

An Experimental Characterization of Automatic Transmission Fluid Flowing Through Air Cooled Microchannel Heat Exchanger

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ABSTRACT

Narrow-channel Heat Exchangers have received diversified research and application interests due to the miniaturized geometry, elevated heat transfer characteristics, and better energy efficiency. Automatic Transmission Fluid (ATF) cooling is one of the most important challenges in recent automotive industries due to the quick response to thermo-physical properties with temperature variations. Although a wide range of investigation on characterization of heat transfer and fluid-flow in Minichannel Heat Exchanger (MICHX) for various fluids is available, literatures examining ATF heat transfer and Fluid-flow behaviors in the MICHX are scarce. In the current study, attempts have been made to justify the suitability of MICHX application in cooling of ATF. Experimental investigations have been conducted in a well equipped closed loop integrated thermal wind tunnel test facility using wavy finned-MCHX as a test specimen. The inlet temperature of the ATF was maintained at a constant 75°C throughout the experiment. Air velocities of 6, 10, 14 and 18 m/s were applied. The entering air temperatures were varied at 15, 22, 29, 36 and 43°C while ATF Reynolds number was varied from 3.00 to 30.00 within the air temperature range.

The effects of ATF Reynolds number on heat transfer coefficient, and Nusselt number, as well as Heat Exchanger effectiveness-NTU for the air-flow Reynolds number of 1450-5200, were examined. ATF Nusselt number correlation with Reynolds number and Prandtl number was established as a power-law function while considering a variable property ratio. The analysis showed an enhanced result in heat transfer characterizations and good agreement with established phenomena of the multi-port Minichannel Heat Exchanges in open literature.

KEY WORDS: Minichannel Heat Exchanger, Automatic Transmission Fluid, Heat transfer coefficient, Thermo-physical properties, Reynolds Number, Nusselt Number, Prandtl Number, Property ratio

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1. INTRODUCTION

Researchers are advancing forward in search of devices that can conserve energy and friendly to the environment & ecology. Compact heat exchangers with narrow channels are able to serve such purposes. Miniaturized geometry and lightweight thermo-fluid systems; which are environment-friendly and capable of holding high-heat flux, find increasing demand among today's industries. Minichannel Heat Exchangers can mitigate needs of elevated-heat flux in the real-world applications such as; cooling of engine oil, engine coolant, and transmission fluid as well as air-conditioning systems in automotive industries. Channel geometry and hydraulic diameter are very important

parameters in terms of heat transfer and fluid flow characterization [1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15]. In the case of a single phase or two-phase liquid-gas heat exchanger, no fundamental change occurs for channel hydraulic diameter up to 200µm. However, a channel hydraulic diameter below 200µm, manufacturing techniques and cleanliness process are very important. One of the important criterions for classifying heat exchanger as a compact one is the hydraulic diameter within 1- 6mm. Although Mehendale et al (2000) and Satish G. K. (2003) made a different scheme for the channel classifications, from the available open literature, it is now established that a channel size with hydraulic diameter range of 3mm > D_h > 200µm are named as minichannel [15,16] although some

authors used the name microchannel in their literature while using a channel hydraulic diameter of 1mm [3, 4, 5, 9, 12]. The nomenclature is still an open question. In the current investigation the channel hydraulic diameter is 1mm, so it has been designated as minichannel.

Motor vehicle performances are usually evaluated in terms of both emissions and fuel consumption. The stringent emission legislation set by the Environmental Protection Agency (EPA) or California Air Resources Board (CARB), leads the manufacturer in difficult situation in view of controlling vehicle emissions and performance. That pressure on manufacturing companies can be increased due to an inefficient power train system. To overcome such problems, efforts are being concentrated into the improvement in system efficiency of a power train. However, investigation dealing with the efficiency improvement in a transmission system especially cooling of the automatic Transmission Fluid (ATF) using Minichannel Heat Exchanger (MICHX) is scarce in open literature [17]. ATF performance is highly dependent on temperature, especially due to its quick response in changing the viscosity. Kemp, S. P. (1990) established a correlation of viscosity and temperature, which has exponential relation as; $Y = 51.565E^{(-0.1651T)}$ [18]. Here Y indicates Brookfield viscosity and T as temperature. The ATF viscosity is usually specified as kinematic viscosity (centistokes) at 100°C (ASTM D445) on the high-temperature range and Brookfield viscosity (centipoises) which is reported as absolute viscosity at -40°C (ASTM D2983) on the low-temperature range [18, 19]. The viscosity of ATF varies within 5.50 - 8.00 cSt at 100°C and generally under 20,000 cP at -40°C. Most essential element properties of the ATF are: friction and friction durability, oxidation resistance, good sealant performance, operating at temperature extremes, non-corrosive to transmission components, and special antifoam properties. Rudnick, L. R. (1999) explained the physical and chemical properties of ATF and its historical review, and also the performance level comparisons [20]. Henderson H. E. et al (1998) conducted an investigation on ATF fluidity at low and high temperatures. At Brookfield viscosity region, significant gelation can occur due to wax crystal growth effects where the ATF behaves as non-Newtonian fluid. However, at the kinematic viscosity region, it behaves as Newtonian fluid. The transition point is known as a cloud point (ASTM D2500) [21]. Although Viscosity Index Improver (VII) helps in stabilizing properties of the fluid at these temperatures, an appropriate cooling method can play a major role in maintaining essential fluid properties at the operating temperature. Semel, R. R. (2001) investigated stand-alone Oil to Air transmission cooling strategy with thermostatic cold flow bypass valve. The study was conducted for improving the efficiency of the transmission by upgrading a warm-up system to minimize energy losses due to extremely viscous fluid at low temperatures [22]. The improved transmission performance allows for enhanced

engine efficiency and emission performance leading to overall vehicle performance and fuel economy.

Few researchers worked on examining the viscous effect of the liquid on temperature variations [23, 24, 25, 26, 27, 28, 29, 30, 31, 32, 33, 34, 35, 36, 37, 38, 39], while few others worked on pressure differential through MICHX [40, 41, 42]. Wang, C. et al (2004) investigated frictional characteristics of highly viscous fluid in minichannels ($Dh = 0.198\text{--}2.01\text{mm}$) while using water and lubricant oil as the working fluids [26]. In their investigation, it was found that the viscosity has negligible influence on the friction factors if the hydraulic diameter is greater than 1.0 mm. While using dimensional analysis, Koo, J. (2004) made an investigation on the effects of viscous dissipation on the temperature field and friction factor. The author experimentally validated computer simulation results using water, methanol and iso-propanol as working fluids [27]. The investigation turned out that, for microconduits, viscous dissipation is a function of the channel aspect ratio, Reynolds number, Eckert number, Prandtl number, and conduit hydraulic diameter. For such cases, even for a low Reynolds number, significant temperature increase may occur due to viscous dissipation for a fluid with high viscosity and low heat capacity. Nonino, C. et al (2006) carried out a parametric investigation on the effects of temperature-dependent viscosity in simultaneously developing laminar flow of a liquid for straight ducts while considering boundary conditions at a uniform wall temperature [30]. Their numerical investigations showed that the viscosity varied exponentially with temperature variation in the entrance region for the laminar forced convection. Sarker, R. et al (2002) conducted investigations to characterize rheological properties of ATF at low and high temperatures. The findings in their investigation showed that the fluids with same kinematic and the Brookfield viscosities undergo a difference in viscosity values up to 40% while the transmission works under the typical conditions [38]. Yang, Y. et al (1995) developed a mathematical model to predict the heat transfer and fluid hydrodynamics during wet clutch engagement while considering viscous dissipation, ATF heat balance, and the effect of other thermo-physical properties [39].

Although a few investigations have been conducted to apprehend the thermo-physical properties of ATF and its heat-transfer fluid-flow performances inside the transmission, the examination on cooling strategies and heat transfer characterization using a minichannel heat exchanger are still very sparse in open literatures. Therefore, the current study entails a detailed investigation on the applicability of a minichannel heat exchanger for transmission fluid cooling strategies.

2. EXPERIMENTAL SETUP

The current experiment was conducted through a well-instrumented experimental setup as shown in [Figure A1 \(Appendix-A\)](#). This experimental setup is capable of

examining heat transfer and fluid flow characteristics of single-phase air to liquid cross-flow minichannel and microchannel heat exchangers under different operating conditions. It consists of a closed loop integrated thermal wind tunnel, test chamber, liquid circuit with heater/chiller for desired heating and cooling modes, and various measuring instruments. The largest component of the experimental setup is the wind tunnel. It is an integrated closed loop thermal wind tunnel of 5440 mm in length, 750 mm width, and 1640 mm height. It is capable of providing up to 30 m/s free stream velocities and 18 m/s with the heat exchanger at the test chamber. A built-in heat exchanger maintains the inlet air temperature. Thermocouples are installed in grids at the upstream and downstream of the test chamber to measure air inlet and outlet temperatures. The test chamber is made of 6.5 mm thick Plexiglas with the dimensions of 305 mm $\times$ 305 mm, and 610 mm along the air flow direction. It is covered with additional insulations to make fully heat resistive.

The wavy-finned serpentine cross-flow minichannel heat exchanger is placed inside the test chamber. The high-quality aluminum heat exchanger consists of three loops. Each loop has a single-piece slab which is bent four times to make five straight slabs with serpentine shape. Each of the loops has 68 parallel channels in the slab. The channel diameter is 1mm and equally spaced over the 100mm width, and 2 mm thick slab as shown in [Figure A2 \(Appendix-A\)](#). The liquid handling system comprises a reservoir tank, piping networks, variable-speed gear pump, inline heater, a recirculation pump, measuring instruments, and monitoring devices. The flow measuring instruments include digital flow meter, impeller flow meter, temperature and pressure gauge, Resistance Temperature Detectors (RTD), and Differential Pressure Transducers (DPTD). A gear pump, which is coupled with a frequency controlled motor of 5.6 kW, maintains a steady flow rate of ATF. In the piping network system, a micro filter is placed before the Digital Flow Meter (DFM) for properly filter the fluid. The DFM is capable of measuring temperature, pressure, and volume flow rate of the liquid.

RTDs and PTDs are installed at the inlet and exit to the heat exchanger to measure the temperature and pressure of the liquid. The liquid handling system can operate with different fluids such as, water, ethylene glycol-water mixture, engine oil, ATF, etc. On the air side, three pressure transducers are installed to measure the differential pressure at different locations such as; top, middle, and bottom through the inlet to the outlet of the heat exchanger specimen.

Hot Automatic Transmission Fluid at a constant 70°C was pumped to flow inside the channels while comparatively cold air was blown to cool the hot liquid. A recirculation pump was added to the heating tank to create a continuous agitating environment similar to that usually exists inside the transmission. The agitation prevents any oil damage from overheating by the heater element due to higher skin temperature. The system of this recirculation pump provides additional function with a better mixing among the cold and

hot fluid inside the heater, and helps maintaining stabilized temperature of the ATF.

The liquid flow through the heat exchanger was a developing flow in the laminar regime with a Reynolds number of $3 \leq Re_l \leq 30$. To ensure accuracy in measurement of the liquid mass flow rate, a bucket and a stop watch were used, and the readings were compared with the digital flow meter readings.

3. EXPERIMENTAL PROCEDURES

To ensure stabilized steady-state condition of the experimental setup including instrument responses for better data, the whole setup was run for a period until it reached satisfactorily stabilized. Before acquiring any data, all conditions of running the experiment and acquiring data were set. The Automatic Transmission Fluid (ATF) steady-state inlet temperature of 75°C was maintained for all the data points. The inlet air temperatures were varied at 15°C, 22°C, 29°C, 36°C, and 43°C with the air velocity of 6, 10, 14, and 18 m/s for each inlet temperature. The condition was considered steady-state when the fluctuation of the readings remained within $\pm 2\%$. The source tank was filled with ATF. An instrumented variable-speed gear pump was used to pump the fluid from the tank to the system. The choice of selecting positive displacement external gear pump was to maintain a steady flow that would help the instrument reading accurately. A vane pump could serve the same purpose but more expensive than gear pumps. The fluid was heated up to $75 \pm 0.5^\circ\text{C}$ and this temperature was kept constant for whole experiment. The cold and hot city water was supplied through a mixing chamber to have the desired air inlet temperature as required for the experiment. A separate recirculation pump was used to make continuous agitation of the ATF inside the heater for creating an environment that exists in the real-world application inside the transmission. This additional pump arrangement prevents the ATF from overheating due to high skin temperature of the heater core elements. The ATF temperature got steady very quickly, but the air temperature took longer time. Once the whole system became steady, all the instruments were again checked carefully so that they read accurately. Four different mass flow rates were chosen for each inlet air temperature and for a particular mass flow rate, four velocities of the inlet air were applied. Therefore, for a particular air temperature, there were $4 \times 4 = 16$ data sets and for five inlet air temperatures total number of data points were $16 \times 5 = 80$ set points. At each liquid mass flow rate, about 162,000 thousands of samples were collected and averaged for subsequent calculations. Based on the selected mass flow rates, and inlet air temperature & velocities, the liquid Reynolds number (Re) varied as $3 \leq Re_l \leq 30$. Due to a highly viscous fluid, and smaller channel hydraulic diameter, the Re_l was very low. The data was collected using Lab View, a 16 bit data acquisition system with 128 channels.

4. DATA REDUCTION

Experimental data were collected at a steady-state condition. The thermo-physical properties of the Automatic Transmission Fluid (ATF) and the heat transfer- fluid flow key parameters of the Minichannel Heat Exchanger (MICHX) cores were appropriately analyzed by using established equations. The basic key parameters include average heat transfer rate ($\dot{Q}$), heat transfer coefficient (h), Effectiveness (ϵ), Number of Transfer Units (NTU), ATF Reynolds number (Re_l), Nusselt number (Nu_l), Prandtl Number (Pr_l), and other parameters relevant to the analysis.

The heat transfer rate can be evaluated from the overall energy balance for the hot ATF and the conditioned inlet air as described below:

$$\dot{Q}_l = \dot{m}_l C_{p_l} (T_{l_i} - T_{l_o}), \quad \text{for liquid} \quad (4.1)$$

$$\dot{Q}_a = \dot{m}_a C_{p_a} (T_{a_i} - T_{a_o}), \quad \text{for air} \quad (4.2)$$

$$\dot{Q} = \frac{\dot{Q}_l + \dot{Q}_a}{2}, \quad \text{for average heat transfer} \quad (4.3)$$

Since both fluids were in motion, the heat transfer occurred mainly due to forced convection. So, the Reynolds number (Re), as a non-dimensional parameter, was taken into account. It is the function of velocity, fluid density & Dynamic Viscosity, and the channel geometry. The heat exchanger under investigation contained three loops while each loop contains 68 channels. Hence, the Re_l for this heat exchanger can be expressed as;

$$Re_l = \frac{\rho_l D_{h_l}}{\mu_l} \left(\frac{4\dot{m}_l}{3(68)\pi\rho_l D^2_{h_l}} \right) = \frac{\dot{m}_l}{51\mu_l \pi D_{h_l}} \quad (4.4)$$

Heat transfer coefficient (h) can be deduced from equation 5. The obtained expression explains the importance on narrowness of channel hydraulic diameter whereas high fluid conductivity in heat transfer enhancement.

$$h_l = \frac{Nu_l k_l}{D_{h_l}} = \frac{Q}{A(T_b - T_{s_i})} \quad (4.5)$$

Another non-dimensional parameter called Nusselt number (Nu) which indicates the temperature gradient at the surface is also very important in a heat transfer phenomenon. It can be expressed as;

$$Nu_l = \frac{\text{Fluid layer resistance}}{\text{Convective resistance}} = \frac{h_l D_{h_l}}{k_l} \quad (4.6)$$

ATF as a viscous fluid, some of the thermo-physical properties dramatically varied with temperature. Although density, thermal conductivity, and the specific heat capacity affect the velocity and thermal distributions in the flow field, their contributions are less as compared to that of the viscosity [25]. Temperature has a high effect on viscosity that affects other non-dimensional parameters, especially Nu . So, the Nu of ATF should be corrected with a variable property ratio for temperature variation. Two common schemes are usually used for compensating viscosity variation effect due to temperature. These are (a) reference temperature (the film average temperature scheme) and (b) the method of lumping the effects into the ratio of mean to wall fluid viscosity (μ_m/μ_w) [47]. Method (b) is more convenient to handle due to simplicity. For the temperature effect on viscosity, Nu of ATF can be corrected as follows;

$$\frac{Nu_c}{Nu_m} = \left(\frac{\mu_m}{\mu_w} \right)^n \quad (4.7)$$

To calculate μ_w the inner wall surface temperature is required that can be calculated as;

$$T_{s_i} = T_{s_o} + (\dot{Q} * R_w) \quad (4.8)$$

The thermal resistance of the wall (R_w), although it has been considered negligible for calculating overall thermal resistance in subsequent equations, can be calculated as;

$$R_w = \frac{\ln\left(\frac{D_o}{D_i}\right)}{2\pi k_{al} L_c} \quad (4.9)$$

The channels are spaced 1.00 mm from each other in the heat exchanger slab, and the thickness of the slab wall is 0.5 mm. So, for easy calculation, the surrounding of a single channel has been considered circular, and the above expression was used.

Prandtl number (Pr) which is the fluid property, indicates the relative effectiveness of momentum as well as energy transport by diffusion in the velocity and thermal boundary layers. It can be expressed as;

$$Pr_l = \frac{\text{Momentum Difusivity}}{\text{Thermal Difusivity}} = \frac{\mu C_p}{k_l} \quad (4.10)$$

For a particular fluid, the heat diffusion occurs quickly when $Pr \ll 1$, and slower when $Pr \gg 1$ [48]. $Pr \ll 1$ indicates that, the thermal boundary layer is thicker compared to that of the velocity boundary layer, and opposite when $Pr \gg 1$. Liquid metal has $Pr \ll 1$ that offers thicker thermal

boundary layers whereas oil has $Pr \gg 1$. So, ATF has a thinner boundary layer with high Pr values.

For a multi-pass heat exchanger with cross flow orientation, the following expression applies.

$$Q = UAF\Delta T_{lmt\delta}, \text{ or } UA = \frac{Q}{F\Delta T_{lmt\delta}} \quad (4.11)$$

Here, UA is the overall conductance which is the measure of how well, heat can be transferred through the heat exchanger. F refers to the log mean temperature difference correction factor for heat exchanger at cross-flow orientation. The LMTD correction factor F is non-dimensional, and it depends on the temperature effectiveness P , and heat capacity rate ratio R .

The heat exchanger is made of high-performance aluminum, which has a higher rate of conductivity. In addition to that, the channel slab thickness is only 0.5mm that allows very quick heat transfer. The wall conductive thermal resistance (R_w) is very low compared to those of the convective resistances. Therefore, R_w can be neglected in calculating overall thermal resistance. So, the overall thermal resistance can be expressed as;

$$R_{tot} = \frac{1}{h_l A_l} + \frac{1}{\eta h_a A_a} \quad (4.12)$$

In equation 12 the first and the second terms indicate the convective thermal resistances for ATF and air respectively. The thermal resistances due to fouling for both fluids were ignored due to a new specimen.

The heat exchangers are usually characterized by their effectiveness (ε) and the Number of Transfer Unit (NTU). The effectiveness is defined as the ratio of the actual heat transfer rate to the maximum possible heat transfer rate. In this investigation, the maximum possible heat transfer rate can be obtained at the ATF side. Therefore, the expression for the effectiveness can be expressed as;

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{\dot{Q}}{(\dot{m}c_p)_l(T_{l_i} - T_{a_i})} \quad (4.13)$$

The number of transfer units (NTU) is defined as a ratio of the overall thermal conductance to the minimum heat capacity rate. It also explains the relative magnitude of the heat transfer rate compared to that of the rate of enthalpy change in the fluid with minimum heat capacity rate [43]. The NTU can be expressed as;

$$NTU = \frac{UA}{c_{min}} = \frac{UA}{(\dot{m}c_p)_l} \quad (4.14)$$

NTU is a non-dimensional design parameter for heat transfer that basically indicates the thermal size of a heat exchanger, not the geometrical or the physical size.

Although ATF inlet temperature was kept constant at a 75°C , the outlet temperatures varied following the inlet air temperatures. To estimate the appropriate representative temperature in considering fluid properties, bulk temperature (T_{bl}) was considered [43, 45, 47, 48].

5. RESULTS AND DISCUSSIONS

5.1. ATF MASS FLOW RATE

A Digital Flow Meter (DFM) recorded the mass flow rate while it was verified using a bucket, stop watch, and weighing scale. For an ideal case, when positive displacement pumps are used, volume flow rate increases with the increase in viscosity as; $Q_{actual} = (Q_{nominal}) - C(P_{discharge})/\mu$, (C = Pump geometrical factor). The MICHX under the study has a very small channel diameter of 1mm; whereas, a highly viscous fluid was forced through it. Due to the high viscosity and low channel diameter, a high rate of shear stress ($\tau = \mu dv/dr$) developed inside the minichannel causing higher system pressure. As a result, the pump volumetric efficiency decreased with the increase in outlet pressure.

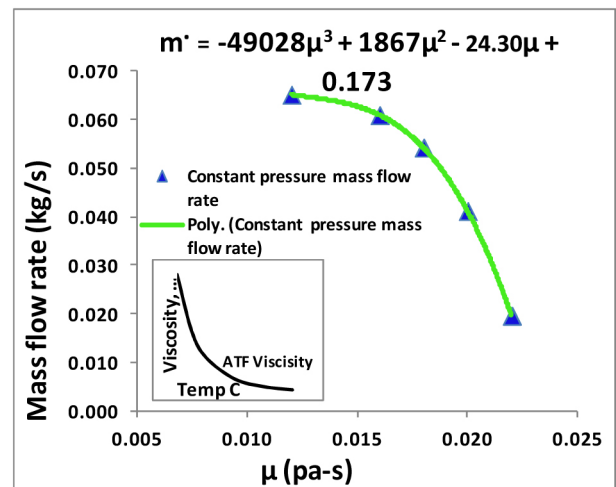


Figure 1. The effect of Dynamic Viscosity (μ) on liquid mass flow rate ($\dot{m}$)

Figure 1 explains the relation between the liquid mass flow rate and viscosity at constant system pressure. The mass flow rate decreases with the increase in viscosity as a result of decreasing temperature.

5.2. HEAT TRANSFER RATE

The heat transfer rate was calculated and plotted against the liquid Reynolds number as shown in Figure 2. The heat transfer rate increases with the increase in liquid Reynolds number.