

Efficient, Active Radiator-Cooling System

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ABSTRACT

A new concept for an efficient radiator-cooling system is presented for reducing the size or increasing the cooling capacity of vehicle coolant radiators. Under certain conditions, the system employs active evaporative cooling in addition to conventional finned air cooling. In this regard, it is a hybrid radiator-cooling system comprised of the combination of conventional air-side finned surface cooling and active evaporative water cooling. The air-side finned surface is sized to transfer required heat under all driving conditions except for the most severe. In the later case, evaporative cooling is used in addition to the conventional air-side finned surface cooling. Together the two systems transfer the required heat under all driving conditions. However, under most driving conditions, only the air-side finned surface cooling is required. Consequently, the finned surface may be smaller than in conventional radiators that utilize air-side finned surface cooling exclusively. Results of this study include details of the hybrid system as well as radiator size and evaporative cooling loads at various evaporation rates. It is shown that a hybrid radiator with a 76-L/hr (20-gal/hr) water flow rate can transfer up to 19% more heat than a conventional radiator with the same finned area, or alternately, it can transfer heat at the same rate with a radiator size of 21% less than that of a conventional radiator.

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INTRODUCTION

Coolant radiators in trucks and automobiles are designed to transfer the maximum heat load at a designated "design condition." An example design condition is a fully-loaded truck climbing up Baker Grade, a stretch of Interstate Highway 15 just east of Baker, California, on the hottest summer day. The coolant system including the radiator is sized to remove 100% of the required heat from the engine at the design condition without boiling the coolant, which results in a large radiator. This condition has two ramifications: first, the radiator is oversized under most driving conditions; and second, in cases where the maximum radiator size is limited by the vehicle frontal area, the engine power may be limited by the radiator size.

The dominant factor affecting the size of a coolant radiator is the heat transfer coefficient of the outside air flowing over it. If this coefficient were increased, the radiator size could be reduced, or more heat could be transferred from

an existing radiator. The latter case would allow for increased engine power when it is limited by the radiator size.

The air-side fin designs of current radiators have been optimized to maximize the effective air-side heat transfer coefficient and heat transfer area. Typical louvered fins are of short fin lengths (close channel spacing) to maximize fin effectiveness. With the fin design at or near optimum, there is little more to be gained in terms of increasing the effective heat transfer coefficient from this aspect of the design. Therefore, an alternate concept, a hybrid system of cooling the outside of a radiator using both finned air cooling and evaporative cooling, was developed and evaluated in this study for increasing the air-side heat transfer coefficient. The evaporative cooling is a more active heat transfer mechanism, and it is envisioned that it would be used only at or very near the thermal design condition for the coolant system. Under most driving conditions, only the more conventional finned air cooling would be used to remove the required engine heat.

Evaporative cooling mechanisms including liquid films, liquid drops, and sprays have been studied for various applications [1, 2, 3, 4]. Any of these types of evaporative cooling could be used in the hybrid radiator-cooling system. In this study, both liquid films and liquid drops were analyzed for gravity flow along the extended surfaces of the radiator coolant channels. Also, as will be discussed subsequently, the geometry for the hybrid radiator-cooling system is quite flexible. As an example, a geometry with important heat transfer advantages was analyzed in this study.

APPROACH

Figure 1b shows the hybrid radiator compared to a conventional radiator of Figure 1a. The example hybrid radiator-cooling system shown in Figure 1b is similar to the conventional radiator with vertical coolant channels and fins between them on the air side. The channels have been extended beyond the fins on the downstream air side of the radiator. Liquid water flows downwards by gravity along the extended surfaces providing evaporative cooling to the engine coolant. In the case of the hybrid radiator, there is a liquid supply and distribution system not shown in Figure 1b.

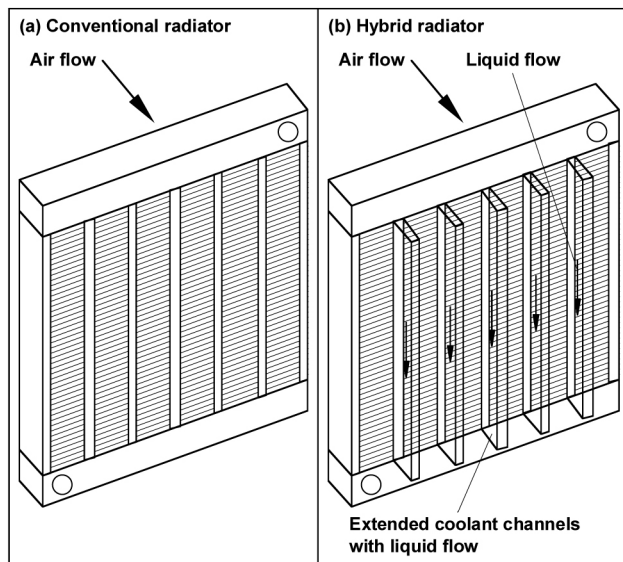


Figure 1. Hybrid radiator system.

Figure 2 shows a top view of a section of the hybrid radiator. In this schematic, the extended channel surfaces are cooled by evaporating water flowing downwards by gravity into the plane of the figure. The combination of the conventional cooling from the finned surfaces and the evaporative cooling from the extended channel surfaces is the total heat transfer from the radiator to the atmosphere. Under the thermal design condition, both cooling mechanisms would be functioning. However, at most thermal loads below the design condition, only the conventional air-side finned surface cooling would be required. Thus, the active cooling of the water evaporation would be used only at or very near the thermal design condition.

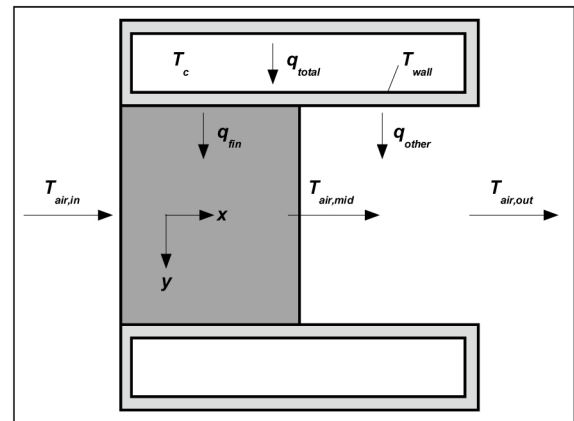


Figure 2. Top View of a section of the hybrid radiator (z direction is into the page).

This limited use, of the active evaporative cooling component of the hybrid radiator-cooling system, is important because evaporative cooling requires a supply of water. Using evaporative cooling only at or very near the thermal design condition serves to optimize the parameters of reduced radiator size (or increased maximum radiator heat transfer) and minimized water use/transport. In the results presented subsequently, an example is given of a hybrid radiator-cooling system with a reduced sized radiator being capable of rejecting all required engine heat under all driving conditions except for rare desert conditions like Baker grade. Under such exceptional driving conditions, the evaporative cooling component of the system would be used in addition to the conventional air-side finned channel heat transfer. It will be shown that evaporative cooling is an excellent means of heat transfer and can have a significant effect on the radiator size or the maximum radiator thermal load. Importantly, the water supply necessary to accomplish this hybrid radiator cooling is reasonable and does not represent any significant technology barrier. Furthermore, the added weight of water is only encountered during exceptional driving conditions and will be partially compensated by the reduced radiator size and weight. It will be shown that adding evaporative cooling, without changing the finned surface area, can increase the heat transfer rate up to 46% for reasonable water flow rates and water usage amount.

COMPONENT GEOMETRY

The geometry and characteristics of the radiator, engine, and coolant pump used in this study were determined from a generic Cummins class 8, 500-hp diesel, truck engine identical to the one used by Sarapella, et al. [5]. The radiator cooling channels of Figure 1 were 564 mm high and 52 mm wide along the direction of air flow. The coolant channel extensions, of the hybrid radiator of Figure 1b, added 20 mm to the channel width for a total of 72 mm. The thermal design condition for the radiator was taken at full engine power occurring at 1700 rpm and 221.8 kW to be rejected in the

radiator at an ambient temperature of 47 °C. This temperature is the highest temperature in a typical meteorological year for Barstow, California (near Baker Grade) [6].

ANALYSIS

Heat transfer increases and benefits were analyzed utilizing evaporating water in the form of a continuous falling film on the extended coolant channel surfaces of the radiator by comparing it to the radiator without evaporative cooling. In addition, heat transfer benefits were considered where the continuous water film was replaced by discrete water droplets falling by gravity on the extended channel surfaces. Results focused on the percentage of evaporation of the water, the increased heat transfer rates attainable, and the decreased radiator size attainable.

Evaporative Cooling from Falling Film

The local heat transfer coefficient, evaporation rate, film thickness, and film velocity for a continuous falling film flowing downwards along the extended channel surface shown in Figure 2 can be determined by the classical Nusselt solution through analogy between condensation and evaporation [7, 8]. The film thickness δ at a given position z is determined as a function of the density ρ and the viscosity μ from a force balance between gravity and the viscous forces at the wall

$$\delta = \left[\frac{3\mu_w \Gamma_w}{\rho_w(\rho_w - \rho_{air})g} \right]^{1/3} \quad (1)$$

where the mass flow rate per unit depth Γ_w , an important parameter in determining evaporation, is a function of the film thickness δ and the average water velocity $\bar{u}$

$$\Gamma_w = \rho_w \bar{u}_w \delta \quad (2)$$

The rate of heat transfer from the wall to liquid surface is controlled by conduction through the liquid film and can be calculated by assuming the water vapor at the water/air interface being at a saturated state with the same temperature as the air

$$\dot{q}'' = k_w \frac{dT}{dy} = k_w \frac{T_{wall} - T_{sur}}{\delta} = h_w (T_{wall} - T_{air}) \quad (3)$$

The fluid flow behavior of the film as it flows downwards along the extended channel surfaces can be characterized through its Reynolds number

$$Re = \frac{\rho_w \bar{u}_w D_h}{\mu_w} = \frac{4\rho_w \bar{u}_w \delta}{\mu_w} = \frac{4\Gamma_w}{\mu_w} \quad (4)$$

The liquid flow is laminar, laminar wavy, and turbulent for the Reynolds number in the ranges of $Re < 30$, $30 < Re <$

1800, and $1800 < Re$, respectively. Combining the above equations yields the classical Nusselt solution for heat transfer through a gravity-controlled laminar falling film with the Reynolds number of $Re < 30$

$$\frac{h_w}{k_w} \left[\frac{\mu_w^2}{\rho_w(\rho_w - \rho_{air})g} \right]^{1/3} = \left(\frac{4}{3Re} \right)^{1/3} \approx \frac{1.10}{Re^{1/3}} \quad (5)$$

When the Reynolds number is in the range of $30 < Re < 1800$, waves appear on the surfaces causing enhanced heat transfer. To account for this enhancement, an empirical factor $0.8(Re/4)^{0.11}$ is used, which results in

$$\frac{h_w}{k_w} \left[\frac{\mu_w^2}{\rho_w(\rho_w - \rho_{air})g} \right]^{1/3} = 0.8 \left(\frac{Re}{4} \right)^{0.11} \left(\frac{4}{3Re} \right)^{1/3} \approx \frac{0.756}{Re^{0.22}} \quad (6)$$

None of the flow rates in this study is in the turbulent region with the Reynolds number of $1800 < Re$ where additional empirical correlations have been developed [9]. The mass flow rate decreases as a function of its vertical position due to the heat transferred from the wall evaporating liquid resulting in

$$\frac{\Delta \Gamma_m}{\Delta z} = - \frac{h_w (T_{wall} - T_{air})}{i_{fg}} \quad (7)$$

Parallel Fin and Evaporative Cooling

The total heat transfer rate leaving the coolant at a given height z is the sum of the heat transfer rates entering the air through the finned portion and the extended portion of the coolant channel surfaces

$$\dot{q}_{total}(z) = \dot{q}_{fin}(z) + \dot{q}_{ext}(z) \quad (8)$$

In Eq. (8), the total heat transfer rate leaving the coolant $\dot{q}_{total}$ can be expressed as a function of the vertical temperature change in the coolant by the energy balance on the coolant side

$$\dot{q}_{total}(z) = \dot{m}_c c_{pc} [T_c(z) - T_c(z + dz)] \quad (9)$$

or as a function of the temperature difference between the coolant and the wall in terms of thermal resistances

$$\dot{q}_{total}(z) = h_c \Delta A_c [T_c(z) - T_{wall}(z)] \quad (10)$$

In obtaining Eq. (10), T_{wall} was assumed to be a function of the height z because the thermal resistance in the aluminum is small.

The heat transfer rate entering the air from the finned portion of the coolant channel surfaces $\dot{q}_{fin}$ can be expressed