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Thermal performance of reciprocating two-phase thermosyphon with nozzle

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A R T I C L E I N F O

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A B S T R A C T

Flow images and thermal performances of the two-phase reciprocating thermosyphons (RT) with an internal nozzle at 50% volumetric filling ratio (FR) were measured. Two sets of flow snapshots and thermal performance data detected from two similar RTs with and without nozzle were compared. Above the critical reciprocating frequencies (f_{cr}), the circulation routes of working fluid in the RTs with and without nozzle were reversed to cause disparities in thermal performances. At the reciprocating frequencies (f) of 1.75, 1.83, 1.92 and 2 Hz with four different heating and cooling duties for each fixed f, the time-mean local and regionally averaged Nusselt numbers (Nu) along the evaporator/condenser centerlines of the RTs with and without nozzle were dominated by reciprocation number (Re_{ci}) but subject to the coupled effects attributed from the heating and cooling duties indexed by boiling number (Bo) and dimensionless condenser thermal resistance ($R_{th,con}$). Local Nu were raised by increasing Re_{ci}, Bo and/or $R_{th,con}$ for both RTs with and without nozzle. Due to the inversed liquid film circulation and jet impingement, the overall thermal resistances (R_{th}) of present RT with nozzle were reduced by 40.06–59.96% from those of RT without nozzle. To add engineering applications, evaporator Nu correlations were devised for present RT with nozzle.

1. Introduction

The capability to consume residual oil for fuel economy has made diesel engines as the main stream of prime movers for merchandized ships and on-board electricity generation. The developments of diesel engine for meeting the energy saving requirements and the emission standards have led to the increased power density with the ever mounting operating temperatures of the hot components of combustion chamber. Among these hot components, the reciprocating piston is prone to thermal damages and the engine manufacturers are constrained by piston crown temperature limits for future engine advancements. The heat transferred from hot working gas to coolant in a piston not only affect material temperatures and the combustion process but also put forth the consequent impacts on fuel consumption and exhaust emission. To serve effective and efficient piston cooling, the various cooling schemes including oil gallery, piston cooling jet, shaker-bore and shaker-jet were devised for land and marine applications.

A piston cooling system has to enable the controls of material temperatures as well as the thermal stresses and deformations within the prescribed limits to ensure the lifespan/integrity of the piston. Knowledge of heat transfer coefficients on the thermal boundaries of a moving piston plays a decisive role for acquiring reliable piston temperature estimations. The heat flux to an engine piston crown and the material temperatures were numerically analyzed [1]. The use of transient boundary condition did not noticeably affect the results of piston thermal analysis; whereas the use of time varying piston temperatures during an engine cycle exhibited negligible influences on the results of combustion analysis [1]. Alternatively, the adoption of spatial and time averaged combustion side boundary condition was effective. An inverse scheme considering the effects of oil film thickness, piston and piston ring secondary motion for estimating the convective heat transfer rates on the thermal boundaries of an articulated piston was proposed to predict the temperature distributions of a piston [2]. Using few measured wall temperature data, this inverse scheme permitted the design of modified piston for moderating the temperature gradients and hence the thermal loads [2]. Without sufficient piston cooling, the development of hot spots on piston crown enhanced local carbon depositions to form thermal barriers and accelerate the carbonization process on piston crown. Such unfavorable carbonization could cause pre-ignition that undermined the combustion performance. Depending on the temperature distributions over the piston crown, an increase of piston temperature from 189 to 227 °C has led to the reduction of unburned hydrocarbon emissions, the increase of smoke opacity but no change of oxides of nitrogen emissions [3]. Once the temperatures at the underside of a piston exceeded the boiling limit of oil, the oil mist was generated upon jet impingement; which contributed to the additional non-tailpipe emissions in the form of un-burnt hydrocarbon [4]. The subsequent computational study for temperature predictions of

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Nomenclature

English symbols

A	total heating area of evaporator (m^2)
a, b, a_s, b_s	correlative coefficients
Bo	boiling number = $QCp_L/(h_{fg}k_L d)$
Cp_L	constant pressure specific of liquid coolant ($Jkg^{-1}K^{-1}$)
d	hydraulic diameter of thermosyphon (m)
d_j	exit diameter of jet nozzle (m)
f	reciprocating frequency (Hz)
f_{CR}	critical reciprocating frequency (Hz)
g	gravitational acceleration = $9.81(mS^{-1})$
h	boiling convective heat transfer coefficient = $Q/[A \times (T_w - T_f)]$
h_{fg}	latent heat of test coolant at measured pressure (Jkg^{-1})
k_L	thermal conductivity of coolant (liquid) ($Wm^{-1}K^{-1}$)
Nu	Nusselt number = $q_f d/[(T_w - T_f)k_L]$
$\overline{Nu}_{eva}$	evaporator centerline averaged Nusselt number = $q_f d/[(\overline{T_w} - T_f)k_L]$
P	pressure of thermosyphon (Nm^{-2})
Q	convective heater power (W)
q_f	convective heat flux (Wm^{-2})
R	rotating radius of crank-wheel mechanism (m)
Re_{ci}	reciprocation number = $2\Omega^2 R/g$
R_{th}	total thermal resistance = $[(\overline{T_w} - T_\infty)/Q]k_L d$
$R_{th,con}$	condenser thermal resistance = $[(T_{f,con} - T_\infty)/Q]k_L d$
T	cycle period of reciprocation (s)
T_f	fluid temperature in thermosyphon (K)
T_{sat}	saturated temperature at test pressure (K)
T_w	local wall temperature (K)
$\overline{T_w}$	centerline averaged evaporator wall temperature (K)
T_∞	cooling airflow temperature (K)
x	centerline coordinate (m)

Greek symbol

θ	crank angle of flywheel (degree)
Ω	angular velocity of flywheel (s^{-1})

Subscripts

con	condensation section
eve	evaporation section
RT	reciprocating thermosyphon

piston under oil jet cooling was performed with the heat transfer coefficient predicted by Steven-Webb relation [5]. At the heat flux of 45 kW/m^2 applied to the piston, the oil jet cooling scheme reduced piston temperatures by 35–40 °C. The period required for reaching steady state temperatures of the piston was lesser at the higher thermal load and further reduced after employing oil jet cooling. Oil-mist generation underneath the piston was initiated when the surface temperature became above 250 °C. As the surface temperature reached 300 °C, a large amount of oil mist/smoke was observed owing to the localized boiling activities [5].

Without taking the advantage of latent heat transmission, the oil jet with gallery is a common strategy for piston cooling. But the thermal fluid phenomena in a partially filled oil gallery were subject to the cocktail shaker effect. In Ref. [6], an oil gallery cooling model that considered the cocktail shaker effect was developed to predict the average heat transfer coefficient in the oil gallery. The distribution of piston temperatures was noticeably affected by the geometrical configurations of the oil cooling jet and the oil flows in gallery. In comparison with the air-cooled piston, the use of oil gallery cooling could further reduce the surface temperature by as much as 300 K. In view of piston surface temperatures, the convective boundary condition under the piston exhibited the higher degree of impact than that caused by the in-cylinder thermal radiation [6]. The heat flow through the oil gallery in a piston was experimentally studied [7] to highlight the effects of Reynolds number, Prandtl number and the relative position between the piston and the oil cooling nozzle on the thermal responses. Optimization of the location for oil gallery in the piston crown was performed with the considerations of material temperature/stress and the fatigue strength of combustion chamber and oil gallery. By lifting the oil gallery upward, the temperatures of bowl rim, top land, first ring groove, second land and gallery were decreased; but the fatigue factor of the oil cooling gallery was decreased [8]. To attack the forced convective piston cooling performance in a partially filled reciprocating oil gallery, the VOF (volume of fluid) multiphase model was deployed to simulate the flow structures and heat transfer properties in a piston cooling gallery [9]. Having applied the predicted transient oil volume fraction and the heat transfer distribution as the boundary conditions for the cooling gallery, the cooling responses on engine speed, diameter of entry port and wall temperature level were reported. While the oil volume fraction increased from the entry port toward the exit port, the heat transfer rates declined along the flow pathway in the gallery. By reducing engine speed to boost the oil filling ratio, the higher heat transfer rates and lower oil exit temperature were obtained. As the entry port shrunk, the oil fill ratio was decreased to undermine the convective heat transfer rates with elevated oil exit temperatures [9]. The instant air-water flow snapshots detected at various crank angles from a reciprocating horizontal gallery with the water filling ratios of 20, 40, 60 or 80% at the reciprocating frequencies of 6, 8, 10, 12 or 14 Hz were analyzed for piston cooling applications [10]. The stagnant liquid film covering the surfaces of the reciprocating gallery oscillated with reduced thickness due to the periodic detachment of water droplets to promote the near-wall heat transmission. As the oscillation frequency increased, the air-water flow inside the partially filled water gallery became more chaotic to boost the fluids mixings for heat transfer enhancements. In view of heat transfer effectiveness, the optimal filling ratios emerged between 40% and 60%, above which the free liquid movements by the "cocktail shaking" effect were restricted; while the lesser water contents reduced the liquid coverage on the reciprocating surfaces [10]. The follow-up work adopted Eulerian multiphase model with geometry reconstruction scheme for disclosing the heat transfer mechanism of reciprocating two-phase flow through the piston cooling gallery [11]. Impacts of engine speed and oil entry velocity on heat transfer properties were examined. As engine speed increased, the heat transfer rates were increased with the distributions evolved toward a similar pattern. The optimal oil entry velocities at various engine speeds were identified [11].

For cooling of medium/high speed pistons, an impinging jet of subcooled lubricant oil normal to the bottom surface of a piston was generally employed [4,5,12,13]. A numerical study that coupled the three-dimensional fluid and solid domains involving two-phase oil-air flows investigated the effects of piston cooling jet (PCJ) on the temperature and heat transfer distributions of a piston [12]. With the PCJ system, the average of piston crown temperature was reduced to about 70 K, but incurred 50 K temperature gradient in piston [12]. The conjugate heat transfer method which coupled the heat transfer solutions between the solid and fluid regions for predicting the heat transfer rates at the piston walls and subsequently the temperature distributions in the piston was performed numerically [13]. While the cooling jet significantly reduced piston temperatures, the incidence of maximum heat transfer rate was shifted away from the stagnation point as the nozzle size increased