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Investigations on Microchannel Heat Exchanger using Numerical, Experimental, and ANN Techniques

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ABSTRACT

Thermal systems, such as high-performance vehicle radiators, require significant heat flux to be removed to maintain consistent performance and prolonged life. Microchannels offer a viable option because they provide a significant heat transfer area-to-volume ratio. The analytical design of a Microchannel heat exchanger (MCHX) with tiny fins above and below is part of the research work. A coolant with a temperature range of 75OC to 85OC and ambient air with a temperature range of 30OC to 35OC are used as test inputs for the MCHX. The estimated MCHX's size for these test inputs is consistent with a 3000 W heat duty. Compared to the Minichannel Hx (Heat Exchanger) for the same heat duty, the size is decreased by 60%. The estimated size is then subjected to numerical analysis using software tools. Analytical and numerical results are found to concur well with one another, with less than 9.64% variance. Regression, followed by common sense and uncertainty analysis, is then used to build the Nusselt-Prandtl (Nu-Pr) correlation with 0.9% of the calculated uncertainty, which helps increase the confidence in the results obtained. For predicting Nu value, an Artificial Neural Network (ANN) model is also created. The experimental Nu values and the values predicted by regression correlation are compared with the Nu values predicted by the ANN model, and it is shown that they are both within 3% of one another in variance.

Keywords: Microchannel, Numerical Analysis, Performance testing, Common-sense analysis, Uncertainty analysis, ANN model

1. INTRODUCTION

The physics of heat transmission and fluid mechanics serve as the foundation for the Basic Principles of Thermal Design. Heat is transferred from high temperature to low temperature. Through heat conduction, heat convection, and heat radiation, heat can be transferred. The method used for the thermal design depends on the data which is available to the designer. It also depends on the complexity of the applications. For example, Heat exchangers used in marine applications are more susceptible to the fouling of various components, and oversizing of the heat exchanger is needed. Since the domestication of fire, heat transfer research has sought to understand how to efficiently transmit heat from one place to another and from one medium to another while dealing with a range of restrictions. Convectional heat transfer offers a mechanism to quickly move heat away from heat exchange surfaces. In heat exchangers, fluids used in engineering systems and processes go through thermal state changes. The fundamental formula for heat transfer by convection is as follows:

$$Q = hA(T_s - T_f) \quad (1)$$

Early innovations from the 19th and 20th centuries were centered on expanding the surface area to support faster heat transfer rates. Through the demands of the transportation industry, including the automotive, aircraft, submarine, and spaceship industries, the era of small heat exchangers started to take off. Equation (1) was reviewed, with a focus on concurrently enhancing the heat transfer coefficient and area considering the additional limitations of the overall volume and weight. Energy conversion, transmission, and utilization are essential for various applications such as automotive, power generation, process industries, manufacturing plants, HVAC, etc (Dasgupta, 2011). Efficient heat transfer equipment (Hx) play an important role in these applications. This saves a significant amount of cost by improving heat transfer capabilities. Energy efficient Hx can save a significant amount of energy by improving power conversion efficiency. This also helps reduce the size and cost. Modern IC engines generate a large amount of heat.

Consequently, the engine body temperatures spike more than 500°C. It is essential to dissipate this heat to surroundings efficiently. A one-third of the heat produced during combustion is transformed into energy that drives the car and its accessories. Through the exhaust system, a further one-third of the heat is released into the atmosphere. The cooling system, specifically the radiator, must dissipate the remaining one-third of the heat produced by the engine. The current research is targeted at replacing conventional radiators with MCHX to rapidly dissipate heat from the radiator. Some of the researchers have demonstrated that single-phase flow for liquids remains unaffected as hydraulic channel diameters are decreased from 200 μm to 10 μm, however, this necessitates a greater comprehension of MCHX (Kandlikar and Grande, 2003). In single-phase and two-phase applications, the employment of new fins, including microfins, became common.

Twisted tapes and other improvement devices were also created (Mehendale et al., 2000). With the launch of its newest branch in microscale heat transfer, the term "micro" was eagerly accepted. Considering that they offer a significant amount of heat transfer surface area per unit of fluid flow volume, Microchannels and Minichannels are ideally suited. Thus, enabling an extremely high heat transfer rate. Microchannel technology has a variety of uses, including capillary pumps, rocket engines, hybrid vehicles, hydrogen storage, and chilled cooling, to mention a few (Khan and Fartaj, 2011). In domestic air conditioners, the Microchannel heat exchanger is more successful at improving performance than the cross-fin and tube heat exchanger. As a heat exchanger for all types of air conditioners, microchannel heat exchangers help to reduce the amount of refrigerant used in residential air-conditioning systems. However, anti-corrosion technology and product application flexibility should be improved to replace all aluminum parallel flow heat exchangers.

Objectives

Given the increasing use of MCHX for cooling applications, an empirical correlation for Nu Number is necessary. This would help other researchers working in a similar research area address cooling-only applications. Therefore, the following objectives have been set for the current research work.

Parametric analysis of heat transfer and fluid flow in MCHX for the required heat duty.

To establish Nu = f (Re, Pr) correlation for single-phase MCHX applications.

This paper covers a detailed review of current literature in the field of MCHX. It also discusses in detail the methodology used for the design of MCHX, followed by analysis using different techniques like Regression, Uncertainty analysis, and ANN. The results and conclusion are discussed at the end of this paper.

Literature Survey

Hayase et al., (2016) have studied the Microchannel heat exchanger for cooling-only applications. An all-aluminum heat exchanger was developed to replace the existing heat exchanger for automobile applications. They have studied 4 types of MCHXs used in two-phase applications. It was observed that MCHX is more effective than cross-fin and tube heat exchanger two-phase heat exchange applications. It is also observed that the use of MCHX results in a reduced mass flow rate of fluid. It was also observed that refrigerant flow maldistribution is one of the main problems that reduces the heat transfer rate. The conventional Microchannel heat exchanger represents non-uniform behavior and also increases pressure loss. A similar trend is found in MCHS, with a low temperature at the inlet and a high temperature at the outlet, and an increasing temperature along the longitudinal direction.

Secondary flows are introduced along the primary flow at many locations to overcome the above problem. Leading to the introduction of a number of passes in MCHX. This reduces the boundary layer thickness and enhances the heat transfer rate. The MCHX is one of the most suitable devices used in electronic cooling. Heat exchangers with asymmetric geometry have been studied (Denkenberger et al., 2012). It was concluded that the MCHX available in the market has higher manufacturing costs despite the inexpensive material costs. This limits the use of MCHX for many applications. A multi-port serpentine Mesochannel heat exchanger's fluid flow and transfer properties have been analyzed quantitatively (Sanaye and Dehghandokht, 2011). As coolants, two liquid working fluids, water and glycol, were studied. It was determined that water's heat transfer properties at low Reynolds numbers are comparable to those of an ethylene glycol water mixture.

The MCHX has a decent chance of being employed as a radiator in cars. A study on the application of water and glycol mixtures was conducted Manik et al., (2019) to determine the heat transfer in the radiator. They studied heat transfer in the Hx utilizing glycol water as a coolant at 4-8 Ltr/Min (LPM). This increased the heat transfer by 11% as compared to conventional fluid. Numerical analysis of heat transfer and water flow Mohammed et al., (2011) have been performed in Wavy Microchannel Heat Sink (WMCHS). The cross-section consisted of wavy crests ranging from 125 to 500 μm. Steady-state fluid flow and heat transfer were assumed. Heat transfer characteristics of copper and alumina nanoparticles in water and ethylene glycol-based fluids Ngiangia and Nwabuzor, (2021) have been studied. It was observed by the authors that thermal conductivity was enhanced.

However, owing to the scope of current research works, nanoparticles are not used in the Glycol. Researchers Zhang et al., (2014) have numerically modeled complex Microchannel systems for thermal systems focused on U-shaped Microchannels. The simulation was run concurrently with an experimental investigation. The numerical solution was considered to have converged after the highest residuals of all parameters were less than 10-8. This was done by an explicit iteration technique. Farsad et al., (2011) have performed a numerical simulation of a copper Microchannel Heat Sink (MCHS). The commercial software program FLUENT was used to create a three-dimensional computational fluid dynamics model. Studies with the MCHX employing supercritical CO_2 as the heating medium and water as the coolant led to the development of empirical correlations between the local HTC and pressure drop.

The study Zou et al., (2016) used modeling of the Microchannel heat exchanger (MCHX) in wet air. The effectiveness-NTU technique was used to calculate heat transfer. Empirical correlations for heat transfer and pressure drop were employed. Unsteady 3-D incompressible flow Lee et al., (2020) was studied, and Navier-Stokes equations were solved with Boussinesq approximation. The numerical model for the fin-tube Hx design was also verified. An unsteady setup was used in the code of ANSYS CFX 18. Traditional methods used in regression analysis do not capture the data's non-linearities and are unable to handle large data sets. This necessitates the use of novel computer science methods like ANN. The paper also presents a comparative study of various models used in ANN (Thike et al., 2020).

Research Gap

The literature survey clearly indicates that a lot of research is being done in the field of Microchannels. Many researchers have accomplished the task of designing MCHX to increase heat transfer rates in the case of two-phase flows. However, because of the slip phenomenon, MCHX for single-phase flow has become a difficult task. Therefore, a golden mean would be to design MCHX with little larger diameters close to 1 mm or more to make the task easier. Given this, the work is carried out to design MCHX and develop correlations for heat transfer.

2. METHODOLOGY

This section discusses the research methodology used in the current work. This includes the MCHX design for its size estimation, followed by a numerical analysis of the MCHX for the estimated size. Fabrication of the MCHX module and the experimental setup is then completed. Nu-Pr correlation is then developed using regression. To ensure the validity of the performance testing, an ANN model is used. The overall methodology of the current work is as stated in (Figure 1).

Parametric considerations

Some of the main parameters to be considered in the design of MCHX are discussed in the following section. The heat duty needed to be conformed is 3000W with an inlet coolant temperature of 85OC and ambient air in the range of 30 OC to 35 OC. A pressure drop of up to 5 kPa is considered. This is also supported by the work presented in the paper by (Harris et al., 2000). The ambient temperature ranges from 30 OC to 40 OC, and for the coolant under consideration range from 80 OC to 85 OC. The diameters of the Microchannels are fixed at 1 mm (Harris et al., 2000). The air absorbs heat only from coolant flowing through the Microchannels. There is no heat transfer between adjacent Microchannels.

The pressure distribution of the coolant is uniform. The overall length and width of the heat exchanger can be varied as per the requirements of the space. The only constraint is the area available for the heat transfer. The pressure head associated with a car traveling at 60 KMPH provides a reliable indicator of the typical pressure drop of the air across the heat exchanger. The methodology used for the estimation of size is shown in (Figure 2). Microsoft Excel is used to create a straightforward model based on the aforementioned formulae and connections. The dimensions of various heat exchanger components, including channel length, channel count, flat tube count, etc., are then determined using straightforward geometric relationships.

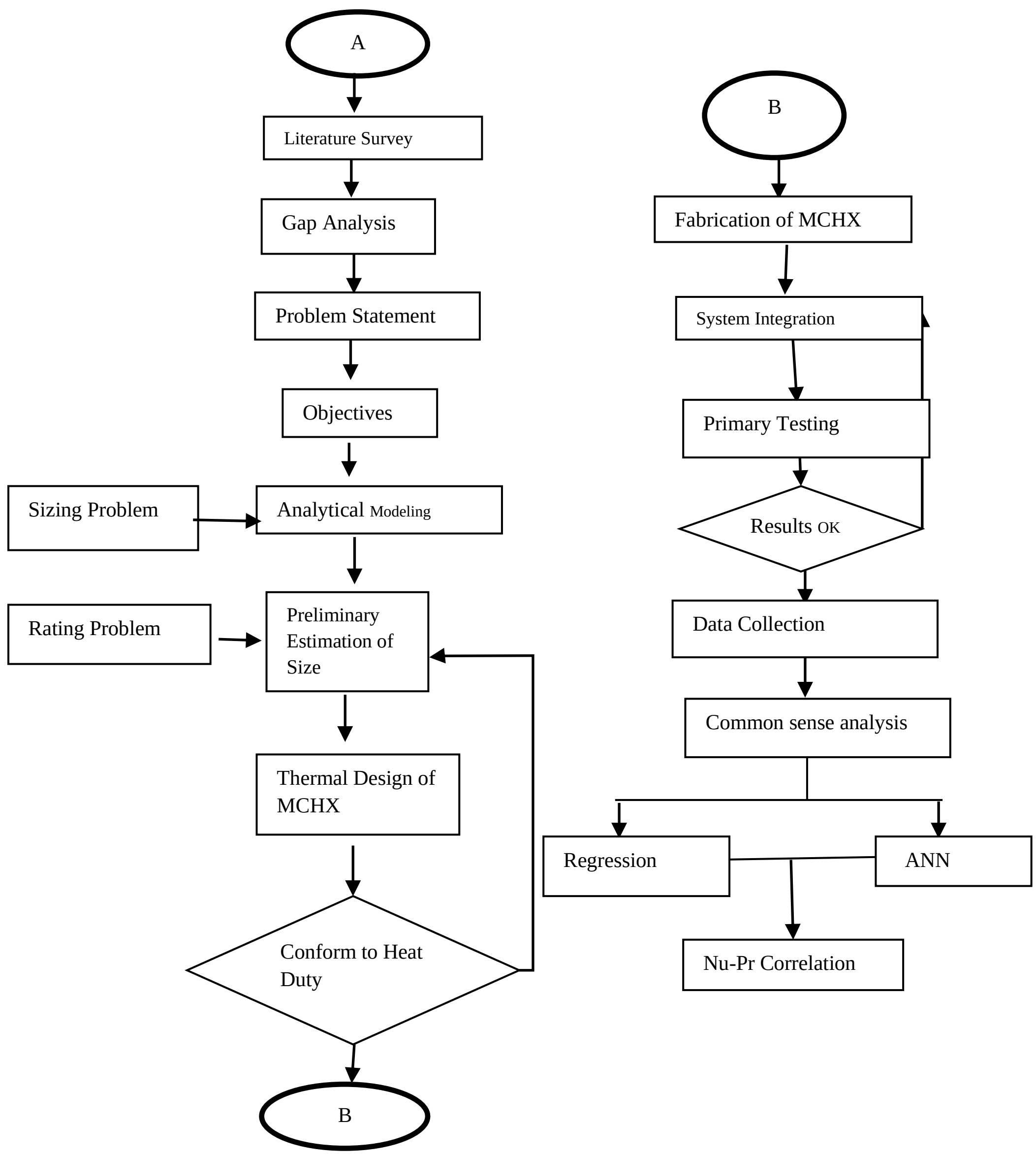

Figure 1 Research Methodology

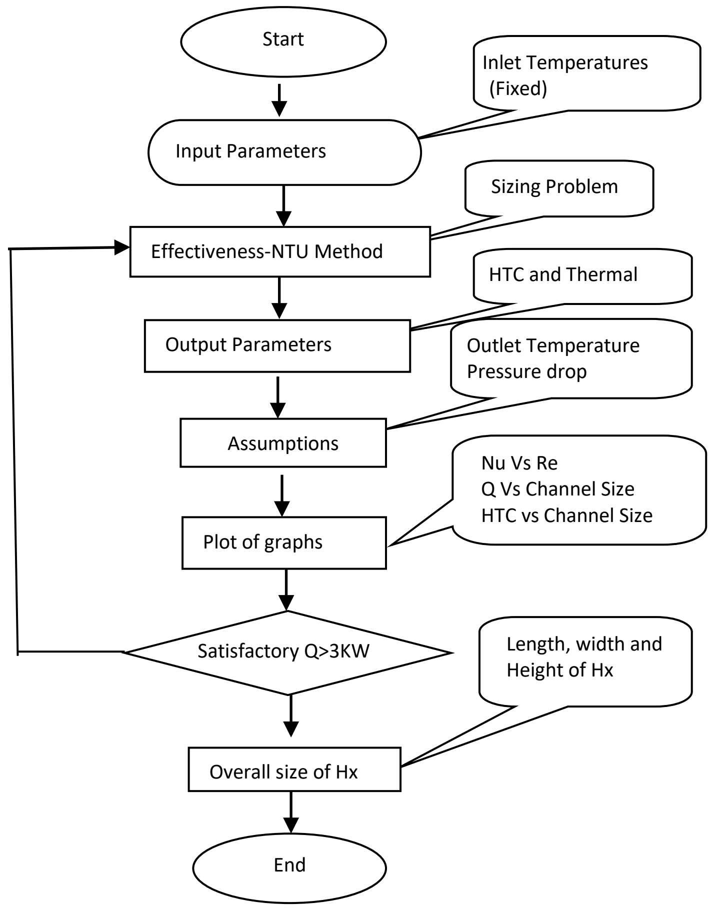

Figure 2 Design Methodology

A simple analytical model is developed using the following set of standard equations.

$Q = \in C_{min}(T_{hi} - T_{ci})$ (2)

$\in = 1 - exp\ [(\frac{1}{C^*})(NTU)^{0.22}\{exp(-C^*(NTU)^{0.78}) - 1\}\ ]$ (3)

$NTU = \frac{UA}{C_{min}}$ (4)

$C^* = \frac{C_{min}}{C_{max}}$ (5)

$C_{min} = minimum\ (C_{min}, C_{max})$ (6)

$C\_max = maximum\ (C\_min, C\_max\ )$ (7)

$UA = \frac{1}{\frac{1}{(1-\frac{A_f}{A_a})(1-\eta_f)h_a A_a} + \frac{1}{h_i A_i}}$ (8)

$h_i = 0.023 Re_{Dh}^{0.8} Pr^{0.4} \frac{k}{D_h}$ (9)

$\Delta P = 2f \frac{l}{{D_h}^3} \frac{\mu_i^2}{\rho_i} {Re_{D_h}}^2$ And $f = 0.0514 Re_{D_h}^{-0.22}$ (10)

$A = A_c x N_{cf} x N_{ft}$ (11)

$A_c = \pi d_o x L$ (12)

$N_{cf} = W / p_t$ (13)

Where the width of MCHX is set at 24 mm. Furthermore, as the microchannel diameter is constant, it should be noted that the Pt (tube pitch) stays constant. The air-side dimensions still need to be determined. The heat exchanger has an overall height of 150 mm. The number of flat tubes would correspond to the number of fin rows on the airside. If so, the number of fins that must be arranged in a row won't be decided until the air-side area is located. The values of length, width, and height of MCHX can be varied in the proportion needed, ensuring the area for heat transfer remains as per the design. The overall size of the MCHX is then finalized. The dimensions of Microchannel tubes and other specifications as per the design are shown in (Table 1). The internal geometry of the Microchannel module is shown in (Figure 3). There is also an enhancement of heat transfer with a decrease in the hydraulic diameter of Microchannels as indicated in (Table 2).

$A_a = N_{ft} x n_{fr} x f_h x W$ (14)

$f_h = H/N_{ft}$ (15)

Table 1 Estimation of the size of MCHX

Sr. No.	Particulars	Value
1	Area of Microchannels	0.3 m2
2	Area of fins	10.7 m2
3	Diameter of microchannels	0.001 m
4	Length of Microchannels	45 m
5	No. of microchannels of 200 mm	225
6	No of channels in a flat tube	15
7	No of flat tubes needed	15
8	Height of Hx	150 mm

Table 2 Outcome of Analytical Modeling

Case	di (m)	Ui (W/m2K)	Re	Nu	Pump (W)
1	0.00025	1522	17033	164	8489
2	0.0005	1252	12067	136	4789
3	0.00075	1097	8778	125	480
4	0.001	969	6633	120	294
5	0.00125	793	5187	117	27
6	0.002	579	3317	112	2
7	0.0025	341	2493	110	0.55119

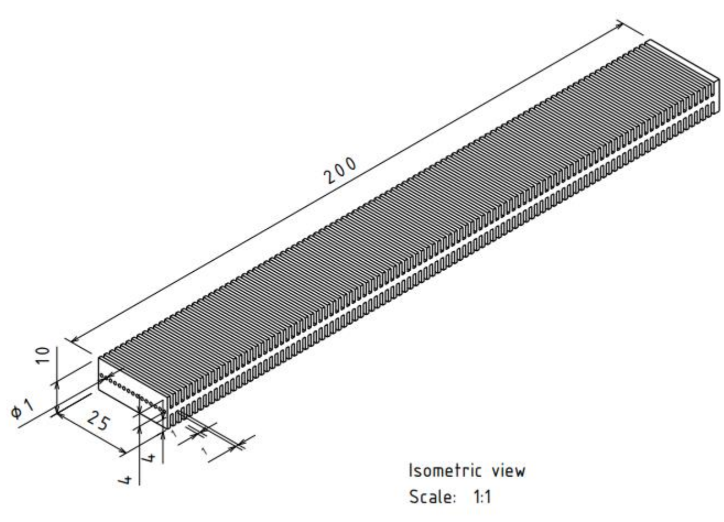

Figure 3 Geometry of MCHX Module

Numerical Model

A numerical model has been studied using software tools (ANSYS Workbench 16.0 and Solidworks) (Barhatte and Lele, 2020). The geometric specifications used for the analysis are depicted in (Figure 3). To achieve a uniform distribution of fluid flow and to avoid any pressure variation at the inlet head, two-pass Microchannels have been used. Considering the circular surface of Microchannels, tetrahedral elements were set for mesh. The minimum size of the element was set to 0.001 mm, and the maximum size was set to the default value of 0.1 mm. Y+ values were set to less than 10 so as to ensure accuracy in prediction near the walls. Post-processing was accomplished using ANSYS Workbench16.0. A total temperature drop of 9OC was observed. The mass flow rate for the coolant was maintained at 0.05 kg/s.

Performance testing

The control parameters are the mass flow rate of the coolant fluid on the Microchannel side and the air velocity on the fin side. Performance testing determines how MCHX works with stability under the heat load of 3000 W. However, the coolant is a mixture of water and Glycol in various concentrations. It is ensured that all the measuring instruments are properly connected to the experimental setup and the whole setup is isolated from electrical and mechanical disturbances. The overall methodology of performance testing is shown in (Figure 4). The experimental setup integrated with the peripheral devices is indicated in the line diagram as shown in (Figure 5).

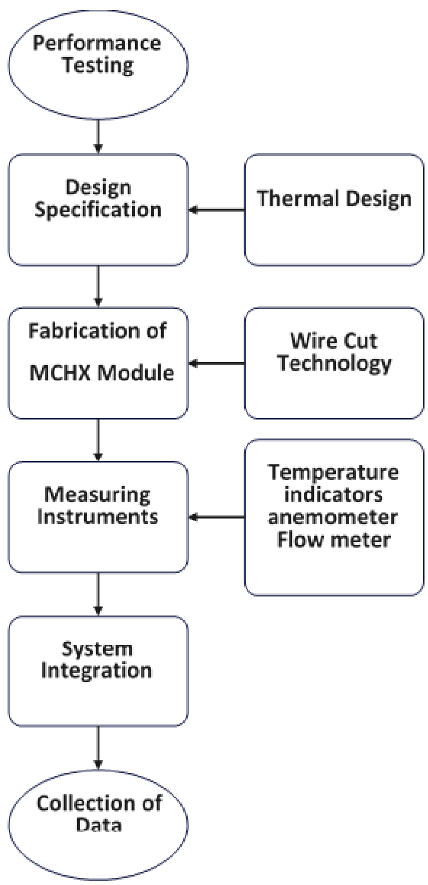

Figure 4 Methodology for Performance Testing

A copper stock of size 200 mm x 20 mm is used for the Microchannel module. The copper stock was exactly cut into two pieces of 200 mm x 10 mm using a wire-cut machine. Semicircular channels of 1 mm diameter were then cut on the face of each stock using a wire-cut machine. The two copper stocks with semicircular channels on their face were then brazed together to give a Microchannel module of 200 mm x 20 mm with 1 mm diameter channels. An electromagnetic flow meter with a measurement sensor having a range of 1-30 LPM is used to measure fluid flow. A 5-blade Aluminium cast blower fan is used for airflow and a variable speed 3-HP 2800 RPM AC motor with a variable frequency drive (VFD) controller regulates the speed of the blower fan. A Digital Hot Wire Anemometer with a range of 2-30 m/s measures air velocity. Five K-type thermocouple sensors are used with a range of up to 350OC.

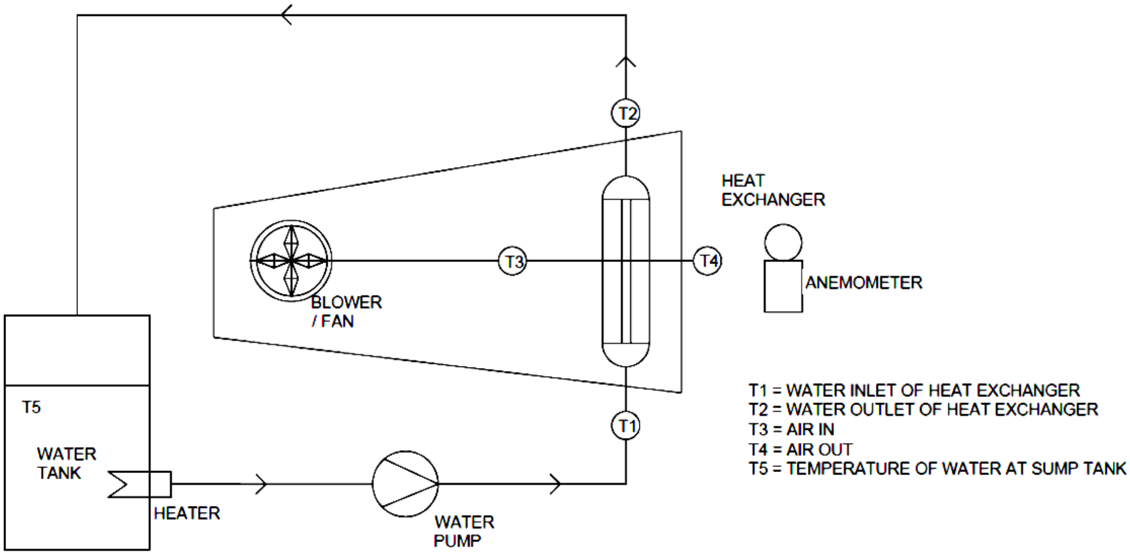

Figure 5 Line Diagram of the Experimental Setup

Data analysis

The accuracy, precision, and general validity of experimental measurements are all evaluated using data analysis. These random errors usually follow a certain statistical distribution (Holman, 2012). This necessitates the analysis of data for proper validation. The detailed methodology of data analysis is shown in (Figure 6). The common-sense analysis is performed using Chauvenet's criterion. The data points with the highest deviation are identified, and such dubious points are eliminated from the data for further analysis.

Regression

The Nu-Pr correlation is not linear. Moreover, the rate of heat transfer and, thus, the change in the temperature are very fast at the initial stage. This warrants the use of logarithmic regression. Therefore, the correlation has to be first reduced to a linear form using a logarithm, as shown below.

$Nu = f(Re, Pr)$ (16)

$Nu = CRe^{m} \times Pr^{n}$ (17)

$Ln\,(Nu) = Ln(C) + mLn(Re) + nLn(Pr)$ (18)

The regression is performed using MS Excel. C, m, and n are found to be 0.1123, 0.8 and 0.008. The Y-intercept is found to be -2.18622, and its antilogarithm is 0.1123, which is the coefficient of Re and Pr in the correlation. Thus, Nu-Pr correlation becomes

$Nu = 0.1123Re^{0.8}Pr^{0.008}$ (19)

Uncertainty analysis

Uncertainty in the predicted value of the Nusselt number depends on the uncertainty in the measurements of temperature and flow velocity. These uncertainties are purely a function of the error in the measurement of these properties.

$Nu = f(T, v)$ (20)

$W_{Nu} = [(W_T)^2 + (W_v)^2]^{1/2}$ (21)

W_{Nu} is the uncertainty in Nusselt number and W_T and W_v the uncertainties in temperature measurement and velocity measurement. The accuracy of temperature measurement is $\pm 3^0$ over a range of 400OC and therefore the uncertainty in temperature measurement is calculated to be 0.75% and that of flow measurement is found to be 0.5%. Using (21), the uncertainty in the Nusselt number calculation is found to be 0.901% signifying that the error in the calculated value of the Nusselt number would be ± 0.9 in the range of 1-100.

ANN Techniques

Preprocessing of data needed for ANN is performed using common-sense analysis. In neural networks, the data is split into two, sometimes three smaller, sets for training and validation. The ratio in which the data is partitioned into two parts is generally 80:20 percent (Qiu et al., 2020). This implies that 80 % of the data is used for training purposes, whereas 20% is used to validate the model obtained by ANN. The scaled Conjugate Gradient (SCG) and Levenberg-Marquardt (LM) algorithms are the two models used for the analysis, which are justified by their fast convergence and relatively good accuracy. The comparative Mean squared error (MSE) using both models is indicated in (Figure 7). SCG performs better than the LM model for the data under study. It is observed that convergence is faster for the SCG model even if the input-output characteristics are nonlinear. The solution converges once the set MSE criterion is achieved or the solution converges once the set number of iterations is achieved. Convergence takes place in 200 Epoch as shown in (Figure 8). The criterion set for the convergence is MSE. For the set MSE, 205 iterations are performed, and then the solution converges.

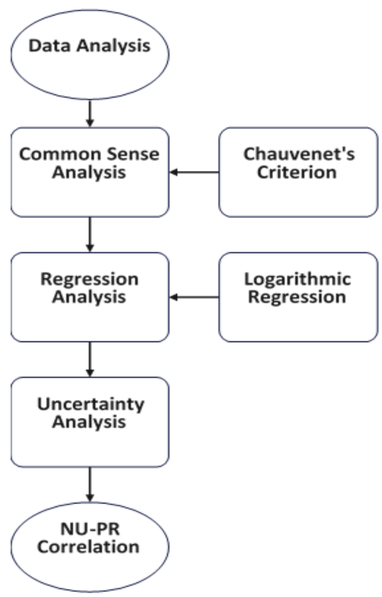

Figure 6 Methodology of Data Analysis

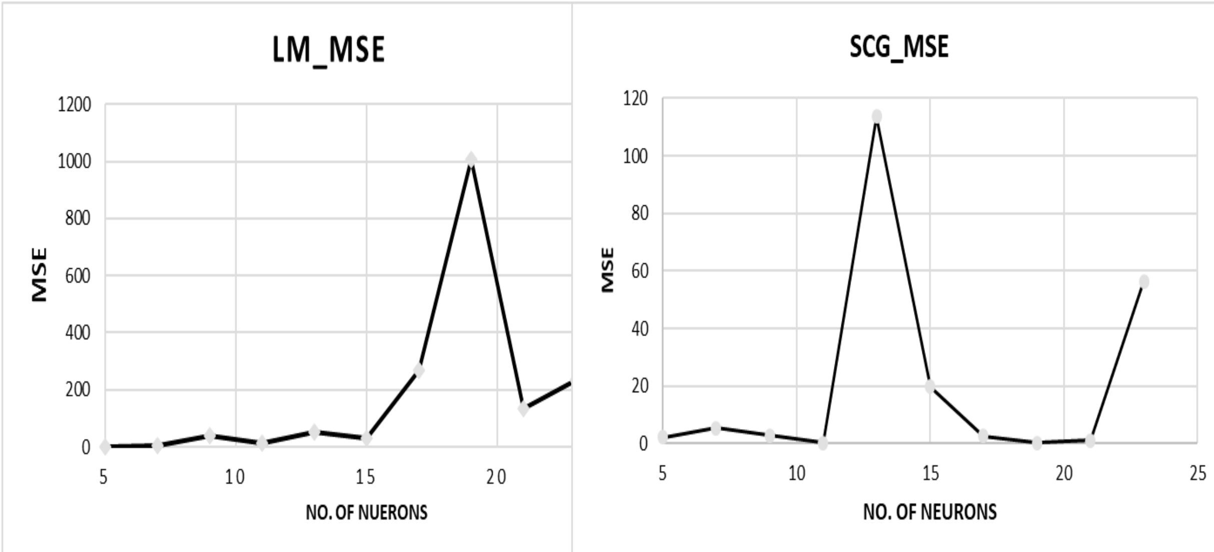

Figure 7 Comparison of LM and SCG Model

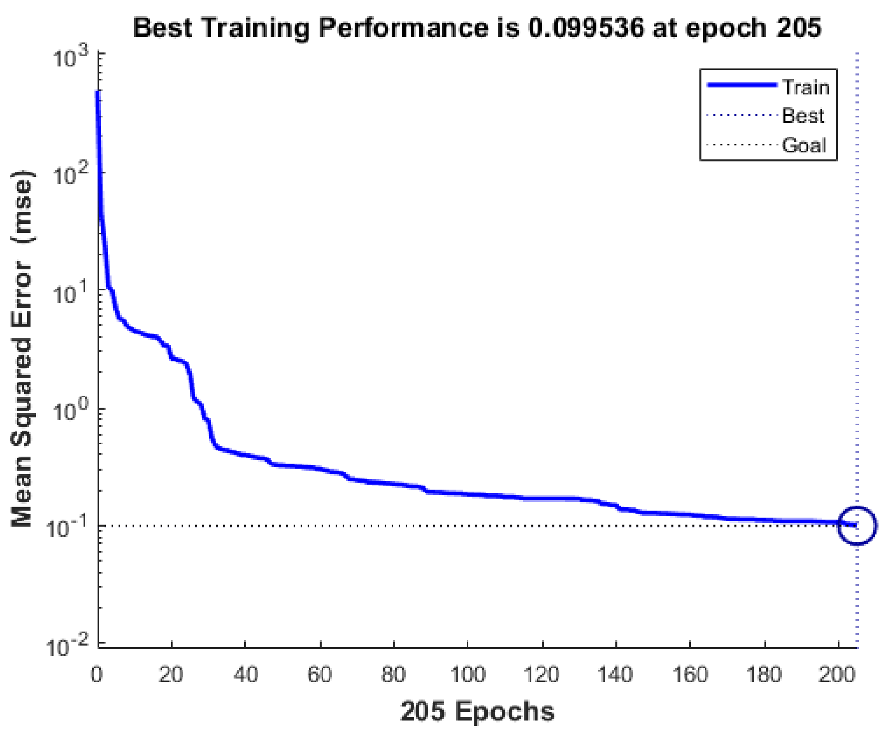

Figure 8 MSE using SCG algorithm

3. RESULTS AND DISCUSSION

The outcome of the computational analysis is in good agreement with the experimental results. There is considerable effect of hydraulic diameter on MCHX performance, and it is also supported by literature (Peng et al., 1995). The outcome of the analytical procedure indicates that heat transfer increases with a decrease in the hydraulic diameter of Microchannels. However, the pressure drop also increases as shown in (Figure 9). It is a known fact that any increase in pressure loss leads to more pumping power requirements. The heat transfer rates go on increasing as the hydraulic diameter is reduced (Masoud et al., 2011). On the other hand, pumping power requirements also increase exponentially.

Therefore, a hydraulic diameter of 0.001m indicates a golden mean between the pumping power requirements and heat transfer as indicated in (Figure 10). A quantitative comparison of the initial size estimation outcomes and the computational analysis outcomes is shown in Table 3 and is in good agreement and the variation is within 9.64%. The outcome of analytical and numerical analysis forms a basis for the performance testing of MCHX as indicated in (Figure 11). The performance testing and regression analysis outcome are validated by the ANN techniques with less than 3% variance. This is understood well from (Table 4).

Table 3 Analytical Vs the numerical results

Sr. No.	Variable	Analytical	Numerical
1	Mass flow rate	0.005 Kg/S	0.005Kg/S
2	Inlet temp	85OC	85OC
3	Outlet Temp	74OC	76OC
4	Pressure drop	1.77 atm	1.2 atm
5	Pumping power	0.6085	0.6755
6	HTC over Microchannels	39473 W/m2K	36000 W/m2K

Table 4 Results obtained by Experiment and ANN

Re	Pr	Expt Nu	Regression	ANN-SCG	ANN-LM	%Variance
6773.2	4.24	131	130.4	130.4227	130.4363	0.432
6456.1	4.6	128	125.5	125.3091	125.4519	2.031
6930	4.6	132	132.8	132.9736	133.0367	-0.779
7198.1	4.6	135	136.9	137.0873	137.2805	-1.661
6389.6	4.95	124	124.4	124.7281	124.5691	-0.457
6487.6	4.95	127	126	125.7798	125.9369	0.844
6720	4.95	129	129.5	129.3312	129.505	-0.390
6795.6	4.95	131	130.7	130.6971	130.7633	0.181
6918.1	4.95	136	132.6	132.8983	132.8683	2.357
6951	4.95	135	133.1	133.4599	133.4358	1.172
7702.8	4.95	136	135.9	134.8194	134.8678	0.839
7702.8	4.95	131	144.5	144.3167	144.3576	-9.253
8251.6	4.95	133	152.7	152.7999	152.6505	-12.873
8323	5.43	148	153.7	153.8087	153.5837	-3.636
6706	5.43	132	129.3	129.1278	129.3454	2.052
6850.2	5.43	133	131.6	131.7583	131.7635	0.938
8038.8	5.43	146	149.5	149.5249	149.8433	-2.565
6439.3	5.45	128	125.2	125.1112	125.3147	2.143
8230	6.07	131	152.4	152.0897	152.5439	-14.123
8235	6.07	126	152.4	152.1696	152.6067	-17.435
6526.1	6.07	131	126.5	126.3323	126.6355	3.447
6846	6.07	137	131.5	131.7976	131.7704	3.969

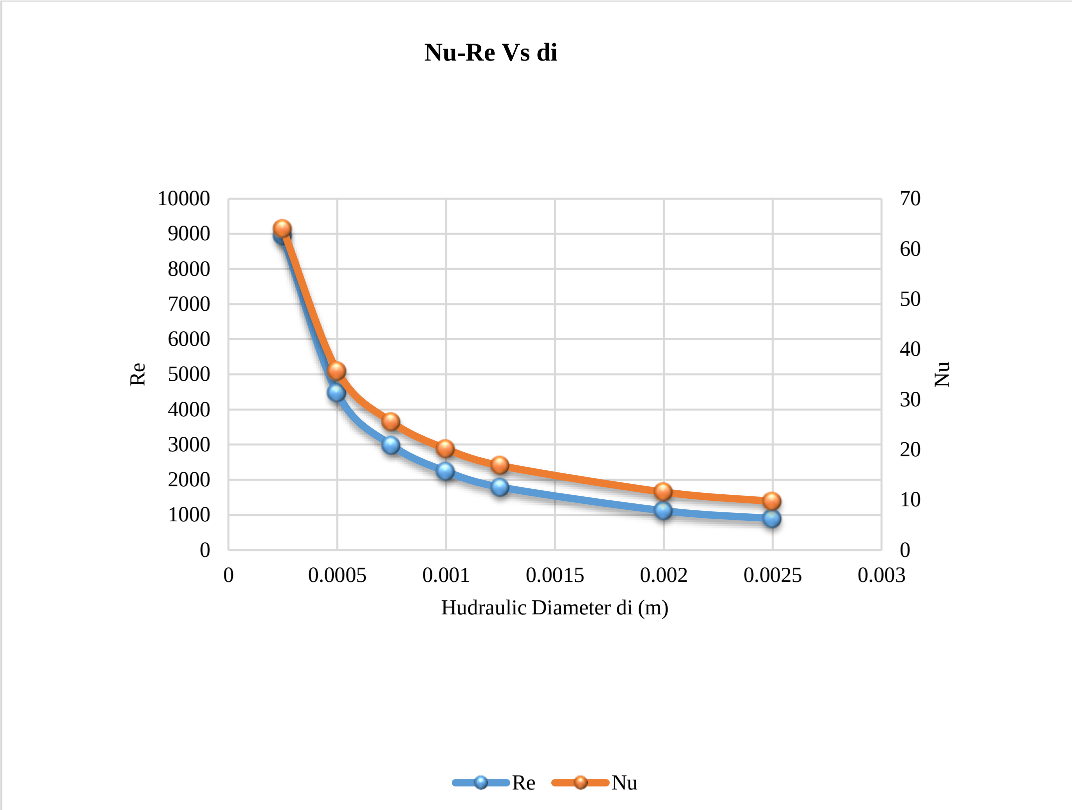

Figure 9 Graph of Re-Nu Vs Hydraulic Diameter

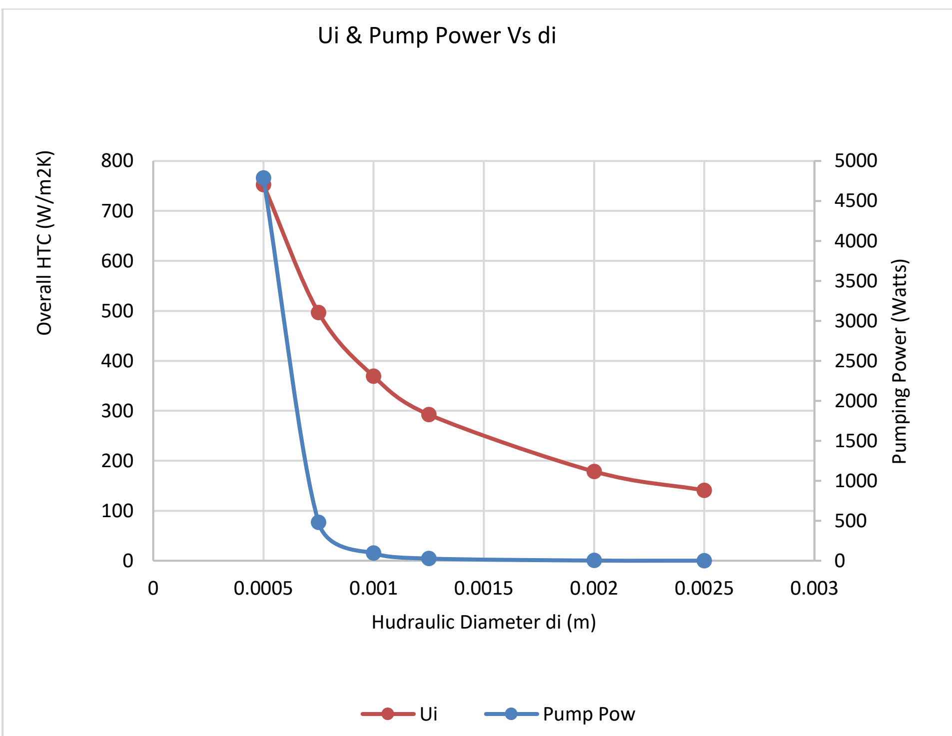

Figure 10 Graph of overall HTC and Pumping Power Vs Hydraulic Diameter

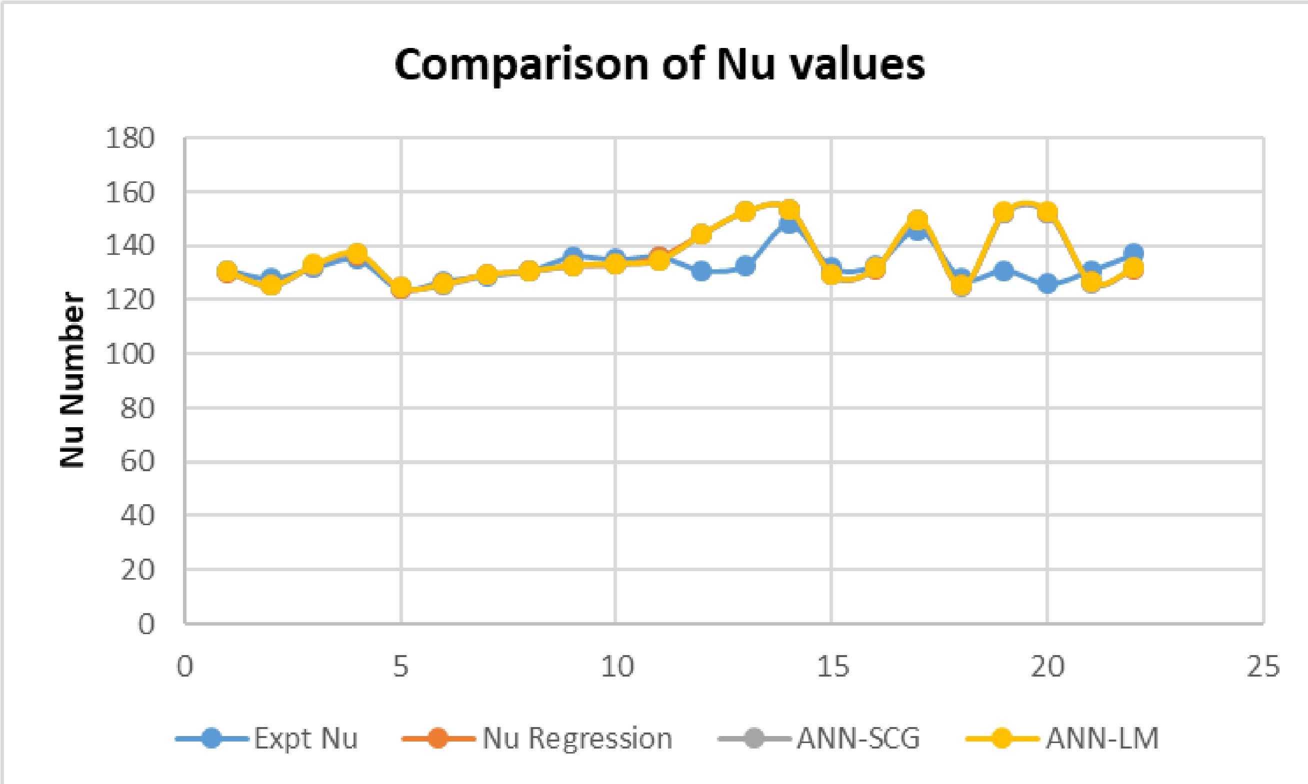

Figure 11 Comparison of Nu Number by experimentation, Regression, and ANN

5. CONCLUSIONS

The heat transfer performance of liquid flowing through Microchannels with circular sections was investigated analytically and numerically. The findings of the computation analysis and the outcomes of the analytical technique closely align; the level of agreement is within 5%. Thus, creating a strong base for the fabrication and experimental investigations. The thermal design-based rerating perfectly satisfies the desired heat duty. The frontal size of an existing heat exchanger is 300 x 250 mm; however, it has been decreased to 150 x 200 mm with the usage of Microchannels. The size of the heat exchanger is decreased by 60% when Microchannels are used in place of standard flat tubes for the same heat duty. The use of MCHX increases heat transfer rates, consequentially reducing the size of the heat exchanger. Thus, they are suitable for various limited-space applications such as automobile radiators and electric vehicle cooling jackets. The Nu-Pr correlation established for the Air-Glycol MCHX in single-phase applications is $Nu = 0.1123Re^{0.8}Pr^{0.008}$

List of abbreviations

Abbreviations			
OC	Degree Centigrade	MCHX	Microchannel Heat Exchanger
A	Area for heat transfer	mm	Millimetre
Ac	Free flow area	η	Efficiency
ANN	Artificial Neural Networks	*Ncf*	Number of Microchannels in flat tube
atm	Pressure in Atmosphere	N*f*	Number of fins
BC	Boundary Conditions	*Nft*	Number of flat tubes
BL	Boundary Layer	NTU	Number of Transfer Units
C	Specific Heat	Nu	Nusselt Number
CFD	Computational Fluid Dynamics	Ø	Diameter
D*h*	Hydraulic Diameter	P	Pressure

ε	Effectiveness	Pa	Pascal
f	Friction factor	Pr	Prandtl Number
H	Height of MCHX	*pt*	Tube pitch
HTC, h	Heat Transfer Coefficient	Q	Heat Transfer Rate
Hx	Heat Exchanger	Re	Reynolds Number
IC	Internal Combustion	S	Seconds
K	Kelvin	T	Temperature
Kg	kilogram	U	Overall HTC
KMPH	Kilometres Per Hour	*W*	Width of MCHX
L	Length of MCHX	W	Watt
LMTD	Log Mean Temperature Difference	WMCHS	Wavy Microchannel Heat Sink
LPM	Liters per minute	µm	Micrometre
MCHS	Microchannel Heat Sink		
Suffix/Prefix			
f	Fin	max	Maximum
ft	Flat Tube	min	Minimum
H	Hot	O	Outlet/outer surface
I	Inlet/inner surface		

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Authors' Contributions

SB is responsible for conceptualization, data curation, research, methodology, writing the first draft, writing the review, and editing. ML oversaw and handled the article writing tasks. Additionally, ML offered the tools required to finish the paper. The final manuscript was read and approved by all writers.

Informed consent

Not Applicable.

Ethical Approval

Not Applicable.

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Conflict of Interest

The author declares that there are no conflicts of interests.

Data and materials availability

All data associated with this study are present in the paper.

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