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ABSTRACT

The cooling system of a Class 8 truck engine was modeled using the Flowmaster computer code. Numerical simulations were performed replacing the standard coolant, 50/50 mixture of ethylene-glycol and water, with nanofluids comprised of CuO nanoparticles suspended in a base fluid of a 50/50 mixture of ethylene-glycol and water. By using engine and cooling system parameters from the standard coolant case, the higher heat transfer coefficients of the nanofluids resulted in lower engine and coolant temperatures. These temperature reductions introduced flexibility in system parameters - three of which were investigated for performance improvement: engine power, coolant pump speed and power, and radiator air-side area.

INTRODUCTION

The objective of this study was to determine the effects of using a nanofluid as the coolant in the cooling system of a Class 8 truck engine. Nanofluids are comprised of small concentrations of nanometer-sized solid particles suspended in liquid base fluids. The particles have been shown to significantly enhance the thermal characteristics compared to the base fluids alone. (See, for example, the early work of Choi [1] and Eastman [2].) Using nanofluids as engine coolants can lead to increased cooling rates, which in turn have positive ramifications on such things as engine materials, radiator size and effectiveness, coolant pump power and size, cooling system size, and truck aerodynamic design. With this large general potential for improvement in mind, the objective of this study was to quantify the effects in these areas.

The approach taken was to utilize the commercial system computer code, Flowmaster [3] to simulate the cooling system of a Class 8 truck engine. The Flowmaster code dates back to 1980. Today it has an automotive version, with dynamic thermo-fluid system simulation, that includes a variety of standard and custom components that can be arranged into a system. Components configured in Flowmaster for this study included the engine cooling passages, coolant pump, radiator, thermostat, cabin heater, and interconnecting piping and hoses. Operating conditions were then simulated using two nanofluid coolants as well as the standard ethylene-glycol and water coolant. Comparisons were made among the nanofluids and the ethylene-glycol and water coolant (50 percent of each component by volume) to quantify any improvements found due to the nanofluids. Specifically, the primary goals of the numerical simulations were to:

- Compare the performance of nanofluid coolants to the standard coolant for a Class 8 truck diesel engine.
- Determine the performance and sizing of the coolant pump and radiator to maximize the cooling effect and minimize power consumption under steady-state conditions.
- Determine the enhancement in cooling capacity of the coolant circuit for a given design.

NANOFLUID COOLANTS

Nanofluids are nanotechnology-based heat transfer fluids that are derived by stably suspending nanometer-sized particles (with typical length scales of 1 to 100 nm) in conventional heat transfer fluids – usually liquids. Review papers [4-7] and most recently [8] with commentary [9] include overviews of various aspects of the field including histories of the technology development. Experimental observations documented in these papers show that nanofluids have enhanced thermal properties compared to the base fluid including higher thermal conductivity and heat-transfer capability. Many researchers have measured enhanced thermal properties and heat transfer rates using many different nanoparticles in a variety of liquids with volume concentrations in the range of 0.5-4%. At these low particle volume concentrations, typical enhancements have been very high - in the 15-40% range over the base fluid. Such nanofluids are good potential coolants for engine systems replacing the standard ethylene-glycol/water (50/50) mixture. They have the potential to enhance cooling rates and affect several under-hood components. In the current study, two nanofluids were considered for such application.

1) 2% by volume of CuO nanoparticles + ethylene-glycol/water (50/50)
2) 4% by volume of CuO nanoparticles + ethylene-glycol/water (50/50)

These two nanofluids were simulated for the purposes of this study and are representative of those used by researchers in the field [8].

In practice, surfactants are sometimes added to nanofluids for stability reasons. In this study, it was assumed that such additions had no effect on the nanofluid thermal properties and heat transfer. In addition, it should be noted that nanofluids are currently in a state of development, and nanofluids for commercial applications must meet important criteria such as long term stability, minimal erosion potential, high volume/low cost availability, and more.

CALCULATION OF NANOFLUID PROPERTIES – Fluid properties needed in the numerical simulations of this study were calculated in the following ways. The equivalent density of the nanofluid (subscript e) is the weighted average of the particle (subscript p) and base fluid (subscript m) densities as is the nanofluid specific heat. The dynamic viscosity of the nanofluid was calculated from the Einstein equation [10]. The equations for these three properties are given below, and they were used in this study to calculate nanofluid properties as a function of temperature.

$$\rho_e = (1 - v_p)\rho_m + v_p\rho_p \qquad (1)$$

$$C_{pe} = \frac{(1 - v_p)(\rho C_p)_m + v_p(\rho C_p)_p}{(1 - v_p)\rho_m + v_p\rho_p} \qquad (2)$$

$$\mu_e = (1 + 2.5 v_p)\mu_m \qquad (3)$$

In Equations 1-3, ρ is density (kg/m^3), C_p is specific heat (kJ/kg K), μ is dynamic viscosity (kg/m s), and v_p is the particle volume concentration in the nanofluid. These properties are compared in Table 1 at 20 C for nanofluid-1, nanofluid-2 and the standard base fluid.

Coolant / Property	Base Fluid (50/50 ethylene glycol/water mixture)	Nanofluid-1 ($v_p = 2\%$ CuO in base fluid)	Nanofluid-2 ($v_p = 4\%$ CuO in base fluid)
ρ kg/m^3	1072	1179	1285
C_p kJ/kg K	3.30	3.00	2.75
μ kg/m s	0.0046	0.0048	0.0050

Table 1 Nanofluid and Base fluid Properties at 20 C

NANOFLUIDS USED IN NUMERICAL SIMULATIONS – To complete the specification of the two nanofluids used in this study, general experimental results were taken from the engineering literature and used to define the thermal conductivities and the heat transfer coefficients. The extensive review of these two parameters [8] showed that, for nanofluids made up of CuO nanoparticles in base fluids of either ethylene glycol or water, 2% concentrations of CuO typically increased the thermal conductivity by about 20%. For a 4% CuO concentration, the thermal conductivity of the nanofluid was generally found to be about 40% higher than the base fluid. Similarly, the Nusselt number of a nanofluid, comprised of a 2% concentration of CuO nanoparticles in a base fluid of either ethylene glycol or water, was typically found to be 20% higher than the base fluid. A 4% CuO concentration produced a 40% increase in Nusselt number. These same enhancements were applied to the two nanofluids used in this study with a base fluid of a 50/50 mixture of ethylene-glycol and water.

COOLING SYSTEM MODELING

The engine cooling system of a Class 8 truck was modeled in the Flowmaster computer code and run in a transient mode. Simulations in the code started with the engine off at atmospheric conditions. Then the engine was started and brought to a steady-state condition over a prescribed time period. This transient analysis was used to understand the performance of the coolant circuit and to arrive at a stable steady-state condition. Attempts at using a direct steady-state solver lead to convergence problems owing to non-linearities in the coolant system. These were primarily due to the temperature dependent properties of the coolant and to the thermostat which redistributed the flow between two paths based on the temperature of the fluid at the inlet. Stability and convergence are discussed further with respect to Figure 2 below.

SYSTEM DESCRIPTION - The coolant system model includes heat rejection from the engine to the coolant circuit and from the coolant to air through the radiator. The coolant pump drives the flow through the network and the thermostat distributes the coolant to the radiator based on the actuation temperature. A complete cooling system block diagram is given in Figure 1. Flowmaster components were combined to simulate the engine, radiator, thermostat, and cabin heater functions in the cooling system as indicated in Figure 1. The salient Flowmaster components are as follows.

1 Expansion tank
2 Coolant recirculation path bypassing radiator
3, 29 Coolant pump
4, 5, 17 Thermostat flow control
6, 24, 28 Coolant hose
7, 10, 19 Engine cooling fan
8, 9, 15 Heat source - engine to coolant including EGR

12 Radiator
13, 20, 21 Cabin air flow
14 Cabin heater
16 Thermostat
18 Cabin heater flow control
30 Coolant path pressure loss

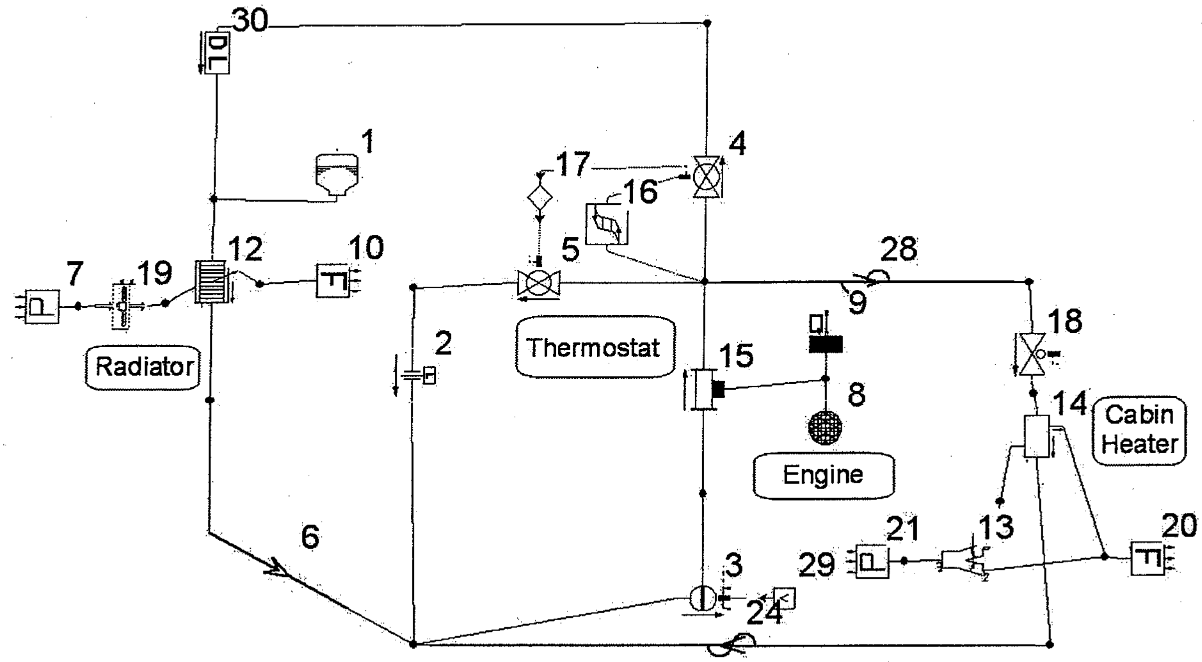

Figure 1. Coolant System Model

DESIGN POINT - Cooling circuits are designed to dissipate a maximum heat load. In practice, such a maximum occurs under summer conditions when the truck, under load, is climbing a steep grade. In Flowmaster, this design point translates into a maximum heat source from the engine to the coolant. In this study, the heat rejection, for a Class 8 truck diesel engine, to the coolant circuit was set to 298.4kW (400HP). During the transient analysis in Flowmaster, the engine heat rejection was increased from zero to its maximum value of 400HP over a few minutes time. It was held constant at this maximum value thereafter for all three coolants studied, the base fluid and the two nanofluids.

As the engine heat rejection increased during the transient, so did the coolant temperature. In Flowmaster, the thermostat was set to open at a temperature of 82 C, and it was completely open at 92 C. The coolant temperature at the pump exit is shown in Figure 2 during the coolant heat up and beyond.

After the initial linear rise in coolant temperature shown in Figure 2, the thermostat started to open causing coolant to flow to the radiator, and the coolant temperature decreased enough to cause the thermostat to close. Subsequently, the coolant temperature rose, the thermostat opened, and the coolant temperature decreased. At least three times during this process, the coolant temperature decreased enough to cause the thermostat to close or nearly close causing the coolant temperature to rise sharply. It was this type of large coolant temperature gradient, due to thermostat position, that caused instabilities in the steady-state simulations attempted with Flowmaster. These conditions are seen in Figure 2 in the time period between about 40 and 150 s. However, with the transient simulation, the coolant temperature eventually reached a steady constant value as seen in Figure 2 at times beyond about 400 s.

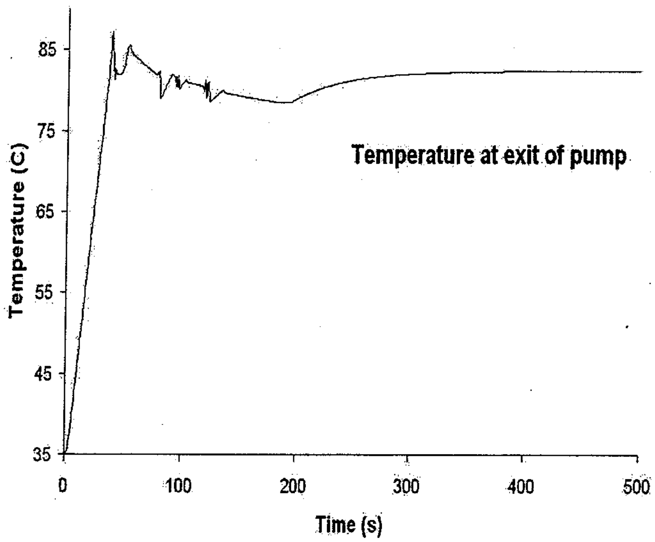

Figure 2. Coolant Temperature during Thermostat Opening

Two coolant–to–air heat exchangers were modeled in Flowmaster. The radiator was a cross-flow heat exchanger with its effectiveness specified as a function of the mass flow rates of both the air and coolant sides. The cabin heater was modeled similarly to the radiator. However, for the coolant heat load design point occurring in the summer, coolant flow to the cabin heater was set to zero for this study using the cabin heater flow control, component 18 in Figure 1.

The heat transfer rate of the radiator was improved when using nanofluid coolants because they have enhanced Nusselt numbers (and thus heat transfer coefficients) compared to the base fluid. This improvement was taken into consideration by incorporating the enhanced heat transfer coefficient for the nanofluid into the overall heat transfer coefficient for the heat exchanger. For the range of parameters of this study, a 40% Nusselt number enhancement for the nanofluid resulted in an increase in the overall heat transfer coefficient of the heat exchanger by a maximum of about 10%.

A similar increase in heat transfer rate was seen in the engine when using nanofluid coolants to replace the base fluid of a 50/50 mixture of ethylene-glycol and water. Here, the increased heat transfer coefficient of the nanofluid in the engine cooling passages was input directly in Flowmaster.

RESULTS

Flowmaster simulations were performed for a Class 8 truck engine reaching steady-state conditions at maximum heat load. The corresponding parameters were as follows.

1) 400 HP engine heat rejection to the coolant
2) Coolant pump design flowrate of 100 gpm at 10 psi (0.68 bar) pressure drop
3) Ambient temperature of 95 F (35 C) with 40 mph ram air entering radiator

Transient simulations were performed in all cases with interest in the steady-state solution for the system. A time step of 0.5 s was found to be near optimum, and the simulations were run for a total time of 500 s after which the system had reached steady state. A series of numerical trials determined the adequacy of these two parameters: 0.5 s step size and 500 s to steady state.

BASELINE CASE – A coolant system simulation was performed for the maximum heat load situation using a 50/50 mixture of ethylene-glycol and water as the coolant. Results are shown in Figures 3 – 5 for the steady-state mass flow rates, pressures and temperatures, respectively, of the coolant at various locations within the system.

It is seen in Figure 3 that essentially all of the coolant flow is through the radiator. The thermostat is fully open, and there is essentially no flow bypassing the radiator. There is no flow through the cabin heater under these summer conditions.

The coolant pressures were calculated in Flowmaster from the pump characteristic and the size and length of the various flow passages. It is seen in Figure 4 that the pump discharge pressure is 1.85 bar while the lowest pressure, at the radiator outlet to the pump suction, is slightly over atmospheric. These pressures are typical of the design point operating conditions.

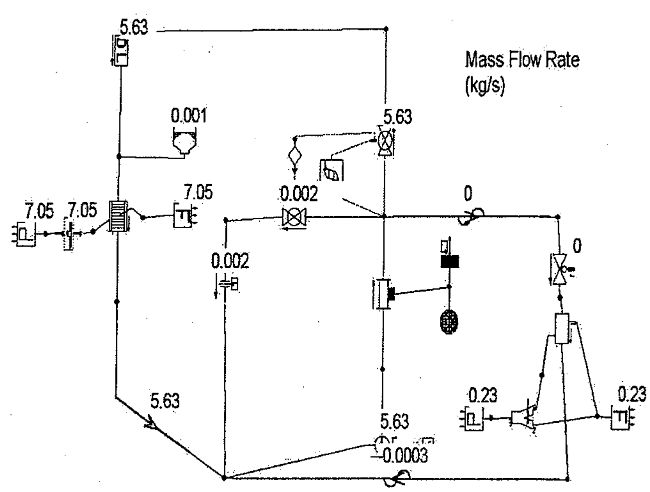

Figure 3. Coolant Flow Rates – Baseline Case

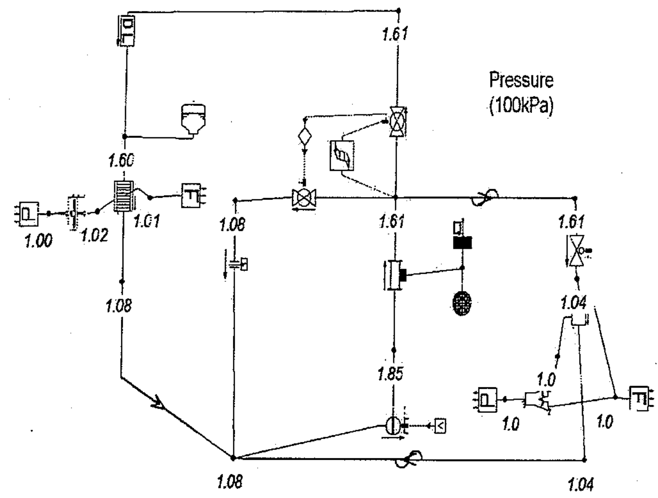

Figure 4. Coolant Pressures – Baseline Case

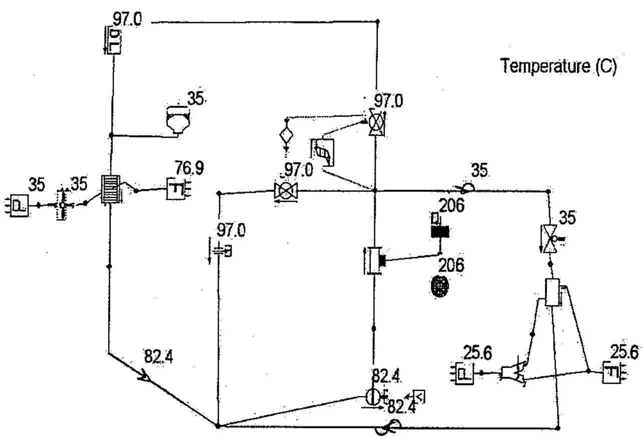

Figure 5. Coolant Temperatures – Baseline Case

The results of Figure 5 show the coolant temperatures at various positions in the coolant circuit. The maximum coolant temperature, as it exits the engine, is seen to be 97 C. The minimum temperature, exiting the radiator, is 82.4 C. (There is very little flow bypassing the radiator, and its temperature is the same as at the engine exit.)

For comparison purposes, an engine temperature of 206 C is shown in Figure 5. This temperature is calculated in Flowmaster as a lumped average block temperature. Although it has no direct physical meaning, it is useful when considering the effects of the various coolants on engine block temperature. Subsequent comparisons of engine temperatures refer to this average block temperature.

NANOFLUIDS IN BASELINE CASE– Two Flowmaster simulations were run with the same parameters as the baseline case with the exception of the coolant. In the first simulation, nanofluid-1 was created by the addition of a 2% concentration (by volume) of CuO nanoparticles to the baseline coolant of a 50/50 mixture of ethylene-glycol and water. In the second simulation, nanofluid-2 was created by the addition of a 4% concentration of CuO in the same base fluid. Results are compared in Table 2 among the two nanofluids and the base fluid.

Coolant	Maximum Coolant Temperature (C)	Mass Flow Rate through Radiator (kg/s)	Engine Block Temperature (C)
Base Fluid (50/50 ethylene glycol/water mixture)	97.0	5.6	206
Nanofluid-1 ($v_p = 2\%$ CuO in base fluid)	95.6	6.4	165
Nanofluid-2 ($v_p = 4\%$ CuO in base fluid)	94.5	7.2	141

Table 2 Baseline Case Results with Nanofluids

The most significant changes in the results using nanofluid coolants are the coolant and engine temperatures. In Table 2, the coolant mass flow rate is seen to increase as the nanoparticle concentration increases in the coolant, but this effect is mainly due to the increased density of the nanofluids, and the coolant volume flow rates through the system are not significantly affected. (The coolant pressures are not significantly affected by the nanofluids.) However, the engine block temperature is seen to decrease from 206 C with the base fluid alone to 141 C with the 4% nanofluid. (The 2% nanofluid produced an intermediate engine block temperature of 165 C.) This temperature reduction is due to the increase in the nanofluid heat transfer coefficient compared to the base fluid, and it allows for considerable flexibility in engine coolant passage design and local temperature control. A large reduction in engine block temperature need not be taken. Instead, redesign can lead to more uniform engine block temperatures, more effective local cooling, optimum engine block temperature, etc.

The same large heat transfer coefficient increase occurred in the radiator, but there the reduction in coolant temperature is less than the engine block temperature reduction, going from 97.0 C with the base fluid to 94.5 C with the 4% nanofluid. The coolant temperature in the radiator is less sensitive to the increased heat transfer coefficient than is the engine temperature because, in the radiator, the heat transfer resistance is dominated by the airside. However, the 2.5 C reduction in radiator coolant temperature is not insignificant owing to the large coolant flow rate – which is the-order-of 100 gpm.

OPTIONS WITH NANOFLUIDS - The results of Table 2 show that replacing the standard (baseline) coolant of a 50/50 mixture of ethylene-glycol and water with a nanofluid results in increased heat transfer rates both in the engine and in the radiator. With all other parameters unchanged, the reduced engine block and coolant temperatures occur per Table 2. However, the improved heat transfer rates of the nanofluids present several options for the truck cooling system to optimize efficiency and performance.

Three Flowmaster simulations were performed varying one parameter in each case with the objective of improving the performance of the system. In each case, the parameter was varied to keep the maximum coolant temperature approximately the same as that for the base fluid case (~ 97 C). The parameters studied were:

1) Increased engine heat rejection to the coolant circuit over the baseline case (400HP for the baseline),
2) Decreased coolant pump speed (reduced power consumption) compared to the baseline case (1600RPM for the baseline),
3) Reduced radiator air-side area compared to the baseline case (0.39 m^2 for the baseline).

In all three cases, maintaining the 97 C maximum coolant temperature resulted in engine block temperatures below the baseline level of 206 C. These levels using the nanofluids did not present high temperature problems for engine materials and, as discussed previously, could be raised and optimized by engine coolant passage re-design.

Increased engine heat rejection – Numerical simulations were performed with nanofluid-1 and nanofluid-2 using the baseline case parameters with the exception of the engine heat rejection to the coolant which was allowed to vary. The heat rejection was increased to raise the maximum coolant temperature to approximately equal the baseline case, ~ 97 C. Results are presented in Table 3.

It is seen in Table 3 that the engine heat rejection increased from 400HP for the base fluid alone to 410HP (305.9kW) for the 2% nanofluid and 420HP (313.3 kW) for the 4% nanofluid. The latter amounts to a 5% increase in engine horsepower rejected to the coolant using the nanofluid compared to the standard (baseline) coolant. (The coolant flow rates and pressures were not significantly changed due to the increased engine heat rejection.) This result introduces the option for increased engine power in the Class 8 truck using the same (baseline case) cooling system and a nanofluid coolant.

Coolant	Base Fluid (50/50 ethylene glycol/water mixture)	Nanofluid-1 ($v_p = 2\%$ CuO in base fluid)	Nanofluid-2 ($v_p = 4\%$ CuO in base fluid)
Engine Heat Rejection to Coolant Circuit	400 HP	410 HP	420 HP

Table 3 Increased Engine Heat Rejection with Nanofluids

Decreased coolant pump speed - Numerical simulations were performed with nanofluid-1 and nanofluid-2 using the baseline case parameters with the exception of the coolant pump speed which was allowed to vary. The pump speed was decreased to raise the maximum coolant temperature to approximately equal the baseline case of ~ 97 C. Results are presented in Table 4.

It is seen in Table 4 that the nanofluid coolants allow a reduction in pump speed, up to a factor of 2, with the same 400HP heat rejection rate as the baseline case. In the nanofluid simulations of Table 4, the reductions in pump speed resulted in reduced nanofluid heat transfer coefficients allowing the maximum coolant temperature to increase to the baseline level of ~ 97 C. This reduction in pump speed resulted in dramatic reductions in pump power, as much as 88% with nanofluid-2, compared to the base fluid alone. This result introduces the option for a reduction in parasitic power consumption in a Class 8 truck.

Coolant	Base Fluid (50/50 ethylene glycol/water mixture)	Nanofluid-1 ($v_p = 2\%$ CuO in base fluid)	Nanofluid-2 ($v_p = 4\%$ CuO in base fluid)
Pump Speed	1600 RPM	1150 RPM	800 RPM
Pump Power	0.75 HP (0.56 kW)	0.266 HP (0.2 kW)	0.09 HP (0.0675 kW)

Table 4 Reduced Pump Speed with Nanofluids

Reduced radiator air-side area - Numerical simulations were performed with nanofluid-1 and nanofluid-2 using the baseline case parameters with the exception of the radiator air-side area which was allowed to vary. Because of the increased heat transfer coefficient of the nanofluids compared to the base fluid alone, it was possible to reduce the surface area of the air-side of the radiator while transferring the same heat until the maximum nanofluid temperature reached the baseline case maximum of ~ 97 C. Reduction in radiator size can be used to reduce aerodynamic drag of the truck increasing fuel efficiency. Results are presented in Table 5.

The results of Table 5 show that the radiator air-side surface area can be reduced to 0.37m^2 from the baseline value of 0.39m^2 using nanofluid-2. This reduction of 5% in area can be used to increase fuel economy by reducing aerodynamic drag if the front end of the tractor is modified accordingly. For a 5% area reduction resulting in a 5% decrease in aerodynamic drag coefficient, typically half of that amount, or 2.5%, is achievable as a fuel consumption reduction [11]. This reduction is considered significant for Class 8 trucks with their relatively high fuel consumption rates accounting for 11-12% of the total US petroleum usage [11].

Coolant	Base Fluid (50/50 ethylene glycol/water mixture)	Nanofluid-1 ($v_p = 2\%$ CuO in base fluid)	Nanofluid-2 ($v_p = 4\%$ CuO in base fluid)
Radiator Air-Side Area (m^2)	0.39	0.38	0.37

Table 5 Reduced Radiator Air-Side Area with Nanofluids

The three cooling system parameters investigated, for improvement/optimization due to the use of the 4% nanofluid coolant, all showed significant improvements. The improvements were in the areas of engine power and energy efficiency, and combinations of these parameter variations are possible as are other parametric variations.

CONCLUSIONS

Comparisons were made among the results of numerical simulations replacing the standard Class 8 truck coolant, a 50/50 mixture of ethylene-glycol and water, with nanofluids comprised of CuO nanoparticles suspended in a base fluid of a 50/50 mixture of ethylene-glycol and water. The high heat transfer coefficients of the nanofluids compared to the standard fluid resulted in decreased engine and coolant temperatures (with the engine block temperatures being reduced the most) with all cooling system parameters unchanged. This situation in the engine introduces flexibility in engine cooling system design to: achieve more uniform block temperatures, improve localized cooling, optimize block temperature, etc. The nanofluid coolant in the radiator allows for the optimization of coolant system parameters. Three of these parameters were simulated numerically. Quantitative results showed increased engine horsepower up to 5%, decreased coolant pump speed and power up to 88%, or decreased radiator air-side surface area up to 5% leading to reductions in aerodynamic drag and fuel consumption.

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REFERENCES

1. Choi, S. U. S., "Enhancing Thermal Conductivity of Fluids with Nanoparticles," Developments and Applications of Non-newtonian Flows, eds. D. A. Singer and H. P. Wang, vol. FED 231, pp. 99-105, American Society of Mechanical Engineers, New York, 1995.
2. Eastman, J. A., "Measuring Thermal Conductivity of Fluids Containing Oxide Nanoparticles," Journal of Heat Transfer, vol. 121, pp. 280-289, American Society of Mechanical Engineers, New York, 1999.
3. Flowmaster, Inc., www.Flowmaster.com
4. Choi, S. U. S., Z. G. Zhang, and P. Keblinski, "Nanofluids," Encyclopedia of Nanoscience and Nanotechnology, ed. H. S. Nalwa, vol. 6, pp. 757-737, American Scientific Publishers, Los Angles, California, 2004.
5. Eastman, J. A., S.R. Phillpot, S. U. S. Choi, and P. Keblinski, "Thermal Transport in Nanofluids," Annual Reviews in Material Research, vol. 34, pp. 219-246, 2004.
6. Keblinski, P., J. A. Eastman, and D. G. Cahill, "Nanofluids for Thermal Transport," Materials Today, pp. 36-44, June 2005.
7. Das, S. K., S. U. S. Choi, and H. E. Patel, "Heat Transfer in Nanofluids – A Review," Heat Transfer Engineering, vol. 27, no. 10, pp. 3-19, 2006.
8. Yu, W., D. M. France, J. L. Routbort, and S. U. S. Choi, "Review and Comparison of Nanofluid Thermal Conductivity and Heat Transfer Enhancements," Heat Transfer Engineering, accepted for publication.
9. Das, S. K., "Nanofluids – The Cooling Medium of the Future," Heat Transfer Engineering, vol. 27, no. 10, pp. 1-2, 2006.
10. Einstein, A., "Eine neue Bestimmung der Moleküldimensionen," Annalen der Physik, vol. 19, pp. 289-306, 1906.
11. Heavy Vehicle Systems Optimization Program, 2005 Annual Progress Report: I.A. and I.E., www.eere.energy.gov/vehiclesandfuels/resources/fcvt_reports.html, January 2006.