    acm.T1_c = ambient.T-5;

    acm.N_c_phy = compressor_speed(acm.P1_c,acm.P2_c,acm.T1_c,acm.
m_dot_c_corr,acm.comp_x_speed,acm.comp_y_speed,acm.comp_z_speed);

    % Find physical flow from compressor corrected flow
    acm.m_dot_c_phy = compressor_flow_phy(acm.P1_c,acm.T1_c,acm.
m_dot_c_corr);

    % Find compressor efficiency.
    acm.eta_c = compressor_efficiency(acm.P1_c,acm.P2_c,acm.T1_c,
acm.m_dot_c_corr,acm.comp_x_eta,acm.comp_y_eta,acm.comp_z_eta);

    % Find compressor outlet temperature.
    acm.T2_c = acm.T1_c + (1/acm.eta_c)*acm.T1_c*((acm.P2_c/acm.
P1_c)^0.285-1);      %Deg R

    % Find required compressor power.
    acm.power_c = compressor_power(acm.P1_c,acm.P2_c,acm.T1_c,acm.
m_dot_c_phy,acm.eta_c);

    % Find pressure drop accross the heat exchanger.
    acm.P_delta = 0.3;      %psia, constant for now...

    acm.P1_t = acm.P2_c - acm.P_delta;

    % Find heat exchanger outlet temperature.
    acm.T_delta_htxr2 = acm.T2_c - ambient.T;

    acm.E_htxr = 0.83;

    acm.T1_t = acm.T2_c - acm.E_htxr*(acm.T2_c - ambient.T);

    % Convert T1_t to T1_t_DAR
    acm.g_sat_t1 = PsychrometricLookup(acm.P1_t,acm.T1_t,1,
PsychrometricTables)*7000;      %grains
    acm.T1_t_DAR = T_DAR((acm.T1_t-460),acm.g_sat_t1,
water_mass_init) + 460;

```

```

    % Find Turbine mass flow rate (physical) from speed and pressure
ratio across turbine.
    acm.N_t_phy = acm.N_c_phy;
    acm.P2_t = ambient.P;

    acm.m_dot_t_phy = turbine_massflow(acm.P1_t,acm.P2_t,acm.
T1_t_DAR,acm.N_t_phy,acm.turb_data); %Actual flow (not corrected)

    % Find corrected turbine flow from m_dot_t
    acm.m_dot_t_corrected = turbine_massflow_corrected(acm.P1_t,
acm.T1_t_DAR,acm.m_dot_t_phy);

    % Find turbine efficiency
    acm.eta_t = turbine_efficiency(acm.P1_t,acm.P2_t,acm.T1_t_DAR,
acm.N_t_phy,acm.turb_data);

    % Find power produced by turbine
    acm.power_t = turbine_power(acm.P1_t,acm.P2_t,acm.T1_t,acm.
m_dot_t_phy,acm.eta_t);

    % Find turbine outlet temperature
    acm.T2_t = acm.T1_t_DAR - acm.T1_t_DAR*(acm.eta_t+0.10)*
(((acm.P1_t/acm.P2_t)^0.285-1)/(acm.P1_t/acm.P2_t)^0.285);
    %acm.T2_t = acm.T1_t_DAR - (acm.power_t*42.4072263)/acm.
m_dot_t_phy/0.241;

    % Find humidity stuff...
    acm.T2_g_sat = PsychrometricLookup(ambient.P,acm.T2_t,1,
PsychrometricTables)*7000; %grains
    acm.T2_t_DAR = T_DAR((acm.T2_t-460),acm.T2_g_sat,
water_mass_init) + 460;

    acm.T2_t_DB = T_DAR_DB((acm.T2_t_DAR-460),acm.T2_g_sat,
water_mass_init) + 460;

    % Find cooling capacity
    acm.Q_dot_DAR = acm.m_dot_t_phy*0.241*(ambient.T - acm.
T2_t_DAR)*60/12000; %Units in tons

% Cleanup time...
clear index;
clear index2;
clear counter;

acm2 = acm;

end %End of function

```

```

%Program by Chris Forster
%July 15, 2008

%Input all humidity ratios in grains/lbm
%Input temperature in degrees F

%Returns Dry Air Rated Temperature in degrees F

function[T_DB] = T_DAR_DB(T_DAR,g_sat,g_tot)
    if (isnan(T_DAR) == 1) | (isnan(g_sat) == 1) | (isnan(g_tot) == 1)
        T_DB = 0;
        %display('T_DAR_DB NaN input warning, T_DB set to zero!')
    else

        % Define g_ent (entrained water)
        if (g_tot >= g_sat)
            g_ent = g_tot - g_sat;
        else
            g_ent = 0;
        end

        % Convert grains to lbm
        g_sat = g_sat/7000;
        g_tot = g_tot/7000;
        g_ent = g_ent/7000;

        % Calculate T_DB from T_DAR assuming above 32F (DB)
        % Check assumption afterwards.

        T_DB_TBD = (T_DAR*(0.24+0.4332*g_tot)+1093*g_ent)/(0.24+0.4332*g_sat+g_ent);

        if (T_DB_TBD <= 32)
            T_DB_TBD = (T_DAR*(0.24+0.4332*g_tot)+1221*g_ent)/(0.24+0.4332*g_sat+0.5*g_ent);
        end

        T_DB = T_DB_TBD;    %Degree F

    end
end
end

```

```

%Program by Chris Forster
%April 16, 2008

%Input all humidity ratios in grains/lbm
%Input temperature in degrees F

%Returns Dry Air Rated Temperature in degrees F

function[T_DAR] = T_DAR(T_DB,g_sat,g_tot)
    if (isnan(T_DB) == 1) | (isnan(g_sat) == 1) | (isnan(g_tot) == 1)
        T_DAR = 0;
        %display('T_DAR NaN input warning, T_DAR set to zero!')
    else

        if g_sat > g_tot
            g_ent = 0;
            T_DAR = T_DB;
            return
        end

        if g_sat < g_tot
            g_ent = g_tot - g_sat;
        end

        if T_DB > 32
            T_DAR = ((0.24+0.4332*g_sat/7000+g_ent/7000)*T_DB-
1093*g_ent/7000)/(0.24+0.4332*g_tot/7000);
        end

        if T_DB <= 32
            T_DAR = ((0.24+0.4332*g_sat/7000+0.5*g_ent/7000)*T_DB-
1221*g_ent/7000)/(0.24+0.4332*g_tot/7000);
        end

    end
end
end

```

```

%Chris Forster

%This function generates the Saturation Psychometric Tables (Source:
EES Tables)
%HumidityRatio = function(Pressure, Temperature)

function[PsychometricTables] = PsychometricTables(nullvar)

P = [10 14.69 20 25 30 40 50 60 70 80 90 100 120 140 160 180 200 220];
T = [460:5:1450];

P_10 = [0.001173
0.00196
0.003207
0.005149
0.007766
0.01142
0.01656
0.02372
0.03362
0.04725
0.06602
0.09197
0.1283
0.18
0.256
0.3733
0.5697
0.9488
1.936
9.871];

P_14_69 = [0.0007981
0.001333
0.002179
0.003496
0.005265
0.00773
0.01118
0.01595
0.0225
0.0314
0.04347
0.05979
0.08192
0.1122
0.154
0.2133
0.3001

```

0.4343
0.6609
1.108
2.34
18.64];

P_20 = [0.000586
0.0009783
0.001599
0.002564
0.003859
0.005659
0.008173
0.01164
0.01637
0.02276
0.03135
0.04282
0.05814
0.07861
0.1061
0.1436
0.1954
0.2691
0.3786
0.5524
0.8599
1.529
3.975];

P_25 = [0.0004687
0.0007824
0.001279
0.002049
0.003083
0.004519
0.006521
0.009277
0.01303
0.01808
0.02483
0.03379
0.04566
0.06134
0.08211
0.1098
0.1471
0.1982
0.27

0.3752
0.5389
0.8199
1.396
3.174];

P_30 = [0.0003905
0.0006519
0.001065
0.001707
0.002567
0.003761
0.005425
0.007711
0.01082
0.01499
0.02055
0.02791
0.03759
0.05029
0.06695
0.08888
0.1179
0.1568
0.2098
0.2841
0.3924
0.5602
0.8466
1.429
3.194
1510];

P_40 = [0.0002929
0.0004888
0.0007986
0.001279
0.001923
0.002817
0.00406
0.005766
0.008078
0.01118
0.01529
0.0207
0.02777
0.03697
0.0489
0.06436

```
0.08443
0.1106
0.1451
0.1913
0.2542
0.3429
0.4737
0.6808
1.049
1.863
5.047];
```

```
P_50 = [0.0002343
0.000391
0.0006388
0.001023
0.001538
0.002251
0.003244
0.004604
0.006445
0.008909
0.01217
0.01645
0.02202
0.02923
0.03851
0.05044
0.06576
0.08546
0.1109
0.1441
0.188
0.2471
0.3289
0.4468
0.6275
0.932
1.539
3.297
54.34];
```

```
P_60 = [0.0001952
0.0003258
0.0005322
0.0008522
0.001281
0.001875
0.002701
```

0.003832
0.005362
0.007406
0.01011
0.01365
0.01824
0.02417
0.03177
0.04147
0.05385
0.06962
0.08978
0.1157
0.1492
0.1931
0.2519
0.3326
0.4477
0.6215
0.9082
1.459
2.91
16.44];

P_70 = [0.0001673
0.0002792
0.0004561
0.0007303
0.001098
0.001606
0.002313
0.003282
0.00459
0.006337
0.008645
0.01166
0.01557
0.0206
0.02703
0.03521
0.04559
0.05874
0.0754
0.09656
0.1236
0.1585
0.2041
0.2648
0.3479

0.4661
0.6441
0.9366
1.495
2.95
15.49];

P_80 = [0.0001464
0.0002443
0.0003991
0.000639
0.0009602
0.001405
0.002023
0.00287
0.004013
0.005538
0.007551
0.01018
0.01358
0.01795
0.02352
0.0306
0.03953
0.0508
0.06499
0.08289
0.1055
0.1344
0.1715
0.22
0.2845
0.3729
0.499
0.6897
1.006
1.621
3.295
24.16];

P_90 = [0.0001301
0.0002171
0.0003547
0.0005679
0.0008534
0.001249
0.001798
0.002549
0.003564

0.004918
0.006703
0.009032
0.01204
0.01591
0.02082
0.02705
0.03489
0.04475
0.0571
0.0726
0.09208
0.1167
0.1479
0.1882
0.2407
0.3108
0.4072
0.5458
0.758
1.117
1.844
4.04];

P_100 = [0.0001171
0.0001954
0.0003192
0.0005111
0.0007679
0.001124
0.001618
0.002293
0.003206
0.004423
0.006026
0.008117
0.01082
0.01428
0.01868
0.02424
0.03123
0.03998
0.05093
0.06459
0.08166
0.1031
0.1301
0.1644
0.2086

0.2664
0.344
0.4516
0.6081
0.8524
1.28
2.204
5.6];

P_120 = [0.00009759
0.0001628
0.000266
0.0004258
0.0006398
0.0009361
0.001347
0.00191
0.002669
0.003681
0.005014
0.00675
0.008989
0.01185
0.01549
0.02007
0.02581
0.03297
0.04187
0.05291
0.06659
0.08358
0.1047
0.1312
0.1646
0.2072
0.2625
0.3357
0.4357
0.5782
0.7942
1.155
1.866
3.886
46.9];

P_140 = [0.00008364
0.0001396
0.000228
0.000365

0.0005483
0.0008022
0.001155
0.001636
0.002287
0.003153
0.004292
0.005776
0.007689
0.01013
0.01323
0.01712
0.02199
0.02805
0.03554
0.0448
0.05622
0.07029
0.08766
0.1092
0.1359
0.1695
0.2122
0.2671
0.3395
0.4375
0.5757
0.7823
1.12
1.76
3.415
17.09];

P_160 = [0.00007319
0.0001221
0.0001995
0.0003193
0.0004797
0.0007018
0.00101
0.001431
0.002
0.002757
0.003753
0.005048
0.006718
0.008848
0.01154
0.01493

0.01916
0.0244
0.03088
0.03885
0.04864
0.06065
0.07538
0.09347
0.1158
0.1435
0.1781
0.2218
0.2781
0.3519
0.4515
0.5915
0.7997
1.138
1.772
3.372
14.77];

P_180 = [0.00006505
0.0001086
0.0001773
0.0002838
0.0004264
0.0006237
0.0008976
0.001272
0.001777
0.002449
0.003333
0.004484
0.005964
0.007852
0.01024
0.01324
0.01697
0.0216
0.0273
0.0343
0.04287
0.05333
0.06611
0.08172
0.1008
0.1243
0.1534

0.1897
0.2355
0.2943
0.3714
0.4755
0.622
0.8404
1.196
1.871
3.608
17.77];

P_200 = [0.00005855
0.0000977
0.0001596
0.0002554
0.0003837
0.0005613
0.0008077
0.001145
0.001599
0.002203
0.002999
0.004032
0.005363
0.007058
0.009201
0.01189
0.01523
0.01937
0.02446
0.0307
0.03832
0.04759
0.05887
0.0726
0.0893
0.1097
0.1347
0.1657
0.2042
0.2529
0.3154
0.3976
0.5089
0.6663
0.903
1.294
2.055

```

4.146
33.49];

P_220 = [0.00005323
0.00008881
0.0001451
0.0002322
0.0003488
0.0005102
0.0007342
0.00104
0.001453
0.002003
0.002725
0.003664
0.004871
0.00641
0.008353
0.01079
0.01382
0.01756
0.02216
0.02778
0.03464
0.04297
0.05307
0.0653
0.08014
0.09816
0.1201
0.147
0.1803
0.2217
0.2741
0.3416
0.4306
0.552
0.7252
0.9895
1.437
2.347
5.165];

%End of data from EES

HumidityRatio = zeros(length(P),length(T));

%P_10
for index = 1:length(P_10)

```

```

        HumidityRatio(1,index) = HumidityRatio(1,index) + P_10(index);
    end

    %P_14.69
    for index = 1:length(P_14_69)
        HumidityRatio(2,index) = HumidityRatio(2,index) + P_14_69(index);
    end

    %P_20
    for index = 1:length(P_20)
        HumidityRatio(3,index) = HumidityRatio(3,index) + P_20(index);
    end

    %P_25
    for index = 1:length(P_25)
        HumidityRatio(4,index) = HumidityRatio(4,index) + P_25(index);
    end

    %P_30
    for index = 1:length(P_30)
        HumidityRatio(5,index) = HumidityRatio(5,index) + P_30(index);
    end

    %P_40
    for index = 1:length(P_40)
        HumidityRatio(6,index) = HumidityRatio(6,index) + P_40(index);
    end

    %P_50
    for index = 1:length(P_50)
        HumidityRatio(7,index) = HumidityRatio(7,index) + P_50(index);
    end

    %P_60
    for index = 1:length(P_60)
        HumidityRatio(8,index) = HumidityRatio(8,index) + P_60(index);
    end

    %P_70
    for index = 1:length(P_70)
        HumidityRatio(9,index) = HumidityRatio(9,index) + P_70(index);
    end

    %P_80
    for index = 1:length(P_80)
        HumidityRatio(10,index) = HumidityRatio(10,index) + P_80(index);
    end

    %P_90

```

```

for index = 1:length(P_90)
    HumidityRatio(11,index) = HumidityRatio(11,index) + P_90(index);
end

%P_100
for index = 1:length(P_100)
    HumidityRatio(12,index) = HumidityRatio(12,index) + P_100(index);
end

%P_120
for index = 1:length(P_120)
    HumidityRatio(13,index) = HumidityRatio(13,index) + P_120(index);
end

%P_140
for index = 1:length(P_140)
    HumidityRatio(14,index) = HumidityRatio(14,index) + P_140(index);
end

%P_160
for index = 1:length(P_160)
    HumidityRatio(15,index) = HumidityRatio(15,index) + P_160(index);
end

%P_180
for index = 1:length(P_180)
    HumidityRatio(16,index) = HumidityRatio(16,index) + P_180(index);
end

%P_200
for index = 1:length(P_200)
    HumidityRatio(17,index) = HumidityRatio(17,index) + P_200(index);
end

%P_220
for index = 1:length(P_220)
    HumidityRatio(18,index) = HumidityRatio(18,index) + P_220(index);
end

rows = length(HumidityRatio(:,1));
cols = length(HumidityRatio(1,:));

length_nonzero = 0;

for index3 = 1:cols
    if HumidityRatio(rows,index3) ~= 0
        length_nonzero = length_nonzero + 1;
    end
end

```

```

for index1 = 1:rows
    for index2 = 1:length_nonzero
        PsychometricTables(index1,index2) = HumidityRatio(index1,
index2);
    end
end

end %End of function

```

```

%Chris Forster

% Temperature - Rankine
% Pressure - psia

function[PsychometricLookup_output] = PsychometricLookup(Pressure,↵
Temperature,RelativeHumidity,PsychometricTables)
P = [10 14.69 20 25 30 40 50 60 70 80 90 100 120 140 160 180 200 220];
T = [460:10:460+38*10];

PsychometricLookup_sat = interp2(T,P,PsychometricTables,Temperature,↵
Pressure,'spline');

PsychometricLookup_output = PsychometricLookup_sat*RelativeHumidity;

end %End of function

```

```

%Chris Forster

% This section of code is for plotting...keeps it clean.

% Get screensize and figure layout
acm.chart.screensize = get(0,'ScreenSize');

    %Figure(1) location and size
    acm.chart.comp_x_loc = 0;
    acm.chart.comp_y_loc = 0;
    acm.chart.comp_width = acm.chart.screensize(3);
    acm.chart.comp_height = acm.chart.screensize(4)*0.95;
    acm.chart.comp_pos = [acm.chart.comp_x_loc, acm.chart.comp_y_loc,
acm.chart.comp_width, acm.chart.comp_height];

    %Figure(2) location and size
    acm.chart.turb_x_loc = acm.chart.screensize(3)/3;
    acm.chart.turb_y_loc = acm.chart.screensize(4)*2/3;
    acm.chart.turb_width = acm.chart.screensize(3)/3;
    acm.chart.turb_height = acm.chart.screensize(4)*0.3;
    acm.chart.turb_pos = [acm.chart.turb_x_loc, acm.chart.turb_y_loc,
acm.chart.turb_width, acm.chart.turb_height];

% Compressor Plots
    figure('Position',acm.chart.comp_pos)
    figure(1)
    subplot(1,2,1)
    contourf(acm.comp_x_eta,acm.comp_y_eta,acm.comp_z_eta)
    title 'Compressor Efficiency'
    xlabel 'Corrected Mass Flow Rate [lbm/min]'
    ylabel 'Pressure Ratio [-]'

    colorbar

    hold on
    comp_point = plot(multi_acm(1).m_dot_c_corr,multi_acm(1).PR_c);
    set(comp_point,'marker','o')
    set(comp_point,'markersize',10)
    set(comp_point,'markeredgecolor','black')
    set(comp_point,'markerfacecolor','black')
    hold off

    subplot(1,2,2)
    surf(acm.comp_x_eta,acm.comp_y_eta,acm.comp_z_eta)
    title 'Compressor Efficiency'
    xlabel 'Corrected Mass Flow Rate [lbm/min]'
    ylabel 'Pressure Ratio [-]'

```

```

% Turbine Plots
figure('Position',acm.chart.turb_pos)
figure(2)
subplot(2,1,1)
for index = 1:length(acm.turb_data.speeds)
    for index2 = 1:length(acm.turb_data.data(:,1,1))-1
        acm.chart.turbine.data(index2,1,index) = acm.␣
turb_data.data(index2,1,index);
        acm.chart.turbine.data(index2,2,index) = acm.␣
turb_data.data(index2,2,index);
        acm.chart.turbine.data(index2,3,index) = acm.␣
turb_data.data(index2,3,index);
        acm.chart.turbine.data(index2,4,index) = acm.␣
turb_data.data(index2,4,index);
    end
end

for index = 1:length(acm.turb_data.speeds)
    hold all
    x_plot = acm.chart.turbine.data(:,2,index);
    y1_plot = acm.chart.turbine.data(:,3,index);
    y2_plot = acm.chart.turbine.data(:,4,index);
    [AX,H1,H2] = plotyy(x_plot,y1_plot,x_plot,y2_plot,'plot');
    hold off
    clear x_plot;
    clear y1_plot;
    clear y2_plot;

    %legend('a','b','c','d','location','eastoutside')

    set(AX(1),'xlim',[1 2.4])
    set(AX(1),'ylim',[0 8])
    set(AX(2),'xlim',[1 2.4])
    set(AX(2),'ylim',[0.4 1.2])

    set(H1(1),'color', get_color(index))
    set(H2(1),'color', get_color(index))

    set(H1(1),'linewidth', 2)
    set(H2(1),'linewidth', 2)

    set(AX(1),'xtick',[1:0.2:2.4])
    set(AX(1),'ytick',[0:1:8])
    set(AX(2),'ytick',[0.4:0.1:1.2])

    set(AX(1),'xminorgrid', 'on')

```



```

        set(AX(2),'xminorgrid', 'on')

        grid on

        set(get(AX(1),'Xlabel'),'String','Pressure Ratio [-]')
        set(get(AX(1),'Ylabel'),'String','Corrected Turbine Flow'
[lbm/min]')
        set(get(AX(2),'Ylabel'),'String','Efficiency [-]')

    end

    %legend('a','b','c','d','location','eastoutside')

    %Draw current operating point on chart...
    hold on

    [AX,H1,H2] = plotyy(multi_acm(1).Pl_t/multi_acm(1).P2_t,
multi_acm(1).m_dot_t_corrected,multi_acm(1).Pl_t/multi_acm(1).P2_t,
multi_acm(1).eta_t,'plot');

    hold off

    set(H1(1),'marker', 'o')
    set(H2(1),'marker', 'o')
    set(H1(1),'markersize', 10)
    set(H2(1),'markersize', 10)
    set(H1(1),'markerfacecolor', 'black')
    set(H2(1),'markerfacecolor', 'black')
    set(H1(1),'markeredgecolor', 'black')
    set(H2(1),'markeredgecolor', 'black')

    set(AX(1),'xlim',[1 2.4])
    set(AX(1),'ylim',[0 8])
    set(AX(2),'xlim',[1 2.4])
    set(AX(2),'ylim',[0.4 1.2])

    set(AX(1),'xtick',[1:0.2:2.4])
    set(AX(1),'ytick',[0:1:8])
    set(AX(2),'ytick',[0.4:0.1:1.2])

    set(AX(1),'xminorgrid', 'on')
    set(AX(2),'xminorgrid', 'on')

    grid on

    set(get(AX(1),'Xlabel'),'String','Pressure Ratio [-]')
    set(get(AX(1),'Ylabel'),'String','Corrected Turbine Flow'
[lbm/min]')
    set(get(AX(2),'Ylabel'),'String','Efficiency [-]')

```

```

% End of current operating points

subplot(2,1,2)

acm.chart.turbine.pr = 1:0.01:3;

for index = 1:length(acm.turb_data.speeds)
    for index2 = 1:length(acm.chart.turbine.pr)
        acm.chart.turbine.flow(index2,index) = acm.turb_data.␣
fits.flow(index,1)*acm.chart.turbine.pr(index2)^2+acm.turb_data.fits.␣
flow(index,2)*acm.chart.turbine.pr(index2)+acm.turb_data.fits.flow␣
(index,3);
        acm.chart.turbine.eff(index2,index) = acm.turb_data.␣
fits.eff(index,1)*acm.chart.turbine.pr(index2)^2+acm.turb_data.fits.␣
eff(index,2)*acm.chart.turbine.pr(index2)+acm.turb_data.fits.eff␣
(index,3);

        if acm.chart.turbine.flow(index2,index) < 0
            acm.chart.turbine.flow(index2,index) = 0;
        end

        if acm.chart.turbine.eff(index2,index) < 0
            acm.chart.turbine.eff(index2,index) = 0;
        end

    end
end

for index = 1:length(acm.turb_data.speeds)
    hold on
    x_plot = acm.chart.turbine.pr;
    y1_plot = acm.chart.turbine.flow(:,index);
    y2_plot = acm.chart.turbine.eff(:,index);
    [AX,H1,H2] = plotyy(x_plot,y1_plot,x_plot,y2_plot,'plot');

    hold off
    clear x_plot;
    clear y1_plot;
    clear y2_plot;

    set(AX(1),'xlim',[1 2.4])
    set(AX(1),'ylim',[0 8])
    set(AX(2),'xlim',[1 2.4])
    set(AX(2),'ylim',[0.4 1.2])

    set(H1(1),'color', get_color(index))
    set(H2(1),'color', get_color(index))

    set(H1(1),'linewidth', 2)

```

```

        set(H2(1),'linewidth', 2)

        set(AX(1),'xtick',[1:0.2:2.4])
        set(AX(1),'ytick',[0:1:8])
        set(AX(2),'ytick',[0.4:0.1:1.2])

        set(AX(1),'xminorgrid', 'on')
        set(AX(2),'xminorgrid', 'on')

        grid on

        set(get(AX(1),'Xlabel'),'String','Pressure Ratio [-]')
        set(get(AX(1),'Ylabel'),'String','Corrected Turbine Flow [lbm/min]')
        set(get(AX(2),'Ylabel'),'String','Efficiency [-]')

        end

%Draw current operating point on chart...
hold on

[AX,H1,H2] = plotyy(multi_acm(1).P1_t/multi_acm(1).P2_t,multi_acm(1).m_dot_t_corrected,multi_acm(1).P1_t/multi_acm(1).P2_t,multi_acm(1).eta_t,'plot');

hold off

set(H1(1),'marker', 'o')
set(H2(1),'marker', 'o')
set(H1(1),'markersize', 10)
set(H2(1),'markersize', 10)
set(H1(1),'markerfacecolor', 'black')
set(H2(1),'markerfacecolor', 'black')
set(H1(1),'markeredgecolor', 'black')
set(H2(1),'markeredgecolor', 'black')

set(AX(1),'xlim',[1 2.4])
set(AX(1),'ylim',[0 8])
set(AX(2),'xlim',[1 2.4])
set(AX(2),'ylim',[0.4 1.2])

set(AX(1),'xtick',[1:0.2:2.4])
set(AX(1),'ytick',[0:1:8])
set(AX(2),'ytick',[0.4:0.1:1.2])

set(AX(1),'xminorgrid', 'on')
set(AX(2),'xminorgrid', 'on')

grid on

```

```
set(get(AX(1),'Xlabel'),'String','Pressure Ratio [-]')
set(get(AX(1),'Ylabel'),'String','Corrected Turbine Flow [lbm/min]')
set(get(AX(2),'Ylabel'),'String','Efficiency [-]')
% End of current operating points
```

```

%Program by Chris Forster
%April 8, 2008

function[PressureRatio] = N_air_rev(N_in,N)

%Isentropic Exponent Lookup and Interpolation
%Reference - Chester Smith

%Accuracy of function is good. The largest difference between the
function and Chester Smith's tables I've seen so far is
%approximately 0.0003

%table_resolution should be a value like 0.01 or 0.001, depending on
the
%table entry step size you want...more means slower, but more
accurate...

%Returns error if you specify table_start less than one

if (N_in <= N(1,length(N(2,:))))
    PressureRatio = interp1(N(2,:),N(1,:),N_in,'spline');
else
    PressureRatio = 0.00000001;
end

end

```

```

%Program by Chris Forster
%April 8, 2008

function[N_air_gen] = N_air_gen(table_start, table_end,
table_resolution)

%Isentropic Exponent Lookup and Interpolation
%Reference - Chester Smith

%Accuracy of function is good. The largest difference between the
function and Chester Smith's tables I've seen so far is
%approximately 0.0003

%table_resolution should be a value like 0.01 or 0.001, depending on
the
%table entry step size you want...more means slower, but more
accurate...

%Returns error if you specify table_start less than one

if (table_start < 1)
    N_air_gen = 'error - table_start less than 1';
    return
end

if (table_end <= table_start)
    N_air_gen = 'error - check table bounds';
    return
end

P2 = 14.69;
C1 = [2.06806, 1; 2.06416, 1.0500; 2.004159781, 1.879];

PR = table_start:table_resolution:table_end;

P1 = P2*PR;

for index = 1:length(PR)
    if (PR(index) <= C1(3,2))
        N_air_gen(1,index) = PR(index);
        N_air_gen(2,index) = sqrt(P2*(P1(index)-P2))/P1(index)*interp1
(C1(:,2),C1(:,1),PR(index),'linear');
    else
        N_air_gen(1,index) = PR(index);
        N_air_gen(2,index) = 1;
    end
end

end

```



```

%Program by Chris Forster
%April 8, 2008

function[N_air] = N_air(PR_in,N)

%Isentropic Exponent Lookup and Interpolation
%Reference - Chester Smith

%Accuracy of function is good. The largest difference between the
function and Chester Smith's tables I've seen so far is
%approximately 0.0003

%table_resolution should be a value like 0.01 or 0.001, depending on
the
%table entry step size you want...more means slower, but more
accurate...

%Returns error if you specify table_start less than one

if (PR_in < 1)
    N_air = 'error - PR less than 1...reverse flow probably not
correct'
    return
end

if (PR_in <= N(1,length(N(1,:))))
    N_air = interp1(N(1,:),N(2,:),PR_in,'spline');
else
    N_air = 1;
end

end

```



```
% Chris Forster

% This function finds the turbine speed.

function[speed] = get_speed(index,acm)

speed = acm.turb_data.speeds(index);

end %End of function
```

```

% Chris Forster

% This function allows indexed colors for easy plot formatting, avoids
adjacent duplicate or hard-to-see colors.

function[color] = get_color(index)

switch index
    case {1}
        color = 'blue';
    case {2}
        color = 'red';
    case {3}
        color = 'green';
    case {4}
        color = 'cyan';
    case {5}
        color = 'magenta';
    case {6}
        color = 'black';
    case {7}
        color = 'blue';
    case {8}
        color = 'red';
    case {9}
        color = 'green';
    case {10}
        color = 'cyan';
    case {11}
        color = 'magenta';
    case {12}
        color = 'black';
    otherwise
        disp('Too many speeds...expand color range.');
```

color = 0;

```

end

end %End of function

```

```
% Chris Forster
% Compressor Outlet Temperature

function[T2] = compressor_temp(P1,P2,T1,comp_data)

PR = P2/P1;

T2 = T1 + T1/comp_data*(PR^0.283-1);

end %End of function
```

```

% Chris Forster
% Compressor Speed

%comp_data(:,1) = Corrected Speed
%comp_data(:,2) = Corrected mass flow rate data
%comp_data(:,3) = Pressure Ratio

% This function returns the physical speed (not corrected)

function[N_phy] = compressor_speed(P1,P2,T1,m_dot,comp_x_speed,
comp_y_speed,comp_z_speed)

PR = P2/P1;

%Removed because it is already receiving corrected compressor flow
%m_dot_corrected = m_dot*sqrt(T1/545)/(P1*2.03625437/28.4);

N_corrected_c = interp2(comp_x_speed,comp_y_speed,comp_z_speed,m_dot,
PR,'linear');

N_phy = N_corrected_c*sqrt(T1/545);

end %End of function

```

```

% Chris Forster
% Compressor Outlet Pressure

function[P2] = compressor_pressure(P1,T1,W_c,m_dot,comp_data)

C_p = 0.24;

P2 = P1*(W_c*comp_data/m_dot/C_p/T1+1)^(1/0.283);

end %End of function

```

```

% Chris Forster

% This function finds the power required from the compressor.

%comp_data(:,1) = Corrected Speed
%comp_data(:,2) = Corrected mass flow rate data
%comp_data(:,3) = Pressure Ratio

function[power_c] = compressor_power(P1,P2,T1,m_dot,eta_c)

PR = P2/P1;

power_c = m_dot*0.241*T1/eta_c*(PR^0.285-1)*0.02358;    %Units are in
HP

end %End of function

```

```

% Chris Forster

% This function finds the power required from the compressor.

%comp_data(:,1) = Corrected Speed
%comp_data(:,2) = Corrected mass flow rate data
%comp_data(:,3) = Pressure Ratio

function[m_dot_phy] = compressor_flow_phy(P1,T1,m_dot_c)

m_dot_phy = m_dot_c*(P1*2.03625437)/28.4/sqrt(T1/545);    %Units in
lbm/min

end %End of function

```

```

% Chris Forster

% The script runs the script to load the compressor data files and
makes
% uniform grids from sparse testing data.

comp_data_loader

%comp_data_eta(:,1) = Corrected mass flow rate data
%comp_data_eta(:,2) = Pressure Ratio
%comp_data_eta(:,3) = Compressor Efficiency

acm.comp_x_eta = min(acm.comp_data.etafile(:,1)):0.05:max(acm.
comp_data.etafile(:,1));
acm.comp_y_eta = min(acm.comp_data.etafile(:,2)):0.01:max(acm.
comp_data.etafile(:,2));
acm.comp_y_eta = acm.comp_y_eta';
acm.comp_z_eta = griddata(acm.comp_data.etafile(:,1),acm.comp_data.
etafile(:,2),acm.comp_data.etafile(:,3),acm.comp_x_eta,acm.
comp_y_eta,'cubic');

%comp_data_speed(:,1) = Corrected Speed
%comp_data_speed(:,2) = Corrected mass flow rate data
%comp_data_speed(:,3) = Pressure Ratio

acm.comp_x_speed = min(acm.comp_data.speedfile(:,2)):0.01:max(acm.
comp_data.speedfile(:,2));
acm.comp_y_speed = min(acm.comp_data.speedfile(:,3)):0.01:max(acm.
comp_data.speedfile(:,3));
acm.comp_y_speed = acm.comp_y_speed';
acm.comp_z_speed = griddata(acm.comp_data.speedfile(:,2),acm.
comp_data.speedfile(:,3),acm.comp_data.speedfile(:,1),acm.
comp_x_speed,acm.comp_y_speed,'cubic');

clc;

```



```
%Chris Forster

%This file will be run from within the simulation (automatically...)

% This script loads the two compressor data files (tab-delimited
format)

acm.comp_data.speedfile = load('comp_data_speed.txt');
acm.comp_data.etafile = load('comp_data_eta.txt');
```

```

% Chris Forster

% This function finds the compressor efficiency from the compressor
data matrix.

%comp_data(:,1) = Corrected Speed
%comp_data(:,2) = Corrected mass flow rate data
%comp_data(:,3) = Pressure Ratio

function[eta_c] = compressor_efficiency(P1,P2,T1,m_dot,comp_x_eta,
comp_y_eta,comp_z_eta)

PR = P2/P1;

eta_c = interp2(comp_x_eta,comp_y_eta,comp_z_eta,m_dot,PR,'linear');

end %End of function

```

```

% Chris Forster
% Turbine Mass Flow Rate

function[m_dot_t_corrected] = turbine_massflow_corrected(P1,T1,m_dot_t)

m_dot_t_corrected = m_dot_t*sqrt(T1/519)/(P1*2.03625437/29.92);

end %End of function

```

Appendix J - ACM Performance Prediction Code in EES

// Chris Forster

// 10/1/2009

// ACM Performance Prediction

// Functions Section

Function T_DAR(T_DB,g_tot,P_amb)

g_sat = humrat(airh2o,T=T_DB,P=P_amb,R=1)

g_ent = g_tot - g_sat

If(g_ent > 0) Then

If((T_DB-459.67) >=32) Then

A:=((0.24+0.4332*g_sat+g_ent)*(T_DB-459.67)-1093*g_ent)/(0.24+0.4332*g_tot)

Else

A:=((0.24+0.4332*g_sat+0.5*g_ent)*(T_DB-459.67)-1221*g_ent)/(0.24+0.4332*g_tot)

EndIf

Else

A:=(T_DB-459.67)

EndIf

T_DAR:=A+459.67

End

g_tot = humrat(airh2o, T=T_amb, P=P_amb, R=R_amb)

// End Functions Section

fluid\$ = 'air'

// debugging stuff

{m_dot = 10}

{N_phy = 95000}

{W_t_c = 6}

// end debugging stuff

// Ambient Conditions

T_amb = 531 [R]

P_amb = 14.69 [psia]

R_amb = 0.40

// ACM Inlet Conditions

T_1c = 526 [R]

{P_1c = 25.5 [psia]}

{PR_ACM = 1.5}

PR_ACM = P_1c/P_2t

// Heat Exchanger Data

E_htr = 0.83

dp_htr = 0.3 [psi]

gamma = 1.4

n = (gamma-1)/gamma

// Compressor Work Calculation

PR_c = P_2c/P_1c

W_dot_c = m_dot*CP(fluid\$, T = T_1c)*T_1c/eta_c*(PR_c^n-1)*0.02358

$$T_{2c} = T_{1c} + T_{1c} \cdot (PR_c^n - 1) / \eta_c$$

// HTXR Calculations

$$P_{1t} = P_{2c} - dp_{htxr}$$

$$T_{1t} = T_{2c} - E_{htxr} \cdot (T_{2c} - T_{amb})$$

// Turbine Work Calculation

$$PR_t = P_{1t} / P_{2t}$$

$$P_{2t} = 14.69$$

$$W_{dot_t} = m_{dot} \cdot CP(\text{fluid}\$, T = T_{1t}) \cdot T_{1t} \cdot \eta_t \cdot (PR_t^n - 1) / PR_t^n \cdot 0.02358$$

$$T_{2t} = T_{1t} - W_{dot_t} \cdot 0.02358 / m_{dot} / CP(\text{fluid}\$, T = T_{1t})$$

// Compressor-Turbine Relationship

$$W_{dot_c} = W_{dot_t}$$

// Compressor and Turbine Map Interpolation

$$W_{c_c} = m_{dot} \cdot \sqrt{T_{1c} / 545} / (P_{1c} \cdot 2.03625437 / 28.4)$$

$$\eta_c = \text{interpolate2D}('c_data', 'W_c', 'PR_c', 'Eta_c', W_c = W_{c_c}, PR_c = PR_c)$$

$$N_{c_c} = \text{interpolate2D}('N_c', 'W_c', 'PR_c', 'N_c', W_c = W_{c_c}, PR_c = PR_c)$$

$$N_{c_c} = N_{phy} / \sqrt{T_{1c} / 545}$$

$$N_{t_c} = N_{phy} / \sqrt{T_{1t} / 519}$$

$$W_{t_c} = m_{dot} \cdot \sqrt{T_{1t} / 519} / (P_{1t} \cdot 2.03625437 / 29.92)$$

$$\eta_t = \text{interpolate2D}('t_data', 'W_t', 'PR_t', 'Eta_t', W_t = W_{t_c}, PR_t = PR_t)$$

$$W_{t_c} = \text{interpolate2D}('t_data', 'N_t', 'PR_t', 'W_t', N_t = N_{t_c}, PR_t = PR_t)$$

// Capacity Calculations

$$Q_{dot_dry} = m_{dot} \cdot CP(\text{fluid}\$, T = T_{amb}) \cdot (T_{amb} - T_{2t}) \cdot 60 / 12000$$

$$T_{2t_DAR} = T_{DAR}(T_{2t_g_tot}, P_{amb})$$

$$Q_{dot_DAR} = m_{dot} \cdot CP(\text{fluid}\$, T = T_{amb}) \cdot (T_{amb} - T_{2t_DAR}) \cdot 60 / 12000$$

// Primary/Source Compressor Calculations

$$PR_{ce} = P_{1c} / P_{amb}$$

$$\eta_{ce} = 0.70$$

$$W_{dot_{ce}} = m_{dot} \cdot CP(\text{fluid}\$, T = T_{amb}) \cdot T_{amb} / \eta_{ce} \cdot (PR_{ce}^n - 1) \cdot 0.02358$$

// COP Calculations

$$COP_{dry} = Q_{dot_dry} \cdot 4.71617735 / W_{dot_{ce}}$$

$$COP_{DAR} = Q_{dot_DAR} \cdot 4.71617735 / W_{dot_{ce}}$$

SOLUTION

Unit Settings: [R]/[psia]/[lbm]/[degrees]

(Table 1, Run 20)

$$COP_{DAR} = 0.6145$$

$$dp_{htxr} = 0.3 \text{ [psi]}$$

$$\eta_{ce} = 0.7$$

$$E_{htxr} = 0.83$$

$$\gamma = 1.4$$

$$\dot{m} = 14.16$$

$$N_{c,c} = 109524$$

$$N_{t,c} = 105396$$

$$PR_c = 1.292$$

$$PR_t = 2.177$$

$$COP_{dry} = 0.4299$$

$$\eta_c = 0.6317$$

$$\eta_t = 0.5873$$

$$\text{fluid}\$ = 'air'$$

$$g_{tot} = 0.006505$$

$$n = 0.2857$$

$$N_{phy} = 107598$$

$$PR_{ACM} = 1.7$$

$$PR_{ce} = 1.7$$

$$P_{1c} = 24.97 \text{ [psia]}$$

$$P_{1t} = 31.98$$

$$P_{2t} = 14.69$$

$$\dot{Q}_{\text{DAR}} = 1.296$$

$$R_{\text{amb}} = 0.4$$

$$T_{1t} = 540.9$$

$$T_{2t} = 477.6$$

$$T_{\text{amb}} = 531 \text{ [R]}$$

$$\dot{W}_c = 5.072 \text{ [Btu/lbm]}$$

$$\dot{W}_t = 5.072$$

$$P_{2c} = 32.28$$

$$P_{\text{amb}} = 14.69 \text{ [psia]}$$

$$\dot{Q}_{\text{dry}} = 0.9068$$

$$T_{1c} = 526 \text{ [R]}$$

$$T_{2c} = 589.3 \text{ [R]}$$

$$T_{2t,\text{DAR}} = 454.7$$

$$W_{c,c} = 7.769$$

$$\dot{W}_{ce} = 9.947$$

$$W_{t,c} = 6.642$$

No unit problems were detected.

EES suggested units (shown in purple) for T_2c W_dot_c .

Appendix K - ACM Single-Point Performance Comparison in EES


```

Function T_DAR(T_DB,g_tot,P_amb)
  g_sat = humrat(airh2o,T=T_DB,P=P_amb,R=1)
  g_ent = g_tot - g_sat

  If(g_ent > 0) Then
    If((T_DB-459.67) >=32) Then
      A:=((0.24+0.4332*g_sat+g_ent)*(T_DB-459.67)-1093*g_ent)/(0.24+0.4332*g_tot)
    Else
      A:=((0.24+0.4332*g_sat+0.5*g_ent)*(T_DB-459.67)-1221*g_ent)/(0.24+0.4332*g_tot)
    EndIf
  Else
    A:=T_DB
  EndIf

  T_DAR:=A+459.67
End

```

```

T_amb = 460 + 71
P_amb = 14.69
R_amb = 0.40

```

```

g_tot = humrat(airh2o, T=T_amb, P=P_amb, R=R_amb)

```

```

//////// User-defined Parameters //////////

```

```

P_ACM_inlet = 26.95
T_ACM_inlet = 460 + 66
W_ACM_inlet = 13.73

```

```

ACM_eta_c = 0.72

```

```

ACM_eta_t = 0.56

```

```

ACM_E_htxr = 0.83

```

```

ACM_dp_htxr = 0.3

```

```

//////// User-defined Parameters //////////

```

"Engine Parameters"

```

eta_c1 = 0.7

```

```

W_bleed = W_ACM_inlet

```

```

boost = P_ACM_inlet - P_amb {Comment this out for parametric table operations...}

```

```

E_htxr1 = 1

```

```

dp_htxr1 = 0

```

"ACM Parameters"

```

eta_c2 = ACM_eta_c

```

```

eta_t2 = ACM_eta_t

```

```

E_htxr2 = ACM_E_htxr

```

```

dp_htxr2 = ACM_dp_htxr

```

"Primary Compressor Power"

```

power_c1 = W_bleed*0.24*T_1c1*(PR_c1^0.285-1)/eta_c1*0.02358

```

"Power Balance"

```

power_c2 = W_bleed*0.24*T_1c2*(PR_c2^0.285-1)/eta_c2*0.02358

```

```

power_t2 = W_bleed*0.24*T_1t2*(PR_t2^0.285-1)/PR_t2^0.285*eta_t2*0.02358

```

power_c2 = power_t2

$T_{1c2}*(PR_{c2}^{0.285}-1)/\eta_{c2} = T_{1t2}*(PR_{t2}^{0.285}-1)/PR_{t2}^{0.285}\eta_{t2}$

"Pressure Ratios"

$PR_{c1} = (P_{amb} + \text{boost})/P_{amb}$

$PR_{c2} = P_{2c2}/P_{1c2}$

$PR_{t2} = P_{1t2}/P_{2t2}$

$PR_{ACM} = P_{1c2}/P_{2t2}$

$P_{2c1} = P_{amb} + \text{boost}$

$P_{1c2} = P_{ACM_inlet}$ {User-defined}

$P_{1t2} = P_{2c2} - dp_{htxr2}$

$P_{2t2} = P_{amb}$

"Temperatures"

$T_{1c1} = T_{amb}$

$T_{2c1} = T_{1c1} + T_{1c1}*(PR_{c1}^{0.285}-1)/\eta_{c1}$

$T_{1htxr1} = T_{2c1}$

$T_{2htxr1} = T_{1htxr1} - E_{htxr1}*(T_{1htxr1} - T_{amb})$

$T_{1c2} = T_{2htxr1}$

$T_{2c2} = T_{ACM_inlet}$ {User-defined}

$T_{1htxr2} = T_{2c2}$

$T_{2htxr2} = T_{1htxr2} - E_{htxr2}*(T_{1htxr2} - T_{amb})$

$T_{1t2} = T_{2htxr2}$

$T_{2t2} = T_{1t2} - \text{power}_{t2}/0.02358/W_{bleed}/0.24$

$T_{2t2_DAR} = T_{DAR}(T_{2t2}, g_{tot}, P_{amb})$

$Q_{DAR} = W_{bleed}*0.241*(T_{amb} - T_{2t2_DAR})*60/12000$

$COP_{DAR} = Q_{DAR}/(\text{power}_{c1}/0.02358)/60*12000$

{"Uncertainty Analysis Section"

"Measurement Uncertainties"

$T_{1c2_u} = 1.7$

$T_{2c2_u} = 1.7$

$P_{1c2_u} = 0.02$

$P_{2c2_u} = 0.02$

"Compressor Efficiency"

}

Appendix L - ACM Component Efficiency Variation Simulation

```

Function T_DAR(T_DB,g_tot,P_amb)
  g_sat = humrat(airh2o,T=T_DB,P=P_amb,R=1)
  g_ent = g_tot - g_sat

  If(g_ent > 0) Then
    If((T_DB-459.67) >=32) Then
      A:=((0.24+0.4332*g_sat+g_ent)*(T_DB-459.67)-1093*g_ent)/(0.24+0.4332*g_tot)
    Else
      A:=((0.24+0.4332*g_sat+0.5*g_ent)*(T_DB-459.67)-1221*g_ent)/(0.24+0.4332*g_tot)
    EndIf
  Else
    A:=T_DB
  EndIf

  T_DAR:=A+459.67
End

T_amb = 530
P_amb = 14.69
R_amb = 0.7

g_tot = humrat(airh2o, T=T_amb, P=P_amb, R=R_amb)

```

"Engine Parameters"

```

eta_c1 = 0.7
W_bleed = 15
{boost = 10}

```

```

E_htxr1 = 1
dp_htxr1 = 0

```

"ACM Parameters"

```

eta_c2 = 0.7
eta_t2 = 0.7

```

```

E_htxr2 = 0.7
dp_htxr2 = 0

```

"Primary Compressor Power"

```

power_c1 = W_bleed*0.24*T_1c1*(PR_c1^0.285-1)/eta_c1*0.02358

```

"Power Balance"

```

power_c2 = W_bleed*0.24*T_1c2*(PR_c2^0.285-1)/eta_c2*0.02358
power_t2 = W_bleed*0.24*T_1t2*(PR_t2^0.285-1)/PR_t2^0.285*eta_t2*0.02358

```

```

power_c2 = power_t2

```

```

" $T_{1c2}*(PR_{c2}^{0.285}-1)/eta_{c2} = T_{1t2}*(PR_{t2}^{0.285}-1)/PR_{t2}^{0.285}*eta_{t2}$ "

```

"Pressure Ratios"

```

PR_c1 = (P_amb+boost)/P_amb
PR_c2 = P_2c2/P_1c2
PR_t2 = P_1t2/P_2t2
PR_ACM = P_1c2/P_2t2

```

```

P_2c1= P_amb+boost

```

$$P_{1c2} = P_{2c1} - dp_{htxr1}$$

$$P_{1t2} = P_{2c2} - dp_{htxr2}$$

$$P_{2t2} = P_{amb}$$

"Temperatures"

$$T_{1c1} = T_{amb}$$

$$T_{2c1} = T_{1c1} + T_{1c1} * (PR_{c1}^{0.285} - 1) / \eta_{c1}$$

$$T_{1htxr1} = T_{2c1}$$

$$T_{2htxr1} = T_{1htxr1} - E_{htxr1} * (T_{1htxr1} - T_{amb})$$

$$T_{1c2} = T_{2htxr1}$$

$$T_{2c2} = T_{1c2} + T_{1c2} * (PR_{c2}^{0.285} - 1) / \eta_{c2}$$

$$T_{1htxr2} = T_{2c2}$$

$$T_{2htxr2} = T_{1htxr2} - E_{htxr2} * (T_{1htxr2} - T_{amb})$$

$$T_{1t2} = T_{2htxr2}$$

$$T_{2t2} = T_{1t2} - power_{t2} / 0.02358 / W_{bleed} / 0.24$$

$$T_{2t2_DAR} = T_{DAR}(T_{2t2}, g_{tot}, P_{amb})$$

$$Q_{DAR} = W_{bleed} * 0.241 * (T_{amb} - T_{2t2_DAR}) * 60 / 12000$$

$$COP_{DAR} = Q_{DAR} / (power_{c1} / 0.02358) / 60 * 12000$$

{"Uncertainty Analysis Section"

"Measurement Uncertainties"

$$T_{1c2_u} = 1.7$$

$$T_{2c2_u} = 1.7$$

$$P_{1c2_u} = 0.02$$

$$P_{2c2_u} = 0.02$$

"Compressor Efficiency"

}

Appendix M - ACM Post-processing Code in Matlab

```

% Chris Forster
% 5/2/2009

clear;
clc;
close all;

%% Ambient Test Conditions
Ambient_Pressure = 14.69;          %psia
% Ambient_Temperature = 83.5;      %Deg. F
Ambient_Temperature = 71;         %Deg. F
Rel_hum = 0.40;
%Tank_Temperature = [83.7 84.1 84.1 84.2 84.2 85.0 85.2 85.3 85.4 85.5 86.1 71]; %Deg F
Tank_Temperature = 65*ones(1,7);  %Deg F
%% Column Assignments - Data

acm.ch.t = 1;          % Timestamp
acm.ch.T1C = 3;        % Compressor Inlet Temperature [Deg. F]
acm.ch.T2C = 2;        % Compressor Outlet Temperature [Deg. F]
acm.ch.T1T = 4;        % Turbine Inlet Temperature [Deg. F]
acm.ch.T2T = 5;        % Turbine Outlet Temperature [Deg. F]
acm.ch.T_IC = 12;      % Intercooler Cold Flow Inlet Temperature [Deg. F]
acm.ch.P2C = 6;        % Compressor Outlet Pressure [psia]
acm.ch.P1C = 8;        % Compressor Inlet Pressure[psia]
acm.ch.P1T = 7;        % Turbine Inlet Pressure[psia]
acm.ch.LFE = 9;        % Laminar Flow Element [scfm]
acm.ch.ACC = 111;      % Accelerometer [g's]
acm.ch.N = 10;         % Turbocharger Speed [RPM]
acm.ch.ICC = 11;       % Intercooler cold flow inlet temperature [Deg. F]
%% Load psychometric table data
PsychometricTables = PsychometricTables(0);

%% Find humidity stuff...
water_mass_init = PsychometricLookup(Ambient_Pressure, Ambient_Temperature+460, Rel_hum, PsychometricTables)*7000; %grains

%% Load Data Files

% acm.files = ['test0001.lvm'; 'test0002.lvm'; 'test0003.lvm'; 'test0004.lvm'; 'test0005.lvm'; 'test0006.lvm'; 'test0007.lvm'; 'test0008.lvm'; 'test0009.lvm'; 'test0010.lvm'; 'test0011.lvm'; 'test0013.lvm']; % Separate files with semi-colon ';'
% acm.files = ['test0001.lvm'; 'test0002.lvm'; 'test0003.lvm'; 'test0004.lvm'; 'test0005.lvm'; 'test0006.lvm'; 'test0007.lvm'; 'test0008.lvm'; 'test0009.lvm']; %Separate files with semi-colon ';'

```

```

acm.files = ['test0001.lvm'; 'test0002.lvm'; 'test0003.lvm'; 'test0004.lvm'; 'test0005.lvm'; 'test0006.lvm'; 'test0007.lvm']; %Separate files with semi-colon ';'

for index2 = 1:length(acm.files(:,1))
    acm.testdata(index2).speedline = dlmread(acm.files(index2,:), '\t', 24);

    %% Column Assignments - Calculations

    % [1] - Compressor Pressure Ratio
    % [2] - Turbine Pressure Ratio
    % [3] - Compressor Efficiency
    % [4] - Compressor Power
    % [5] - Turbine Efficiency
    % [6] -
    % [7] -
    % [8] -
    % [9] -
    % [10] -
    % [11] -

    %% Calculations

    % Insert manually collected IC cold flow inlet temperatures
    acm.testdata(index2).speedline(:,11) = Tank_Temperature(index2);

    length_dataset(index2) = length(acm.testdata(index2).speedline(:,1));

    % Remove erroneous turbocharger speed points
    speed_tol = 0.05; % Percent difference/100

    for index = 1:length_dataset(index2)
        speed_mode(index2) = mode(acm.testdata(index2).speedline(:,10));
        if (abs((acm.testdata(index2).speedline(index,10)-speed_mode)/speed_mode) >= speed_tol)
            acm.testdata(index2).speedline(index,10) = speed_mode(index2);
        end
    end

    % Compressor Pressure Ratio
    for index = 1:length_dataset(index2)
        acm.calcs(index2).speedline(index, 1) = acm.testdata(index2).speedline(index,acm.ch.P2C)/acm.testdata(index2).speedline(index,acm.ch.P2C);
    end
end

```



```

ch.PlC);
    end

    acm.avg(1,index2) = mean(acm.calcs(index2).speedline(:,1));

    % Turbine Pressure Ratio
    for index = 1:length_dataset(index2)
        acm.calcs(index2).speedline(index, 2) = acm.testdata(index2).
speedline(index,acm.ch.PlT)/Ambient_Pressure;
    end

    acm.avg(2,index2) = mean(acm.calcs(index2).speedline(:,2));

    % Compressor Efficiency
    for index = 1:length_dataset(index2)
        acm.calcs(index2).speedline(index, 3) = (acm.testdata(index2).
speedline(index,acm.ch.TlC)+460)*(acm.calcs(index2).speedline(index,1)
^0.285-1)/((acm.testdata(index2).speedline(index,acm.ch.T2C)+460)-
(acm.testdata(index2).speedline(index,acm.ch.TlC)+460));
    end

    acm.avg(3,index2) = mean(acm.calcs(index2).speedline(:,3));

    % Mass Flowrate
    for index = 1:length_dataset(index2)
        acm.calcs(index2).speedline(index, 4) = acm.testdata(index2).
speedline(index,acm.ch.LFE)*acm.testdata(index2).speedline(index,acm.
ch.PlC)/(acm.testdata(index2).speedline(index,acm.ch.TlC)+460)*2.
699504268*viscosity_correction(acm.testdata(index2).speedline(index,
acm.ch.TlC));
    end

    acm.avg(4,index2) = mean(acm.calcs(index2).speedline(:,4));

    % Compressor Power
    for index = 1:length_dataset(index2)
        acm.calcs(index2).speedline(index, 5) = acm.calcs(index2).
speedline(index,4)*0.241*(acm.testdata(index2).speedline(index,acm.ch.
TlC)+460)/acm.calcs(index2).speedline(index,3)*(acm.calcs(index2).
speedline(index,1)^0.285-1)*0.02358;
    end

    acm.avg(5,index2) = mean(acm.calcs(index2).speedline(:,5));

    % Turbine Efficiency (WITH BEARING LOSSES)
    for index = 1:length_dataset(index2)
        acm.calcs(index2).speedline(index, 6) = acm.calcs(index2).
speedline(index,5)/acm.calcs(index2).speedline(index,4)/0.241/(acm.
testdata(index2).speedline(index,acm.ch.TlT)+460)/(acm.calcs(index2).

```

```

speedline(index,2)^0.285-1)*acm.calcs(index2).speedline(index,2)^0.285/0.02358;
end

acm.avg(6,index2) = mean(acm.calcs(index2).speedline(:,6));

% Cooling Load --no moisture--
for index = 1:length_dataset(index2)
    acm.calcs(index2).speedline(index, 7) = acm.calcs(index2).speedline(index,4)*0.241*(acm.testdata(index2).speedline(index,acm.ch.ICc)-acm.testdata(index2).speedline(index,acm.ch.T2T))*60/12000; % tons of cooling
end

acm.avg(7,index2) = mean(acm.calcs(index2).speedline(:,7));

% Corrected Compressor Flowrate
for index = 1:length_dataset(index2)
    acm.calcs(index2).speedline(index, 8) = acm.calcs(index2).speedline(index,4)*sqrt((acm.testdata(index2).speedline(index,acm.ch.T1C)+460)/545)/(acm.testdata(index2).speedline(index,acm.ch.P1C)*2.03625437/28.4);
end

acm.avg(8,index2) = mean(acm.calcs(index2).speedline(:,8));

% Corrected Turbine Flowrate
for index = 1:length_dataset(index2)
    acm.calcs(index2).speedline(index, 9) = acm.calcs(index2).speedline(index,4)*sqrt((acm.testdata(index2).speedline(index,acm.ch.T1T)+460)/519)/(acm.testdata(index2).speedline(index,acm.ch.P1T)*2.03625437/29.92);
end

acm.avg(9,index2) = mean(acm.calcs(index2).speedline(:,9));

% ACM Power Consumption (power from air tank or Perkins)
Eta_c_assumed = 0.70;

for index = 1:length_dataset(index2)
    %acm.calcs(index, 10) = acm.calcs(index,4)/(acm.testdata(index,acm.ch.P1T)/(acm.testdata(index,acm.ch.T1C)+460)*2.699504268)*(acm.testdata(index,acm.ch.P1C)-Ambient_Pressure)*0.00436363636;
    acm.calcs(index2).speedline(index, 10) = acm.calcs(index2).speedline(index,4)*0.241*(Ambient_Temperature+460)/Eta_c_assumed*((acm.testdata(index2).speedline(index,acm.ch.P1C)/Ambient_Pressure)^0.285-1)*0.02358;
end

```

```

acm.avg(10,index2) = mean(acm.calcs(index2).speedline(:,10));

% ACM Coefficient of Performance (COP) --DRY AIR--
for index = 1:length_dataset(index2)
    acm.calcs(index2).speedline(index, 11) = acm.calcs(index2).speedline(index,7)*4.71617735/acm.calcs(index2).speedline(index,10);
end

acm.avg(11,index2) = mean(acm.calcs(index2).speedline(:,11));

% ACM moist air saturation stuff
for index = 1:length_dataset(index2)
    acm.calcs(index2).speedline(index, 12) = PsychometricLookup(Ambient_Pressure,acm.testdata(index2).speedline(index,acm.ch.T2T)+460,1,PsychometricTables)*7000;    %grains
end

acm.avg(12,index2) = mean(acm.calcs(index2).speedline(:,12));

% ACM moist air DAR Temperature stuff (T_dar is stored here...)
for index = 1:length_dataset(index2)
    acm.calcs(index2).speedline(index, 13) = T_DAR(acm.testdata(index2).speedline(index,acm.ch.T2T),acm.calcs(index2).speedline(index,12),water_mass_init);
end

acm.avg(13,index2) = mean(acm.calcs(index2).speedline(:,13));

% ACM moist air T2T Saturation stuff (not used anymore...)
for index = 1:length_dataset(index2)
    acm.calcs(index2).speedline(index, 14) = PsychometricLookup(Ambient_Pressure,acm.testdata(index2).speedline(index,acm.ch.T2T)+460,1,PsychometricTables)*7000;    %grains
end

acm.avg(14,index2) = mean(acm.calcs(index2).speedline(:,14));

% Cooling Load --DAR--
for index = 1:length_dataset(index2)
    acm.calcs(index2).speedline(index, 15) = acm.calcs(index2).speedline(index,4)*0.241*(acm.testdata(index2).speedline(index,acm.ch.ICc)-acm.calcs(index2).speedline(index,13))*60/12000;    %tons of cooling
end

acm.avg(15,index2) = mean(acm.calcs(index2).speedline(:,15));

% Corrected Compressor Speed
for index = 1:length_dataset(index2)

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        acm.calcs(index2).speedline(index, 16) = acm.testdata(index2).speedline(index,acm.ch.N)/sqrt((acm.testdata(index2).speedline(index,acm.ch.T1C)+460)/545);
    end

    acm.avg(16,index2) = mean(acm.calcs(index2).speedline(:,16));

    % ACM Coefficient of Performance (COP) --DAR--
    for index = 1:length_dataset(index2)
        acm.calcs(index2).speedline(index, 17) = acm.calcs(index2).speedline(index,15)*4.71617735/acm.calcs(index2).speedline(index,10);
    end

    acm.avg(17,index2) = mean(acm.calcs(index2).speedline(:,17));

    % ACM IC Effectiveness Estimate (without cold flow inlet temp, assumed to be ambient...)
    %     for index = 1:length_dataset(index2)
    %         acm.calcs(index2).speedline(index, 18) = (acm.testdata(index2).speedline(index,acm.ch.T2C)-acm.testdata(index2).speedline(index,acm.ch.T1T))/(acm.testdata(index2).speedline(index,acm.ch.T2C)-acm.testdata(index2).speedline(index,acm.ch.T1C));
    %     end

    % ACM IC Effectiveness Estimate Using Measured IC Cold Flow Inlet Temperatures
    for index = 1:length_dataset(index2)
        acm.calcs(index2).speedline(index, 18) = (acm.testdata(index2).speedline(index,acm.ch.T2C)-acm.testdata(index2).speedline(index,acm.ch.T1T))/(acm.testdata(index2).speedline(index,acm.ch.T2C)-acm.testdata(index2).speedline(index,acm.ch.ICc));
    end

    acm.avg(18,index2) = mean(acm.calcs(index2).speedline(:,18));

    % Corrected Turbine Speed
    for index = 1:length_dataset(index2)
        acm.calcs(index2).speedline(index, 19) = acm.testdata(index2).speedline(index,acm.ch.N)/sqrt((acm.testdata(index2).speedline(index,acm.ch.T1T)+460)/519);
    end

    acm.avg(19,index2) = mean(acm.calcs(index2).speedline(:,19));

    % Turbine Efficiency (WITHOUT BEARING LOSSES)
    for index = 1:length_dataset(index2)
        %acm.calcs(index2).speedline(index, 20) = (1-(acm.calcs(index2).speedline(index,13)+460)/(acm.testdata(index2).speedline(index,acm.ch.T1T)+460))/((acm.calcs(index2).speedline(index,2)^0.285-

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```

1)/acm.calcs(index2).speedline(index,2)^0.285);
    acm.calcs(index2).speedline(index, 20) = (1-(acm.testdata(
(index2).speedline(index,acm.ch.T2T)+460)/(acm.testdata(index2).
speedline(index,acm.ch.T1T)+460))/((acm.calcs(index2).speedline(index,
2)^0.285-1)/acm.calcs(index2).speedline(index,2)^0.285);
    end

    acm.avg(20,index2) = mean(acm.calcs(index2).speedline(:,20));

    % Overall Cycle Pressure Ratio
    for index = 1:length_dataset(index2)
        acm.calcs(index2).speedline(index, 21) = acm.testdata(index2).
speedline(index,acm.ch.P1C)/Ambient_Pressure;
    end

    acm.avg(21,index2) = mean(acm.calcs(index2).speedline(:,21));

    % Bearing Efficiency
    for index = 1:length_dataset(index2)
        acm.calcs(index2).speedline(index, 22) = 1-(acm.calcs(index2).
speedline(index, 20)-acm.calcs(index2).speedline(index, 6));
    end

    acm.avg(22,index2) = mean(acm.calcs(index2).speedline(:,22));

    % Turbine Power (before bearing losses)
    for index = 1:length_dataset(index2)
        acm.calcs(index2).speedline(index, 23) = acm.calcs(index2).
speedline(index, 4) * 0.241 * (acm.testdata(index2).speedline(index,
acm.ch.T1T)-acm.testdata(index2).speedline(index,acm.ch.T2T)) *
0.02358;
    end

    acm.avg(23,index2) = mean(acm.calcs(index2).speedline(:,23));

    % Bearing Efficiency (Recalculated using power)
    for index = 1:length_dataset(index2)
        acm.calcs(index2).speedline(index, 24) = 1 - (acm.calcs(
(index2).speedline(index, 23) - acm.calcs(index2).speedline(index, 5))
/acm.calcs(index2).speedline(index, 23);
    end

    acm.avg(24,index2) = mean(acm.calcs(index2).speedline(:,24));

    % Compressor Outlet Temperature vs. overall ACM PR
    for index = 1:length_dataset(index2)
        acm.calcs(index2).speedline(index, 25) = acm.testdata(index2).
speedline(index, 2);
    end

```

```

acm.avg(25,index2) = mean(acm.calcs(index2).speedline(:,25));

for index3 = 1:10
    acm.avgcalc(index3,index2) = mean(acm.testdata(index2).
speedline(:,index3));
end

end

%% Plots

% Plot smoothing
datap = 100;
scatter_bit = 0;      % 0 scatter off / 1 scatter on

for index = 1:length(acm.avg(:,1))
    acm.xi(index,:) = min(acm.avg(index,:)):(max(acm.avg(index,:))-min
(acm.avg(index,:)))/datap:max(acm.avg(index,:));
end

for index = 1:length(acm.avgcalc(:,1))
    acm.xicalc(index,:) = min(acm.avgcalc(index,:)):(max(acm.avgcalc
(index,:))-min(acm.avgcalc(index,:)))/datap:max(acm.avgcalc(index,:));
end

% Compressor Locus of Operating Points
figure(1)
plot(acm.avg(8,:),acm.avg(1,:))
grid on
xlabel('Corrected Compressor Flow Rate [lbm/min]')
ylabel('Compressor Pressure Ratio [-]')
hold on
if scatter_bit == 1
    for index = 1:length(acm.files(:,1))
        scatter(acm.calcs(index).speedline(:,8),acm.calcs(index).
speedline(:,1))
    end
else
    scatter(acm.avg(8,:),acm.avg(1,:))
end

% Turbine Locus of Operating Points

```

```

figure(2)
plot(acm.avg(2,:),acm.avg(9,:))
grid on
xlabel('Turbine Pressure Ratio [-]')
ylabel('Corrected Turbine Flow Rate [lbm/min]')
hold on
if scatter_bit == 1
    for index = 1:length(acm.files(:,1))
        scatter(acm.calcs(index).speedline(:,2),acm.calcs(index).
speedline(:,9))
    end
else
    scatter(acm.avg(2,:),acm.avg(9,:))
end

% COP vs. Overall Cycle Pressure Ratio
figure(3)
plot(acm.xi(21,:),interp1(acm.avg(21,:),acm.avg(17,:),acm.xi
(21,:),'linear'))
grid on
xlabel('Overall ACM Pressure Ratio [-]')
ylabel('DAR COP [-]')
hold on
if scatter_bit == 1
    for index = 1:length(acm.files(:,1))
        scatter(acm.calcs(index).speedline(:,21),acm.calcs(index).
speedline(:,17))
    end
else
    scatter(acm.avg(21,:),acm.avg(17,:))
end

compressor_map_spdlms
hold on
plot(acm.avg(8,:),acm.avg(1:), 'black', 'LineWidth', 2)

% ACM Physical Mass Flow Rate vs. Overall Cycle Pressure Ratio
figure(5)
plot(acm.xi(21,:),interp1(acm.avg(21,:),acm.avg(4,:),acm.xi
(21,:),'linear'))
grid on
xlabel('Overall ACM Pressure Ratio [-]')
ylabel('ACM Physical Flow Rate [lbm/min]')
hold on
if scatter_bit == 1
    for index = 1:length(acm.files(:,1))
        scatter(acm.calcs(index).speedline(:,21),acm.calcs(index).
speedline(:,4))
    end
end

```

```

else
    scatter(acm.avg(21,:),acm.avg(4,:))
end

Fan_FlowEstimation

% Intercooler Effectiveness vs. ACM Physical Mass Flowrate
figure(7)
plot(acm.xi(4,:),interp1(acm.avg(4,:),acm.avg(18,:),acm.xi(
(4,:), 'linear'))
grid on
xlabel('ACM Physical Flow Rate [lbm/min]')
ylabel('Intercooler Effectiveness [-]')
hold on
if scatter_bit == 1
    for index = 1:length(acm.files(:,1))
        scatter(acm.calcs(index).speedline(:,4),acm.calcs(index).
speedline(:,18))
    end
else
    scatter(acm.avg(4,:),acm.avg(18,:))
end

%Temporary fix because two datapoints had the same speed...
%acm.avgcalc(10,10) = acm.avgcalc(10,10)+5;
%The above operation is an insignificant change in speed to make
interp1 work for this dataset

% Bearing Efficiency vs. Turbocharger Physical Speed
figure(8)
plot(acm.xicalc(10,:),interp1(acm.avgcalc(10,:),acm.avg(22,:),acm.
xicalc(10,:), 'linear'))
grid on
xlabel('ACM Physical Speed [RPM]')
ylabel('Bearing Efficiency [-]')
hold on
if scatter_bit == 1
    for index = 1:length(acm.files(:,1))
        scatter(acm.testdata(index).speedline(:,10),acm.calcs(index).
speedline(:,22))
        %scatter(acm.avgcalc(10,:),acm.avg(22,:))
    end
else
    scatter(acm.avgcalc(10,:),acm.avg(22,:))
end
end
%xlim([0 150e3])
%ylim([0.5 1])

% Cooling Capacity (DAR) vs. ACM Pressure Ratio

```



```

figure(9)
plot(acm.xi(21,:),interp1(acm.avg(21,:),acm.avg(15,:),acm.xi(
(21:),'linear'))
grid on
xlabel('ACM Pressure Ratio [-]')
ylabel('DAR Cooling Capacity [ton]')
hold on
if scatter_bit == 1
    for index = 1:length(acm.files(:,1))
        scatter(acm.calcs(index).speedline(:,21),acm.calcs(index).
speedline(:,15))
    end
else
    scatter(acm.avg(21,:),acm.avg(15,:))
end

% Compressor Power vs. ACM Pressure Ratio
figure(10)
plot(acm.xi(21,:),interp1(acm.avg(21,:),acm.avg(5,:),acm.xi(
(21:),'linear'))
grid on
xlabel('ACM Pressure Ratio [-]')
ylabel('Compressor Power [hp]')
hold on
if scatter_bit == 1
    for index = 1:length(acm.files(:,1))
        scatter(acm.calcs(index).speedline(:,21),acm.calcs(index).
speedline(:,5))
    end
else
    scatter(acm.avg(21,:),acm.avg(5,:))
end

% Turbine Power(before bearing losses) vs. ACM Pressure Ratio
figure(11)
plot(acm.xi(21,:),interp1(acm.avg(21,:),acm.avg(23,:),acm.xi(
(21:),'linear'))
grid on
xlabel('ACM Pressure Ratio [-]')
ylabel('Turbine Power [hp]')
hold on
if scatter_bit == 1
    for index = 1:length(acm.files(:,1))
        scatter(acm.calcs(index).speedline(:,21),acm.calcs(index).
speedline(:,23))
    end
else
    scatter(acm.avg(21,:),acm.avg(23,:))
end

```

```

% Bearing Efficiency vs. Turbocharger Physical Speed
figure(12)
plot(acm.xicalc(10,:),interp1(acm.avgcalc(10,:),acm.avg(24,:),acm.xicalc(10,:), 'linear'))
grid on
xlabel('ACM Physical Speed [RPM]')
ylabel('Bearing Efficiency [-]')
hold on
if scatter_bit == 1
    for index = 1:length(acm.files(:,1))
        scatter(acm.testdata(index).speedline(:,10),acm.calcs(index).speedline(:,24))
        %scatter(acm.avgcalc(10,:),acm.avg(22,:))
    end
else
    scatter(acm.avgcalc(10,:),acm.avg(24,:))
end
%xlim([0 150e3])
ylim([0 1])

% Bearing Efficiency vs. Compressor Pressure Ratio
figure(13)
plot(acm.xi(1,:),interp1(acm.avg(1,:),acm.avg(24,:),acm.xi(1,:), 'linear'))
grid on
xlabel('Compressor Pressure Ratio [-]')
ylabel('Bearing Efficiency [-]')
hold on
if scatter_bit == 1
    for index = 1:length(acm.files(:,1))
        scatter(acm.calcs(index).speedline(:,1),acm.calcs(index).speedline(:,24))
    end
else
    scatter(acm.avg(1,:),acm.avg(24,:))
end

% Compressor Outlet Temperature vs. ACM Pressure Ratio
figure(14)
plot(acm.xi(21,:),interp1(acm.avg(21,:),acm.avg(25,:),acm.xi(21,:), 'linear'))
grid on
xlabel('ACM Pressure Ratio [-]')
ylabel('Compressor Outlet Temperature [deg. F]')
hold on
if scatter_bit == 1
    for index = 1:length(acm.files(:,1))
        scatter(acm.calcs(index).speedline(:,21),acm.calcs(index).speedline(:,25))
    end
else
    scatter(acm.avg(21,:),acm.avg(25,:))
end

```

```

speedline(:,25))
    end
else
    scatter(acm.avg(21,:),acm.avg(25,:))
end

% Compressor Efficiency vs. ACM Pressure Ratio
figure(15)
plot(acm.xi(21,:),interp1(acm.avg(21,:),acm.avg(3,:),acm.xi(
(21,:),'linear'))
grid on
xlabel('ACM Pressure Ratio [-]')
ylabel('Compressor Efficiency [-]')
hold on
if scatter_bit == 1
    for index = 1:length(acm.files(:,1))
        scatter(acm.calcs(index).speedline(:,21),acm.calcs(index).
speedline(:,3))
    end
else
    scatter(acm.avg(21,:),acm.avg(3,:))
end
ylim([0 1])

% Turbine Efficiency vs. ACM Pressure Ratio
figure(16)
plot(acm.xi(21,:),interp1(acm.avg(21,:),acm.avg(20,:),acm.xi(
(21,:),'linear'))
grid on
xlabel('ACM Pressure Ratio [-]')
ylabel('Turbine Efficiency Without Bearing Losses')
hold on
if scatter_bit == 1
    for index = 1:length(acm.files(:,1))
        scatter(acm.calcs(index).speedline(:,21),acm.calcs(index).
speedline(:,20))
    end
else
    scatter(acm.avg(21,:),acm.avg(20,:))
end
ylim([0 1])

```

```

%Program by Chris Forster
%July 15, 2008

%Input all humidity ratios in grains/lbm
%Input temperature in degrees F

%Returns Dry Air Rated Temperature in degrees F

function[T_DB] = T_DAR_DB(T_DAR,g_sat,g_tot)
    % Define g_ent (entrained water)
    if (g_tot >= g_sat)
        g_ent = g_tot - g_sat;
    else
        g_ent = 0;
    end

    % Convert grains to lbm
    g_sat = g_sat/7000;
    g_tot = g_tot/7000;
    g_ent = g_ent/7000;

    % Calculate T_DB from T_DAR assuming above 32F (DB)
    % Check assumption afterwards.

    T_DB_TBD = (T_DAR*(0.24+0.4332*g_tot)+1093*g_ent)/(0.24+0.4332*g_sat+g_ent);

    if (T_DB_TBD <= 32)
        T_DB_TBD = (T_DAR*(0.24+0.4332*g_tot)+1221*g_ent)/(0.24+0.4332*g_sat+0.5*g_ent);
    end

    T_DB = T_DB_TBD;    %Degree F
end

```

```

%Program by Chris Forster
%April 16, 2008

%Input all humidity ratios in grains/lbm
%Input temperature in degrees F

%Returns Dry Air Rated Temperature in degrees F

function[T_DAR] = T_DAR(T_DB,g_sat,g_tot)
    if g_sat > g_tot
        g_ent = 0;
        T_DAR = T_DB;
        return
    end

    if g_sat < g_tot
        g_ent = g_tot - g_sat;
    end

    if T_DB > 32
        T_DAR = ((0.24+0.4332*g_sat/7000+g_ent/7000)*T_DB-
1093*g_ent/7000)/(0.24+0.4332*g_tot/7000);
    end

    if T_DB <= 32
        T_DAR = ((0.24+0.4332*g_sat/7000+0.5*g_ent/7000)*T_DB-
1221*g_ent/7000)/(0.24+0.4332*g_tot/7000);
    end

end

```

```

% Chris Forster

x_fan = [1 2 3 4 5 6];
y_fan = [2375 2360 2350 2325 2280 2220];
p_fan = polyfit(x_fan,y_fan,2);
x2_fan = 1:50;
y2_fan = polyval(p_fan,x2_fan);

figure(6)
subplot(1,2,1)
scatter(x_fan,y_fan)
hold on
plot(x2_fan,y2_fan)
ylim([0 2500])
grid on
xlabel('Pressure Deficit [in. H2O]')
ylabel('Volumetric Flowrate [CFM]')
title('Zirgo Fan P/N ZF14')
legend('Location','NorthEast','Zirgo Datapoints','2nd-Order Fit')

% Calculate fan power requirements (absolute minimum)

power_fan = x2_fan.*y2_fan/27.7/229*745/12;

subplot(1,2,2)
plot(x2_fan,power_fan)
grid on
xlabel('Pressure Deficit [in. H2O]')
ylabel('Current Draw [A]')
title('Zirgo Fan P/N ZF14, 12V Supply')
ylim([0 250])

clear x_fan;
clear y_fan;
clear x2_fan;
clear y2_fan;
clear p_fan;
clear power_fan;

```

```

comp_data_loader

%comp_data_eta(:,1) = Corrected mass flow rate data
%comp_data_eta(:,2) = Pressure Ratio
%comp_data_eta(:,3) = Compressor Efficiency

acm.comp_x_eta = min(acm.comp_data.etafile(:,1)):0.05:max(acm.
comp_data.etafile(:,1));
acm.comp_y_eta = min(acm.comp_data.etafile(:,2)):0.01:max(acm.
comp_data.etafile(:,2));
acm.comp_y_eta = acm.comp_y_eta';
acm.comp_z_eta = griddata(acm.comp_data.etafile(:,1),acm.comp_data.
etafile(:,2),acm.comp_data.etafile(:,3),acm.comp_x_eta,acm.
comp_y_eta,'cubic');

%comp_data_speed(:,1) = Corrected Speed
%comp_data_speed(:,2) = Corrected mass flow rate data
%comp_data_speed(:,3) = Pressure Ratio

acm.comp_x_speed = min(acm.comp_data.speedfile(:,2)):0.01:max(acm.
comp_data.speedfile(:,2));
acm.comp_y_speed = min(acm.comp_data.speedfile(:,3)):0.01:max(acm.
comp_data.speedfile(:,3));
acm.comp_y_speed = acm.comp_y_speed';
acm.comp_z_speed = griddata(acm.comp_data.speedfile(:,2),acm.
comp_data.speedfile(:,3),acm.comp_data.speedfile(:,1),acm.
comp_x_speed,acm.comp_y_speed,'cubic');

clc;

```

```
%Chris Forster

%The file format is a tab-delimited text file

%This file will be run automatically...

acm.comp_data.speedfile = load('comp_data_speed.txt');
acm.comp_data.etafile = load('comp_data_eta.txt');
```



```

% Chris Forster
% 6/20/2009

% This function is a viscosity correction factor for LFE measurements.
% (Source: www.meriam.com)

function [up] = viscosity_correction(T)

v_coll = 50:159;
v_col2 = [1.03034 1.02877 1.02720 1.02564 1.02408 1.02253 1.02099
1.01945 1.01792 1.01639 1.01487 1.01336 1.01185 1.01035 1.00885
1.00736 1.00588 1.00440 1.00292 ...
1.00146 1.00000 0.99854 0.99709 0.99564 0.99420 0.99277 0.99134
0.98992 0.98850 0.98709 0.98568 0.98428 0.98288 0.98149 0.98010
0.97872 0.97734 0.97597 0.97461 0.97325 ...
0.97189 0.97054 0.96919 0.96785 0.96651 0.96518 0.96386 0.96253
0.96122 0.95991 0.95860 0.95729 0.95600 0.95470 0.95341 0.95213
0.95085 0.94957 0.94830 0.94704 ...
0.94578 0.94452 0.94327 0.94202 0.94077 0.93953 0.93830 0.93707
0.93584 0.93462 0.93340 0.93219 0.93098 0.92977 0.92857 0.92737
0.92618 0.92499 0.92380 0.92262 ...
0.92144 0.92027 0.91910 0.91794 0.91678 0.91562 0.91446 0.91331
0.91217 0.91103 0.90989 0.90875 0.90762 0.90650 0.90537 0.90425
0.90314 0.90203 0.90092 0.89981 ...
0.89871 0.89761 0.89652 0.89543 0.89434 0.89326 0.89218 0.89110
0.89003 0.88896];

v_table = [v_coll' v_col2'];

up = interp1(v_table(:,1),v_table(:,2),T);

end

```

```

% Chris Forster

comp_data_loader
comp_data_process

figure(4)
hold off
contourf(acm.comp_x_eta,acm.comp_y_eta,acm.comp_z_eta,20)
xlabel('Corrected Compressor Flowrate [lbm/min]')
ylabel('Compressor Pressure Ratio [-]')
%title('GT1244 Turbocharger')
h = colorbar('vert');
ylabel(h,'Compressor Efficiency [-]')
grid on
xlim([2.5 14.5])
ylim([1 2.5])

hold on

scatter3(acm.comp_data.speedfile(:,2),acm.comp_data.speedfile(:,3),
acm.comp_data.speedfile(:,1))

x = 1:0.1:15;

% N = 120krpm
p120 = polyfit(acm.comp_data.speedfile(1:7,2),acm.comp_data.speedfile(
(1:7,3),2);
spd120 = polyval(p120,x);

% N = 140krpm
p140 = polyfit(acm.comp_data.speedfile(8:14,2),acm.comp_data.speedfile(
(8:14,3),2);
spd140 = polyval(p140,x);

% N = 160krpm
p160 = polyfit(acm.comp_data.speedfile(15:21,2),acm.comp_data.
speedfile(15:21,3),2);
spd160 = polyval(p160,x);

% N = 180krpm
p180 = polyfit(acm.comp_data.speedfile(22:28,2),acm.comp_data.
speedfile(22:28,3),2);
spd180 = polyval(p180,x);

% N = 200krpm
p200 = polyfit(acm.comp_data.speedfile(29:35,2),acm.comp_data.
speedfile(29:35,3),2);
spd200 = polyval(p200,x);

```

```

hold on

plot(x,spd120,x,spd140,x,spd160,x,spd180,x,spd200)

g = legend('Location','NorthEast','Compressor Efficiency','Actual &
Speed Points','120krpm','140krpm','160krpm','180krpm','200krpm');

hold off

```

Appendix N - Perkins Diesel Post-processing Code in Matlab

```

%Chris Forster - Added turbocharger calculations, additional engine
%calculations and corrections

%Drew Lozano - Original program
%3/6/08
%
%Digalog Data Processing
%
%©2008

%clear everything
clear;
clc;
close all
%

%␣
setup*****␣
****

%filename
%file = 'TC1_Baseline_Engine_Map_001.prn';
file = '    0V_ACM.prn';
file2 = 'Perkins_Trial.prn';
%struct for data
diesel = struct;
Willans = struct;
%sets the RPM and Torque tolerances for data parsing
RPM_Tolerance = 100;
RPM_Minimum = 500;
Torque_Tolerance = 40;
Torque_Minimum = 25;

%constants
Displacement = 6;           %[L]
Stroke = 5.0;               %[in]
Bore = 3.937;               %[in]
CompressionRatio = 16.0;    %[-]
Cylinders = 6;              %[-]
Q_lhv = 18000;              %[btu/lbm]
Ambient_Pressure = 14.7;    %[Psi]
Dew_Point = 14.6999;        %[°C]
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%␣
%%%%%

%Import␣
Data%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%reads in Perkins DAQ files, removes empty line at end, transposes␣

```

```

data
data1 = dlmread(file,'\t',[10 3 26 2402]);    %2400,8400,14000
data2 = dlmread(file2,'\t',[10 3 26 200]);

% Remove last 200 pts from data1 because of bearing failure
data1(:,2201:2400) = [];

data = [data1 data2];
%data = data1;
data(18,:) = 0;

data(18,:) = [];
data = transpose(data);
%stores dimensions of the data array
data_dims = size(data);
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

%Data
Format%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% Data Array Column Headings
% 1 - Torque [ft-lbf]
% 2 - Dyno Speed [RPM]
% 3 - Engine Coolant Temperature [F]
% 4 - Engine Oil Pressure [psi]
% 5 - Engine Oil Temperature [F]
% 6 - Fuel Temperature [F]
% 7 - Fuel Mass Flow Rate [kg/hr]
% 8 - Ambient Air Temperature [F]
% 9 - Intercooler Charge Temperature [F]
% 10 - Intake Charge Pressure [psi]
% 11 - Intake Manifold Temperature [F]
% 12 - Intercooler Coolant Inlet Temperature [F]
% 13 - Intercooler Coolant Outlet Temperature [F]
% 14 - Exhaust Manifold Temperature [F]
% 15 - Exhaust Manifold Pressure [psi]
% 16 - Exhaust Temperature [F]
% 17 - Laminar Flow Element Meter [CFM]
% CALCS
% 18 - Power [hp]
% 19 - BSFC [lbm/(hp hr)]
% 20 - Fuel Conversion Efficiency [%]
% 21 - Intercooler Effectiveness [-]
% 22 - Atmospheric Volumetric Efficiency [%]
% 23 - Manifold Volumetric Efficiency [%]
% 24 - Air Mass Flow Rate [lbm/min]
% 25 - Mean Piston Speed [ft/sec]
% 26 - BMEP [psi]

```

```

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%

```

```

% index = 1;
% count = 1;
% count2 = 1;
% flag = true;
%
% %Parse
Data%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% %Data is parsed into sets of data based on RPM and Torque values
% %
% while(flag)
%
%     if ( (index + count) <= data_dims(1))
%         temp = [data(index,1) data(index,2)];
%         if (((data(index + count,1) < temp(1) + Torque_Tolerance) &&
% (data(index + count,1) > temp(1) - Torque_Tolerance)) && ((data(index
% + count,2) < temp(2) + RPM_Tolerance) && (data(index + count,2) > temp
% (2) - RPM_Tolerance)))
%             count = count + 1;
%         else
%             diesel(count2).data = data(index:index+count-1,:);
%             diesel(count2).samples = count;
%             index = index + count;
%             count = 1;
%             count2 = count2 + 1;
%         end
%     else
%         diesel(count2).data = data(index:index+count-1,:);
%         diesel(count2).samples = count;
%         index = index + count;
%         flag = false;
%     end
% end
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%

```

```

Torque = 1;
Speed = 2;
ECT = 3;
EOP = 4;
EOT = 5;
FT = 6;
FFR = 7;

```

```

Base_Index = 1;
Sample_Count = 1;

```

```

Data_Set = 1;
flag = true;
bad_data = false;

Sample_Maximum = 200;
Sample_Minimum = 20;

%Parse
Data%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%Data is parsed into sets of data based on RPM and Torque values
%
while(flag)

    if ( (Base_Index + Sample_Count) < data_dims(1))
        if ((data(Base_Index,Torque) > Torque_Minimum) && (data
(Base_Index,Speed) > RPM_Minimum) && (Sample_Count < Sample_Maximum))
            Operating_Point(Data_Set,:) = [data(Base_Index,Torque)
data(Base_Index,Speed)];
            if (((data(Base_Index + Sample_Count,1) < Operating_Point
(Data_Set,Torque) + Torque_Tolerance) && (data(Base_Index +
Sample_Count,Torque) > Operating_Point(Data_Set,Torque) -
Torque_Tolerance)) && ((data(Base_Index + Sample_Count,Speed) <
Operating_Point(Data_Set,Speed) + RPM_Tolerance) && (data(Base_Index +
Sample_Count,Speed) > Operating_Point(Data_Set,Speed) -
RPM_Tolerance)))
                Sample_Count = Sample_Count +1;
                if (data(Base_Index + Sample_Count,FFR) < 0)
                    bad_data = true;
                end
            else
                if ((Sample_Count >= Sample_Minimum) && (bad_data ==
false))
                    diesel(Data_Set).data = data(Base_Index:
Base_Index+Sample_Count-1,:);
                    diesel(Data_Set).samples = Sample_Count;
                    Data_Set = Data_Set + 1;
                else
                    bad_data = false;
                end
                Base_Index = Base_Index + Sample_Count;
                Sample_Count = 1;
            end
        elseif (Sample_Count == 1)
            Base_Index = Base_Index+1;
        else
            if ((Sample_Count >= Sample_Minimum) && (bad_data ==
false))
                diesel(Data_Set).data = data(Base_Index:

```



```

Base_Index+Sample_Count-1,:);
        diesel(Data_Set).samples = Sample_Count;
        Data_Set = Data_Set + 1;
    else
        bad_data = false;
    end
    Base_Index = Base_Index + Sample_Count;
    Sample_Count = 1;

end

else
    diesel(Data_Set).data = data(Base_Index:
Base_Index+Sample_Count-1,:);
    diesel(Data_Set).samples = Sample_Count;
    Base_Index = Base_Index + Sample_Count;
    flag = false;
end

end

end

diesel_dims = size(diesel);

%% Append ACM Test Data to the Perkins Dataset //Added by Chris
Forster
file_acm = 'acm_avg.txt';
data_acm = dlmread(file_acm,'\t');

% Remove bad dataset for bearing failure
data_acm(:,12)= [];

for count = 1:diesel_dims(2)
    diesel(count).data(:,18) = data_acm(1,count);
    diesel(count).data(:,19) = data_acm(2,count);
    diesel(count).data(:,20) = data_acm(3,count);
    diesel(count).data(:,21) = data_acm(4,count);
    diesel(count).data(:,22) = data_acm(5,count);
    diesel(count).data(:,23) = data_acm(6,count);
    diesel(count).data(:,24) = data_acm(7,count);
    diesel(count).data(:,25) = data_acm(8,count);
    diesel(count).data(:,26) = data_acm(9,count);
    diesel(count).data(:,27) = data_acm(10,count);
    diesel(count).data(:,28) = data_acm(11,count);
    diesel(count).data(:,29) = data_acm(12,count);
    diesel(count).data(:,30) = data_acm(13,count);
    diesel(count).data(:,31) = data_acm(14,count);
    diesel(count).data(:,32) = data_acm(15,count);
    diesel(count).data(:,33) = data_acm(16,count);

```

```

        diesel(count).data(:,34) = data_acm(17,count);
        diesel(count).data(:,35) = data_acm(18,count);
        diesel(count).data(:,36) = data_acm(19,count);
        diesel(count).data(:,37) = data_acm(20,count);
        diesel(count).data(:,38) = data_acm(21,count);
        diesel(count).data(:,39) = data_acm(22,count);
    end

% Note: The ACM GT1244 turbocharger had a bearing failure at
count=12...

%
Calculations%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%

%Power Calculations
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,1) = diesel(count).data(count2,1) *
diesel(count).data(count2,2) / 5252;
    end
end

%BSFC Calculations
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,2) = diesel(count).data(count2,7) *
2.204624418059/ diesel(count).calcs(count2,1);
    end
end

%Fuel Conversion Efficiency Calculations
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,3) = 2545 / (diesel(count).calcs
(count2,2) * Q_lhv);
    end
end

%Intercooler Effectiveness Calculations
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,4) = (diesel(count).data(count2,9)
- diesel(count).data(count2,11)) / (diesel(count).data(count2,9) -
diesel(count).data(count2,12));
    end
end

%Atmospheric Volumetric Efficiency Calculations

```

```

for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,5) = 2 * diesel(count).data(count2,17) / (0.035314666919 * Displacement * diesel(count).data(count2,2));
    end
end

%Inlet Volumetric Efficiency Calculations
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        %diesel(count).calcs(count2,6) = 2304 * diesel(count).data(count2,17) / (Stroke * Bore^2 * pi * diesel(count).data(count2,2)) * (Ambient_Pressure / (Ambient_Pressure + diesel(count).data(count2,10))) * (diesel(count).data(count2,11) + 459.67) / (diesel(count).data(count2,8) + 459.67);
        diesel(count).calcs(count2,6) = Ambient_Pressure/(diesel(count).data(count2,10)+Ambient_Pressure)*(diesel(count).data(count2,11)+460)/(diesel(count).data(count2,8)+460)*diesel(count).data(count2,17)/(0.25*pi*Bore^2*Stroke*Cylinders/1728*diesel(count).data(count2,2)/2);
    end
end

%Air Mass Flow Rate Calculations
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,7) = 2.699504268 * diesel(count).data(count2,17) * Ambient_Pressure / ( diesel(count).data(count2,8) + 459.67);
    end
end

%Mean Piston Speed Calculations
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,8) = 2 /60 * (Stroke/12) * diesel(count).data(count2,2);
    end
end

%BMEP Calculations
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,9) = diesel(count).calcs(count2,1) * 2 * 396000 / (6 * Stroke * pi * (Bore^2 /4) *diesel(count).data(count2,2));
    end
end

```

```

%% Turbocharger Calculation - Chris Forster

% Compressor PR Calcs
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,10) = (diesel(count).data(count2,10)+Ambient_Pressure)/Ambient_Pressure;
    end
end

% Turbine PR Calcs
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,11) = (diesel(count).data(count2,15)+Ambient_Pressure)/Ambient_Pressure;
    end
end

% Compressor Efficiency Calcs
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,12) = (diesel(count).data(count2,8)+460)*(diesel(count).calcs(count2,10)^0.285-1)/(diesel(count).data(count2,9)-diesel(count).data(count2,8));
    end
end

% Compressor Power Calcs    (((Bad data)))
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,13) = diesel(count).calcs(count2,7)*0.241*(diesel(count).data(count2,8)+460)/diesel(count).calcs(count2,12)*(diesel(count).calcs(count2,10)^0.285-1)*0.02358;
    end
end

% Turbine Efficiency Calcs    (((Bad data)))
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,14) = diesel(count).calcs(count2,13)/(diesel(count).calcs(count2,7)+diesel(count).data(count2,7)*2.20462262/60)/0.285/(diesel(count).data(count2,14)+460)/(diesel(count).calcs(count2,11)^0.254-1)*diesel(count).calcs(count2,11)^0.254/0.02358;
    end
end

% Corrected Compressor Flow Calcs    (((Bad data)))
for count = 1:diesel_dims(2)

```

```

        for count2 = 1:diesel(count).samples
            diesel(count).calcs(count2,15) = diesel(count).calcs(count2,7)✓
            *sqrt((diesel(count).data(count2,8)+460)/545)/(Ambient_Pressure*2.✓
03625437/28.4);
        end
    end

% Corrected Turbine Flow Calcs    (((Bad data)))
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,16) = diesel(count).calcs(count2,7)✓
        *sqrt((diesel(count).data(count2,14)+460)/545)/((diesel(count).data✓
(count2,15)+Ambient_Pressure)*2.03625437/28.4);
    end
end

% Defining an assumed pressure differential across air filter element,
% since it is not measured.

% filter_dp = 0.108;          % Units of psi

% Collect LFE inlet pressure data at each operating point and load✓
into
% following table

% Data collected by Chris Forster 6/24/2009
% Some points are slightly modified to allow griddata to work✓
properly,
% without cutting off the edges of the maps.

Filter_data = [1250 50 0.4; ...
    1250      100      0.4; ...
    1750      50-20    0.7; ...
    1750      150      0.8; ...
    1750      400      1.4; ...
    2500+20    50-20    1.4; ...
    2500+20    150      1.8; ...
    2500+20    350      2.8; ...
    2500+20    375+20    3.1; ...
    2250      400+20    2.6; ...
    2000      420+20    2.0; ...
    1750      425+20    1.5; ...
    1500      450+20    1.0; ...
    1250      430+20    0.7; ...
    1000-25    380+20    0.4; ...
    1000-25    50-20     0.1; ...
    0          0        0];

% Convert in-H2O to psi

```

```

Filter_data(:,3) = Filter_data(:,3)/27.67;

% Expand data grid for filter data

Filter.gd.RPM = 900:20:2600;
Filter.gd.Torque = 30:2:500;
Filter.gd.Torque = Filter.gd.Torque';
Filter.gd.dp = griddata(Filter_data(:,1),Filter_data(:,2),Filter_data(:,3),Filter.gd.RPM,Filter.gd.Torque,'linear');

% Air Filter Pressure Drops
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        LFE_temp(count).calcs(count2,1) = interp2(Filter.gd.RPM,Filter.gd.Torque,Filter.gd.dp,diesel(count).data(count2,2),diesel(count).data(count2,1));
    end
end

% Updated LFE Flowrates
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,17) = diesel(count).data(count2,17) * (70+459.67)/(diesel(count).data(count2,8)+459.67) * (Ambient_Pressure)/14.69*viscosity_correction(diesel(count).data(count2,8));
        % diesel(count).calcs(count2,17) = diesel(count).data(count2,17) * (70+459.67)/(diesel(count).data(count2,8)+459.67) * (Ambient_Pressure-LFE_temp(count).calcs(count2,1))/14.69*viscosity_correction(diesel(count).data(count2,8));
        % diesel(count).calcs(count2,17) = diesel(count).data(count2,17) * (70+459.67)/(diesel(count).data(count2,8)+459.67) * (Ambient_Pressure-P_LFE(count))/14.69*viscosity_correction(diesel(count).data(count2,8));
    end
end

% Mass flowrate using updated LFE flowrates
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,18) = 2.699504268 * diesel(count).calcs(count2,17) * (Ambient_Pressure) / (diesel(count).data(count2,8) + 459.67);
        % diesel(count).calcs(count2,18) = 2.699504268 * diesel(count).calcs(count2,17) * (Ambient_Pressure-LFE_temp(count).calcs(count2,1)) / (diesel(count).data(count2,8) + 459.67);
    end
end

```

```

% Intake Air Density using updated LFE flowrates (Compressor Inlet)
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,19) = 2.699504268 *
(Ambient_Pressure) / ( diesel(count).data(count2,8) + 459.67);
        % diesel(count).calcs(count2,19) = 2.699504268 *
(Ambient_Pressure-LFE_temp(count).calcs(count2,1)) / ( diesel(count).
data(count2,8) + 459.67);
    end
end

% Manifold Volumetric Efficiency using updated LFE flowrates
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,20) = diesel(count).calcs(count2,
18)/((0.25*pi*Bore^2*Stroke*Cylinders/1728*diesel(count).data(count2,
2)/2)*2.699504268*(diesel(count).data(count2,10)+Ambient_Pressure) / (
diesel(count).data(count2,8) + 459.67));
    end
end

% Exhaust Manifold Exhaust Density Calculations
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,21) = 2.699504268*(1.122/0.8456) *
(Ambient_Pressure+diesel(count).data(count2,15)) / ( diesel(count).
data(count2,14) + 459.67);
    end
end

% Exhaust Manifold Volumetric Flowrate
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,22) = (diesel(count).calcs(count2,
18)+diesel(count).data(count2,7)*0.0367437104)/diesel(count).calcs
(count2,21);
    end
end

% Exhaust Manifold - Pumping Losses (hp)
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,23) = 0.00436363636 * diesel
(count).calcs(count2,22)*diesel(count).data(count2,15);
    end
end

% Compressor Efficiency Calcs (using updated LFE values...doesn't
affect this one)

```

```

for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,24) = (diesel(count).data(count2,8)✓
+460)*(diesel(count).calcs(count2,10)^0.285-1)/(diesel(count).data✓
(count2,9)-diesel(count).data(count2,8));
    end
end

% Compressor Power Calcs (using updated LFE values)
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,25) = diesel(count).calcs(count2,✓
18)*0.241*(diesel(count).data(count2,8)+460)/diesel(count).calcs✓
(count2,12)*(diesel(count).calcs(count2,10)^0.285-1)*0.02358;
    end
end

% Turbine Efficiency Calcs (using updated LFE values)
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,26) = diesel(count).calcs(count2,✓
25)/(diesel(count).calcs(count2,18)-diesel(count).data(count2,21)✓
+diesel(count).data(count2,7)*2.20462262/60)/0.285/(diesel(count).data✓
(count2,14)+460)/(diesel(count).calcs(count2,11)^0.254-1)*diesel✓
(count).calcs(count2,11)^0.254/0.02358;
    end
end

% Corrected Compressor Flow Calcs (using updated LFE values)
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,27) = diesel(count).calcs(count2,✓
18)*sqrt((diesel(count).data(count2,8)+460)/545)/(Ambient_Pressure*2.✓
03625437/28.4);
    end
end

% Corrected Turbine Flow Calcs (using updated LFE values)
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,28) = (diesel(count).calcs(count2,✓
18)-diesel(count).data(count2,21)+diesel(count).data(count2,7)*2.✓
20462262/60)*sqrt((diesel(count).data(count2,14)+460)/545)/((diesel✓
(count).data(count2,15)+Ambient_Pressure)*2.03625437/28.4);
    end
end

% Engine Pressure Differential (exhaust - intake)
for count = 1:diesel_dims(2)

```



```

        for count2 = 1:diesel(count).samples
            diesel(count).calcs(count2,29) = diesel(count).data(count2,15) - diesel(count).data(count2,10);
        end
    end

% Intake Manifold Air Density
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,30) = 2.699504268 * (Ambient_Pressure+diesel(count).data(count2,10)) / (diesel(count).data(count2,11) + 459.67);
    end
end

% Engine mass flow rate (total - acm)
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,31) = diesel(count).calcs(count2,18) - diesel(count).data(count2,21);
    end
end

% Engine Volumetric Flow Rate (@ intake manifold)
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,32) = diesel(count).calcs(count2,31)/diesel(count).calcs(count2,30);
    end
end

% Exhaust manifold volumetric flow rate using only the engine m_dot
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,33) = (diesel(count).calcs(count2,31)+diesel(count).data(count2,7)*0.0367437104)/diesel(count).calcs(count2,21);
    end
end

% Engine Overall Pumping Losses
for count = 1:diesel_dims(2)
    for count2 = 1:diesel(count).samples
        diesel(count).calcs(count2,34) = 0.00436363636 * (diesel(count).calcs(count2,33)*diesel(count).data(count2,15)-diesel(count).calcs(count2,32)*diesel(count).data(count2,10));
    end
end

```

```

%% End Turbocharger Calculations

temp = size(diesel(1).calcs);
%Calculate average data for sets of data
for count = 1:diesel_dims(2)
    for count2 = 1:39
        diesel(count).average(count2) = mean(diesel(count).data(:,
count2));
    end
    for count2 = 40:(39+temp(2))
        diesel(count).average(count2) = mean(diesel(count).calcs(:,
count2-39));
    end
end
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

%% Temp
averages = [];

for count = 1:diesel_dims(2)
    averages = [averages; diesel(count).average];
end

%
Plots%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

%Data Parsing - BSFC contour map
%RPM, Torque axes

%creates RPM,Torque, and BSFC vectors based on averaged data
for count = 1:diesel_dims(2)
    RPM(count) = diesel(count).average(2);
    Torque(count) = diesel(count).average(1);
    BSFC(count) = diesel(count).average(19+22);
    TC.PR_C(count) = diesel(count).average(27+22);
Added by Chris Forster
    TC.PR_T(count) = diesel(count).average(28+22);
Added by Chris Forster
    TC.ETA_C(count) = diesel(count).average(29+22);
Added by Chris Forster
    TC.P_C(count) = diesel(count).average(30+22);
Added by Chris Forster
    TC.ETA_T(count) = diesel(count).average(31+22);
Added by Chris Forster
    TC.W_C_C(count) = diesel(count).average(32+22);

```

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Added by Chris Forster
    TC.W_T_C(count) = diesel(count).average(33+22);           %✓
Added by Chris Forster
    TC.TlT(count) = diesel(count).average(14);                 %Added✓
by Chris Forster
    IC_eff(count) = diesel(count).average(21+22);              %✓
Added by Chris Forster
    IC_Tout(count) = diesel(count).average(11);                 %Added✓
by Chris Forster
    TC.W_phy(count) = diesel(count).average(24+22);            %✓
Added by Chris Forster
    TC.VE_man(count) = diesel(count).average(23+22);            %✓
Added by Chris Forster
    TC.W_fuel(count) = diesel(count).average(7)*0.0367437104;   %Added✓
by Chris Forster
    TC.Boost(count) = diesel(count).average(10);                %Added✓
by Chris Forster
    TC.T_amb(count) = diesel(count).average(8);                 %Added✓
by Chris Forster
    TC.W_phy_updated(count) = diesel(count).average(35+22);     %✓
Added by Chris Forster
    TC.VE_man_updated(count) = diesel(count).average(37+22);    %✓
Added by Chris Forster
    TC.EGD(count) = diesel(count).average(38+22);               %✓
Added by Chris Forster
    TC.Pumping(count) = diesel(count).average(39+22);           %✓
Added by Chris Forster
    TC.EGP(count) = diesel(count).average(15);                  %Added✓
by Chris Forster
    TC.ETA_C_updated(count) = diesel(count).average(41+22);     %✓
Added by Chris Forster
    TC.P_C_updated(count) = diesel(count).average(42+22);       %✓
Added by Chris Forster
    TC.ETA_T_updated(count) = diesel(count).average(43+22);     %✓
Added by Chris Forster
    TC.W_C_C_updated(count) = diesel(count).average(44+22);     %✓
Added by Chris Forster
    TC.W_T_C_updated(count) = diesel(count).average(45+22);     %✓
Added by Chris Forster
    TC.Vflow_updated(count) = diesel(count).average(34+22);     %✓
Added by Chris Forster
    TC.EDP(count) = diesel(count).average(46+22);               %✓
Added by Chris Forster
    TC.Pumping_dp(count) = diesel(count).average(51+22);        %✓
Added by Chris Forster
    ACM.DARcap(count) = diesel(count).average(32);               %✓
Added by Chris Forster

end

```

```

%Determines the RPM set points used
%Creates a vector of RPM set points
clear temp;
temp = sort(RPM);
count = 1;
for count2 = 1:diesel_dims(2)
    if ((count == 1) && (temp(count2) > RPM_Minimum) )
        Base_RPM(count) = temp(count2);
        count = count + 1;
        count2 = count2 + 1;
    else if ((count > 1) && (temp(count2) > Base_RPM(count-1) +
RPM_Tolerance))
        Base_RPM(count) = temp(count2);
        count = count + 1;
    end
end
end

%Determines the Torque set points used
%Creates a vector of Torque set points
clear temp;
temp = sort(Torque);
count = 1;
for count2 = 1:diesel_dims(2)
    if ((count == 1) && (temp(count2) > Torque_Minimum) )
        Base_Torque(count) = temp(count2);
        count = count + 1;
        count2 = count2 + 1;
    else if ((count > 1) && (temp(count2) > Base_Torque(count-1) +
Torque_Tolerance/2))
        Base_Torque(count) = temp(count2);
        count = count + 1;
    end
end
end

%Determines the dimensions of the useful averaged data matrix
RPM_Points = size(Base_RPM);
Torque_Points = size(Base_Torque);

%Determines the position of the BSFC data point in the RPM/Torque
matrix
%based on its RPM and Torque set points
for count = 1:diesel_dims(2)
    for count2 = 1:RPM_Points(2)
        for count3 = 1:Torque_Points(2)
            if ( ((RPM(count) > (Base_RPM(count2) - RPM_Tolerance/2))
&& (RPM(count) < (Base_RPM(count2) + RPM_Tolerance/2))) && ((Torque

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(count) > (Base_Torque(count3) - Torque_Tolerance/2)) && (Torque_Tolerance/2))
(count) < (Base_Torque(count3) + Torque_Tolerance/2))))
        BSFC_Torque_Contour(count3,count2) = BSFC(count);
    end
end
end

%index1 = RPM_Points(2);
%index2 = Torque_Points(2);
%for count2 = 1:index1
%    for count3 = 1:index2
%        if ( (count3 == 1) && (BSFC_Torque_Contour(count3,count2)~=0))
%            BSFC_Torque_Contour(count3,count2) == BSFC_Torque_Contour
(count3+1,count2);
%        else if (BSFC_Torque_Contour(count3,count2) == 0)
%            BSFC_Torque_Contour(count3,count2) = BSFC_Torque_Contour
(count3-1,count2);
%        end
%    end
% end
%end

% %BSFC Contour map
% figure(1)
% clabel(contour(Base_RPM,Base_Torque,BSFC_Torque_Contour,50));
% title('BSFC Contours');
% xlabel('Engine Speed (RPM)');
% ylabel('Torque (lbf-ft)');
% [C,h] = contour(Base_RPM,Base_Torque,BSFC_Contour);
% clabel(C,h)

%Data Parsing - BSFC Contour Map
%SP_ave, BMEP axes

%creates RPM,Torque, and BSFC vectors based on averaged data
for count = 1:diesel_dims(2)
    SP_ave(count) = diesel(count).average(25+22);
    BMEP(count) = diesel(count).average(26+22);
end

%Determines the SP_ave set points used
%Creates a vector of SP_ave set points
clear temp;
temp = sort(SP_ave);

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count = 1;
for count2 = 1:diesel_dims(2)
    if ((count == 1) && (temp(count2) > (2/60*(Stroke/12)*
RPM_Minimum)))
        Base_SP_ave(count) = temp(count2);
        count = count + 1;
        count2 = count2 + 1;
    else if ((count > 1) && (temp(count2) > Base_SP_ave(count-1) +
2/60*(Stroke/12)*RPM_Tolerance ))
        Base_SP_ave(count) = temp(count2);
        count = count + 1;
    end
end
end

%Determines the Torque set points used
%Creates a vector of Torque set points
clear temp;
temp = sort(BMEP);
count = 1;
for count2 = 1:diesel_dims(2)
    if ((count == 1) && (temp(count2) > (75.4*2*Torque_Minimum)/(6 *
Stroke * pi * (Bore^2 /4))))
        Base_BMEP(count) = temp(count2);
        count = count + 1;
        count2 = count2 + 1;
    else if ((count > 1) && (temp(count2) > Base_BMEP(count-1) + (75.4
*2*Torque_Tolerance/2)/(6 * Stroke * pi * (Bore^2 /4)) ))
        Base_BMEP(count) = temp(count2);
        count = count + 1;
    end
end
end

SP_ave_Points = size(Base_SP_ave);
BMEP_Points = size(Base_BMEP);

%Determines the position of the BSFC data point in the SP_ave/BMEP
matrix
%based on its RPM and Torque set points
for count = 1:diesel_dims(2)
    for count2 = 1:SP_ave_Points(2)
        for count3 = 1:BMEP_Points(2)
            if ( ((SP_ave(count) > (Base_SP_ave(count2) - (2/60*
(Stroke/12) * RPM_Tolerance/2))) && (SP_ave(count) < (Base_SP_ave
(count2) + (2/60*(Stroke/12) *RPM_Tolerance/2)))) && ((BMEP(count) >
(Base_BMEP(count3) - (75.4*2*Torque_Tolerance/2)/(6 * Stroke * pi *
(Bore^2 /4)))) && (BMEP(count) < (Base_BMEP(count3) + (75.4

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*2*Torque_Tolerance/2)/(6 * Stroke * pi * (Bore^2 /4))))))
        BSFC_BMEP_Contour(count3,count2) = BSFC(count);
    end
end
end
end

%index1 = SP_ave_Points(2);
%index2 = BMEP_Points(2);
%for count2 = 1:index1
%    for count3 = 1:index2
%        if ( (count3 == 1) && (BSFC_BMEP_Contour(count3,count2)==0))
%            BSFC_BMEP_Contour(count3,count2) = BSFC_BMEP_Contour(
(count3+1,count2);
%        else if (BSFC_BMEP_Contour(count3,count2) == 0)
%            BSFC_BMEP_Contour(count3,count2) = BSFC_BMEP_Contour(
(count3-1,count2);
%        end
%    end
%end
%end

```

```

% %BSFC Contour map
% figure(2)
% clabel(contour(Base_SP_ave,Base_BMEP,BSFC_BMEP_Contour,50));
% title('BSFC Contours');
% xlabel('Average Piston Speed (ft/sec)');
% ylabel('BMEP (psi)');
% % [C,h] = contour(Base_RPM,Base_Torque,BSFC_Contour);
% % clabel(C,h)

```

```

for count = 1:diesel_dims(2)
    FuelConversion_Efficiency(count) = diesel(count).average(20);
    mdot_fuel(count) = diesel(count).average(7)*2.2;
    Power(count) = diesel(count).average(18);
    Volumetric_man_Efficiency(count) = diesel(count).average(23);
    Volumetric_atm_Efficiency(count) = diesel(count).average(22);
    Boost(count) = diesel(count).average(10);
end

```

```

%Determines the position of the BSFC data point in the RPM/Torque
matrix
%based on its RPM and Torque set points
for count = 1:diesel_dims(2)
    for count2 = 1:RPM_Points(2)

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```

        for count3 = 1:Torque_Points(2)
            if ( ((RPM(count) > (Base_RPM(count2) - RPM_Tolerance/2)) &&
            && (RPM(count) < (Base_RPM(count2) + RPM_Tolerance/2))) && ((Torque
            (count) > (Base_Torque(count3) - Torque_Tolerance/2)) && (Torque
            (count) < (Base_Torque(count3) + Torque_Tolerance/2))))
                Volumetric_man_Efficiency_Contour(count3,count2) =
Volumetric_man_Efficiency(count);
                Volumetric_atm_Efficiency_Contour(count3,count2) =
Volumetric_atm_Efficiency(count);
            end
        end
    end
end

```

```

index1 = RPM_Points(2);
index2 = Torque_Points(2);
for count1 = 1:index1
    for count2 = 1:index2
        if ( (count2 == 1) && (Volumetric_man_Efficiency_Contour
(count2,count1)==0))
            Volumetric_man_Efficiency_Contour(count2,count1) =
Volumetric_man_Efficiency_Contour(count2+1,count1);
            else if (Volumetric_man_Efficiency_Contour(count2,count1) ==
0)
                Volumetric_man_Efficiency_Contour(count2,count1) =
Volumetric_man_Efficiency_Contour(count2-1,count1);
            end
        end
    end
end
end

```

```

index1 = RPM_Points(2);
index2 = Torque_Points(2);
for count1 = 1:index1
    for count2 = 1:index2
        if ( (count2 == 1) && (Volumetric_atm_Efficiency_Contour
(count2,count1)==0))
            Volumetric_atm_Efficiency_Contour(count2,count1) =
Volumetric_atm_Efficiency_Contour(count2+1,count1);
            else if (Volumetric_atm_Efficiency_Contour(count2,count1) ==
0)
                Volumetric_atm_Efficiency_Contour(count2,count1) =
Volumetric_atm_Efficiency_Contour(count2-1,count1);
            end
        end
    end
end
end

```



```

%         if ((count1 == 1) && (count2 ==1) && (Param(count2,count1) == 0))
%             Param(count2,count1) = (Param(count2+1,count1) + Param(count2,count1+1))/2;
%         else if ((count1 == 1) && (count2 > 1) && (count2 < index2) && (Param(count2,count1) == 0))
%             Param(count2,count1) = (Param(count2-1,count1) + Param(count2+1,count1) + Param(count2,count1+1))/3;
%         else if ((count1 == 1) && (count2 == index2) && (Param(count2,count1) == 0))
%             Param(count2,count1) = (Param(count2-1,count1) + Param(count2,count1+1))/2;
%         else if

% %Volumetric Efficiency (manifold) Contour map
% figure(3)
% subplot(2,1,1)
% clabel(contour(Base_RPM,Base_Torque,Volumetric_man_Efficiency_Contour,75));
% title('Volumetric Efficiency (manifold) Contours');
% xlabel('Engine Speed (rpm)');
% ylabel('Torque (lbf ft)');
%
% subplot(2,1,2)
% clabel(contour(Base_RPM,Base_Torque,Volumetric_atm_Efficiency_Contour,50));
% title('Volumetric Efficiency (atmospheric) Contours');
% xlabel('Engine Speed (rpm)');
% ylabel('Torque (lbf ft)');
%
%
% figure(4)
% subplot(3,1,1)
% scatter(Power,Power.*mdot_fuel);
% title('Willans');
% xlabel('Power (hp)');
% ylabel('Power*Mdot (hp lbm/hr)');
% subplot(3,1,2)
% scatter(RPM,Volumetric_man_Efficiency);
% title('Volumetric Efficiency (manifold)');
% xlabel('Engine Speed (rpm)');
% ylabel('Volumetric Efficiency');
% subplot(3,1,3)
% scatter(RPM,Volumetric_atm_Efficiency);

```

```

% title('Volumetric Efficiency (atmospheric)');
% xlabel('Engine Speed (rpm)');
% ylabel('Volumetric Efficiency');

%Boost Surface
for count = 1:diesel_dims(2)
    for count2 = 1:RPM_Points(2)
        for count3 = 1:Torque_Points(2)
            if ( (RPM(count) > (Base_RPM(count2) - RPM_Tolerance/2)) && (RPM(count) < (Base_RPM(count2) + RPM_Tolerance/2)) && (Torque(count) > (Base_Torque(count3) - Torque_Tolerance/2)) && (Torque(count) < (Base_Torque(count3) + Torque_Tolerance/2)))
                Boost_Contour(count3,count2) = Boost(count);
            end
        end
    end
end

% figure(5)
% surf(Base_RPM,Base_Torque,Boost_Contour);
% title('Boost');
% xlabel('Engine Speed (rpm)');
% ylabel('Torque (lbf ft)');

count = 1;
for index = 1:RPM_Points(2)
    for index2 = 1:diesel_dims(2)
        if( RPM(index2)> (Base_RPM(index) - RPM_Tolerance/2)) && (RPM(index2) < (Base_RPM(index) + RPM_Tolerance/2))
            Willans(index).data(count,1) = diesel(index2).average(26);
            Willans(index).data(count,2) = diesel(index2).average(7)/1000*3600;
            count = count + 1;
        end
    end
end
count = 1;

% figure(6)
% hold on
% for count = 1:RPM_Points(2)
%     scatter(Willans(count).data(:,1),Willans(count).data(:,2));

```

```

% end
% legend ('1000 RPM', '1200 RPM','1500 RPM','1800 RPM','2200
RPM','2400 RPM','2500 RPM', 'Location', 'EastOutside')
% xlabel ('BMEP (psi)')
% ylabel ('Fuel Flow Rate (g/s)')
% title ('Willans Plot')

%% Plots - Chris Forster

% Griddata interval parameters
TC.gd.int = 2;

% Griddata for compressor PR contours
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.PR_C = griddata(RPM,Torque,TC.PR_C,TC.gd.RPM,TC.gd.
Torque,'linear');

% Griddata for turbine PR contours
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.PR_T = griddata(RPM,Torque,TC.PR_T,TC.gd.RPM,TC.gd.
Torque,'linear');

% Griddata for compressor efficiency contours
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.ETA_C = griddata(RPM,Torque,TC.ETA_C,TC.gd.RPM,TC.gd.
Torque,'linear');

% Griddata for compressor power contours
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.P_C = griddata(RPM,Torque,TC.P_C,TC.gd.RPM,TC.gd.
Torque,'linear');

% Griddata for turbine efficiency contours
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.ETA_T = griddata(RPM,Torque,TC.ETA_T,TC.gd.RPM,TC.gd.
Torque,'linear');

% Griddata for corrected compressor flowrate contours
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);

```

```

TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.W_C_C = griddata(RPM,Torque,TC.W_C_C,TC.gd.RPM,TC.gd.
Torque,'linear');

% Griddata for corrected turbine flowrate contours
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.W_T_C = griddata(RPM,Torque,TC.W_T_C,TC.gd.RPM,TC.gd.
Torque,'linear');

% Griddata for turbine inlet temperature contours
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.TlT = griddata(RPM,Torque,TC.TlT,TC.gd.RPM,TC.gd.
Torque,'linear');

% Griddata for engine BSFC contours
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
engine.gd.BSFC = griddata(RPM,Torque,BSFC,TC.gd.RPM,TC.gd.
Torque,'linear');

% Griddata for engine compressor efficiency contours...over W*-PR
TC.gd.WPR.W_C_C = min(TC.W_C_C):TC.gd.int/10:max(TC.W_C_C);
TC.gd.WPR.PR_C = min(TC.PR_C):TC.gd.int/100:max(TC.PR_C);
TC.gd.WPR.PR_C = TC.gd.WPR.PR_C';
TC.gd.WPR.ETA_C = griddata(TC.W_C_C,TC.PR_C,TC.ETA_C,TC.gd.WPR.W_C_C,
TC.gd.WPR.PR_C,'linear');

% Griddata for engine turbine efficiency contours...over W*-PR
TC.gd.WPR.W_T_C = min(TC.W_T_C):TC.gd.int/10:max(TC.W_T_C);
TC.gd.WPR.PR_T = min(TC.PR_T):TC.gd.int/100:max(TC.PR_T);
TC.gd.WPR.PR_T = TC.gd.WPR.PR_T';
TC.gd.WPR.ETA_T = griddata(TC.W_T_C,TC.PR_T,TC.ETA_T,TC.gd.WPR.W_T_C,
TC.gd.WPR.PR_T,'linear');

% Griddata for physical mass flowrate contours
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.W_phy = griddata(RPM,Torque,TC.W_phy,TC.gd.RPM,TC.gd.
Torque,'linear');

% Griddata for manifold volumetric efficiency contours
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);

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TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
engine.gd.VE_man = griddata(RPM,Torque,TC.VE_man,TC.gd.RPM,TC.gd.
Torque,'linear');

% Griddata for fuel flow rate contours
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
engine.gd.W_fuel = griddata(RPM,Torque,TC.W_fuel,TC.gd.RPM,TC.gd.
Torque,'linear');

% Griddata for Boost contours
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.Boost = griddata(RPM,Torque,TC.Boost,TC.gd.RPM,TC.gd.
Torque,'linear');

% Griddata for ambient temperature contours
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.T_amb = griddata(RPM,Torque,TC.T_amb,TC.gd.RPM,TC.gd.
Torque,'linear');

% Griddata for physical mass flow rate contours (using updated LFE
calcs)
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.W_phy_updated = griddata(RPM,Torque,TC.W_phy_updated,TC.gd.RPM,
TC.gd.Torque,'linear');

% Griddata for manifold VE contours (using updated LFE calcs)
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.VE_man_updated = griddata(RPM,Torque,TC.VE_man_updated,TC.gd.
RPM,TC.gd.Torque,'linear');

% Griddata for exhaust gas density
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.EGD = griddata(RPM,Torque,TC.EGD,TC.gd.RPM,TC.gd.
Torque,'linear');

% Griddata for exhaust manifold pumping losses

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TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.Pumping = griddata(RPM,Torque,TC.Pumping,TC.gd.RPM,TC.gd.
Torque,'linear');

% Griddata for exhaust manifold pumping losses
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.EGP = griddata(RPM,Torque,TC.EGP,TC.gd.RPM,TC.gd.
Torque,'linear');

% Griddata for intercooler effectiveness contours
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.IC_eff = griddata(RPM,Torque,IC_eff,TC.gd.RPM,TC.gd.
Torque,'linear');

% Griddata for intercooler discharge temperature contours
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.IC_Tout = griddata(RPM,Torque,IC_Tout,TC.gd.RPM,TC.gd.
Torque,'linear');

% Griddata for compressor efficiency using updated LFE values
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.ETA_C_updated = griddata(RPM,Torque,TC.ETA_C_updated,TC.gd.RPM,
TC.gd.Torque,'linear');

% Griddata for compressor power using updated LFE values
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.P_C_updated = griddata(RPM,Torque,TC.P_C_updated,TC.gd.RPM,TC.
gd.Torque,'linear');

% Griddata for turbine efficiency (including bearing losses) using
updated
% LFE values
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.ETA_T_updated = griddata(RPM,Torque,TC.ETA_T_updated,TC.gd.RPM,
TC.gd.Torque,'linear');

```

```

% Griddata for corrected compressor flow using updated LFE values
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.W_C_C_updated = griddata(RPM,Torque,TC.W_C_C_updated,TC.gd.RPM,TC.gd.Torque,'linear');

% Griddata for corrected turbine flow using updated LFE values
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.W_T_C_updated = griddata(RPM,Torque,TC.W_T_C_updated,TC.gd.RPM,TC.gd.Torque,'linear');

% Griddata for volumetric flowrate using updated LFE values
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.Vflow_updated = griddata(RPM,Torque,TC.Vflow_updated,TC.gd.RPM,TC.gd.Torque,'linear');

% Griddata for engine pressure differential
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.EDP = griddata(RPM,Torque,TC.EDP,TC.gd.RPM,TC.gd.Torque,'linear');

% Griddata for pumping losses using engine dp
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
TC.gd.Pumping_dp = griddata(RPM,Torque,TC.Pumping_dp,TC.gd.RPM,TC.gd.Torque,'linear');

% Griddata for ACM DAR Cooling Capacity
TC.gd.RPM = min(RPM):TC.gd.int:max(RPM);
TC.gd.Torque = min(Torque):TC.gd.int:max(Torque);
TC.gd.Torque = TC.gd.Torque';
ACM.gd.DARcap = griddata(RPM,Torque,ACM.DARcap,TC.gd.RPM,TC.gd.Torque,'linear');

speed.min = 1000;
speed.max = 1600;
tq.min = 50;
tq.max = 300;

figure(7)

```

```

contourf(TC.gd.RPM,TC.gd.Torque,engine.gd.BSFC,25)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('Engine BSFC')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[lbm/hp-hr]')

figure(8)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.TlT,20)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('Engine Exhaust Gas Temperature (EGT)')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[deg. F]')

figure(9)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.ETA_C,20)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('TC Compressor Efficiency')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[-]')

figure(10)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.ETA_T,20)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('TC Turbine Efficiency')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[-]')

figure(11)
contourf(TC.gd.WPR.W_C_C,TC.gd.WPR.PR_C,TC.gd.WPR.ETA_C,(0.40:0.05:1.0))

```



```

0.70))
axis square
grid on
title('TC Compressor Efficiency')
xlabel('Corrected Compressor Flowrate [lbm/min]')
ylabel('Compressor Pressure Ratio [-]')
h = colorbar('vert');
ylabel(h, '[-]')

figure(12)
contourf(TC.gd.WPR.W_T_C,TC.gd.WPR.PR_T,TC.gd.WPR.ETA_T,20)
axis square
grid on
title('TC Turbine Efficiency')
xlabel('Corrected Turbine Flowrate [lbm/min]')
ylabel('Turbine Pressure Ratio [-]')
h = colorbar('vert');
ylabel(h, '[-]')

figure(13)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.W_C_C)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('TC Corrected Compressor Flowrate')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[lbm/min]')

figure(14)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.W_T_C,20)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('TC Corrected Turbine Flowrate')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[lbm/min]')

figure(15)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.IC_eff,
[0.7,0.72,0.74,0.76,0.78,0.8,0.82,0.84,0.86])
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square

```

```

grid on
title('Intercooler Effectiveness')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[-]')
ylim(h, [min(abs(IC_eff)) max(IC_eff)])

figure(16)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.IC_Tout,20)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('Intercooler Discharge Temperature')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[Deg. F]')

figure(17)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.W_phy,20)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('TC Physical Flowrate')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[lbm/min]')

figure(18)
contourf(TC.gd.RPM,TC.gd.Torque,engine.gd.VE_man,20)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('Engine Manifold Volumetric Efficiency')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[-]')

figure(19)
contourf(TC.gd.RPM,TC.gd.Torque,engine.gd.W_fuel,20)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square

```

```

grid on
title('Fuel Flowrate')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[lbm/min]')

figure(20)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.Boost,15)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('Boost Map')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[psig]')

figure(21)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.W_phy/0.074456,20)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('SCFM Map')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[SCFM]')

figure(22)
contourf(TC.gd.RPM,TC.gd.Torque,engine.gd.VE_man,20)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('Engine Manifold Volumetric Efficiency')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[-]')

figure(23)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.T_amb,20)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on

```

```

title('Ambient Room Temperature')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[Deg. F]')

figure(24)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.W_phy_updated,20)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('Air Mass Flowrate') %Using updated LFE values
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[lbm/min]')

figure(25)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.W_phy,20)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('Mass Flowrate (not updated)')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[lbm/min]')

figure(26)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.VE_man_updated,20)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('Manifold Volumetric Efficiency ') %Using updated LFE Data
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[-]')

figure(27)
contourf(Filter.gd.RPM,Filter.gd.Torque,Filter.gd.dp)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('Air Filter Pressure Drop')

```

```

xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[psi]')

figure(28)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.EGD)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('Exhaust Manifold Gas Density')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[lbm/ft^3]')

figure(29)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.Pumping)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('Exhaust Manifold Pumping Losses')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[hp]')

figure(30)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.EGP)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('Exhaust Manifold Pressure')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[psig]')

figure(31)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.ETA_C_updated, 20)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('Compressor Efficiency')
xlabel('Engine Speed [RPM]')

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```

ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[-]')

figure(32)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.P_C_updated)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('Compressor Power')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[hp]')

figure(33)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.ETA_T_updated)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('Turbine Efficiency')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[-]')

figure(34)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.W_C_C_updated,20)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('Corrected Compressor Flow')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[lbm/min]')

figure(35)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.W_T_C_updated,20)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('Corrected Turbine Flow')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')

```

```

h = colorbar('vert');
ylabel(h, '[lbm/min]')

figure(36)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.EDP,20)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('Engine Pressure Differential')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[psi]')

figure(37)
contourf(TC.gd.RPM,TC.gd.Torque,TC.gd.Pumping_dp,20)
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('Engine Pumping Losses')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[hp]')

figure(38)
contourf(TC.gd.RPM,TC.gd.Torque,ACM.gd.DARcap,(0:max(max(ACM.gd.DARcap))/15:max(max(ACM.gd.DARcap))))
xlim([speed.min speed.max])
ylim([tq.min tq.max])
axis square
grid on
title('ACM DAR Cooling Capacity')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[ton]')
caxis([0 max(max(ACM.gd.DARcap))])
ylim(h,[0 max(max(ACM.gd.DARcap))])

% figure(15)
% contourf(TC.gd.RPM,TC.gd.Torque,engine.gd.VE_man,(0.8:0.025:1.25))
% xlim([1000 2500])
% ylim([50 450])
% axis square

```

```

% grid on
% title('Manifold Volumetric Efficiency and Boost Comparison Map')
% xlabel('Engine Speed [RPM]')
% ylabel('Engine Load [lbf-ft]')
% h = colorbar('vert');
% ylabel(h, '[-]')
% hold on
% contour(TC.gd.RPM,TC.gd.Torque,TC.gd.Boost)

%% End Plots - Chris Forster
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%
%%

%cleanup
%clear Ambient_Pressure Bore Stroke Cylinders CompressionRatio data1
data2;
%clear file1 file2 Dew_Point Displacement Q_lhv RPM_Tolerance
Torque_Tolerance;
%clear diesel_dims count count2 file flag index temp RPM Torque BSFC
BMEP;
%clear count3 Torque_Minimum RPM_Minimum BMEP_Points RPM_Points
Torque_Points;
%clear SP_ave_Points;

```



```

% Chris Forster
% 6/20/2009

% This function is a viscosity correction factor for LFE measurements.

function [up] = viscosity_correction(T)

v_coll = 50:159;
v_col2 = [1.03034 1.02877 1.02720 1.02564 1.02408 1.02253 1.02099 1.01945 1.01792 1.01639 1.01487 1.01336 1.01185 1.01035 1.00885 1.00736 1.00588 1.00440 1.00292 ...
1.00146 1.00000 0.99854 0.99709 0.99564 0.99420 0.99277 0.99134 0.98992 0.98850 0.98709 0.98568 0.98428 0.98288 0.98149 0.98010 0.97872 0.97734 0.97597 0.97461 0.97325 ...
0.97189 0.97054 0.96919 0.96785 0.96651 0.96518 0.96386 0.96253 0.96122 0.95991 0.95860 0.95729 0.95600 0.95470 0.95341 0.95213 0.95085 0.94957 0.94830 0.94704 ...
0.94578 0.94452 0.94327 0.94202 0.94077 0.93953 0.93830 0.93707 0.93584 0.93462 0.93340 0.93219 0.93098 0.92977 0.92857 0.92737 0.92618 0.92499 0.92380 0.92262 ...
0.92144 0.92027 0.91910 0.91794 0.91678 0.91562 0.91446 0.91331 0.91217 0.91103 0.90989 0.90875 0.90762 0.90650 0.90537 0.90425 0.90314 0.90203 0.90092 0.89981 ...
0.89871 0.89761 0.89652 0.89543 0.89434 0.89326 0.89218 0.89110 0.89003 0.88896];

v_table = [v_coll' v_col2'];

up = interp1(v_table(:,1),v_table(:,2),T);

end

```

Appendix O - Perkins Post-processing Comparison in Matlab

```

% Chris Forster

clear
clc
close all;

load -mat comparison_data

compare.gd.rpm = 1000:5:2500;
compare.gd.torque = (50:2:500)';

% Regrid ACM data to common grid for comparison
compare.acm.bsfc = interp2(acm.gd.rpm,acm.gd.torque,acm.bsfc,compare.gd.rpm(ones(1,length(compare.gd.torque)),:),compare.gd.torque(:,ones(1,length(compare.gd.rpm))), 'linear',NaN);
compare.acm.fuel = interp2(acm.gd.rpm,acm.gd.torque,acm.fuel,compare.gd.rpm(ones(1,length(compare.gd.torque)),:),compare.gd.torque(:,ones(1,length(compare.gd.rpm))), 'linear',NaN);
compare.acm.pumping = interp2(acm.gd.rpm,acm.gd.torque,acm.pumping,compare.gd.rpm(ones(1,length(compare.gd.torque)),:),compare.gd.torque(:,ones(1,length(compare.gd.rpm))), 'linear',NaN);
compare.acm.boost = interp2(acm.gd.rpm,acm.gd.torque,acm.boost,compare.gd.rpm(ones(1,length(compare.gd.torque)),:),compare.gd.torque(:,ones(1,length(compare.gd.rpm))), 'linear',NaN);
compare.acm.DARcap = interp2(acm.rem.rpm,acm.rem.tq,acm.DARcap,compare.gd.rpm(ones(1,length(compare.gd.torque)),:),compare.gd.torque(:,ones(1,length(compare.gd.rpm))), 'linear',NaN);

% Regrid Baseline data to common grid for comparison
compare.baseline.bsfc = interp2(baseline.gd.rpm,baseline.gd.torque,baseline.bsfc,compare.gd.rpm(ones(1,length(compare.gd.torque)),:),compare.gd.torque(:,ones(1,length(compare.gd.rpm))), 'linear',NaN);
compare.baseline.fuel = interp2(baseline.gd.rpm,baseline.gd.torque,baseline.fuel,compare.gd.rpm(ones(1,length(compare.gd.torque)),:),compare.gd.torque(:,ones(1,length(compare.gd.rpm))), 'linear',NaN);
compare.baseline.pumping = interp2(baseline.gd.rpm,baseline.gd.torque,baseline.pumping,compare.gd.rpm(ones(1,length(compare.gd.torque)),:),compare.gd.torque(:,ones(1,length(compare.gd.rpm))), 'linear',NaN);
compare.baseline.boost = interp2(baseline.gd.rpm,baseline.gd.torque,baseline.boost,compare.gd.rpm(ones(1,length(compare.gd.torque)),:),compare.gd.torque(:,ones(1,length(compare.gd.rpm))), 'linear',NaN);
compare.baseline.VE_man = interp2(baseline.gd.rpm,baseline.gd.torque,baseline.VE_man,compare.gd.rpm(ones(1,length(compare.gd.torque)),:),compare.gd.torque(:,ones(1,length(compare.gd.rpm))), 'linear',NaN);

% Regrid stock turbocharger data to common grid for comparison
compare.stock.VE_man = interp2(stock.gd.rpm,stock.gd.torque,stock.VE_man,compare.gd.rpm(ones(1,length(compare.gd.torque)),:),compare.gd.torque(:,ones(1,length(compare.gd.rpm))), 'linear',NaN);

```

```

compare.stock.bsfc = interp2(stock.gd.rpm,stock.gd.torque,stock.bsfc,
compare.gd.rpm(ones(1,length(compare.gd.torque)),:),compare.gd.torque
(:,ones(1,length(compare.gd.rpm))), 'linear',NaN);

% Comparing difference in pumping losses
compare.diff.pumping = compare.acm.pumping - compare.baseline.pumping;

% Calculating power based on difference
compare.diff.bsfcpower = (compare.acm.fuel - compare.baseline.fuel).
/compare.baseline.bsfc*60;

% Calculate percent difference in methods
compare.diff.method = (compare.diff.pumping - compare.diff.bsfcpower).
/compare.diff.bsfcpower*100;

% Compare difference in boost
compare.diff.boost = compare.acm.boost - compare.baseline.boost;

% Calculate COP based on BSFC differences
compare.acm.DARcop1 = compare.acm.DARcap*4.71617735./compare.diff.
bsfcpower;

% Calculate COP based on BSFC differences
compare.acm.DARcop2 = compare.acm.DARcap*4.71617735./compare.diff.
pumping;

% Compare stock turbo and GT      VE_man data (percent difference)
compare.diffstock.VE_man = (compare.baseline.VE_man - compare.stock.
VE_man)./compare.stock.VE_man*100;

% Compare stock turbo and GT      VE_man data (difference only)
compare.diffstock.VE_man2 = (compare.baseline.VE_man - compare.stock.
VE_man);

% Compare stock turbo and GT      bsfc data
compare.diffstock.bsfc = (compare.baseline.bsfc - compare.stock.bsfc).
/compare.stock.bsfc*100;

% Arrange boost-percent difference in method data into a vector for 2d
plot
compare.diff.boostsize = size(compare.diff.boost);
compare.diff.methodsize = size(compare.diff.method);
compare.twod.boostlength = compare.diff.boostsize(1)*compare.diff.
boostsize(2);
compare.twod.methodlength = compare.diff.methodsize(1)*compare.diff.
methodsize(2);

compare.twod.boost = reshape(compare.diff.boost,compare.twod.
boostlength,1);

```

```

compare.twod.method = reshape(compare.diff.method,compare.twod.␣
methodlength,1);

% Plots
figure(1)
contourf(compare.gd.rpm,compare.gd.torque,compare.diff.pumping,20)
xlim([1000 1600])
ylim([50 300])
axis square
grid on
title('Net Engine Pumping Losses')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[hp]')

figure(2)
scatter(compare.twod.boost,compare.twod.method)
grid on
ylim([-75 75])

figure(3)
contourf(compare.gd.rpm,compare.gd.torque,compare.diff.method,(-75:10:␣
75))
xlim([1000 1600])
ylim([50 300])
axis square
grid on
title('Percent Difference Relative to BSFC Method')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, ' [%]')

figure(4)
contourf(compare.gd.rpm,compare.gd.torque,compare.acm.DARcop1,[-0.1 0␣
0.1 0.2 0.3 0.4 0.5 0.525 0.55 0.575 0.6 0.65])
xlim([1000 1600])
ylim([50 300])
axis square
grid on
title('ACM COP Based on BSFC Method')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, ' [-]')

```

```

figure(5)
contourf(compare.gd.rpm,compare.gd.torque,compare.acm.DARcop2,20)
xlim([1000 1600])
ylim([50 300])
axis square
grid on
title('ACM COP Based on Pumping Losses Method')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[-]')

figure(6)
contourf(compare.gd.rpm,compare.gd.torque,compare.diffstock.VE_man,20)
xlim([1000 2500])
ylim([50 450])
axis square
grid on
title('Percent Difference in stock and GT      Intake Manifold VE')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, ' [%]')

figure(7)
contourf(compare.gd.rpm,compare.gd.torque,compare.diffstock.VE_man2*100,20)
xlim([1000 2500])
ylim([50 450])
axis square
grid on
title('Difference in stock and GT      Intake Manifold VE')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, ' [%]')

figure(8)
contourf(compare.gd.rpm,compare.gd.torque,compare.diffstock.bsfc,20)
xlim([1000 2500])
ylim([50 450])
axis square
grid on
title('Percent Difference in stock and GT      BSFC')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, ' [%]')

```

```
figure(9)
contourf(compare.gd.rpm,compare.gd.torque,compare.diff.bsfcpower,20)
xlim([1000 1600])
ylim([50 300])
axis square
grid on
title('Power Required by ACM using BSFC Method')
xlabel('Engine Speed [RPM]')
ylabel('Engine Load [lbf-ft]')
h = colorbar('vert');
ylabel(h, '[hp]')
```