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

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ABSTRACT

The Reverse-Brayton cycle has been used for aircraft cabin cooling for many decades. However, air-cycle cooling hasn't been popular in the automotive field yet. This study demonstrates that air-cycle technology can provide sufficient cooling for certain applications. The primary focus is a novel forced induction engine control system, where compressor bleed is used both to provide engine boost control and air-conditioning. The bleed-air drives an air-cycle machine (ACM) consisting of typical automotive components: a turbocharger, heat exchanger, and ducting. The components of an ACM system are lightweight and compact compared to those of a typical vapor compression system; both qualities are critical in high performance applications, where such a system seems to make most sense. The ACM was tested first on a test stand and then directly on an engine, in a bootstrap-cycle configuration. The turbocharged test engine's intake manifold pressure was controlled by bleeding air from the outlet of the engine's intercooler and feeding the ACM compressor inlet. Once the compressed air was supplied to the ACM it was further compressed by the ACM, cooled by the secondary intercooler, and expanded through the ACM turbine. The engine's turbocharger was resized to compensate for the increased air flow during ACM operation. The results show that a dry-air-rated (DAR) coefficient of performance (COP) of 0.73 and a DAR cooling capacity of 1.5 tons are possible on a test stand, and a DAR COP of 0.56 and a DAR cooling capacity of 0.72 tons are possible on-engine. The data available from the on-engine testing was limited to lower ACM pressure ratios due to a bearing failure before full testing was complete; performance would likely increase with higher inlet pressures, as shown by the compressed air test stand results. The test results strongly suggest that continued development and in-vehicle testing will provide adequate air-conditioning and engine performance, using only the most benign and environmentally friendly working fluid: air.

INTRODUCTION

Automotive racing and high performance applications generally require components that are light-weight, compact, and create minimum loss of power available at the wheels. Thus, air-conditioning is typically considered an unnecessary luxury, since it adds weight, bulk and parasitic load on the powerplant. The inclusion of air-conditioning is usually forgone. As a result of the exclusion of air-conditioning, driver fatigue is increased due to the high temperatures experienced in the drivers' cockpit. However, an air-conditioning system based on the Reverse-Brayton cycle can provide a significant amount of cooling with minimum impact on engine performance and fuel consumption.

The air-conditioning system based on the Reverse-Brayton cycle uses air as the working fluid. This has benefits of safety, weight, and space requirements since air is non-toxic, non-flammable, and doesn't require containment in a closed-system. The air-cycle machine (ACM) consists of a compressor, heat exchanger, and turbine. The ACM has been used in aircraft applications for many decades and has proven to be reliable and effective in applications sensitive to weight-associated fuel costs and a readily-available source of compressed air. In these types of applications an air-cycle can be more effective in reducing fuel consumption than typical vapor-compression air-conditioning systems despite the lower coefficient of performance (COP) associated with the air cycle [1]. A detailed comparison of COP, total system weight, and overall fuel consumption for several types of air-conditioning systems can be found in reference [1] by Scofield.

The passive ACM operation is based on a novel forced induction engine control system. A turbocharged engine generally requires some form of regulation of intake manifold pressure (i.e. boost pressure). Typically this is achieved by a turbine-bypass valve to limit power available to the compressor, resulting in lower boost pressure. Turbine by-pass operation is often called waste-gating, due to the energy lost in free expansion occurring across the valve opening that could theoretically have been used for useful work. The ACM operation presented here is one application that can make use of this available energy. The novel forced induction engine control system utilizes an air-bleed valve to feed the inlet of the ACM compressor using air from the outlet of the engine's intercooler, *upstream* of the engine. This provides the coolest possible air charge to the ACM while imposing no additional power consumption from the engine's compressor, while still limiting engine boost pressures to the same acceptable levels that waste-gating provides. Thus, the bleed-air configuration eliminates the need for the waste-gate for boost control, although it may be a good safety feature to leave in place to limit boost pressure spikes over the design limit. Since the power derived from limiting engine boost pressures would have been wasted before incorporating the ACM, it can be considered a "free" source of power for the ACM. However, any power consumption beyond the losses associated with waste-gating are considered the cost of operating the ACM.

The ACM utilizes typical automotive turbocharger components that are readily available and cost-effective to implement. The main components are an automotive turbocharger, intercooler, and ducting. The ACM is configured in a bootstap-cycle that further compresses the air supplied from the engine before cooling it through a secondary intercooler and expanding it through the cooling turbine. The additional pressure rise that the ACM compressor provides is necessary in the relatively low-pressure automotive applications compared to typical aircraft systems. Most automotive turbocharger systems are single-stage, as opposed to multi-stage compression standard in aircraft jet engines. The additional pressure rise creates a larger pressure ratio across the cooling turbine, providing lower ACM outlet temperatures and increasing cooling capacity. The power required by the ACM compressor is supplied by a direct coupling to the cooling turbine, consisting of the turbine stage of the automotive turbocharger.

BACKGROUND

ACM OPTIONS and PERFORMANCE PARAMETERS

There are several types of ACM's that have been used for various aircraft applications, such as the simple cycle, two-wheel bootstrap, three-wheel bootstrap, and modified cycles with evaporative cooling [2]. Each has their own advantages and disadvantages. The two-wheel bootstrap cycle was chosen because it can provide lower turbine outlet temperatures than comparable configurations while still easily lending itself to the adaption of an automotive turbocharger as the core component of the system. Simple and three-wheel cycles were ruled out because the addition of an on-shaft cooling fan for the secondary intercooler increases the complexity of the modified turbocharger. The two-wheel bootstrap-cycle configuration provides adequate performance with minimal complexity. A schematic of the implementation of the aircraft two-wheel bootstrap cycle is shown in Figure 1.

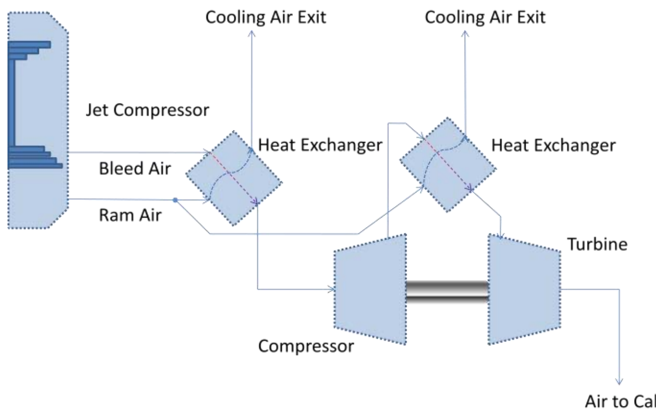

Figure 1. Aircraft schematic of a typical two-wheel bootstrap ACM configuration.

Traditionally air-conditioning cycle COP has been defined as the ratio of the cooling capacity to the power required by the system.

$$COP_{traditional} = \frac{\dot{Q}_{cooling}}{\dot{W}_{compressor,fans}}$$

This is a good representation of relative performance between systems where there are no energy sources with zero cost associated with them. In the situation with the ACM configured with a turbocharged engine, there is an energy source with no associated cost. This is the energy that would have been lost due to waste-gating to control engine boost pressures. The energy is lost as heat in the exhaust stream during waste-gating. An "effective COP" that more accurately represents the true cost of the ACM operation in the present system is proposed, with the work rate term being defined by the change in fuel flow rate with brake-specific fuel consumption (bsfc) considered.

$$COP_{effective} = \frac{\dot{Q}_{cooling}}{\dot{W}_{ACM\ Operation}}$$

$$\dot{W}_{ACM\ Operation} = \frac{\Delta \dot{m}_{fuel,ACM}}{bsfc}$$

This work rate term accounts for the increase in engine power consumption due to ACM operation. The change in fuel consumption provides a direct measure of the overall impact of

ACM operation on engine operation. In the results presented in this paper, however, the experimental ACM apparatus tested never operated in the modulated engine power regime where the 'traditional' and 'effective' COP would differ, for reasons outlined below. The amount of 'free' energy from avoiding waste-gate operation increases with engine load, where waste-gate operation typically runs at a higher duty cycle.

EFFECT OF HUMIDITY ON ACM PERFORMANCE

Ambient humidity levels affect the performance of all air-conditioning systems. Dry-air rated (DAR) analysis is used here for determining the cooling capacity of the system [3]. DAR temperature is the equivalent dry-bulb temperature of the ambient air if the entrained water or ice in the air exiting the cooling turbine is adiabatically evaporated. Dry air is assumed for most of the analysis, except for the cooling turbine. The changes in enthalpy across most of the components in the system have a minimal dependence on moisture content of the air; however, this is not true for the cooling turbine since the temperature reaches a low enough temperature for moisture to drop out as liquid water or ice.

ACM COMPONENT SELECTION

The turbine-compressor matching process for the ACM turbocharger is different than a matching process performed for an engine turbocharger. This is primarily due to the difference in turbine inlet conditions. Corrected parameters, such as flow and speed, for the turbine depend on temperature. The turbine inlet temperature is significantly lower in an ACM application than the exhaust gas temperature of a gasoline or diesel engine. This presents a challenge for designing the test setup because commercial turbochargers are designed for engine applications. Thus, the ACM turbocharger was developed by matching the compressor wheel of one engine turbocharger with the turbine wheel of another engine turbocharger, and a commercial housing is adapted to fit both wheels and shaft. The difference in turbocharger conditions can be seen on the T-s diagram in Figure 2 of both the Brayton and Reverse-Brayton cycles, for engine and ACM applications respectively.

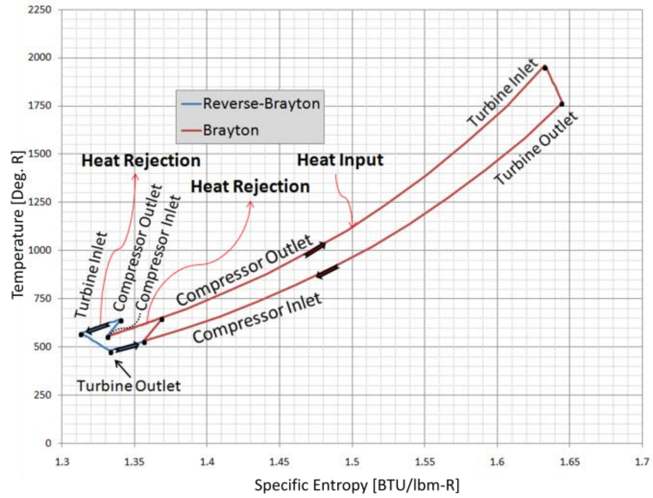

Figure 2. Comparison of the Brayton and Reverse-Brayton cycles.

Before assembling the ACM for testing, a parametric study of theoretical performance was performed and is summarized in Table 1. Efficiency and effectiveness values are based on realistic performance projections using modern turbocharger and heat exchanger components; the compressor pressure ratio selected is a representative design point for such a machine. The values of COP_{DAR} presented below were calculated using the traditional method and not the proposed effective COP method. Note, however, that the same value would be obtained with the "effective COP" method, if the particular running operating point considered below does not involve waste-gate operation of the turbocharger.

Table 1. Tabulated data from theoretical parametric study of ACM performance with 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

The results of the parametric study indicate the turbine efficiency has the greatest impact on overall ACM performance. Compressor efficiency and intercooler effectiveness have a similar impact on overall ACM performance, although much less than that of turbine efficiency. This information is used to guide the turbine-compressor matching process in terms of suitable compromises based on the availability of existing commercially-produced components.

PRIOR WORK

The closest previous work recently published in this research area was performed by a group at Queen's University, Belfast, in 2005. Their work focused on providing cooling for refrigerated tractor-trailers, specifically for the trailer refrigeration system [4,5]. In their work, the primary compressor is gear-driven from the crankshaft and feeds a bootstrap-cycle ACM. Instead of using an air-bleed system, the supercharger is dedicated to supplying air for the ACM. This configuration somewhat decouples engine and ACM operation, making it easier to control and quantify ACM performance. However, it has a severe shortcoming in that it will always be a parasitic power loss from the engine's crankshaft, rather than "free" power from a turbocharger-based system that is demonstrated in the current work. The theoretical COP of this group's ACM was 0.294, corresponding to the $COP_{traditional}$ defined above, while an optimized model showed a possibility for a COP of 0.62. Their performance goal was not achieved due to excessively low efficiency of the gearbox driving the supercharger, heat exchanger performance, and excessively large turbomachinery bearing losses. They termed their ACM as a "Cold Air Unit," and described it as mechanically equivalent to a turbocharger. However, they determined a standard turbocharger would not meet the requirements of their application. Their turbine rotor was substantially modified and vaned-nozzles were incorporated in the turbine inlet. Their ACM required over three times more power than required to drive the vapor-cycle unit it was replacing. But their setup differed significantly in several key aspects to the system presented here, and so their results cannot easily be compared directly to those presented here. More importantly, there is no range of operation where their ACM does not directly impact the performance of the engine.

EXPERIMENTAL SETUP

Two testing configurations were utilized for validating performance predictions, including the turbine-compressor matching process. The first configuration consisted of the ACM supplied with compressed air from the engine test facility's air compressor and storage tanks, henceforth referred to as the “shop air” source. In the second configuration the ACM was connected to a Perkins diesel engine mounted on a dynamometer. Figure 3 indicates the sensor types and locations for the shop air test stand configuration.

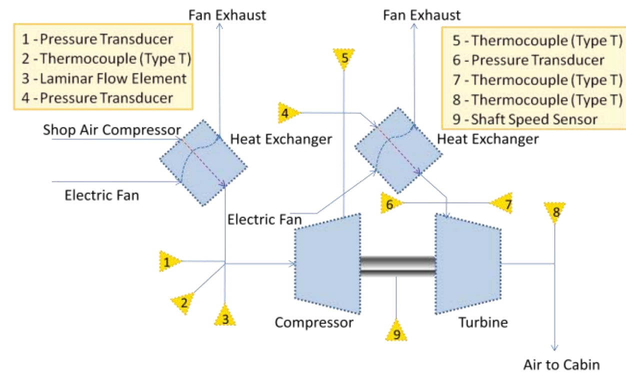

Figure 3. The ACM configured for shop air supply.

The test facility uses a 75-hp reciprocating compressor to maintain two large storage tanks at 125 psig. This supply is regulated down to 11 psig before reaching the ACM. While the 11 psig is representative of the boost pressures seen on the Perkins diesel engine, it creates a large load on the shop compressor because it has to supply the same mass flow rate that the ACM consumes but at 125 psig. With the air source configuration in mind, there is not an exact method for finding primary compressor power for COP calculations. To represent 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:

$$\dot{W}_C = \dot{m}c_p \frac{T_1}{\eta_c}\left[PR_C^{\frac{\gamma-1}{\gamma}} - 1\right]$$

The pressure ratio, PR, is determined using the regulated pressure at the ACM inlet and the primary compressor inlet pressure. The temperature, T_1, is the ambient or primary compressor inlet temperature. The mass flow rate, specific heat at constant pressure, isentropic compressor efficiency, and ratio of specific heats are indicated as $\dot{m}, c_p, \eta_c$, and γ, respectively. This assumption provides a basis for comparison of performance between an unmodified turbocharger and the turbocharger modified for the improved turbine-compressor match.

The testing procedure ensured that steady-state conditions were met before recording data. This provides more conservative data, since the intercooler performance was artificially high during the transient portion of operation. The intercooler has a large mass and requires a significant amount of time to reach a steady-state or "heat-saturated" condition where all the heat removed from the hot air stream is transferred to the cold air stream. During transient operation the aluminum making up the intercooler core and end-caps can store the heat removed from the hot air stream, overestimating the intercooler effectiveness and ACM performance. The steady-state operation is confirmed by monitoring shaft speed, pressures, and temperatures prior to recording data. With the turbine-compressor match verified and the performance of the ACM optimized, the ACM was then tested on-engine to provide more realistic operating conditions.

PERKINS ENGINE TEST

The on-engine testing was performed using a turbocharged six-liter Perkins diesel engine, which is a four-stroke, inline six-cylinder engine. The speed range of the Perkins diesel engine is lower than that of a similar-sized gasoline engine. For engines with larger displacement and higher target speed range, the mass air flow consumed by the ACM will be a smaller percentage of the overall airflow, introducing a smaller effect on primary compressor sizing and engine performance. This indicates that the Perkins diesel will be more impacted by ACM operation than similar gasoline engines, providing a lower-limit for engine load and speed to effectively operate the ACM. The airflow to the ACM was controlled using a Y-pipe and valve located after the engine's intercooler. The on-engine configuration is shown in Figure 4.

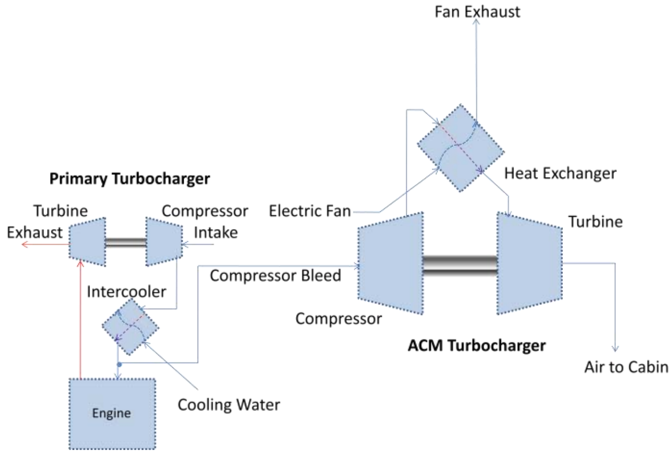

Figure 4. Automotive on-engine schematic of a two-wheel bootstrap ACM configuration.

In order to benchmark the engine before any modifications were carried out, the Perkins diesel was first tested with the stock primary engine turbocharger. This set a baseline for comparison of the turbocharger sized for operation of the Perkins with the ACM. The method of sizing the new compressor splits the full-boost operating points symmetrically about the line drawn through the center of the efficiency contours up the map, shown in Figure 5. Corrected ACM compressor and turbine speeds associated with the operating points in Figure 5 are given in Table 2.

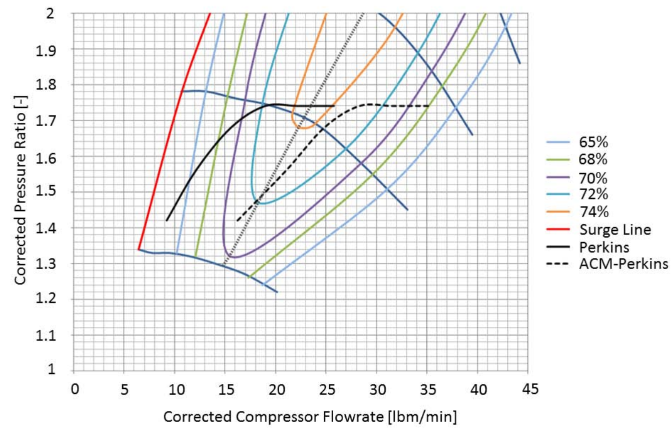

Figure 5. Primary compressor map with operating lines for the Perkins alone and with the ACM operating.

Table 2. Corrected ACM speeds along the engine full-load boost profile.

ACM Pressure Ratio	ACM Corrected Compressor Speed	ACM Corrected Turbine Speed
1.42	87711	86785
1.56	101619	100546
1.68	110080	108917
1.74	113765	112563

The compressor efficiency contour splitting is an attempt to provide the best average efficiency of the compressor. For testing it was assumed that the ACM duty cycle would be approximately 50%. The compressor matching was limited to available compressors, so an exact split was not found, and the nearest match was used. This is an application specific parameter, and a higher average efficiency may be achieved by tailoring this to a particular application by shifting the center of the efficiency contours closer to either the ACM-on or ACM-off curves.

To provide some insight into how often waste-gating occurs during engine operation, the intake manifold pressure map over the engine speed and load range in shown in Figure 6. The waste-gate actuator is set to open at 11 psig, and this corresponds to the highest contour value in Figure 6. It should be noted that the engine data will depend on the final application the ACM is designed for.

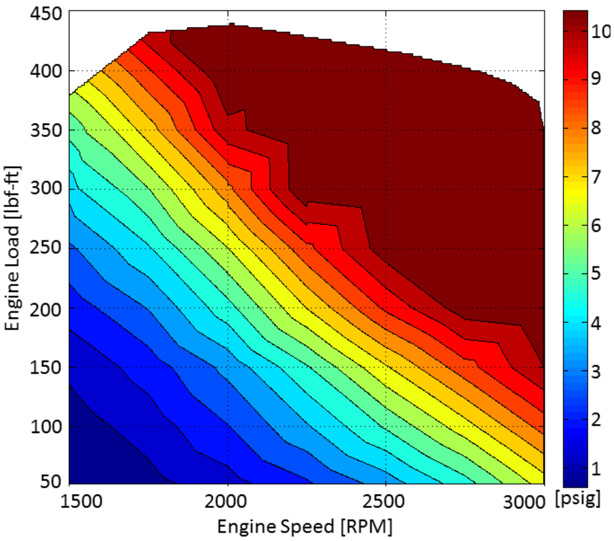

Figure 6. Engine intake manifold pressure baseline on stock turbocharger without the ACM.

RESULTS AND DISCUSSION

The unmodified baseline ACM turbocharger performed better than expected on the compressed air stand; especially, since the turbine operating points were off the manufacturer's performance map. This was due to the corrected turbine speed being higher-than-normal due to the unusually-low turbine inlet temperature for turbocharger applications. Fortunately the turbine efficiency did not drop sharply due to the excessive corrected speed caused by pairing the turbine with the relatively small compressor wheel. The ACM performance was limited by lower-than-expected intercooler performance. The problem with the intercooler was fixed for later runs, but the modified ACM turbocharger was initially tested in the same configuration for an appropriate comparison. The results of the baseline turbocharger can be seen in Figures 7-9.

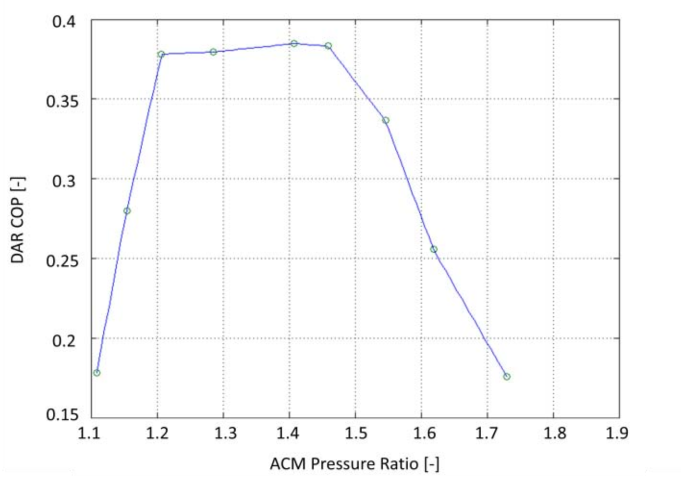

Figure 7. Dry-air-rated COP for the baseline turbocharger.

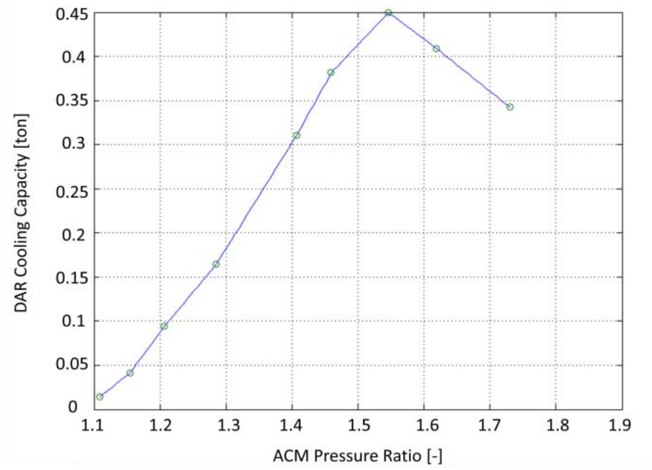

Figure 8. Dry-air-rated ACM cooling capacity for baseline turbocharger.

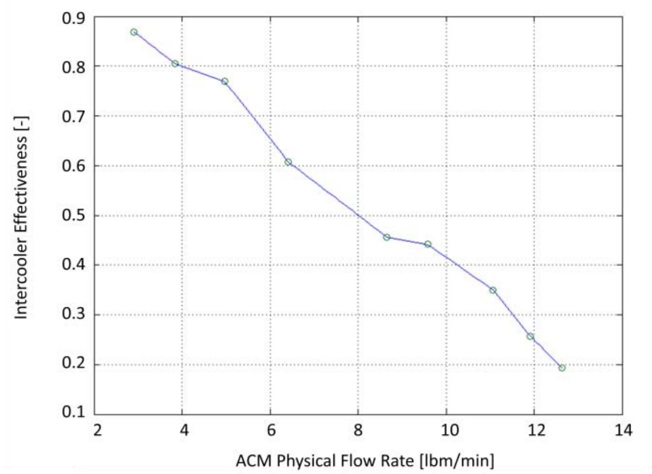

Figure 9. Intercooler effectiveness over operating range of ACM.

The DAR COP for the baseline drops off at a relatively low pressure ratio, which is not ideal for racing applications where the pressure ratio across the ACM can be moderate to high. The cooling capacity reaches a peak at a pressure ratio of 1.55 and begins to decrease. This is due to the somewhat mismatched turbine and compressor. However, it does show that an off-the-shelf turbocharger can be run in an ACM configuration.

The improved turbine-compressor match provided stronger performance over a larger operating range. The DAR COP is maintained up to a higher pressure ratio and then drops off more slowly. The DAR cooling capacity continues to rise instead of peaking out at a low pressure ratio. The results for the modified ACM turbocharger can be seen in Figures 10-12.

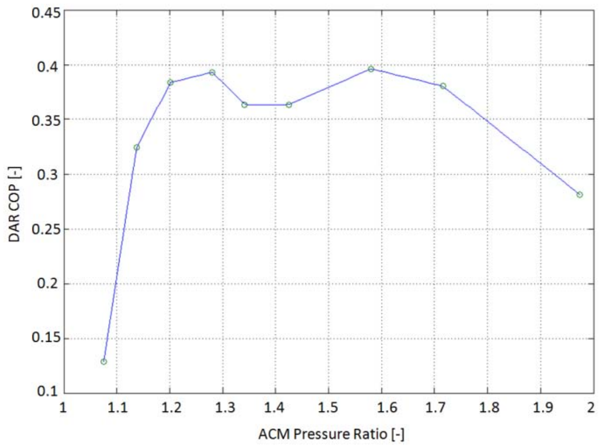

Figure 10. Dry-air-rated COP for the modified ACM turbocharger.

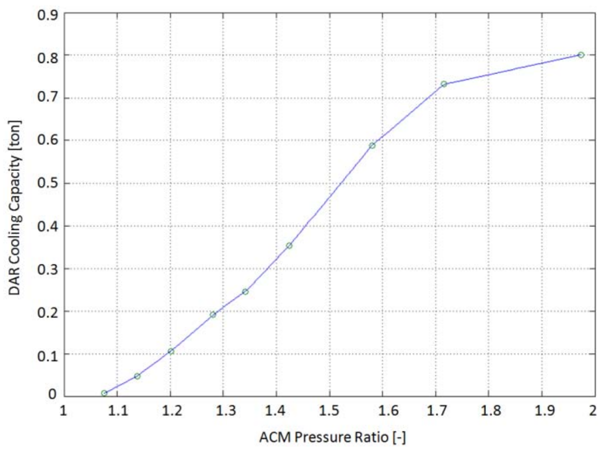

Figure 11. Dry-air-rated ACM cooling capacity.

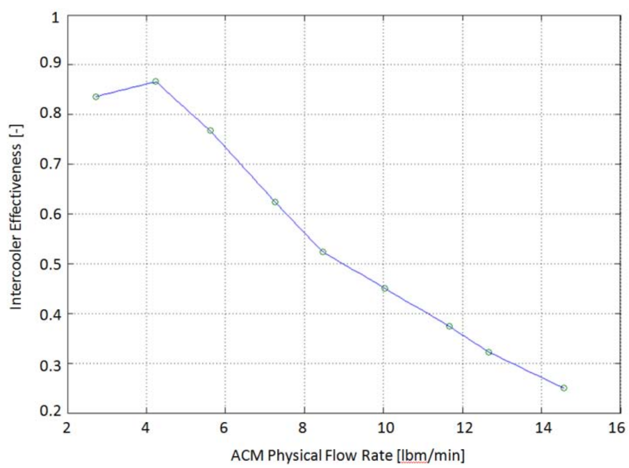

Figure 12. Intercooler effectiveness over operating range of ACM.

The rapid decline of intercooler effectiveness severely limited performance in the baseline comparison, but it does show the relative improvement of the modified ACM turbocharger. The intercooler performance was low due to the electric fan selection and shroud configuration. Due to time-constraints, this was remedied by supplying the intercooler's cold-flow from the shop compressed air supply. The flow rate was matched to levels attainable from a properly-sized electric fan. This was determined to be approximately 350 CFM, and this agrees with volumetric flow rates observed in R134a system condensers [6]. Based on intercooler cold flow entrance area, this results in an average air velocity of 8.5 ft/s perpendicular to the face of the intercooler core. The actual application will have an electric fan mounted to assist in low vehicle speed operation and the fan will be assisted by the oncoming wind when the vehicle is at speed. Note that the COP figures presented here do not account for fan power usage. The electric fan used required 100W to operate, but the actual power varied during the testing using the compressed air source for the heat exchanger cold flow. The electric fan flow rate is unknown due to unreliable manufacturer's data; a cost penalty of 100W may be assumed as a best-estimate of the cold flow power requirement. The improved intercooler and ACM performance is shown in Figures 13-15.

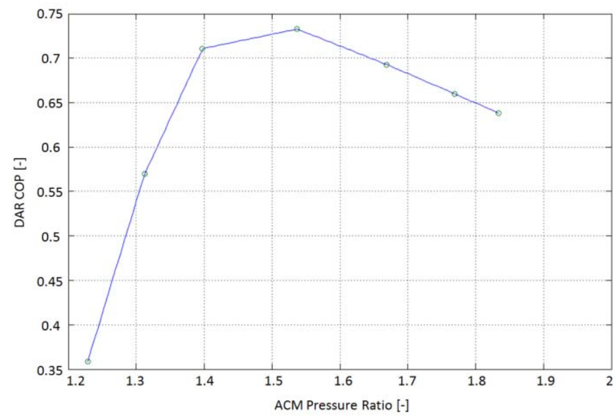

Figure 13. Dry-air-rated COP for the modified ACM turbocharger.

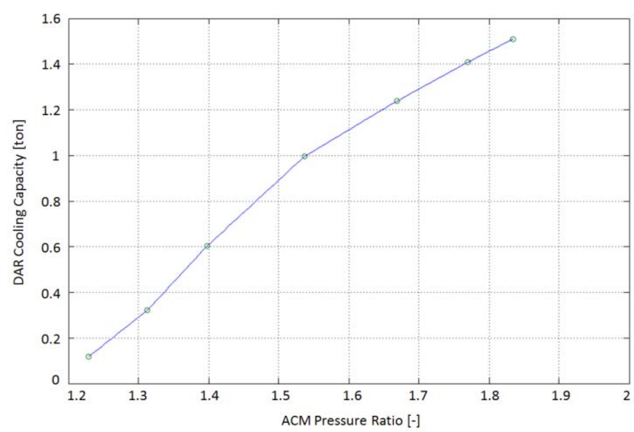

Figure 14. Dry-air-rated ACM cooling capacity.

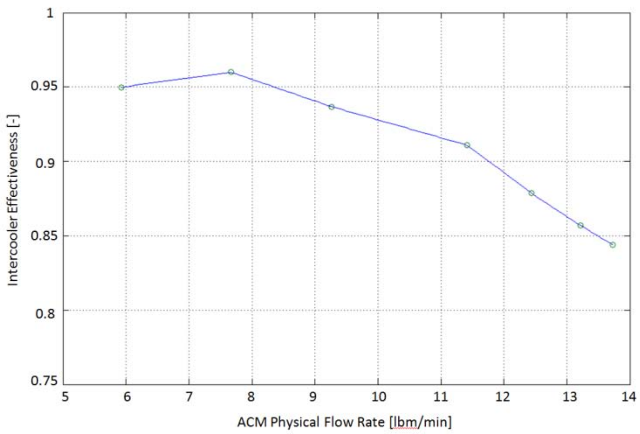

Figure 15. Intercooler effectiveness over operating range of ACM.

The theoretical performance analysis was shown to be a good approximation of the actual ACM performance. Performance simulations were run in both Matlab and Engineering Equation Solver (EES). A single operating point comparison from compressed air stand testing is presented in Tables 3-4. The uncertainty presented is absolute uncertainty with the same units as the base value. A single point of comparison is offered between the theoretical and actual performance due to the limited range of turbine and compressor data available from the turbocharger manufacturer. The corrected turbine speeds and pressure ratios seen in ACM operation are significantly higher than the hot exhaust gas application the turbine was originally intended for.

Table 3. Operating Conditions for single point comparison of theoretical and actual performance.

Ambient and Operating Conditions	Value	Units	Uncertainty [+/-]
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	[lb_m/min]	0.6
ACM Turbine Inlet Temperature	536	[Deg. R]	1.7
ACM Turbine Efficiency	0.56	[-]	0.02

Table 4. Results of single point comparison of theoretical and actual performance.

Performance Parameter	Theoretical Operation	Actual Operation	Uncertainty [+/-]
COP, Dry Air Rated [-]	0.68	0.64	0.04
Cooling Capacity, Dry Air Rated [ton]	1.6	1.5	0.06

Further ACM performance improvements could be made by designing a compressor and turbine for this application. An important first step is to reduce the corrected turbine speed to improve turbine efficiency, since the ACM operating conditions shift this away from the original turbine design points. A turbine designed for higher pressure ratio operation is desired because of the pressure the ACM compression stage provides in addition to the engine intake manifold supply pressure. Also, an U/C0 analysis comparing blade tip speed to spouting velocity would help determine if components are well-matched to their new duties.

On-engine testing was cut short because of a bearing failure in the ACM turbocharger. However one operating point was tested at sufficient load and speed to provide encouraging results. At that operating point COP based on change of fuel flow rates and bsfc indicates a $COP_{effective}$ of 0.56, with a corresponding DAR cooling capacity of 0.72 tons. At the same ACM operating point from on-engine and compressed air stand testing, the DAR cooling capacity dropped from 0.90 to 0.72 tons and DAR COP decreased from 0.73 to 0.56. While this operating point fell short of the goal of one ton of DAR cooling capacity, it suggests that higher pressure ratio operation can meet the target cooling capacity. The shop-air source provided lower inlet temperatures than those available during on-engine testing. The decrease in performance is due to higher ACM compressor inlet temperatures, and compressor power requirements are directly proportional to the absolute compressor inlet temperature. Increasing ACM compressor inlet temperatures led to a decrease in pressure ratio across the compressor. A lower pressure ratio across the compressor will cause a lower pressure ratio across the cooling turbine, resulting in lower system performance. While on-engine testing produced higher ACM compressor inlet temperatures than the shop-air tests, the limited on-engine testing wasn't able to cover higher pressure ratio operation where ACM performance is expected to improve.

The COP of a typical automotive vapor-compression system is approximately 1.75 [7]. At low engine speed operation, the power difference between the two systems, providing 0.72 tons of cooling, is approximately 4.5 hp. A general trend with vapor-compression systems is that COP tends to drop off at higher speeds, where the ACM will be most effective. At high engine load and speed, more exhaust energy is being lost due to waste-gating, and this is where the compressor bleed is intended to reclaim energy from the exhaust stream to power the ACM.

In addition to calculating $COP_{effective}$ from bsfc for the ACM configuration, the engine intake to exhaust pressure ratio can provide some indication of lower-loop losses associated with ACM operation. The change in bsfc calculated from constant speed and load engine measurements accounts for the lower-loop losses due to ACM operation. It is important to note that the on-engine ACM testing did not reach the target boost pressure for normal waste-gate operation, so the ACM is relying on increased exhaust backpressure for power. Table 5 includes a few operating points with the corresponding intake and exhaust pressure ratios.

Table 5. Pi/Pe data for baseline engine operation and during ACM operation.

Engine Speed [RPM]	Engine Torque [lbf-ft]	Pi/Pe (Baseline)	Pi/Pe (ACM)
1200	250	1.00	0.77
1400	200	0.98	0.79
1595	180	0.97	0.81

CONCLUSIONS

The shop-air testing successfully demonstrated that an ACM can be designed around automotive turbocharger components. The use of automotive turbocharger components is critical in reducing costs of developing commercial systems for automotive use. An off-the-shelf turbocharger can provide adequate performance for some applications. The baseline ACM turbocharger provided a peak DAR COP of 0.38 and a DAR cooling capacity of 0.45 tons with low intercooler effectiveness limiting system performance. It is expected that the baseline turbocharger would experience a similar increase in performance by improving intercooler effectiveness, as seen in later compressed air-stand testing of the modified ACM turbocharger. The modified ACM turbocharger provided a peak DAR COP of 0.73 with a peak DAR cooling capacity of 1.5 tons. The DAR COP figures from the shop-air stand were calculated assuming a primary compressor efficiency of 70%. The primary compressor used in the Perkins-ACM testing had an isentropic efficiency of 70% or greater over a corrected flow rate range of approximately 16-32 lb_m/min. The on-engine testing that was completed indicates a DAR COP of 0.56 and a DAR cooling capacity of 0.72 tons. This operating point is below the 11 psig the waste-gate on the original engine turbocharger is set at, so the full benefit of the bleed-air configuration, operation during typical waste-gating conditions, could not be demonstrated. ACM operation during typical waste-gating conditions would utilize the power that would have been lost during free expansion of the exhaust gases through the waste-gate orifice, instead of creating additional exhaust backpressure to allow the primary engine turbocharger to provide additional airflow for ACM operation outside of regular boost-controlled operation. The COP presented for the on-engine test is still calculated from the change in fuel flow rate and engine bsfc, and is thus a lower limit of what this system is capable of achieving. Operation of the ACM at higher engine load and speed will allow for improved system performance due to the availability of excess turbine power at times when the waste-gate would normally be open.

While a considerable amount of work remains in designing an ACM for production use, most of it can be addressed within the standard design guidelines of turbocharger manufacturers. The conditions at the ACM inlet are similar to those of a second turbocharger in a series turbocharger configuration used for efficiently handling high-pressure ratio diesel engine applications. In-vehicle testing is the next step towards the production-ready ACM.

Some design aspects that need to be addressed are bearing system capabilities. The cooling turbine is creating much less power than it would in an application involving hot exhaust gases; the bearing losses remain constant even though the turbine is producing less power, and the losses become a larger percentage of the overall power being transmitted through the shaft to the compressor. There is a need for increased bearing efficiency, while maintaining stability. The thrust bearing in the ACM failed during on-engine testing. This is likely due to the higher pressure differential experienced between the compressor and turbine sides of the rotating assembly. An axial force is generated by the pressure over the projected area of the compressor wheel. The net force is higher in the direction of the turbine, since the ACM compressor is being fed by a compressed air source instead of drawing air at atmospheric pressure. The larger compressor wheel may have added to the bearing load and increased concerns about bearing stability. A larger bearing system can fix the stability and load problems but will increase drag on the rotating assembly. Alternate bearing types may be considered since the ACM turbocharger application creates a less-harsh environment than that of a turbocharger in an on-engine application; the temperature is much lower and there is no soot contamination from the turbine side. Air, magnetic, or ceramic bearings may provide the load capabilities and stability required for reliable operation with minimal losses [8,9].

Transient intake manifold pressure control needs a more detailed assessment and will be studied in future work. The basic idea is to have a pressure regulator to allow excess intake manifold pressure to bleed off to the ACM inlet automatically. The hope is that a simple spring-loaded actuator, similar to a waste-gate actuator, will provide adequate pressure control and prevent over-pressurization of the intake tract.

Clearly, there are many additional engineering aspects to the design of a functioning, commercially viable automotive ACM that remain to be addressed at the end of this study, not least of which is the issue of air quality to the cabin. The outlet of a turbocharger compressor is not designed to provide occupant air, and it is certainly contaminated, by oil and particulates, in the process described in this article. Additional filtering and air quality monitoring would be required before the cooled air provided by the process described here may be ducted through an actual passenger cabin. While similar problems are currently addressed in the aerospace industry, they were not considered in this study, and may further affect the overall performance of the system.

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