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Preprint

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SURFACE TEMPERATURE EFFECT ON CONVECTIVE HEAT TRANSFER COEFFICIENTS FOR JET IMPINGEMENT COOLING OF ELECTRIC MACHINES WITH AUTOMATIC TRANSMISSION FLUID

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ABSTRACT

In this study, the results of NREL's continued work on experimental characterization of the thermal performance of free-surface jets of automatic transmission fluid impinged on a heated target surface are presented. The measured heat transfer coefficients are useful for understanding factors influencing performance of driveline fluid-based cooling systems for electric machines and help designers in developing high-performance, power-dense and reliable machines. Experiments were carried out for different fluid and target surface temperatures (50°C, 70°C, and 90°C for the fluid and 90°C, 100°C, 110°C, and 120°C for the target surface). Impinging jet velocities (0.5 m/s to 7.5 m/s) and the jet position on the target surface (center versus edge) were also varied. The impinging angle was kept at 90° relative to the target surface. It was found that higher target surface temperature increased heat transfer coefficients, namely, increasing surface temperature from 90°C to 120°C enhanced heat transfer coefficient values at higher impinged jet velocities (7.5 m/s) by up to 15%.

Keywords: Jet impingement, automatic transmission fluid, electric machines, electric motors, thermal management

NOMENCLATURE

A	Target impingement surface area
h	Heat transfer coefficient
k	Thermal conductivity
d_1	Distance between two thermocouples embedded in the target
d_2	Distance from upper thermocouple to target's top (cooled) surface
d_o	Nozzle orifice diameter
D	Target impingement surface diameter
Q	Heat dissipated through target's top surface
S	Distance between nozzle and target surface
T_{fluid}	Fluid temperature at nozzle inlet
$T_{surface}$	Target surface temperature
T_{lower}	Lower thermocouple temperature
T_{upper}	Upper thermocouple temperature

INTRODUCTION

Thermal management of electric machines (electric motors and generators) is becoming increasingly important as the transition to vehicles with fully electric propulsion systems in the automotive industry picks up pace. With increasing power density of electric traction drives without sacrificing performance or reliability, the challenges associated with thermal management for electric machines increase. Appropriate thermal design is as important as electromagnetic and mechanical design of the electric machine [1]. In addition to optimizing the passive thermal design (geometric layout of components, material selection, thermal interfaces affecting the heat spreading capabilities within the electric machine) for high-power electric motor or generator, it is necessary to actively remove heat from key components of the machine. The convective cooling mechanism transfers heat from those components to an intermediary fluid and ultimately rejects the heat to the ambient environment. Cooling of the electric machine in a vehicle can be accomplished by circulating coolant, such as water-ethylene glycol (WEG), through the stator case or impinging automatic transmission fluid (ATF) jets onto the machine's copper windings. The latter cooling method is advantageous because the windings are typically the most temperature-sensitive component in the machine. Also, because in many cases the electric machines are housed within the vehicle's transmission or transaxle where ATF is readily available, no additional cooling fluid is required. Besides, ATF is a dielectric and can be in direct contact with electrically active machine components.

Direct cooling of end-windings

A very effective method for removing heat from the electric machine is directly cooling the end-windings of a stator with dielectric fluid impinged on the windings. The winding materials such as insulation are an example of temperature-sensitive components in the machine. In this study, ATF is used and evaluated as a cooling medium. Thermal (heat capacity) and

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physical (highly temperature-dependent viscosity) properties of ATF are inferior to those of WEG, which is generally used in water jacket cooling incorporated in the stator casing. However, ATF is readily available within a vehicle's transmission where the electric machine is often housed, making it an attractive single cooling and lubricating fluid option for the electric vehicle traction drive designers and manufacturers. Besides the ability to directly cool and remove heat generated within the windings, direct cooling with ATF avoids the conduction path thermal resistance through the passive stack of other materials present in a water jacket WEG cooling scenario [2].

Oil cooling performance

Understanding and characterizing heat transfer using oil jets, specifically ATF jet impingement cooling are critical for automotive original equipment manufacturers. There is little publicly available information on this topic. However, some limited data on thermal performance of viscous, high-Prandtl-number fluids (e.g., oils) and their applications for cooling electric machines, internal combustion engine pistons and transformer windings have been reported in the literature [3–14]. A detailed overview of the literature with relevant experimental and theoretical work is provided in the National Renewable Energy Laboratory's (NREL's) previous publication [15] on this topic.

The current work is a continuation of NREL's efforts for characterization of thermal performance of free-surface jet impingement cooling of electric machines with ATF [2, 15]. Previous work focused on the influence of target surface topography and physical enhancements on the heat transfer performance, where four different target samples were evaluated [15]. Baseline (flat surface) sample performance was also compared against results obtained from literature correlations and was found to be within the range of those correlations, although clearly diverging from correlation predictions towards the higher flow velocities. During recent experiments, it was found that the target surface temperature significantly affects convective heat transfer coefficients.

EXPERIMENTAL SETUP AND PROCEDURES

Fluid test loop

Heat transfer measurements were conducted using a fluid test loop designed for characterization of forced convective thermal performance of fluids. A general view of the experimental setup can be seen in *Figure 1*.

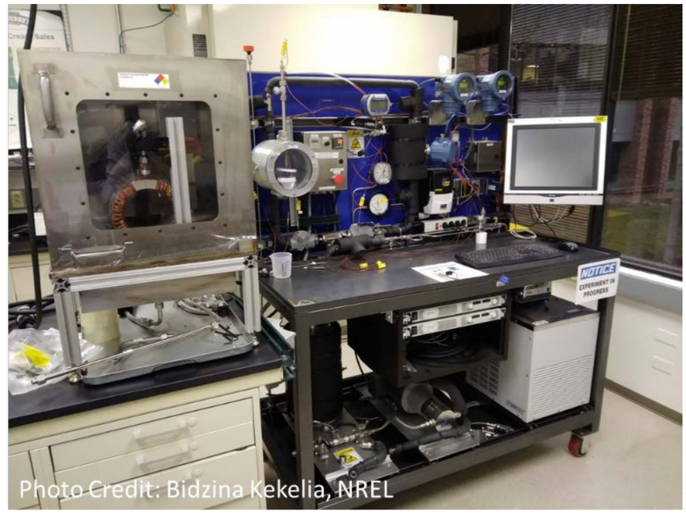

Figure 1. General view of experimental fluid test loop.

The test loop can be configured for jet impingement or channel flow experiments and can accommodate a variety of test articles: from small heated targets to entire electric machines. A schematic of the experimental test loop is provided in Figure 2.

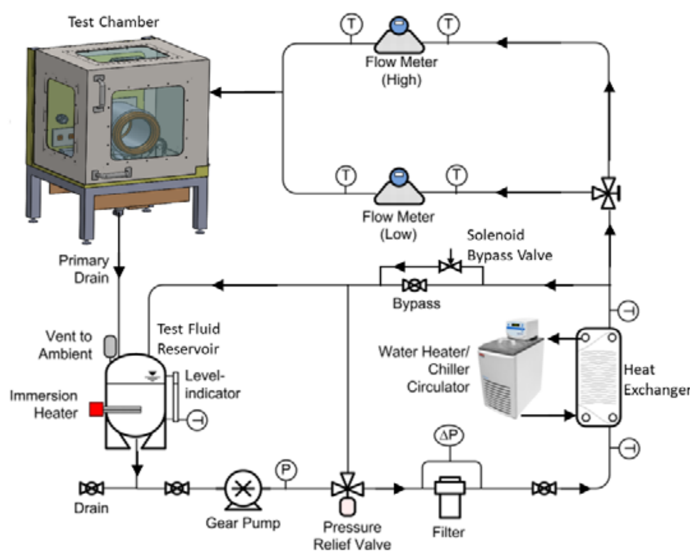

Figure 2. Fluid test loop schematic.

Ford Motor Company's Mercon LV automatic transmission fluid was evaluated in this study because of its use in hybrid electric vehicles. Ford provided thermal and physical properties of the fluid. The fluid is circulated through the loop via a variable-frequency-drive-controlled gear pump. System fluid temperatures are controlled and held constant using a heater/chiller bath circulator and flat-plate heat exchanger system as well as an immersion heater located within the reservoir tank. Two Coriolis mass flow meters (0 – 2 lpm and 2 – 20 lpm) measure fluid-flow rates. Temperatures within the loop are measured with calibrated (±0.09°C) K-type thermocouples.

Jet impingement test section and heated target

The ATF jet impingement experiments were conducted in a test chamber containing an electric machine with exposed end-windings and heated target assembly (*Figure 3*a) mounted within a sectional cutout in the end-windings (*Figure 4*). The electric machine stator is used for holding and positioning (rotating) the heated target assembly under vertically impinging fluid jet.

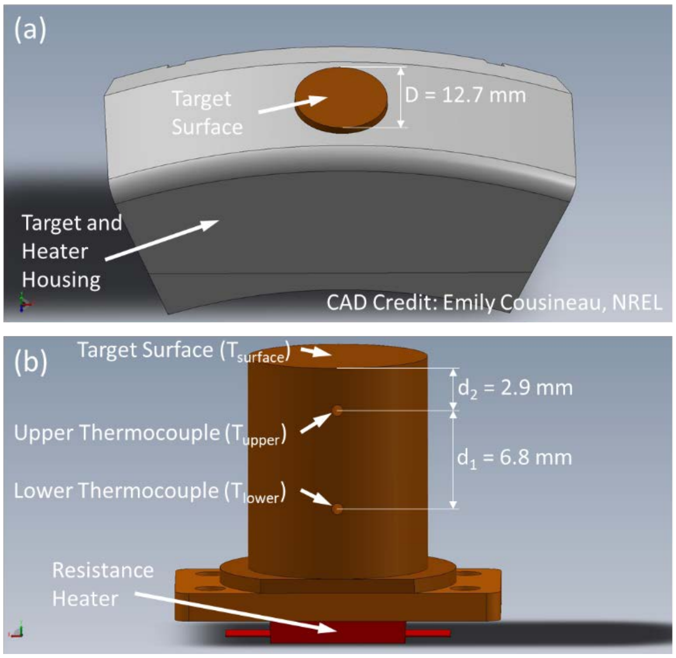

Figure 3. Heated target assembly: (a) assembled housing with exposed target resembling a section of electric machine end-windings, (b) copper target with thermocouple locations and resistance heater mounted on the bottom.

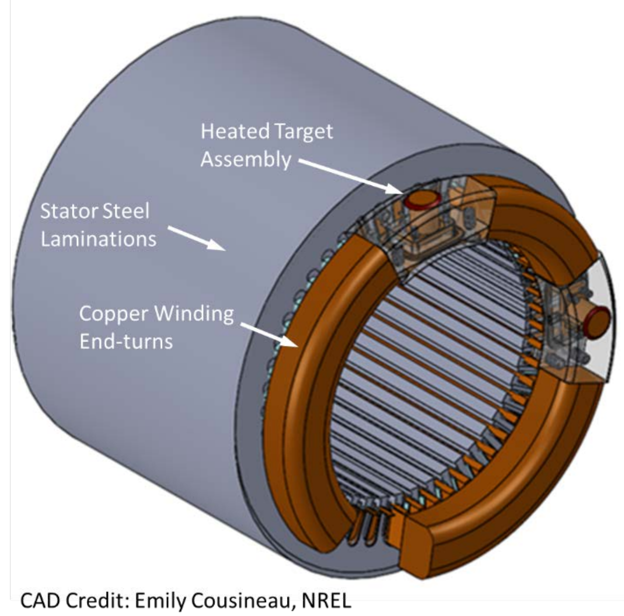

Figure 4. Electric machine stator with mounted heated target assembly within the end-windings cutout.

The target was fabricated from oxygen-free copper with a diameter of D = 12.7 mm and 600-grit sandpaper-polished impingement (cooled) surface (*Figure 3*b). A resistance heater was attached to the bottom of the copper target and enclosed in a polyetheretherketone (PEEK) housing for support and thermal insulation. Two calibrated (±0.09°C) K-type thermocouples were embedded within the target in order to measure heat fluxes from the heater towards the cooled top surface. Copper wires were epoxied on the top surface of the target housing (adjacent to the exposed target surface) to replicate the texture of end windings and directing fluid towards other windings after flowing over the heated target.

An orifice nozzle with d_o = 2.06 mm diameter was positioned at S = 10 mm distance above the circular target and aimed at a 90° angle towards the flat target surface. Fluid temperature (T_{fluid}) was measured with a calibrated (±0.09°C) K-type thermocouple installed at the inlet of the nozzle. *Figure 5* shows the test section with ATF jet impinging on the heated target. A pressure drop across the nozzle was not recorded as the data for the same orifice nozzle and the same ATF flow rates (velocities) was already reported in NREL's previous publication [15].

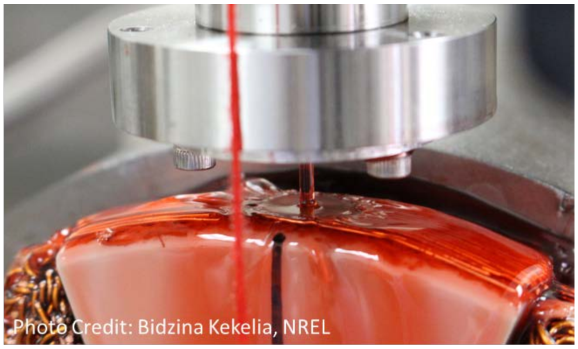

Figure 5. ATF jet impinging on a heated target surface.

Experimental procedures

The experiments were conducted at 50°C, 70°C, and 90°C fluid temperatures (as measured at the inlet of the nozzle). The test fluid was pre-heated with a direct immersion heater and then the desired fluid temperature was maintained with a silicon bath water heater/chiller through a flat heat exchanger (see *Figure 2*). The flow rate was controlled by adjusting manual and solenoid bypass valves. The target surface temperature for each experiment was set sequentially at 90°C, 100°C, 110°C, and 120°C. The surface temperatures were achieved by controlling a direct-current power supply connected to the resistance heater on the bottom of the heated target. The experiments were monitored and controlled with a LabVIEW program via National Instrument's CompactDAQ data acquisition system.

After reaching steady temperature and flow conditions, data were recorded (45 data points at 2-sec intervals) for each test case and repeated a minimum of three times. The heat transfer coefficient values were calculated for each data point and recorded by the LabVIEW control program. Average heat transfer coefficients ($\bar{h}$) for the target surface were defined according to *Equation 1*:

$$\bar{h} = \frac{Q}{A(T_{surface} - T_{fluid})} \quad (1)$$

where Q is the heat dissipated through the top surface (with surface area of A) of the target, T_{surface} is the target's average impingement surface temperature and T_{fluid} is the fluid temperature at the inlet of the nozzle. Due to the highly conductive properties (k) of the oxygen-free copper target and the low conductivity of the PEEK housing, one-dimensional heat transfer can be assumed within the cylindrical body of the target. During steady-state conditions, variations in temperature in cross-sectional planes of the target would be relatively insignificant with T_{surface}, T_{upper} and T_{lower} representing average temperatures in respective cross-sectional planes (see *Figure 3b* for reference). Considering the assumptions mentioned above and neglecting losses to the sides, heat flow from the bottom of the target towards the top surface can be calculated as in *Equations 2 and 3* below:

$$Q = -kA\frac{T_{upper} - T_{lower}}{d_1} \quad (2)$$

$$Q = -kA\frac{T_{surface} - T_{upper}}{d_2} \quad (3)$$

As *Equation 3* is equal to *Equation 2*, the target surface temperature can be calculated as shown in *Equation 4*.

$$T_{surface} = T_{upper} + \frac{d_2(T_{upper} - T_{lower})}{d_1} \quad (4)$$

Finally, after substitutions, *Equation 1* for the heat transfer coefficients could be expressed as *Equation 5*, where all terms are either known or measured during the experiments:

$$\bar{h} = k\frac{T_{lower} - T_{upper}}{d_1(T_{upper} - T_{fluid}) - d_2(T_{lower} - T_{upper})} \quad (5)$$

RESULTS AND DISCUSSION

In order to quantify the target surface temperature effect on ATF jet impingement convective heat transfer coefficients, a series of experiments were performed. The matrix of key parameters varied during the experiments is presented in *Table 1*.

Table 1. Parameters varied during the experiments.

Parameter	Values
Fluid temperature (T_{fluid})	50°C, 70°C, 90°C
Fluid flow rate (average jet velocity)	0.1 lpm (0.5 m/s)* 0.15 lpm (0.75 m/s)** 0.25 lpm (1.25 m/s) 0.5 lpm (2.5 m/s) 1.0 lpm (5.0 m/s) 1.5 lpm (7.5 m/s)
Target surface temperature ($T_{surface}$)	90°C*, 100°C, 110°C, 120°C
Jet incidence location on target surface	center, edge
*Only for experiments with 50°C and 70°C fluid temperatures. **Only for experiments with 90°C fluid temperature.	

Despite insulation, a significant amount of heat was lost to the ambient environment from the experimental fluid loop bench. This was especially evident at higher fluid temperatures and low flow rates (low heat carrying capability). As the fluid path from the heat exchanger to the nozzle inlet was over 4 meters, at the lowest flow rates (0.1 – 0.15 lpm) the heated fluid cooled down by more than 20° – 25°C before reaching the nozzle inlet. Maintaining a 90°C nozzle inlet temperature required elevating fluid temperatures in the reservoir and at the outlet of the heat exchanger up to 115° – 120°C (with even higher heater temperature settings), approaching safety and operational limits of the setup. This prevented us from performing 90°C fluid temperature experiments below 0.15 lpm flow rate.

The results of the experiments are shown in *Figure 6*, *Figure 7,* and *Figure 8*, where the average heat transfer coefficients are plotted against average jet velocities for 50°C, 70°C and 90°C fluid temperatures. The experiments were performed for the jet impinging in the center and at the edge of a round target surface.

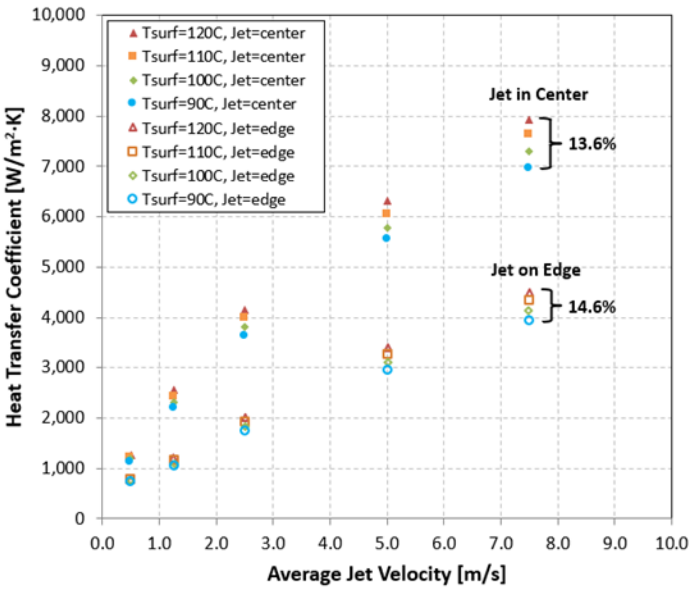

Figure 6. Heat transfer coefficients for 50°C fluid temperature.

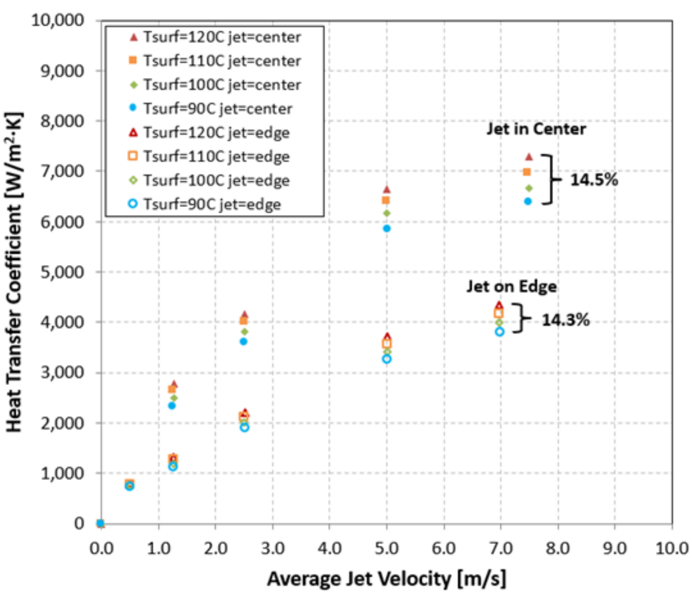

Figure 7. Heat transfer coefficients for 70°C fluid temperature.

As expected, the heat transfer coefficients increased with increasing impinging jet velocity. The heat transfer coefficients are higher for jets impinging in the center of the target compared to those impinging on the edge. This is also expected, as in the latter case only a fraction of impinged fluid flows over the target surface, decreasing its cooling effect. As in NREL's previous studies [15], the current experimental results showed negligible, if any, influence of fluid temperature (measured at the nozzle inlet) on heat transfer coefficients.

From *Figure 6* and *Figure 7,* it can be seen that at the highest tested velocities (7.0–7.5 m/s), a target surface temperature variation from 90°C to 120°C yielded 13%–15% variation in heat transfer coefficient values. For the 90°C fluid temperature (*Figure 8*) case, the heat transfer coefficient variation is smaller, within 11%–12%, which is due to a narrower range of target surface temperatures tested (100°C to 120°C). At lower flow velocities, the surface temperature impact on the heat transfer coefficient values decreases although it is still clearly distinguishable. The observed heat transfer enhancement phenomenon is likely due to increased ATF film temperature near the heated surface, resulting in reduced viscosity (strongly temperature-dependent property for ATF), in turn increasing fluid flow and heat transfer in the vicinity of the surface.

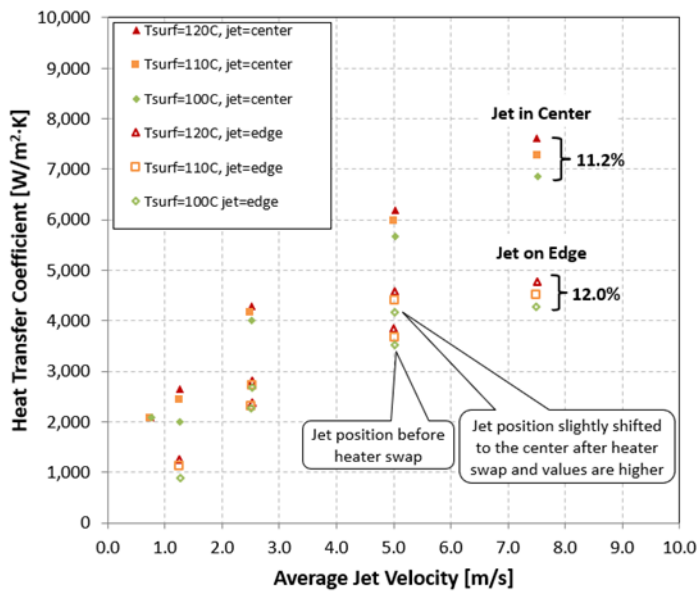

Figure 8. Heat transfer coefficients for 90°C fluid temperature.

In *Figure 8,* two sets of data for target edge jet impingement at 2.5 m/s and 5.0 m/s flow velocities can be seen. This illustrates the sensitivity of the jet incidence location alignment with the edge of the target. After replacing a burnt heater in the target assembly, the position of the impinged jet on target surface had slightly shifted towards the center (≈ 1 mm), resulting in a larger fraction of fluid flowing over the target surface versus away from the surface. A similar small alignment mismatch for target center jet impingement would not have had a noticeable effect (as all impinged fluid still would have flowed over the target surface), whereas in the case of the target edge impingement, it noticeably increased the heat transfer. Namely, the larger amount of fluid carrying away more heat increased measured heat transfer coefficient values. To prevent future errors in edge impingement experiments, a more accurate jet-to-target alignment mechanism design is planned.

The uncertainties for key data points are reported in *Table 2* and *Table 3*. In *Table 2* uncertainties in average jet velocities for the lowest and highest velocities (flow rates) are reported. In *Table 3* the uncertainties in heat transfer coefficients for lowest and highest jet speeds (flow rates) and surface temperatures (for target center impingement experiments) are provided.

Table 2. Uncertainties in average jet velocities.

Uncertainties in Average Jet Velocity	0.5 m/s (0.1 lpm)	0.75 m/s (0.15 lpm)	7.5 m/s (1.5 lpm)
U_{95} at T_{fluid} = 50°C	±1.8%	N/A	±1.0%
U_{95} at T_{fluid} = 70°C	±1.3%	N/A	±1.0%
U_{95} at T_{fluid} = 90°C	N/A	±1.2%	±1.0%

Table 3. Uncertainties in heat transfer coefficients for highest and lowest jet velocities and target surface temperatures.

Uncertainties in Heat Transfer Coefficient (target center impingement)	0.5 m/s (0.1 lpm)	0.75 m/s (0.15 lpm)	1.25 m/s (0.25 lpm)	7.5 m/s (1.5 lpm)
U_{95} at T_{fluid} = 50°C and $T_{surface}$ = 90°C	±16.7%	N/A	±8.8%	±3.0%
U_{95} at T_{fluid} = 50°C and $T_{surface}$ = 120°C	±8.6%	N/A	±4.4%	±1.6%
U_{95} at T_{fluid} = 70°C and $T_{surface}$ = 90°C	N/A	N/A	±16.4%	±6.4%
U_{95} at T_{fluid} = 70°C and $T_{surface}$ = 120°C	N/A	N/A	±5.5%	±3.25%
U_{95} at T_{fluid} = 90°C and $T_{surface}$ = 100°C	N/A	±37.4%	±37.1%	±11.8%
U_{95} at T_{fluid} = 90°C and $T_{surface}$ = 120°C	N/A	±12.2%	±9.8%	±3.7%

As it can be seen from *Table 3*, the highest uncertainties are at low jet velocities (flow rates) with the lowest target surface temperatures. During these experiments, upward heat flux through the target body is relatively low, resulting in small temperature difference (less than 1°C) between the lower and upper thermocouples (see *Figure 3b*), which in turn yields relatively high uncertainty in the respective heat transfer coefficient calculations. The lowest uncertainties at high jet velocities (flow rates) with the highest target surface temperatures reflect higher heat fluxes, thus, higher temperature differences between the lower and upper thermocouples, hence, smaller relative errors. The reported lower uncertainties can also be attributed to the fact that most experimental runs were performed back to back with several minutes apart and with little variation in experimental conditions, resulting in closely matched values of calculated heat transfer coefficients.

CONCLUSIONS

This paper provides experimental data on free-jet impingement convective heat transfer coefficients applicable for end-winding cooling of electric machines with ATF present in the traction drives of electric and hybrid vehicles. The data are expected to be directly useful to researchers and motor manufacturers in the design and development of power-dense, high-performance electric machines. It was found that the target surface temperature has a significant effect on jet impingement cooling with ATF. The impact was quantified for jet velocities from 0.5 m/s to 7.5 m/s, and the results showed that increasing target surface temperature from 90°C to 120°C enhanced heat transfer coefficient values at higher impingement jet velocities (7.5 m/s) by up to 15%. The observed heat transfer enhancement phenomenon is likely due to increased ATF film temperature near the heated surface, resulting in reduced viscosity (a strongly temperature-dependent property for ATF) and consequently, increased fluid flow and heat transfer in the vicinity of the surface.

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