Introduction

Charge cooling is a popular way to improve the performance of a supercharged engine. In a previous paper, we explored some of the engineering concepts behind supercharger performance, including a few of the basic thermodynamic processes involved. One of these processes is the inherent temperature rise of charge air upon compression. We termed this property "heat of compression". Since compressing air causes a temperature rise, and temperature rise is associated with air density loss, a properly installed heat exchanger system will cool the charge air and therefore increase charge air density. Since an engine consumes a fixed volume of air at a given speed, any air density increase afforded the inlet of the engine will allow it to consume a greater mass flow of air. Finally, the increased air mass flow can be combined with the proportionally correct amount of additional fuel to result in increased power output.

Terminology Defined

We have all heard the term "intercooler" and assume this means a heat exchanger is positioned in the air passage between an engine and a supercharger or turbo. Although this modern understanding seems universal, it is technically incorrect. Strictly speaking, the term intercooler has traditionally meant a heat exchanger placed between two compressor stages of a multistage compressor system. This has always been referred to as "inter-stage cooling", with the obvious foreshortened term, "intercooling". It wasn't until Volvo popularized the use of charge cooling on production sedans back in the seventies that intercooling became a fashionable term. Indeed, Volvo's emblazoning of "intercooler" on thousands of deck lids seems to have created a phenomenon that has stuck to this day.

So, you may ask, what is the correct terminology? We prefer the more descriptive term "charge air cooling", sometimes abbreviated, CAC. We will also use the term "charge cooler" in this paper.

Charge Coolers in Detail

Many have heard of various terms to describe charge cooler or more generically, heat exchanger performance. Terms like efficiency and effectiveness are commonplace. Some even claim that cooler "efficiencies", and even superchargers themselves, can achieve or exceed 100% "efficiency" under certain circumstances.[1,2] Such notions are, of course, ludicrous. A definitive understanding of heat exchanger performance and how to objectively evaluate and compare various types of charge cooler products begins with exploring some of the engineering basics.

Is it Efficiency or Effectiveness?

When we talk about "efficiency" of a thermodynamic process, we are primarily concerned with the quantity of work output relative to the cost of energy input. Certainly, heat exchanger function involves the thermodynamic processes of heat and mass transfer. But the notion of work output relative to energy input simply does not apply, nor can such a relation be defined for a heat exchanger. Rather, a heat exchanger is a device that transports energy from one fluid stream to another; no work is performed and no energy need be invested to make the transfer happen. The measure of how well this is accomplished is called effectiveness, which is a commonly accepted engineering term for heat exchanger performance. Fortunately, it is also a parameter that can be thermodynamically defined, which means that charge coolers can be tested for effectiveness performance.

Charge Cooler Thermal Performance

With the notion of effectiveness at hand, an understanding of its definition is fundamental. Strictly speaking effectiveness may be defined as:

$$\frac{\textbf{The actual heat transfer}}{\textbf{The maximum } \textbf{\textit{possible}} \textbf{ heat transfer}}$$

Actual heat transfer may be determined by evaluating the change in enthalpies ($C_p \Delta T$) of either the hot or cold side fluid, as the heat rejected from the hot gas is equal to the heat absorbed by the cooling medium.

The maximum possible heat transfer is the energy transfer that occurs when one of the fluids undergoes the maximum possible temperature change, i.e., if the incoming hot air leaves the charge cooler at the same temperature as the entering cooling medium (i.e., air or water) or if the cooling medium leaves the heat exchanger at the same temperature as the entering hot air. Clearly, it is not possible that the hot air can be cooled to a lower temperature than the entering cooling medium, and vice-versa. So it makes sense that a given charge cooler has limited heat transfer capability, depending on the operating conditions. Further, heat transfer is dependent upon the flowrates. The maximum possible heat transfer, then, will be the lower of:

$$(\dot{m}\ C_p)_{hot\ gas} * (Th_{in} - Tc_{in}),\ or\ldots$$

$$(\dot{m}\ C_p)_{cooling\ air} * (Th_{in} - Tc_{in})$$

With $\dot{m}$ being the mass flowrate. Now, if conditions on the cold side limits the heat transfer capability, charge cooler effectiveness is found from:

$$\varepsilon = (\dot{m}\ C_p)_{hot} * (Th_{in} - Th_{out}) / (\dot{m}\ C_p)_{cold} * (Th_{in} - Tc_{in})$$

From the conservation of energy principle, this can also be written as:

$$\varepsilon = (\dot{m}\ C_p)_{cold} * (Tc_{out} - Tc_{in}) / (\dot{m}\ C_p)_{cold} * (Th_{in} - Tc_{in})$$

Which reduces to:

$$\varepsilon = (Tc_{out} - Tc_{in}) / (Th_{in} - Tc_{in}) \qquad [1]$$

Conversely, the hot side conditions may limit the heat transfer, in which case effectiveness is found from:

$$\varepsilon = (\dot{m}\ C_p)_{hot} * (Th_{in} - Th_{out}) / (\dot{m}\ C_p)_{hot} * (Th_{in} - Tc_{in})$$

Which reduces directly to:

$$\varepsilon = (Th_{in} - Th_{out}) / (Th_{in} - Tc_{in}) \qquad [2]$$

It follows then, that the total heat transfer will be given by:

$$\boldsymbol{Q} = \varepsilon * (\dot{m}\ C_p)_{min} * (Th_{in} - Tc_{in}) \qquad [3]$$

Before getting overwhelmed at all of this and in due consideration of the above, let's make some general statements regarding effectiveness performance of charge coolers:

- Since effectiveness performance depends on hot and cold side mass flows and resulting temperature changes of the two fluids, it is entirely dependent on:
  - Heat exchanger design
  - The operating point
- Per the above, effectiveness is a measure of steady state thermal performance and, therefore, ***can only be evaluated under steady state, fully stabilized operating conditions.*** Per equations 1 and 2, then, effectiveness can be determined from the entering and exit fluid temperatures during a steady-state test.
- Although not evident from the above, but as will be shown later, effectiveness is strongly influenced by the mass flowrates, but weakly influenced, if at all by the entering hot gas (charge air) and cooling medium (air or water) temperatures. Total heat transfer is affected, however, as it is influenced by effectiveness, temperature difference, and mass flowrates. But generally speaking, we can say that for a given (hot) charge air inlet temperature and flowrate, and inlet cold temperature, the charge cooler with the best effectiveness performance will reject the most heat, and therefore be the best choice from a ***thermal performance perspective.***

Pressure Drop Performance

The last statement above might lend a clue to this next topic as it turns out that pure thermal performance does not necessarily conclude the best CAC system. One highly important but often overlooked parameter is the pressure loss incurred across the charge cooler, when inserted into a supercharging system. It must be understood that even the most extraordinarily engineered cooling systems will incur some amount of pressure drop, which necessarily amounts to an engine manifold boost loss. Recognizing that pressure loss performance is equally as important as thermal performance, any test data should include both effectiveness and pressure drop measurements. In addition, there are the effects of piping and ducting losses associated with the charge cooler installation. In many automotive installations, air-to-air type charge cooler systems are particularly prone to these additional losses as the ducting runs required to get charge air to the typically front-mounted charge air cooler and back can easily equal if not exceed the losses experienced across the bare cooler core. An air-to-water system, on the other hand, may offer considerable packaging advantages and can often be inserted directly between the supercharger and the engine intake due to its generally compact design. This minimizes if not eliminates any additional ducting or piping and the losses associated. Presentation of test results later in this paper will discuss this in more detail.

Putting it Together

It may seem evident by now that the "best" performing CAC system will be the one that maximizes thermal performance (effectiveness) while at the same time minimizing pressure loss. One way of evaluating this 'optimum' and determining the best performing system is to examine the change in air density across the CAC system.

Density Ratio

In a companion paper, we examined the importance of charge air density as it relates to supercharged engine performance, and recognized that maximizing charge-air density resulted in the best attainable performance. Charge cooler systems can also be evaluated in a similar manner – the primary objective is to cool the charge air, which increases air density at the engine intake. This follows from the familiar generalized gas law:

$$\rho = P/RT \qquad [4]$$

Clearly, as temperature drops, air density will increase. Any pressure drop, however, results in a density loss, which contradicts our purpose. As a performance measurement, then, we can consider the density ratio performance across the charge cooler system, and define this as:

$$\frac{\rho_{out}}{\rho_{in}} = \frac{\left(P/_{RT}\right)_{out}}{\left(P/_{RT}\right)_{in}} \qquad [5]$$

Which simplifies to:

$$\frac{\rho_{out}}{\rho_{in}} = \frac{P_{out}T_{in}}{P_{in}T_{out}} \qquad [6]$$

Recognizing that we are talking about performance on the hot gas or charge air side, we now have an objective evaluation criteria against which we can compare performance of various available charge cooler systems and products – the system that maximizes density ratio performance over the range of flowrates of interest will, generally, be the "best" system choice. Further, we can conclude the following based on density ratio performance:

- If the density ratio across the system is greater than 1.0, then the cooler system is providing a performance benefit.
- If the ratio drops below 1.0, then there is a net loss in charge air density across the cooler system – performance with the cooler system is worse than without it!
- There will be a maximum flowrate point where the density ratio will be approximately 1.0. This represents a sensible peak flowrate, and hence upper horsepower capability limit for the system. Such is useful for evaluating various claims from equipment manufacturers as to what performance level a system is capable of.

Real Systems

Armed with some of the engineering fundamentals behind charge cooling, we can now look at the two most popular system technologies, air-to-air and air-to-water, more critically.

Air-to-Air Systems

These systems are by now quite well known and are the most popular by far, primarily due to their use on production turbocharged vehicles. These systems benefit from their ultra simplicity and passive operation, reliability, not to mention low cost.

Thermal Performance

Referring back to equation [3], heat transfer performance will be maximized if the product: $(\dot{m}\ C_p)_{min}$ is on the hot side of the exchanger, meaning the cooling air flow must exceed the charge air flow. This illustrates the importance of attaining adequate cooling airflow through the cooler under operating conditions. Further, the nature of the two convective heat transfer coefficients on the hot and cold side inure towards a relatively large core size in order to obtain acceptable performance.

Pressure Loss Performance

Considering again the typical turbocharged vehicle, the air-to-air charge cooler system incorporates piping and ducting to the front-mounted charge cooler, with return back to the engine intake. Pressure losses will occur, however, the waste-gated operation of the turbocharger compensates for this pressure drop as it controls to a manifold pressure 'set point'. Although this built-in compensation mechanism 'masks' the pressure drop effect, this doesn't mean that pressure losses are insignificant or should be ignored – if the system pressure drop is excessive, then additional pumping losses in the engine will result in order for the turbo to overcome this. The turbine pressure ratio can rise considerably creating significant net power losses.

A supercharged system, on the other hand, does not benefit from the automatic waste-gated control of manifold pressure when using the air-to-air charge cooler. Rather, increased blower speed and pressure ratio is required to overcome any charge cooler system pressure loss, and this can only happen at the expense of increased crankshaft parasitic power loss. As will be later shown, the effects can be quite significant. Further, and as will be shown later in test data, the piping losses alone can equal the pressure drop across the charge cooler core, essentially doubling the loss for the system. Now, if there were a way to eliminate the piping loss altogether…

Air-to-Water Systems

Though less popular, air-to-water systems have recently attracted more serious consideration due to the performance potential attainable. This seems to be recognized by the automotive OE community as air-water charge coolers are becoming the system of choice on supercharged power trains. These systems comprise a separate liquid cooling loop, typically a weak glycol water mix, on the cold side of the charge cooler, with a small separate auxiliary cooler mounted at the front of the vehicle. In operation, heat is rejected to the coolant, transported to the front mounted auxiliary cooler, then rejected to the atmosphere. Coolant is then returned to a reservoir to be re-circulated through the system. A low-pressure recirculating pump keeps the liquid moving continuously, at a constant 3 or more gallons per minute flowrate.

Thermal Performance

As both the density of water and specific heat are much greater than air, the product $(\dot{m}\ C_p)$ will almost always be much greater on the cold side than the charge air side. Add to this the fact that water is approximately 23 times more conductive than air – 0.384 Btu/hr-ft-°F versus 0.0164 Btu/hr-ft-°F. Together, these factors result in a heat exchanger of not only high effectiveness performance, but also of very compact design. Effectiveness performance can, in fact exceed the effectiveness of considerably larger air-to-air charge coolers, at similar operating conditions.

Pressure Loss Performance

The air-to-water charge cooler can be quite compact in size so it can, in many cases, be installed under the hood, directly between the supercharger discharge and the engine throttle body. This all but eliminates any additional ducting or piping required to transport charge air to a front-mounted charge cooler and return it to the engine. In the worst of cases, ducting runs are very short and to the point. Thus, only the pressure loss across the charge cooler core itself will result, and the total system pressure loss will typically be much less than a comparable air-to-air system. It should also be noted that the losses through the core itself are less than a comparable air-to-air core. As will be seen, these factors offer significant advantages for maximizing density ratio performance, and hence power gain potential.

Notes on Cooling Air Flow and Thermal Transitions

As previously discussed, cooling airflow across the air-to-air charge cooler directly affects thermal performance. Thus in testing, appropriate cooling airflow must be provided such that test data obtained in the laboratory is representative of actual in-service conditions. Unfortunately, the determination of cold-side airflows experienced in vehicular use for an air-to-air charge cooler is very difficult to determine. Even so, some have fallen into the mistaken belief that a cooling air "face velocity" of 60 MPH (5,280 ft/min) is the appropriate test condition for a charge cooler. This is furthered by the proposition that such a cooling air face velocity produces an actual volumetric flowrate of 5,280 ft^3/min – per square foot of core area through the charge cooler. Such notions cannot be supported either by test data or prudent engineering analysis by considering the following facts:

- Actual flow area is much less than the total projected core face area. This is due to the considerable flow area blockage presented by the tube and fin construction of the charge cooler.
- These myths further ignore the flow restriction presented by the typical punched or louvered fin construction. This restrictive labyrinth flow path necessarily creates a pressure drop across the cold side, which reduces flowrate accordingly.
- Charge coolers are typically mounted behind vehicle bumpers, fascia, grillework, etc., all of which inhibit exposure to a clean and direct source of airflow, with attendant flowrate reduction. A perfect, homogeneous cooling air face velocity is almost never available.
- Total cooling air flowrate will also depend on the size of the charge cooler – clearly, increasing exposed core area will result in increased air flow.

Per these factors, the average face velocity and the total flowrate through the cold side of the air-to-air charge cooler will be much less than the theoretical ideals proposed. The actual flowrate and average face velocity will also be much lower than the vehicular speed.

So, one might ask: what is the appropriate cooling airflow needed for laboratory testing? Fortunately, this can answered by assessing charge air temperatures during road testing, and comparing against laboratory tests. For example the charge-air temperature attained at the end of a high-speed acceleration run can be compared to a chassis or engine dynamometer sweep test. Cooling airflow can then be adjusted such that approximately equivalent exit air (or throttle body) temperatures are achieved in the laboratory. This validates the cooling airflow supply; maintaining this same cooling airflow supply for subsequent lab tests becomes compulsory.

The air-to-water system, on the other hand, is much easier to assess. The cold side liquid flowrate is generally known, or easily obtained. The only additional unknown, then, is the liquid supply temperature to the charge cooler, which is also easily obtained. These conditions can then be replicated in the laboratory.

Lastly, we must consider and understand the notion of thermal transitions in actual use. Some (perhaps the same individuals) claim that air-to-water systems, in the final analysis, are nothing more than air-to-air systems, are prone to heat soak, and therefore cannot match the effectiveness of air-to-air systems. What is conveniently ignored is the fact that all charge cooler systems operate in a continuous cooling, intermittent heating mode. This means that forward vehicle motion provides for constant cooling airflow, and the system operates in a continuous cooling mode. Typically, and only infrequently does high performance (and hence significant thermal load) operation occur, and this generally lasts 20 seconds or less. Given this typical operation and owing to the availability of several pounds of liquid coolant available, the coolant supply temperature remains virtually unchanged for these intermittent 20 second performance runs. This type of operation takes advantage of thermal transitions and describes the notion of intermittent heating, with continuous cooling. It should lastly be recognized that stability of the cooling liquid supply temperature in vehicular use is consistent with the stable operating condition requirements for effectiveness testing in the laboratory.

System Testing

Testing of cooler systems generally consist of laboratory, engine and chassis dynamometer, and road tests. In the laboratory, operating conditions can be held continuous and stabilized so that accurate effectiveness and system pressure loss measurements can be recorded. Chassis and dynamometer tests are conducted to confirm power gain potentials for various systems, while road tests determine actual operating conditions and charge-air temperatures, which confirm dynamometer and laboratory test conditions. Lastly, no “ice-water” or other artificial cold-coolant means are employed in air-to-water system tests. Rather, liquid temperatures are stabilized at ambient, or in the case of dynamometer and road tests, at temperatures experienced during actual in-service use.

Effectiveness and Pressure Loss Tests

Laboratory testing utilizes a supercharger mounted to the compressor test stand, which in turn provides the air flow and thermal loading needed to test an charge cooler. This can be considered quite representative of actual system conditions when the same or similar components are installed in a vehicle.

Figure 1 depicts results of lab tests, and shows a comparison of a Vortech air-to-water charge cooler with 18 sq.-in. cold side flow area, and a comparison air-to-air charge cooler product with 308 sq.-in cold side flow area. The two different curves for each show effectiveness and pressure loss performance, plotted against supercharger flowrate. Note that these tests are of the heat exchanger core itself – per our discussion above, the air-to-water results are quite representative of the losses that may be expected in an actual installation. The air-to-air results must be considered “best possible case” since additional plumbing losses will likely result in actual installation. Even so, one can immediately deduce the advantages offered by the very small (by comparison) air-to-water heat exchanger versus the air-to-air; better thermal performance and less pressure loss.

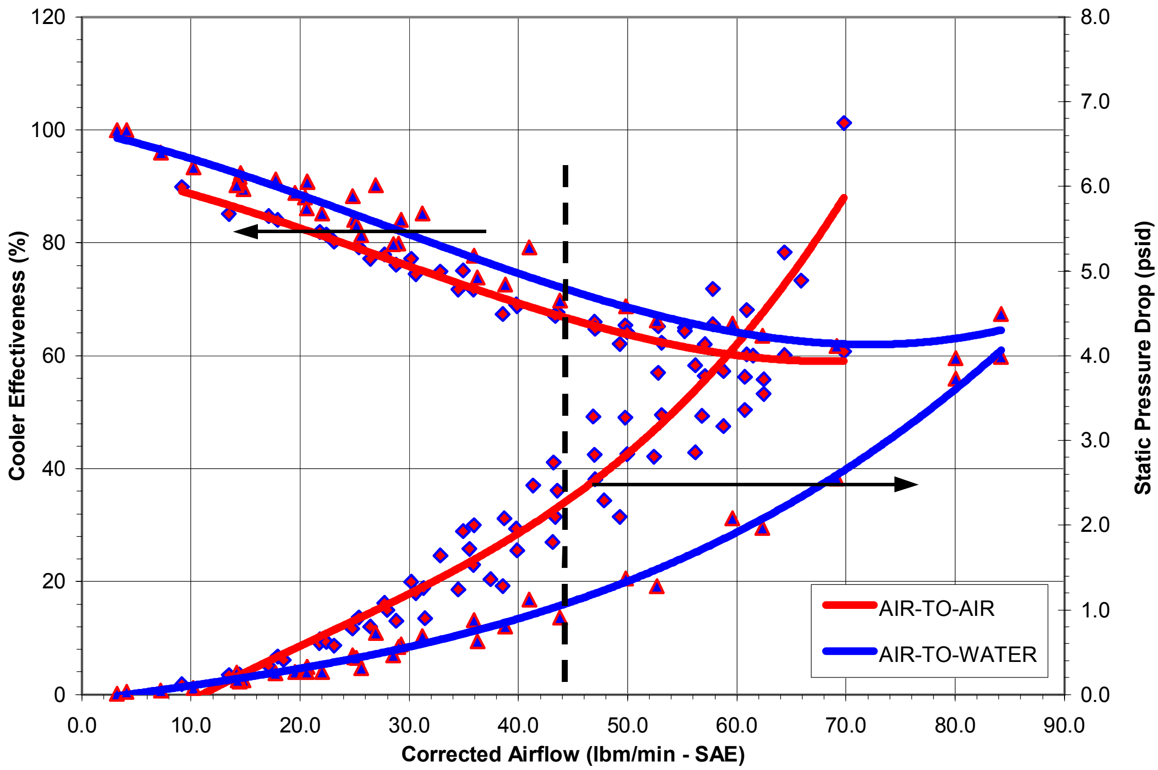

Figure 1 – Effectiveness and pressure loss performance. Air-to-Water (blue): Vortech 4.5" x 9" x 4" deep, single unit w/ air manifolds. Cold side: 18 in^2 flow area, 3.0 GPM; 20% glycol solution. Air-to-air (red): Comparison two-pass, 11" tall x 28" wide (308 sq.-in) core area x 3" deep, w/ air manifolds. 1740CFM cooling air flow on cold side provided by Ø36" ducted fan – same fan used on engine and chassis dynamometer tests. Fully stabilized for all test points. 44 lbm/min flowrate approximates typical operating point of 5.0L V-8 Mustang engine at 5900 RPM.

Figure 2 presents additional laboratory data of other comparison charge cooler systems. In these cases, complete system performance is depicted with pressure loss for the comparison air-to-air systems including the entire piping/ducting system, as supplied by the OE manufacturer. The Figure 1 data for the air-to-water system is repeated in this overlay plot for comparison.

As can be seen in this plot, the additional losses presented by the ducting, piping, fittings and the like needed for installation in the vehicle contributes substantially to the overall system pressure loss. This is especially true for the "2-Core" system which exhibits substantial pressure loss performance, even at moderate flowrates.

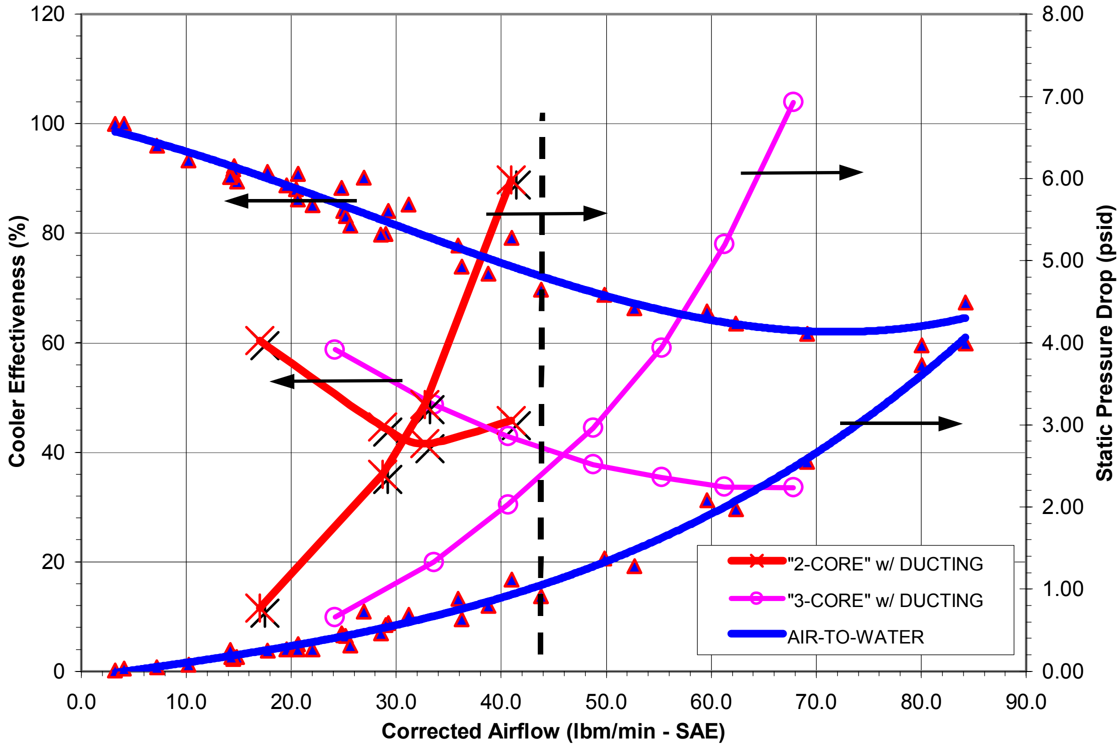

Figure 2 – Effectiveness and pressure loss performance for comparison systems. Air-to-water (blue): Same as Figure 1. Air-to-air (red): Competitive "2-Core" system, with OE supplied Ø2.25" piping system, single-pass, 6" tall x 18" wide (108 sq.-in) core area x 3" deep. 500CFM cooling air flow on cold side provided by Ø36" ducted fan – same fan used on engine and chassis dynamometer tests. Air-to-air (pink): Competitive "3-Core" system, with OE supplied Ø3.00" piping system, single-pass, 6" tall x 27" wide (162 sq.-in) core area x 3" deep. 1260CFM cooling air flow on cold side provided by Ø36" ducted fan. Fully stabilized for all test points. 44 lbm/min flowrate approximates typical operating point, 5.0L V-8 Mustang engine at 5900 RPM.

Density Ratio Performance

As previously discussed, a way to assess the overall effects of thermal performance (effectiveness) and pressure loss is to evaluate the change in air density across the charge cooler system (equation [6]). This is easily accomplished in the laboratory as static temperatures and pressures are available from the instrumented test sections, positioned immediately upstream and downstream of the charge cooler system.

Figure 3 presents density ratio performance for the two charge cooler systems tested in Figure 1. As is clearly shown, the superior performance of the air-to-water system with regard to both thermal and pressure drop is revealed by a density ratio curve above the comparison air-to-air unit. This plot further indicates approximate maximum airflow capacity which can be related to the upper performance limitations when applied to any given engine – for the air-to-water system, performance is enhanced up to 54 lbm/min airflow. The air-to-air system is good to about 41 lbm/min. Based on this metric alone, the much smaller air-to-water system has almost 1/3rd greater flowrate capacity.

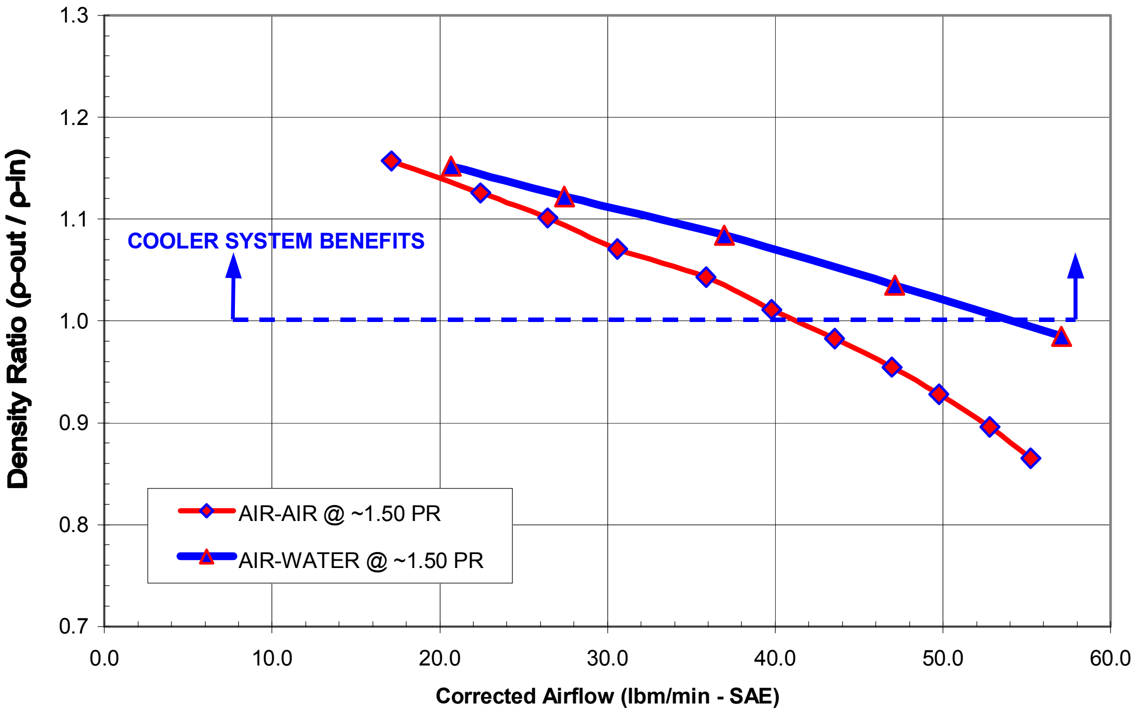

Figure 3 – Density Ratio performance for 18 sq.-in. air-to-water and comparison 308 sq.-in. air-to-air charge coolers (cold side flow areas). Density ratio indicates change in air density across the charge cooler system, which incorporates effects of both thermal and pressure loss performance – Ratio of ≥ 1.0 indicates performance enhancing potential. Upper flowrate limit/system capacity is reached at 1.0 ratio. Supercharger pressure ratio held constant for both systems resulting in equivalent thermal load.

Engine and Chassis Dyno Tests

Engine and chassis dynamometer tests of charge cooler systems can be quite useful for:

- Confirming power gain potential of an charge cooler system, and…
- Confirming in-service use conditions, such as charge air temperatures

Figure 4 presents engine dynamometer test results of a supercharged 5.0 liter V-8 engine (1986-1993 M/Y Mustang) with a Vortech V1S-Trim supercharger and air-to-water charge cooler system, and a comparison supercharger system with a “2-Core” air-to-air charge cooler, from a competing manufacturer, installed on the same engine[3]. As can be seen, the comparison system produced the most pressure at the supercharger discharge (upper curve) but the least delivery pressure to the engine throttle body (lower curve). The Vortech system, on the other hand, produced supercharger and throttle body pressures within 0.5 psi of each other. These results are consistent with the Figures 1 and 2 laboratory data.

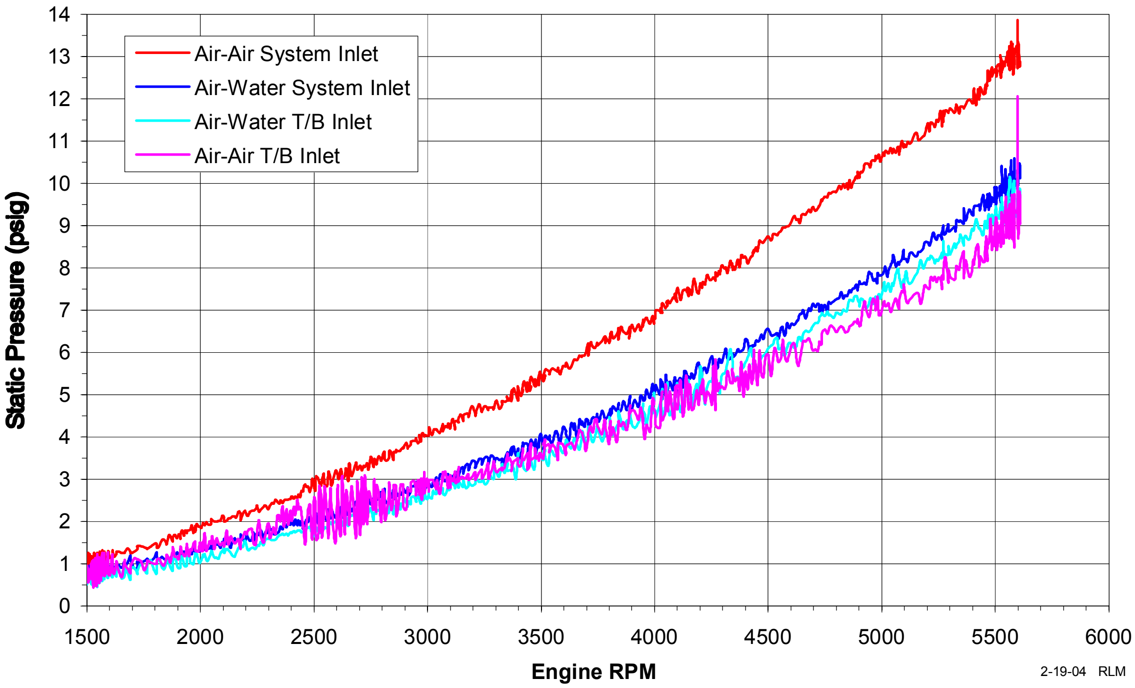

Figure 4 – Engine Dynamometer Testing – Inlet and discharge pressures of charge cooler systems. “T/B” is taken at connection to engine throttle body. Red + pink is “2-core” air-to-air data; blue + light blue is air-water. Tests are average of “best” 3 runs for each. 5.0L Mustang (1986-1993 M/Y) engine. 91 RON octane fuel, 10-degrees BTDC static timing. Ø36” ducted fan used for cooling air supply; tests conducted after warm-up and thermal stabilization of system components. Air-air looses almost 4 psi at 5000 engine RPM. Results: Air-Water system – 342.1 HP @ 5400 RPM; Air-Air system – 312.6 HP @ 5200 RPM (Corrected HP).

Figure 5 shows the same comparison systems from the same tests, but depicts charge air temperatures instead. Even though the air-to-water system produces the coolest throttle body temperatures, little can be concluded relative to thermal performance as the thermal load for the comparison system is much greater due to the increased supercharger pressure and lower compressor efficiency. Charge temperatures of 127°F @ 5250 RPM for the air-to-air system, and 98°F @ 5250 RPM for the air-to-water system will be compared to chassis dyno and road test data later.

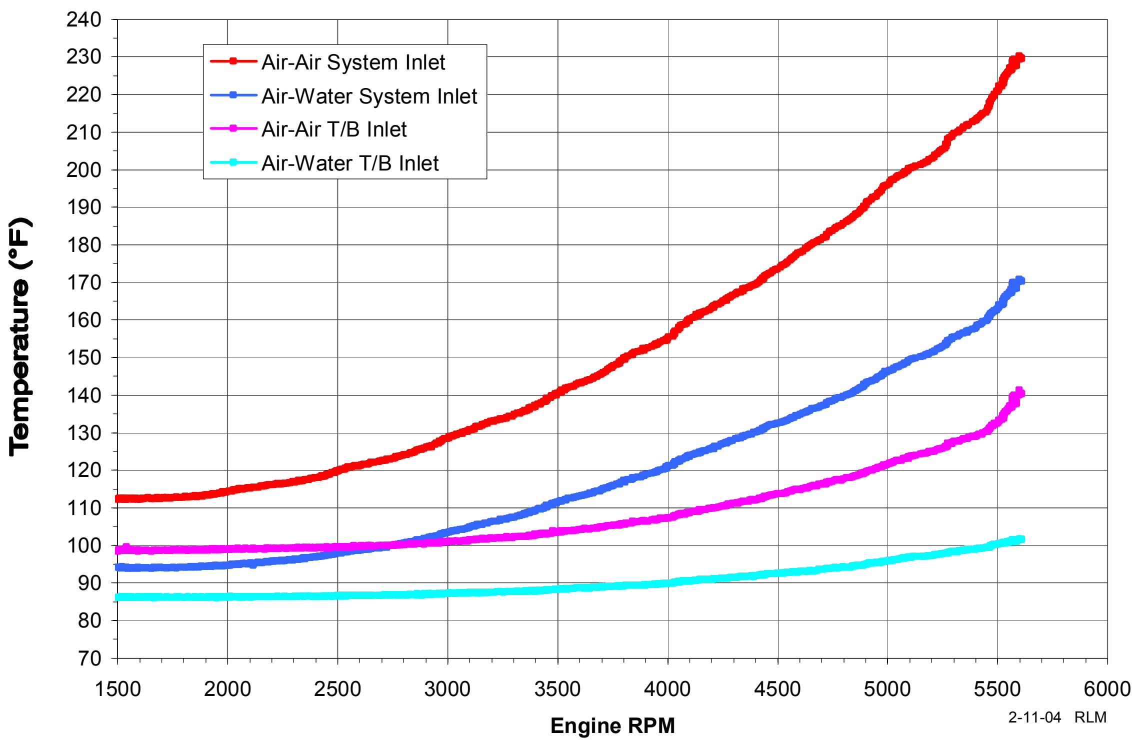

Figure 5 – Engine Dynamometer Testing – Inlet and discharge temperatures of charge cooler systems. Red + pink is "2-core" air-to-air data; blue + light blue is air-water. Averaged data from same test runs as Figure 4. 5.0L Mustang (1986-1993 M/Y) engine. Ø36" ducted fan used for cooling air supply; tests conducted after warm-up and thermal stabilization of system components.

Perhaps the most telling performance difference is shown in Figure 6 – this is the density ratio performance across each charge cooler system undergoing the same tests. As previously discussed, density ratio combines the effects of both thermal performance and pressure loss. In this plot, the air-to-water system shows a performance-improving density gain at all speeds, while the air-to-air system shows a net density loss at speeds above 1700 RPM. We would expect, then, that the air-to-water equipped engine would produce the most power, and this is indeed the case: 342.1 HP at 5400 RPM vs. 312.6 HP at 5200 RPM (corrected to 60°F and 29.92 in-Hg).

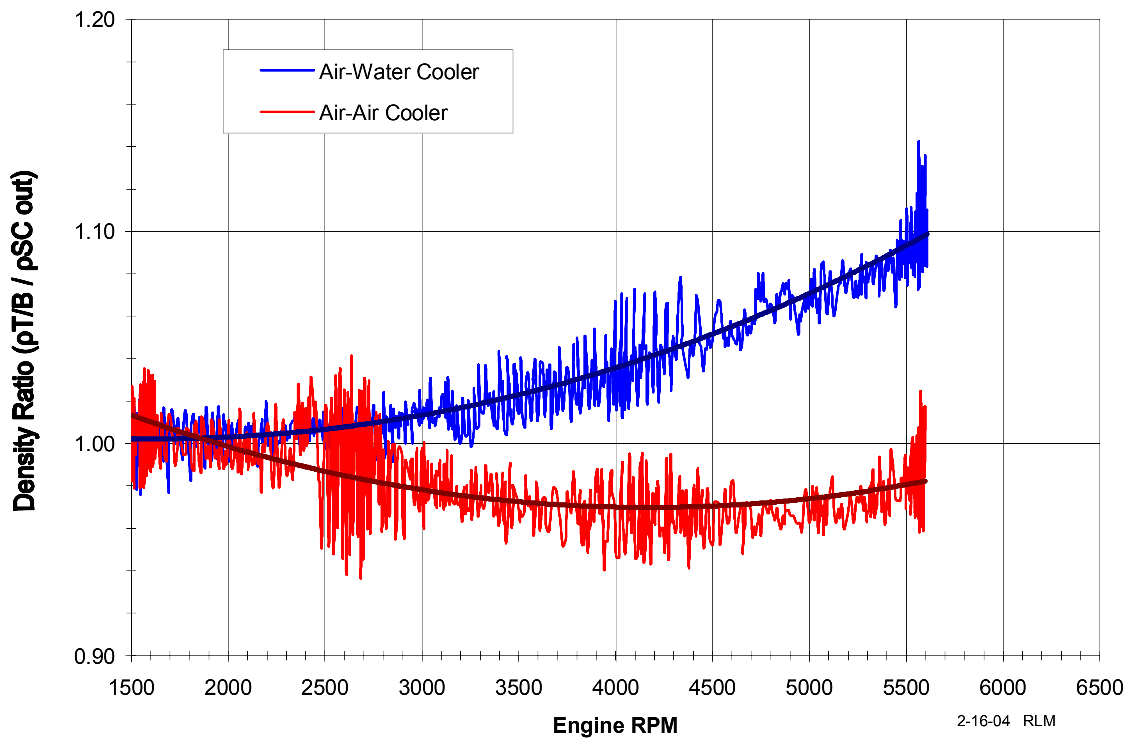

Figure 6 – Density Ratio performance for air-to-water and comparison "2-core" air-to-air charge cooler systems. 5.0 L Mustang engine dynamometer test. Air-to-water results: 342.1HP at 5400 RPM; air-to-air results: 312.6HP at 5200 RPM (Corrected HP).

Chassis Dyno Tests

Figure 7 shows temperature performance obtained from chassis dyno tests of the air-to-water and air-to-air systems, overplotted for comparison. Clearly, chassis dynamometer tests yield similar results with the air-to-water system delivering consistently cooler charge temperatures and more power. On the engine dyno, the Vortech V1S with air-water charge cooler system developed 9.4% more power than the competing supercharger with air-air system. On the chassis dyno, 9.8% more power was developed.

Road Testing

Finally, we consider the performance of these systems under actual in-service, road test conditions. Of key interest is to determine charge-air temperatures during actual use, and confirm that laboratory and dynamometer tests are consistent with in-service use.

Figure 8, then, presents results from the instrumented vehicle (1991 Mustang 5 Liter) equipped with the competing supercharger and air-to-air charge cooler system. Figure 9 is the same vehicle equipped with the Vortech supercharger and air-to-water charge cooler system.

To make for easy comparison of charge temperatures, Table 1 summarizes the results of engine dyno, chassis dyno, and road tests. Along with the logged temperatures, the engine speeds at which these temperatures occur are also logged. It should also be recognized that these tests are dynamic but similar in nature – engine and chassis dyno are ‘sweep tests’ conducted at 300 RPM/second while the road test is a full throttle acceleration of approximately the same time duration. As can be seen in Table 1, the charge air temperatures achieved during engine and chassis dyno tests are comparable to those attained during in-service use, especially with the air-to-air system.

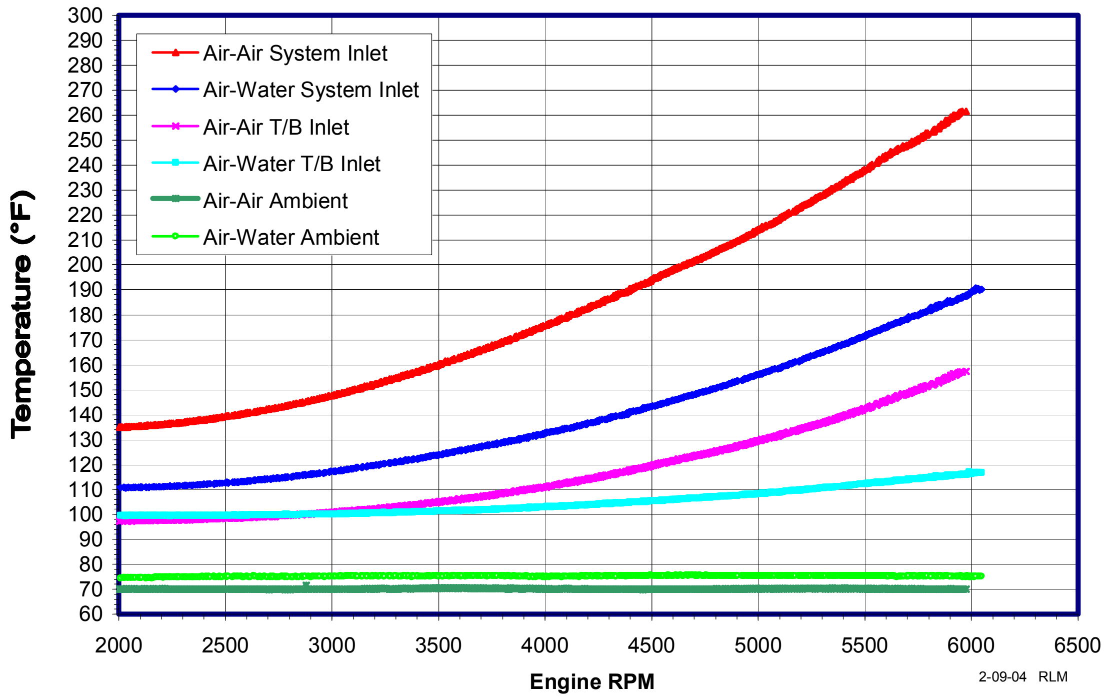

Figure 7 – Chassis Dynamometer Testing – Thermal performance of charge cooling systems. Competing supercharger with “2-core” air-to-air charge cooler system (red + pink) and Vortech V1S supercharger with air-to-water (light & dark blue) charge cooler. Bottom curves (light & dark green) are ambient temps for both tests. Ø36” ducted fan used for cooling air supply for all tests. 1991 5.0L Mustang. Vortech V1S with air-to-water system: 250.8 RWHP at 5450 RPM; Competing supercharger with air-to-air system: 228.5 RWHP at 5300 RPM (Corrected HP). Tests conducted after 15 minute (min.) warm-up street drive. Note slightly cooler ambient conditions for air-air test.

Table 1 – Comparison of charge air temperatures, taken at engine throttle body.

System	TEST					
	Engine Dyno		Chassis Dyno		Road	
	T/B Temp @ RPM		T/B Temp @ RPM		T/B Temp @ RPM	
Air-Air	127°F	5250	134°F	5250	128°F	5200
Air-Water	98°F	5250	110°F	5250	110°F	5000

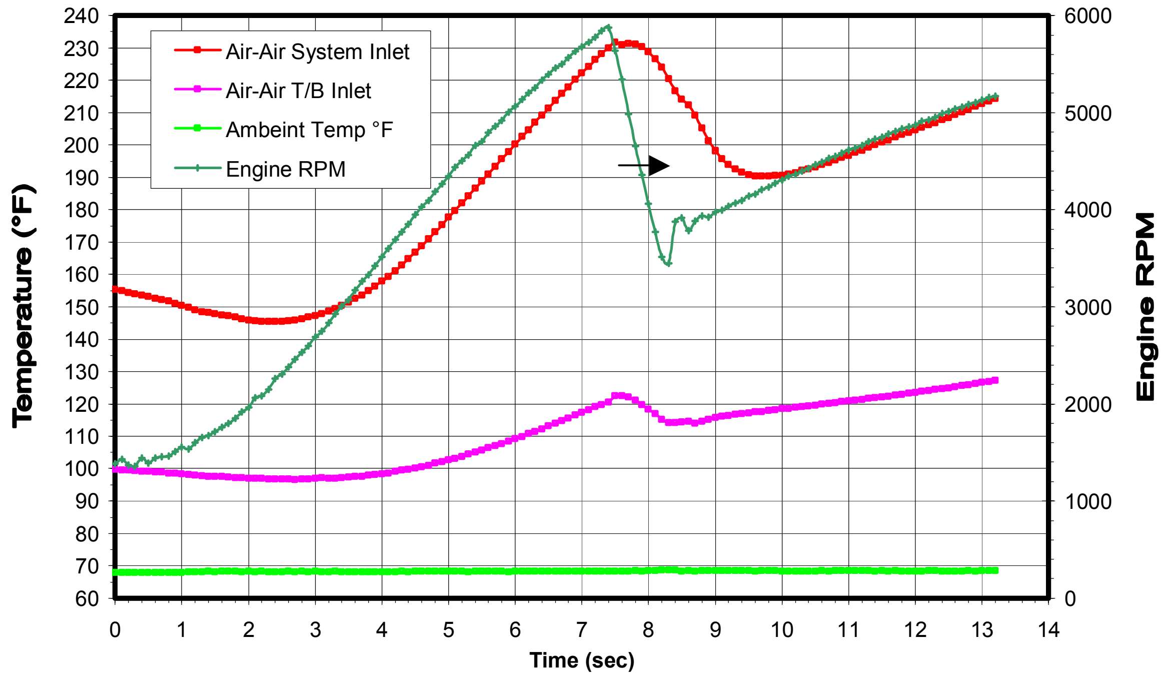

Figure 8 – WOT Acceleration Road Test – competing supercharger with "2-core" air-to-air charge cooler system. 1991 Mustang 5.0L. Tests conducted after 15 minute (min.) warm-up drive.

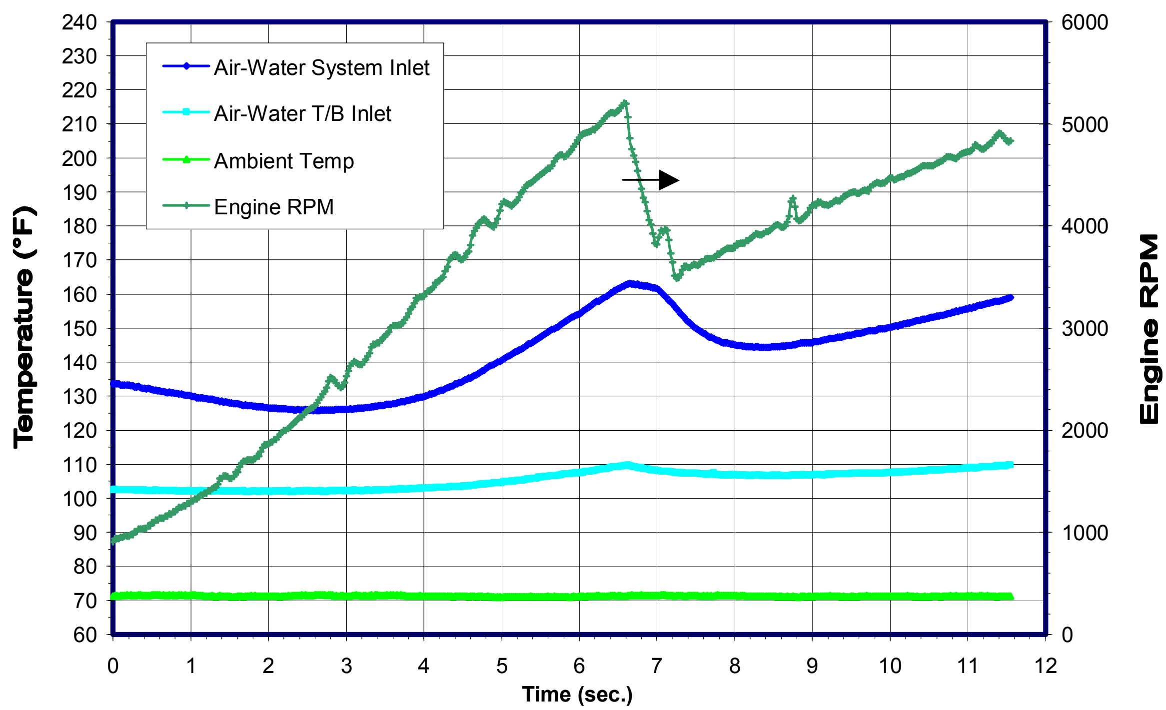

Figure 9 – WOT Acceleration Road Test – Vortech V1S supercharger with air-to-water charge cooler system. 1991 Mustang 5.0L. Tests conducted after 15 minute (min.) warm-up drive.

Closure

Charge cooler system testing is vital to the design and development process, and can afford the consumer with meaningful information for evaluating product performance. Effectiveness testing provides a means for evaluating thermal performance of a heat exchanger. Generally, and given equivalent flow, pressure, and thermal loading conditions at the inlet, the cooler with the best effectiveness performance will be the best at cooling charge air.

Effectiveness alone, however, does not provide the complete story as system pressure loss is critical. The change in air density across the charge cooler system incorporates the effects of both thermal performance and pressure loss. Thus the system with the best density ratio performance will be the best performing charge-air cooling system, and provide the greatest potential for power increase.

Air-to-air charge coolers are quite common and benefit from their simplicity in design and low cost. Air-to-water charge cooler systems are compact and can typically be installed directly between the supercharger and engine throttle body, minimizing or even eliminating extra piping and associated losses. This latter feature offers significant pressure loss and hence density ratio advantages over the typical air-to-air system.

The conductivity of water and improved heat transfer coefficient of the air-to-water cooler produces thermal performance advantages that air-to-air coolers cannot match. Claims that ice-water baths are required for achieving equal performance to the air-to-air cooler are closer to folklore than fact, and are simply not supportable. Further, no ice-water bath is used nor is appropriate to use for tests concerning products intended for the general consumer. Even so, for the consumer in search of the ultimate in performance enhancing potential, the advantages of the air-to-water system makes the choice clear.

References and Endnotes

[1] Bell, Corky, ***Maximum Boost,*** 1997, Robert Bentley, Inc., pp 67.

[2] www.procharger.com, 2004, Accessible Technologies, Inc.

[3] 1991 M/Y Mustang used for all tests. No modifications to the engine or vehicle other than installation of manufacturers' supercharger equipment and charge cooler system; i.e., MAF, exhaust, etc., is OE.

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