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Subcooled flow boiling of ethylene glycol/water mixtures in a bottom-heated tube

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A B S T R A C T

Coolant subcooled boiling in the cylinder head regions of heavy-duty vehicle engines is unavoidable at high thermal loads due to high metal temperatures. However, theoretical and experimental studies of coolant boiling under these specific application conditions are generally lacking. In the present study, subcooled flow boiling heat transfer experiments were performed with water and ethylene glycol/water mixtures at volume ratios of 40/60 and 50/50 in turbulent flow in a specifically designed and fabricated test facility with its experimental test section simulating the heating conditions of the coolant channels in the cylinder head regions of heavy-duty vehicle engines. Boiling curves and subcooled flow boiling heat transfer coefficients for the tested fluids were determined based on the experimental results. Comparisons between the experimental data and the predicted values from existing correlation equations in the engineering literature are presented.

1. Introduction

Currently, the engine cooling systems in heavy-duty vehicles are designed to use approximately 50/50 ethylene glycol/water (EG/W) mixture in the liquid state. Boiling is usually a phenomenon that has been avoided in conventional engine cooling systems in heavy-duty vehicles. However, while the conventional engine cooling systems in heavy-duty vehicles are designed to eliminate coolant saturation boiling, coolant subcooled boiling in the cylinder head regions is unavoidable at high thermal loads due to high metal temperatures. Because of its order-of-magnitude higher heat transfer rates, there is interest currently in using controllable nucleate-boiling precision cooling instead of conventional single-phase forced convection cooling in vehicle cooling systems under certain conditions or in certain areas to remove ever increasing heat loads, to eliminate potential hot spots in engines, or to further optimize the parasitic losses of the coolant pump [1–3]. Theoretical, numerical, and experimental investigations have been conducted on the potentials and the practical applications of nucleate boiling cooling systems in heavy-duty vehicles [4–18]. Consequently, there is great interest in the knowledge of flow boiling heat transfer rates and limitations under these application conditions.

One of the unique characteristics of coolant subcooled boiling in the cylinder head regions of heavy-duty vehicle engines is that boiling generally occurs only on the cooling channel side facing the flame plate because of the one-sided heating condition. Although there have been many investigations into subcooled flow boiling with one-sided heating, most of the effort was focused on very high heat flux fusion reactor system cooling with water as the coolant. See for example Refs. [19,20]. Despite its importance in practical applications, theoretical and experimental studies of coolant boiling under engine cooling application conditions are generally lacking. Among the few studies conducted, Steiner et al. [21] experimentally investigated subcooled flow boiling in a rectangular channel with one-sided heating. Based on the experimental data, they developed a wall heat transfer model by changing the modification factor for the nucleate boiling component in

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Nomenclature

A	area (m^2)
d_i	inside diameter (m)
d_o	outside diameter (m)
E	voltage drop (V)
h	heat transfer coefficient (W/m^2 K)
I	current (A)
k	thermal conductivity (W/m K)
L	length (m)
p	pressure (Pa)
Pr	Prandtl number
Q	volumetric flow rate (m^3/s)
$\dot{q}$	heat flow rate (W)
$\dot{q}''$	heat flux (W/m^2)
Re	Reynolds number
T	temperature (°C)
Greek symbols	
ΔT_{sat}	wall superheat $\Delta T_{sat} = T_w - T_{sat}$ (°C)
ΔT_{sub}	liquid subcooling $\Delta T_{sub} = T_{sat} - T_f$ (°C)
Subscripts	
amb	ambient
b	boiling
bot	bottom
f	fluid
in	inlet
l	liquid
loss	loss
M	mixture
out	outlet
sat	saturation
sub	liquid subcooling
top	top
tot	total
W	water
w	wall

the Chen correlation [22] and by introducing two suppression factors accounting for the effects of the flow forces and the subcooling of the thermal boundary layer. Because it requires only local input quantities, the model is well suited to computational fluid dynamic simulations of geometrically very complex coolant flows where the definition of global length or velocity scales would be impractical. The correlation predicts experimental data well in the partially developed boiling region but with notable deviations in the vicinity of the fully developed boiling region. Kroes et al. [23] assessed several correlations for nucleate subcooled boiling heat transfer in engine cylinder head cooling ducts using one-dimensional modeling and compared the results with four experimental data sets showing that the Chen correlation [22] predicted the measurements best, in particular for the two data sets with heat fluxes in the range of those in a diesel engine.

For applications to heavy-duty vehicle engines, it is seen from the above discussion that the subcooled boiling investigations of one-sided heating lack the use of the proper coolant (a 50/50 EG/W mixture) and the geometry of the engine cylinder head region. Especially important are the coolant subcooled flow boiling characteristics and the corresponding heat transfer coefficients. The objective of this study was to provide such information through experimental investigations of subcooled flow boiling of water and EG/W mixtures on a specifically designed and fabricated test facility with its experimental test section simulating the heating conditions of the coolant channels in the cylinder head regions of heavy-duty vehicle engines.

2. Experimental apparatus

The experimental apparatus used in this study is shown in Fig. 1. It consists of a closed-loop system with main components of a pump, two preheaters, an experimental test section, a heat exchanger (cooler), and a flowmeter. The system was designed and fabricated to study the heat transfer of subcooled flow boiling of water and EG/W mixtures with heat supplied only to the bottom half surface of the experimental test section.

As shown in the schematic diagram of the experimental apparatus in Fig. 1, the test fluid was pumped through the test loop by a turbine pump and the system was open to the atmosphere through the fill port at the flowmeter. The turbine pump was driven by an alternating-current adjustable-frequency driver, which made it possible to fine adjust flowrates through the experimental test section. Exiting the pump, the test fluid flowed through two preheaters arranged in series, in which, for a given test, the fluid temperature was raised to the desired subcooled level and monitored through two in-stream thermocouples ($T_{preheater1}$ and $T_{preheater2}$). Each preheater was heated by passing current through its wall using a direct-current power supply. As a safety precaution for protecting the preheaters from overheating, each preheater was provided with a temperature interlock. At the end of each preheater, the wall temperature ($T_{interlock}$) was measured and then fed to a high-temperature limit switch that would terminate power to the preheater when a preset upper-temperature limit was reached. After passing through the preheaters, the fluid entered the horizontal experimental test section.

The selection of the experimental test section heating method used in this study was based on the combined considerations of measurement accuracy, control precision, and thermal insulation. The experimental test section was heated with a direct-current power supply by passing current through a 1.6256-mm diameter AISI type 304 stainless steel heating wire attached to the bottom half of the experimental test section surface shown schematically in Fig. 2. The output power could be regulated from 0 to 18 kW with the maximum voltage drop and the maximum current of 40 volts and 450 amperes, respectively. The voltage drop across the heating wire (E) was measured directly, and the current through the heating wire (I) was determined from a measurement of the voltage drop across a shunt resistor. The heat input to the experimental test section was calculated using the product of the voltage drop and the current. Electrical isolation for eliminating ground loops was provided for the preheaters and the experimental test section by short high-pressure hoses, designated ISO in Fig. 1. The test fluid out from the experimental test section was cooled in the compact plate-and-frame heat exchanger that used laboratory water as a heat rejection fluid. The volumetric flowrate (Q) of the test fluid was measured by an electromagnetic flowmeter. A thermocouple probe ($T_{flowmeter}$) just upstream from the flowmeter provided a means to determine the density of the fluid and subsequently the mass flowrate of the fluid. Flowing out of the flowmeter, the test fluid returned to the pumps to close the test loop.

The experimental test section, shown schematically in Fig. 1, was fabricated from a 10.9-mm-inside-diameter (d_i) and 12.7-