

Integrated Low Temperature Cooling System Development in Turbo Charged Vehicle Application

Xu Song, John Myers, and Scott Sarnia
Hyundai-Kia America Technical Center Inc.

ABSTRACT

The Low Temperature Cooling (LTC) system is commonly developed for secondary cooling function requirements, such as forced induction air cooling, and HEV power electronics module cooling. The large heat transfer capacity of coolant allows for very compact water-cooled heat exchangers to be installed remotely for better underhood aerodynamic characteristics and more compact packaging design. An integrated LTC loop developed on a Hyundai 2.0L Turbo Charged vehicle extends a traditional WCAC (Water-cooled charged air cooler) application to include a water-cooled condenser (WCOND) module. Unlike other published LTC system design approaches, this research project emphasizes underhood airflow improvement strategy and focuses on heat transfer efficiency. This paper discusses the integrated LTC loop configuration, Low Temperature Radiator (LTR) design, coolant flow control, and others. The LTC system prototype design allowed vehicle performance data to prove that a simplified front end cooling module package allows better airflow cooling efficiency.

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INTRODUCTION

A typical Low Temperature Cooling (LTC) loop consists of low temperature radiator (LTR), electrical pump, degas reservoir, functional heat exchanger(s), and associated plumbing. As the primary role of powertrain cooling system is to prevent the engine and transmission from overheat damage, a LTC loop is designed to provide the possible 'coldest' coolant flow to subsystems where low coolant temperature is desired. Many such subsystems, forced air induction and electric vehicle power electronics, just to name a few, will perform better if system operating temperatures are kept cool. The water-cooled charged air cooler (WCAC) is one of the LTC loop applications in a forced air induction passenger vehicle. Since coolant flow cooling capacity rate is much greater than cooling airflow, i.e. air to air intercooler found standard in many vehicles today, it allows a very compact intercooler to be installed in close proximity or inside of the engine intake manifold. Compared to an air-cooled CAC (ACAC) in either full face or brick configuration, a WCAC has a smaller internal volume and a short overall duct hose length from the turbo outlet to the throttle body. In addition, the internal airflow pressure drop across the WCAC is also much lower than a traditional ACAC. The combined characteristics of small internal volume and low pressure drop of the charged air system with WCAC results in reduced turbo lag and decreased engine pumping power for spinning the turbine. Theoretically a

turbo-charged vehicle equipped with a WCAC exhibits full boost power and faster acceleration response, as well as better fuel economy over those equipped with a traditional ACAC [1].

In design practice, a LTC loop is normally treated as an add-on subsystem which simply replaces ACAC with a WCAC mounted at or in the engine intake manifold. The LTR is placed into the frontal cooling package space where the ACAC is removed from. In such LTC application, three heat exchangers, including condenser, low temperature radiator and high temperature radiator (HTR), construct a very thick stack-up because of the fixed frontal area. The thick stack-up induces two design downsides. First, it raises airflow resistance and requires more fan power for proper cooling airflow. Second, it squeezes frontal plenum space and unevenly redistributes cooling airflow. The frontal plenum is an airflow free flow space from the grill opening to heat exchanger core face. In order to maximize the WCAC advantages but ease LTC loop design downsides, the integrated LTC system has been proposed to apply one single LTC loop for multiple cooling functions, such as refrigerant sub-cooling by the WCOND (water-cooled condenser) along with charged-air cooling by WCAC. The system can also use a WCOND along with power electronics module cooling in HEV cars. When an integrated LTC system removes the air-cooled condenser from frontal cooling package, it allows LTR and HTR to utilize a full-face cooling area to reduce stack-up thickness. Therefore, integrated LTC can potentially solve design issues addressed above.

Several integrated LTC system development projects have been carried out by heat exchanger suppliers. Valeo applies their Ultimate Cooling System on a demo Mercedes C220 CDI to achieve desired refrigerant sub-cooling and charged-air cooling with some fuel consumption improvement [2]. Hara et al presented a partial and full WCOND with WCAC concept to improve refrigerant sub-cooling [3]. Vestrellis et al compared vehicle test and simulation results of a secondary cooling system for water cooled CAC and condenser developed by Fiat and Denso TS [4]. Although it is reported to achieve system design target, these systems are so complex and costly that it is not feasible to adapt them into current production applications. The system analysis shows that the greatest challenge in integrated LTC system development is to get coolant temperature low enough for sufficient refrigerant sub-cooling [5]. Since the temperature difference of a LTC loop between coolant flow and ambient cooling airflow is relatively small, the traditional LTC system design approach may not be capable to meet cooling efficiency required by integrated LTC system applications. However, taking a systematic approach that focuses on radiator tube-fin portfolio efficiency, optimal coolant flow design, proper cooling airflow control and other technologies, it is possible to achieve low temperature cooling requirement with a relatively simple and low cost design.

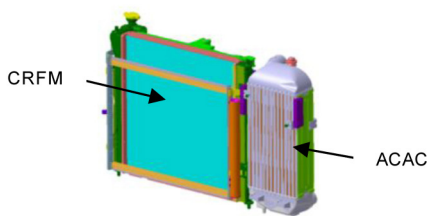


Figure 1. Current production powertrain cooling module with a side-by-side ACAC. There is no package space for engine cooling fan shroud to cover ACAC.

This research paper highlights several technical details in the Hyundai-Kia America Technical Center Inc. (HATCI) integrated LTC system developed on a Hyundai 2.0L Turbo-charged vehicle. As showed in Figure 1, the current production turbo-charged vehicle adapts a brick type of air-cooled charged air cooler (ACAC) which is installed in side-by-side position to condenser/radiator/fan module (CRFM). Since the ACAC is so thick that the engine cooling fan shroud cannot cover it within package space, ACAC is solely dependent on ram cooling airflow. In other words, ACAC does not have desired cooling capability to charged air when vehicle speed is low. Previous vehicle performance benchmark study suggested that ACAC housing heat build-up and turbo lag might impact vehicle acceleration performance. Therefore, the integrated LTC system is a good fit to this vehicle application. The discussion in this paper include: 1) underhood aerodynamic consideration to improve cooling airflow uniformity, which attempts eliminate hot air recirculation and increases airflow cooling efficiency with less fan power; 2) use appropriate coolant flow plumbing strategy for effective cooling control; 3) low temperature radiator tube-fin definition for ultimate heat transfer capability; 4) water-cooled intercooler sizing methods for determining

component specification. The paper also analyzes vehicle validation data in perspective on LTR performance limitations, A/C system operational conditions, charged air cooling issues, coolant flow control and electrical power consumption.

LTC SYSTEM DESIGN

This section discusses LTC loop systematic design, including underhood airflow control method, coolant plumbing comparison, heat exchanger sizing and heat transfer efficiency analysis.

Underhood Airflow Management

In the past several years, the HATCI vehicle thermal system engineering team has conducted underhood airflow research for the purpose of engine cooling module (ECM) optimization design. The research concludes that one-dimensional (1D) underhood airflow model with a single duct flow path could over simplify the underhood airflow problem and mislead ECM design. The single duct flow path model assumes that all of the airflow entering the grill openings will pass through the ECM completely. In design practice, the single duct flow requires a physical duct that forms a sealed airflow channel between grill opening and heat exchanger core faces. Although the sealed cooling airflow duct design is seen in race cars and some of the high-end sport cars, it is not a typical design option for most passenger cars. In order to properly model underhood cooling airflow conditions, the underhood airflow 1D model with a bypass channel has been developed (as shown in Fig 2). The airflow bypass channel represents air free flow space connecting the front plenum and engine compartment, including large gaps and open space around ECM.

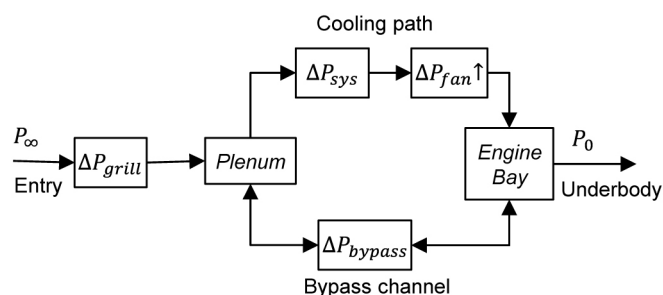


Figure 2. Underhood airflow path schematics

The single duct flow model is described with an extended Bernoulli's Equation, in which all viscous effects are combined in a loss term : ΔP_{sys} :

$$P_{\infty} - P_0 = \frac{1}{2} \rho (v_{\infty}^2 - v_r^2) - \Delta P_{sys} + \Delta P_{fn} \quad (1)$$

where P_{∞} , v_{∞} are total pressure and airflow velocity at grill openings respectively, v_r is averaged airflow speed across heat changers, P_0 is total pressure exiting engine compartment, loss term ΔP_{sys} mainly consists of the heat exchanger pressure drop and fan shield resistance, and ΔP_{fn} is the fan pressure loss from cooling fan operation.

For the airflow model with bypass channel, underhood airflow is split to effective cooling path and bypass path. In effective cooling paths, airflow passes through heat exchangers with direct cooling effect. In contrast, along the bypass path, airflow simply skips around heat exchangers without direct cooling impact. The loss term in bypass channel consists of viscous effect from complex underhood structural blockages and is simplified to an orifice valve:

$$\Delta P_{bypass} = \frac{1}{2} \rho K_{byp} \dot{V}_{byp}^2 \quad (2)$$

where K_{byp} is a resistant factor possibly determined from CFD simulation, $\dot{V}_{byp}$ is volumetric flow rate through bypass channel. To describe the bypass airflow 1D model, additional equations are proposed in the following. Equation 3a states a recirculation condition that the total fresh cooling airflow pulled from grill openings is actually less than airflow through radiator core area. Equation 3b expresses that excessive fresh cooling airflow will exhaust through the bypass channel without going across the radiator core. In equation 4, it is said that fan static pressure balances static pressures between cooling path and bypass channel.

$$v_r \cdot A_r - \dot{V}_{byp} = v_{\infty} \cdot A_g \quad (3a)$$

$$v_r \cdot A_r + \dot{V}_{byp} = v_{\infty} \cdot A_g \quad (3b)$$

$$\Delta P_{sys} - \frac{1}{2} \rho K_{byp} \dot{V}_{byp}^2 = \Delta P_{fn} \quad (4)$$

where A_r is radiator core area, A_g is grill opening area. Equations 1, 3 and 4 construct a complete one dimensional underhood airflow model with a bypass channel.

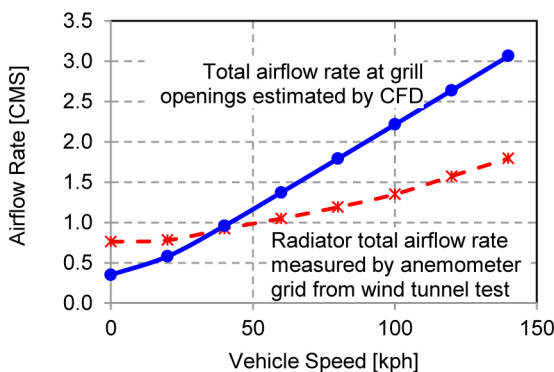


Figure 3. Underhood airflow study from a similar vehicle demonstrated bypass airflow effect: the actual airflow across radiator is greater than grill opening airflow at low vehicle speed, but is less at high vehicle speed. Fan speed was set at ~2100 RPM during the testing and simulation

The comparison study between radiator core airflow and grid opening airflow from a Hyundai car was carried out to verify the one dimensional bypass channel theory. The radiator core airflow was measured by a 3×5 anemometer grid at various vehicle speeds from 0 kph up to 140 kph. The total airflow of grill openings was calculated from vehicle CFD simulation. As shown in Figure 3, the radiator core airflow is greater than the total grill opening airflow when vehicle speed is below 30 kph. In contrast, there is more airflow from grill openings than that at the radiator core when vehicle speed exceeds 40 kph. This phenomenon has been explained as such: When vehicle speed is below a threshold, said 30 kph for this particular car, the ram airflow static pressure is so low that the cooling fan creates negative static pressure in the front plenum but positive static pressure in the engine bay. This static pressure differential drives partial hot air discharged from ECM back to the front plenum through the bypass channel. The static pressure differential forms hot air recirculation situations in which ECM only receives partial fresh cooling airflow from the outside ambient. At idle, less than 50% of radiator core airflow comes from fresh ambient and the rest comes from the ECM's exhaust hot air. In opposite, when vehicle speed exceeds 40 kph, the ram airflow creates positive static pressure in the front plenum and pushes excess cooling airflow around the ECM through surrounding open spaces. At 100 kph, up to 40% of fresh ambient airflow does not pass through radiator for system cooling.

Both hot air recirculation and cooling airflow bypass leakage have negative impacts to ECM performance. At idle to low vehicle speed conditions, hot air recirculation can significantly heat up cooling airflow before condenser and reduce A/C system performance. On the other hand, the bypass may cause insufficient heat exchanger cooling airflow that the powertrain needs as more cooling capacity is required during high speed and heavy load situations. In general, the engineering remedy for fixing these two problems is to increase fan power for more cooling airflow and/or to increase heat exchanger thickness/fin density for larger cooling capacity. From a cooling efficiency stand point, increasing the heat exchanger capacity and/or fan power are not effective design solutions. Our research findings suggest that eliminating hot air recirculation and reducing cooling airflow bypass leakage is an effective design approach to improve airflow cooling efficiency. For instance, the condenser inlet airflow measurement from a research vehicle showed the averaged cooling temperature could be reduced by 18 °C and A/C system COP could be increased by 27% under the condition of 45 °C ambient, no wind and idle.

A simple way to eliminate the airflow bypass channel is to apply appropriate underhood sealing apparatus, or air duct (in Fig 4). This method has been tested on several Hyundai production vehicles. The test data shows that underhood sealing can effectively increase the ram airflow cooling effect and prevent hot air recirculation from occurring. In the integrated LTC system development, the LTR/HTR configuration reduced overall ECM thickness that provides a larger frontal free space for easier sealing structure design.

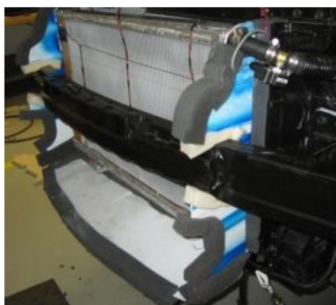


Figure 4. Underhood air duct sealing design

LTC Loop Configuration

Two coolant loop flow configurations have been considered: the first one involves series connection between components while the second one uses parallel connections. The series design places the WCAC downstream of the WCOND in a single flow loop (see Fig 5a). Although the series loop is simple in design and control, three design issues should be addressed: first, it requires more pumping power to transport coolant due to high coolant flow resistance in the plumbing loop; second, if the A/C system operates, the WCOND will heat up the downstream coolant flow that will degrade WCAC cooling performance by increasing charged air outlet temperature; lastly, when A/C system is shut off, it is impossible to cut off the coolant flow across WCOND and it wastes coolant flow pumping power. In contrast, since the parallel loop has lower system resistance by splitting coolant flow to two independent paths (shown in Fig 5b), the coolant pump is capable to deliver more coolant flow rate than series loop with the same pumping power. As illustrated in Figure 6, pump curve at 12 volts intersects with two system resistance curves respectively and predicts 40% more coolant flow rate from parallel system. Furthermore, there is little thermal interference between the WCOND and WCAC so that charged air temperature can be maintained consistently. However, the total coolant flow rate and coolant split ratio can influence heat transfer performance of the LTR, WCOND and WCAC. It is also necessary to develop three-way flow control valve along with associated control logic. Therefore, parallel loop design is more complex and expensive than series loop system.

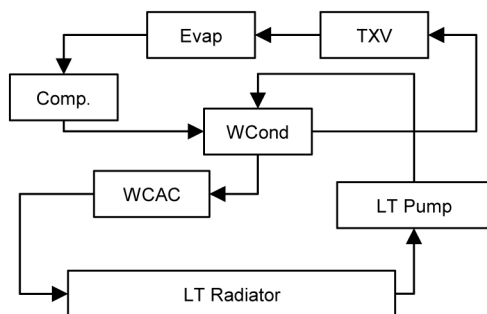


Figure 5a. LTC series loop schematic

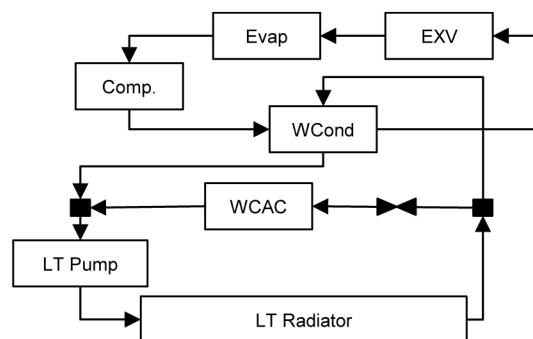


Figure 5b. LTC parallel loop schematic

For system and component characteristic evaluation purpose, the series loop was applied in this project phase because it is simple to control and easy to test. However, the parallel loop design study has been planned for a followed project phase in which production intent design requirements, coolant flow control logic and cost-performance trade-off study will be included.

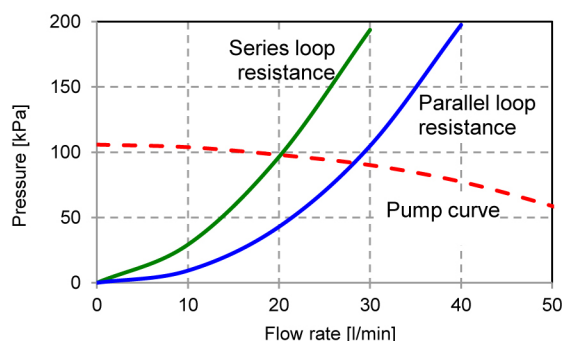


Figure 6. Pump curve intersects with Series and Parallel loop system resistance curves for estimating system working points. The pump curve showed here is specified at 12 volts. Pump curves at 9 and 13.5 volts can be applied to determine various pump working points.

Low Temperature Radiator Design

The primary LTR design goal of an integrated LTC system is to cool the coolant temperature down as close to ambient as possible so that the desired refrigerant sub-cool can be achieved at WCOND. Radiator cooling capacity can be determined by effectiveness method as in Equation (5)

$$\dot{q} = \varepsilon (mC_p)_{\min} (T_{h,In} - T_{c,In}) \quad (5)$$

where is radiator effectiveness explained by Kays and London [4], $(mC_p)_{\min}$ is minimum heat transfer capacity rate between hot and cold fluids, $T_{h,In}$ and $T_{c,In}$ are inlet temperatures of hot and cold fluids respectively. In high temperature cooling applications, such as a normal powertrain cooling radiator, since items of mC_p and $T_{h,In} - T_{c,In}$ are relative large, ε has less effect to radiator heat transfer rate. Therefore, the radiator portfolio is not extremely critical in high temperature radiator design. However, in the low temperature cooling application,