

Effect of Parameters on Thermal and Fluid Flow Behavior of Battery Thermal Management System

Asif Afzal^{1*}, Awatef Abidi^{2,3,4}, Mohammed Samee AD⁵, Abdul Razak RK¹, Manzoore Elahi M Soudagar⁶, Ahamed Saleel C⁷

¹Department of Mechanical Engineering, P. A. College of Engineering (Affiliated to Visvesvaraya Technological University, Belagavi), Mangaluru, INDIA.

²Physics Department, College of Sciences Abha, King Khalid University, Saudi Arabia.

³Research Laboratory of Metrology and Energy Systems, National Engineering School, Energy Engineering Department, Monastir University, Monastir City, Tunisia.

⁴Higher school of Sciences and Technology of Hammam Sousse, Sousse University, Tunisia.

⁵Department of Mechanical Engineering, N.K.Orchid College of Engineering and Technology, Solapur, Maharashtra, INDIA.

⁶Department of Mechanical Engineering, Faculty of Engineering, University of Malaya, 50603, Kuala Lumpur, Malaysia

⁷Department of Mechanical Engineering, College of Engineering, King Khalid University, PO Box 394, Abha 61421, Kingdom of Saudi Arabia

^{1*}Corresponding author: asif.afzal86@gmail.com

Abstract

In modern electric vehicles the thermal stability problems associated with Lithium-ion (Li-ion) battery system is of major concern. Proper battery thermal management systems (BTMS) is required to ensure safety and efficient performance of battery cells. A realistic conjugate heat transfer and fluid flow analysis of Li-ion prismatic battery cell is performed. The flow of air as coolant, is laminar, flowing between the heat generating battery cells. The effect of few important working parameters like volumetric heat generation (\bar{S}_q), conduction-convection parameter (ζ_{cc}), Reynolds number (Re), Aspect ratio (Ar), and spacing between the cells (\bar{W}_f) is investigated in this work. For the wide range of parameters considered, the temperature variations in battery cell and coolant is carried out. Focusing mainly on effect of Re and \bar{W}_f , behavior of local Nusselt number (Nu_x), local friction coefficient ($C_{f,x}$), average Nusselt number (Nu_{avg}), average friction coefficient ($C_{f,avg}$), maximum temperature, mean fluid temperature, heat removed from the lateral surface of cell are discussed. Nu_{avg} increased with increase in Re but decreased with increase in \bar{W}_f , whereas $C_{f,avg}$ decreased with increase in Re and \bar{W}_f . It is also found that their exists an upper and lower limit on Re and \bar{W}_f above and below which the change in $C_{f,avg}$ and Nu_{avg} is negligible. Maximum temperature is significantly influenced at low Re and for all \bar{W}_f . From the lateral surface of battery over which the coolant flows, more than 96% of heat generated in cell is removed.

Keywords: *Nusselt number; friction coefficient; spacing; Battery cell; heat generation; thermal behavior; Reynolds number.*

1. Introduction

Recent development in automobile technology has given the most reliable and clean source of energy in the form of electrochemical energy [1]–[3]. Lithium ion (Li-ion) batteries are the most widely used source of electrochemical form of energy. But again, this source of electrochemical energy requires better thermal management system, so that it can operate safely without creating damage to the devices and vehicles in which it is being used [4]–[6]. Thermal management of heat generating Li-ion batteries plays a vital role for improved performance, high efficiency, long service life and safer operation. If the heat generated in the battery cell is not removed appropriately, additional heat-generating exothermic reactions are triggered by the temperature rise. This results in an increased temperature of the battery cell which

causes a thermal runaway situation. Therefore, the heat generated in the battery cell should be adequately removed, adopting an effective cooling mechanism [7], [8].

Chen and Evan [9] investigated the problem of thermal management of Li-ion battery cell having even heat generation rate. Heat transfer characteristic of three-dimensional battery cell was demonstrated with some suitable cooling method. Baker and Verbrugge [9] proposed a two dimensional model of Li-ion battery cell to study the causes for thermal runaway. In this study they mainly focused on determining the strength and location of maximum temperature in a battery cell body during the process of discharging and charging. Ramadass et al. [10] performed capacity fade analysis of Sony 18650 Li-ion battery cell. Capacity fade of 30 and 36% were obtained at room temperature and 45°C respectively. After 490 cycles, capacity fade of 70% was obtained at 70°C. They concluded that, increase in temperature affects the capacity fading of Battery cells. Mahamud and Park [11] performed thermal management analysis of cylindrical Li-ion cells with reciprocating air flow. Lumped capacitance model was considered for the analysis of Li-ion cells assuming uniform heat losses. Results obtained show that reciprocating air flow arrangement reduces maximum temperature by 1.5°C. Tran et al. [12] used the combined method of heat pipe cooling of battery module for better thermal performance and high efficiency. Natural convection effects are insufficient to maintain the temperature of battery within the operating range. Combination of heat pipes along with ventilation provided better thermal management of battery cells [13], [14].

Rao et al. [15] investigated a cooling method for battery system in which they used coupled technique of PCM and minichannels. The obtained results from this study suggest that the combined effects of PCM/minichannels give more satisfactory results as compared to PCM alone. Chen et al. [16] compared four different methods, direct air cooling, direct liquid cooling, indirect liquid cooling and fin cooling of battery pack having Li-ion cells. The best and optimal method is suggested in this three dimensional numerical simulation in terms of controlling the maximum range of temperature within the battery pack, temperature uniformity, and power efficiency. Basu et al. [17] suggested a correlation for predicting the temperature of each individual cell in a three dimensional packed battery module which can eliminate the usage of temperature sensors to greater extent. The study also conclude that the conductivity of a conduction element which acts as a screen between liquid coolant and battery cell plays very prime role in efficient heat transfer. Chalise et al. [18] investigated a conjugate heat transfer problem pertinent to liquid cooling of Li-ion battery pack with analytical approach. Different fluid and flow velocity were considered to determine the efficient performance of battery pack. Li et al. [19] analyzed thermal behavior of Li-ion battery system cooled by air. Velocity of air up to 30m/s was considered for analysis and a reduced-order model was developed to predict the maximum temperature in battery module. The operating parameters of battery module and their influence on the thermal and fluid flow behavior of different battery models have been discussed specifically in the literature [20]–[23].

Thermal management of battery cells considering some active/passive cooling techniques are commonly reported as seen from the above literature review. Using commercial software like Ansys-Fluent most of the research works were carried to analyze the coolant flow behavior and its arrangement. While the experimental studies mainly focused on charging and discharging effect on thermal behavior of battery cells considering only the solid cell. However, the realistic conjugate condition at the solid battery cell and coolant interface is not reported, except Chalise et al. [18]. Chalise et al. [18] performed a conjugate analysis of battery cell considering few coolants and flow velocity by analytical method and external flow. Boundary layer equation was used by Chalise et al. [18] to conduct the analytical analysis focusing on maximum temperature and temperature distribution keeping the parallel cell plate far away from the vicinity of considered battery cell. Giving consideration to some of the above stated facts and taking motivation from them, the present investigation is intended to perform a numerical simulation by

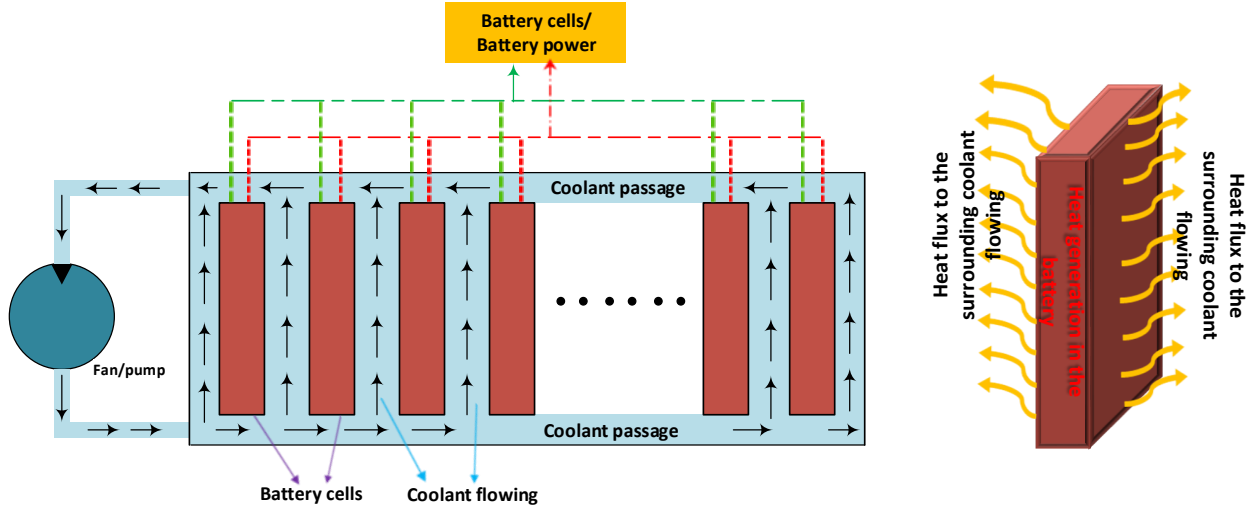
considering the parametric effect on a realistic conjugate thermal behavior of prismatic battery cell cooled by air. The parameters considered are Spacing between cells (\bar{W}_f), volumetric heat generation (\bar{S}_q), conduction–convection parameter (ζ_{cc}), Reynolds number (Re), and Aspect ratio (Ar). Accordingly, an indigenous Finite Volume Method (FVM) based code was developed to perform the analysis. The effect of \bar{W}_f and Re are given a special attention on the variation of temperature distribution, friction coefficient, Nusselt number, maximum temperature and heat removed from the lateral surface of cell.

Nomenclature	
Ar	Aspect ratio of battery cell
C_f	Dimensionless friction coefficient
L	Length of battery cell (m)
k	Thermal conductivity ($W\ m^{-1}\ k^{-1}$)
l_o	Length of extra outlet fluid domain (m)
l_i	Length of extra fluid domain (m)
h	Convective heat transfer coefficient ($W\ m^{-2}\ k^{-1}$)
L_o	Dimensionless length of extra outlet fluid domain
L_i	Dimensionless length of extra inlet fluid domain
Nu	Nusselt number
q'''	Volumetric heat generation ($W\ m^{-3}$)
\bar{q}	Non-dimensional heat flux
\bar{S}_q	Dimensionless volumetric heat generation
Pr	Prandtl number
Re	Reynolds number
T	Temperature ($^{\circ}C$)
T_o	Maximum allowable temperature of battery ($^{\circ}C$)
\bar{T}	Non-dimensional temperature
u	Velocity along the axial direction (ms^{-1})
U	Non-dimensional velocity along the axial direction
u_{∞}	Free stream velocity (ms^{-1})
v	Velocity along the transverse direction (ms^{-1})
V	Non-dimensional velocity along the transverse direction
w	Half width (m)
\bar{W}	Non-dimensional width
x	Axial direction
X	Non-dimensional axial direction
y	Transverse direction
Y	Non-dimensional transverse direction
Greek symbols	
α	Thermal diffusivity of fluid ($m^2\ s^{-1}$)
ν	Kinematic viscosity of fluid ($m^2\ s^{-1}$)
ρ	Density of fluid ($kg\ m^{-3}$)
ζ_{cc}	Conduction-convection parameter
μ	Dynamic viscosity (Nm^2s^{-1})
Subscripts	
c	Center
f	Fluid domain
avg	Average
s	Solid domain (battery cell)
$surf$	Surface
∞	Free stream
m	Mean

2. Numerical analysis

2.1 Physical model

A parallel channel with liquid coolant flow is employed to cool the battery cells during operation. The developed thermal model is then used to analyze the thermal behavior of the battery cell for various parameters, as shown in Figure 1. The fluid flow inside the channel is commonly in laminar regime owing to the low velocity of the flow inside the channel. Based on some previous literature survey, the heat generation is assumed uniform [24], [25], [26]. The governing equations were solved by considering steady state as the heat generated during charging or discharging routes taken over a long time is assumed constant.



(a) Compact battery module with stack of batteries and coolant passages

(b) Heat flux from the battery surface

Figure 1. Schematic view of arrangement of battery cells with air circulation fan and heat generation in batteries.

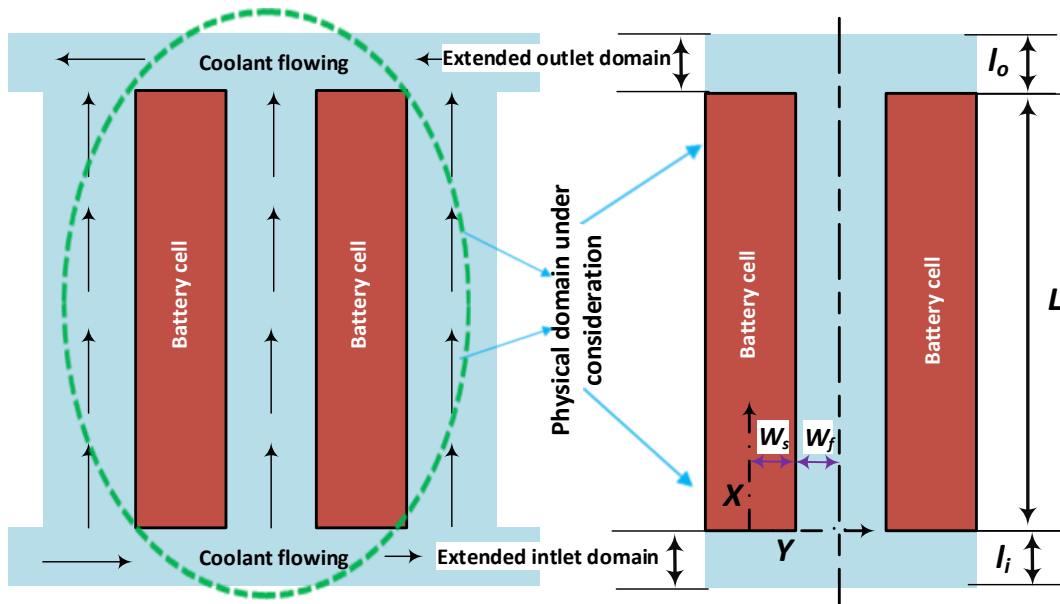


Figure 2. The symmetric battery (prismatic cell) and coolant flow domain considered for computational analysis.

2.2 Mathematical model

The governing equations describing the heat transfer process when discharging/charging the Li-ion battery cell is given by

$$k_s \nabla^2 T + q''' = 0 \quad (1)$$

Where q''' is volumetric heat generation term.

The governing equations for two-dimensional, steady, incompressible, laminar, forced convection flow in the fluid domain are continuity equation, x and y momentum equations and equation of energy, which are as follows:

$$\nabla u = 0 \quad (2)$$

$$(u\nabla u) = -\frac{1}{\rho}\nabla p + \mu\nabla^2 u \quad (3)$$

$$u\nabla T = \alpha\nabla^2 T \quad (4)$$

The above equations are non-dimensionalized using the following set of normalizing parameters:

$$\begin{aligned} \bar{S}_q &= \frac{q'' w_s^2}{k_s(T_o - T_\infty)}, & C &= 4Ar^2, & \bar{T} &= \frac{T - T_\infty}{T_0 - T_\infty}, & L_i &= \frac{l_i}{L}, & L_o &= \frac{l_o}{L}, & X &= \frac{x}{L}, \\ U &= \frac{u}{u_\infty}, & V &= \frac{v}{u_\infty}, & P &= \frac{p}{\rho u_\infty^2}, & A_r &= \frac{L}{2w_s}, & Y_s &= \frac{y_s}{w_s}, & Y_f &= 1 + \frac{y_f}{L}, \\ \bar{W}_f &= \frac{w_f}{L}, & \zeta_{cc} &= \frac{k_f}{k_s} \left[\frac{w_s}{L} \right], & Re &= \frac{u_\infty L}{\nu}, & Pr &= \frac{\nu}{\alpha} \end{aligned} \quad (5)$$

The final set of non-dimensionalized governing equations turns out to be:

$$\frac{\partial^2 \bar{T}_s}{\partial X^2} + C \frac{\partial^2 \bar{T}_s}{\partial Y_s^2} + C\bar{S}_q = 0 \quad (6)$$

$$\nabla U = 0 \quad (7)$$

$$U\nabla U = -\nabla P + \frac{1}{Re}\nabla^2 U \quad (8)$$

$$U\nabla \bar{T}_f = \frac{1}{RePr}\nabla^2 T_f \quad (9)$$

The normalized governing equations given above are coupled partial differential equations (elliptic in nature) and thus require specification of boundary conditions on all the boundaries. At the inlet, the axial component of the velocity and the inlet temperature distribution are understood to be uniform.

$$X = 0; \quad 0 \leq Y_s \leq 1, \quad U = 0, \quad V = 0, \quad \bar{T}_s = 0$$

At the left side, thermal symmetry boundary condition. Hence,

$$Y_s = 0; \quad 0 \leq X \leq 1, \quad \frac{\partial \bar{T}_s}{\partial X} = 0$$

At the interface, continuity of temperature boundary condition is used, i.e.

$$Y_s = 1; \quad 0 \leq X \leq 1, \quad \bar{T}_s = \bar{T}_f$$

At the top of the domain, heat transfer is assumed to be negligibly small, as at the exit of the cell cathode and anode current collector are present [27]. Therefore,

$$X = 1; \quad 0 \leq Y_s \leq 1, \quad \frac{\partial \bar{T}_s}{\partial Y_s} = 0,$$

An enlarged extended computational domain of finite section, as shown in Figure 2, has been used in this analysis to mimic the free-stream conditions of the fluid flow far away from the region of thermal disturbance induced by the battery cells and to capture the diffusive effects of the low-Prandtl fluids. Its

height is chosen as l_i at the inlet and l_o at the exit, based on a few preliminary studies performed. So the most appropriate boundary condition at the extended domain is as follows:

$$\begin{aligned}
Y_f = 1; \quad -L_i \leq X \leq 0 \quad \text{and} \quad L \leq X \leq L_o, \quad \frac{\partial \bar{T}_f}{\partial Y_f} = 0, \quad \frac{\partial U}{\partial Y_f} = 0, \quad V = 0 \\
Y_f = 1; \quad 0 \leq X \leq L, \quad \frac{\partial T_f}{\partial Y_f} = \frac{1}{\zeta_{cc}} \frac{\partial T_s}{\partial Y_s}, \quad U = 0, \quad V = 0 \\
Y_f = 1 + \bar{W}_f; \quad -L_i \leq X \leq (L + L_o), \quad \frac{\partial \bar{T}_f}{\partial Y_f} = 0, \quad V = 0, \quad \frac{\partial U}{\partial Y_f} = 0 \\
X = -L_i; \quad 1 \leq Y_f \leq (1 + \bar{W}_f), \quad \bar{T}_s = 0, \quad U = 1, \quad V = 0 \\
X = L + L_o; \quad 1 \leq Y_f \leq (1 + \bar{W}_f), \quad \frac{\partial \bar{T}_s}{\partial X} = 0, \quad \frac{\partial U}{\partial Y_f} = 0, \quad V = 0
\end{aligned} \tag{10}$$

Coolant average Nusselt number (Nu_{avg}) is calculated using Equation 12. Based on Newton's law of cooling at the cell and coolant interface:

$$-k \frac{\partial T}{\partial Y} \Big|_{y_s=w_s} = h(T_{y_s=w_s} - T_{f,m})$$

Where $\bar{T}_{f,m}$ is the mean temperature of the fluid domain and \bar{T}_s is the cell surface temperature. Using appropriate dimensionless parameters, the local Nu_x is obtained which is integrated over the cell surface length to get Nu_{avg} .

$$\begin{aligned}
Nu_x &= -\frac{1}{(\bar{T}_{y_s=w_s} - \bar{T}_{f,m})} \frac{\partial \bar{T}}{\partial Y} \Big|_{Y_s=W_s} \\
Nu_{avg} &= -\int_0^1 \frac{1}{(\bar{T}_{y_s=w_s} - \bar{T}_{f,m})} \frac{\partial \bar{T}}{\partial Y} \Big|_{Y_s=W_s} dX
\end{aligned} \tag{12}$$

The friction coefficient (C_f) variation at the cell and coolant interface along the channel length is estimated based on the Newton's law of viscosity given by

$$\tau_s = \mu \frac{du}{dy} \Big|_{y=w_s}$$

And according to the practical definition of the shear stress (τ_s) at the surface is given by

$$\tau_s = C_f \frac{\rho U_m^2}{2}$$

Equating the above equations of τ_s and using the appropriate non-dimensional terms the following Equation 13 is obtained for local friction coefficient ($C_{f,x}$)

$$C_{f,x} = \frac{2}{Re U_m^2} \frac{dU}{dY_s} \Big|_{Y_s=\bar{W}_s} \tag{13}$$

U_m is the mean velocity of the coolant in the channel. The average friction coefficient ($C_{f,avg}$) is further computed using Equation 14

$$C_{f,avg} = \frac{2}{ReU_m} \int_0^1 \frac{dU}{dY_s} \Big|_{Y_s=\bar{W}_s} dX$$

2.3 Numerical solution

The numerical solution of the conjugate problem consisting of energy and momentum equations is obtained by employing staggered grid method of finite volume method (FVM). SIMPLE algorithm is used to solve the coupled momentum and continuity equation to obtain velocity and pressure components. Discretization was done by central differencing scheme for continuity and momentum equations. The diffusion equation of cell domain and energy equation of fluid domain are coupled as the conjugate condition at the cell-fluid interface should be satisfied. Hence they are solved simultaneously using line-by-line gauss seidel method. To evaluate the accuracy of the numerical results, grid-independence tests to check the dependence of the obtained results on the grid resolution was conducted. The considered mesh systems have grid size of 42x82, 62x122, 82x162 in fluid domain and grid size of 82x82, 82x122 and 122x122 in the solid domain respectively. For typical cases calculated in the present work, the numerical results obtained for non-dimensional temperature for different mesh system was less than 5%. The %deviation for the mesh system tested is mentioned in Table 1. The %deviation between mesh 2 and mesh 3 is lower than between mesh 1 and mesh 2. However, to reduce the computational time without affecting the accuracy, a grid size of 62x122 for solid domain and fluid domain are used.

Table 1. Removed heat from the lateral surface for different grid sizes.

Mesh No	Mesh 1	Mesh 2	% deviation	Mesh 3	% deviation
Mesh size	42x82	62x122		82x162	
Heat removed	0.478451	0.484558	1.27%	0.489045	0.92

2.4 Validation

In order to validate the heat transfer and fluid flow results, the problem has been solved for the the external flow problem on nuclear fuel element (Jahangeer et al. [13] and Ramis and Jilani [14]) which is close to present work available in literature as shown Figure 3, having uniform and non-uniform heat generation term was analyzed. A fairly good agreement was obtained of the temperature distribution between their results and the present FVM code was obtained. The temperature along the width of fuel element for uniform ($\bar{S}_q = 0.5$ and 0.75) and non-uniform ($\bar{S}_q = 0.75$) heat generation is plotted. The temperature variation obtained from the both the work is having the same nature. For uniform heat generation the temperature reduction is smooth along the width. While for axial temperature there is sharp rise reaching to peak and then fall as the top portion of fuel element is reached indicating the non-uniform heat generated at the axial center location. Note that the flow was considered to be external flow with $Re=2500$, $Ar=15$, and $\zeta_{cc}=0.35$.

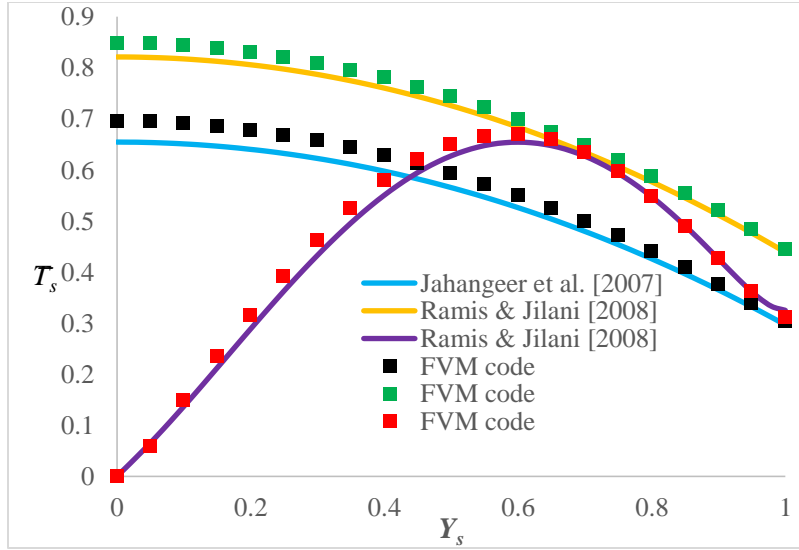
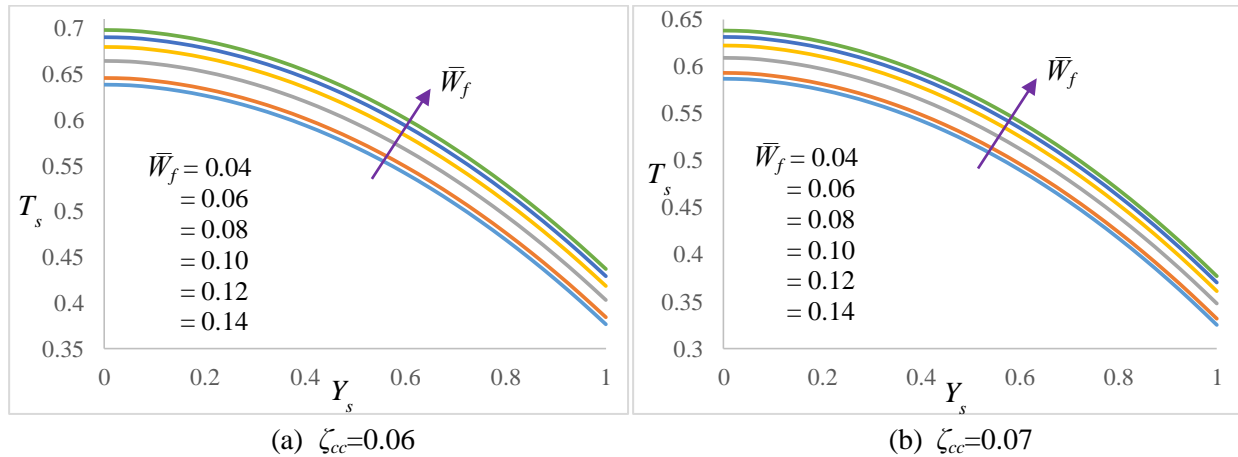


Figure 3. Present FVM code results in agreement with results from Jahangeer et al. [28] and Ramis and Jilani [29].

3. Results and Discussions

3.1. Effect of parameters on temperature distribution in cell

The temperature distribution along the transverse direction of cell at center $X=0.5$ for change in \bar{W}_f and ζ_{cc} is shown Figure 4. The temperature is maximum at the center and smoothly reduces and becomes minimum at the surface of cell. The nature of temperature distribution with increase in \bar{W}_f from 0.04 to 0.14 and $\zeta_{cc}=0.06$ is monotonous which is evident from Figure 5.a. When \bar{W}_f is increased from 0.04 to 0.06, the increase in temperature distribution is least. On further increase in \bar{W}_f from 0.06 to 0.08 the temperature distribution significantly increases and there onwards from $\bar{W}_f=0.08$ to 0.14 the increase in temperature distribution constantly reduces and becomes nearly equal at 0.12 and 0.14. The fact behind this increase in temperature distribution is that at fixed Re when the spacing between cells (\bar{W}_f) is less the coolant flow velocity increases and when the spacing between cells increases the coolant flow velocity reduces as the criteria of continuity of mass should be satisfied.



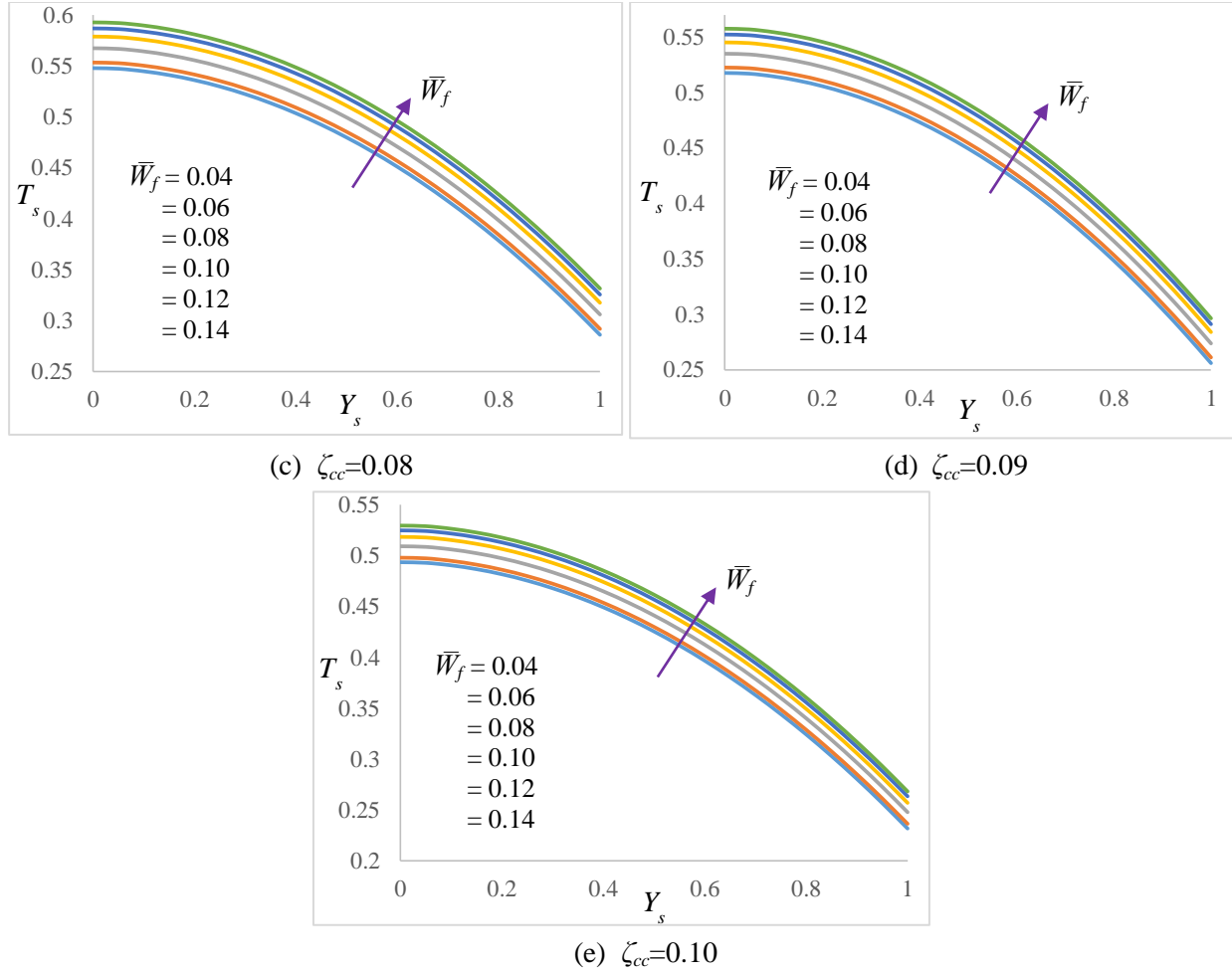


Figure 4. Temperature distribution along the width of cell for changing spacing and ζ_{cc}

At minimum spacing ($\bar{W}_f=0.04$) the heat transfer coefficient of coolant increases carrying more heat away from the surface of cell which causes reduced temperature in cell. There are two ways to look at this result obtained: firstly, when \bar{W}_f is reduced from 0.08 to 0.04 the temperature distribution significantly reduces and then remains almost equal. Hence there exists a lower limiting spacing between the cells below which the temperature reduction is least and only causes more pressure drop. Secondly, when \bar{W}_f is increased from 0.04 to 0.14 the temperature distribution significantly increases and then again remains very nearly same. Hence there exists an upper limiting spacing between the cells above which the increase in temperature is also least and causes rise in overall size of battery module.

From Figure 4.a-4.e it can be seen that with increase in ζ_{cc} from 0.06 to 0.10 the decrease in temperature is of similar trend at each \bar{W}_f . With increase in \bar{W}_f the temperature increases and with increase in ζ_{cc} the temperature reduces. According to the definition of ζ_{cc} , mentioned earlier, an increase in ζ_{cc} at fixed Ar and thermal conductivity of cell means increase in thermal conductivity of coolant which causes more heat dissipation. But it is worth noticing that irrespective of increase or decrease in \bar{W}_f , the decrease in temperature with ζ_{cc} reduces and becomes nearly same for all spacing's at $\zeta_{cc}=0.10$. Thus, if ζ_{cc} is further increased, the decrease/increase in temperature for all spacing's will be negligible. It is worth mentioning at this point that firstly, at any fixed spacing \bar{W}_f , with increase in ζ_{cc} the temperature reduces and becomes nearly same at higher value of ζ_{cc} . Hence there exists an upper limiting value of ζ_{cc} beyond which the decrease in temperature becomes negligible. The reason behind this heat transfer phenomena is that at

fixed \bar{S}_q , ζ_{cc} , Re , Ar , and \bar{W}_f the heat dissipation from the cell to coolant is limited beyond which it cannot be increased. Secondly, the increase in temperature at $\zeta_{cc}=0.10$ with increase in \bar{W}_f is comparatively less. Hence it can be deduced that with further increase in ζ_{cc} above 0.10 will cause negligible decrease in temperature and simultaneous increase in \bar{W}_f will cause negligible increase in temperature. On the other side, if ζ_{cc} is reduced below 0.06, the temperature in cell increases and may cross its permissible limit causing thermal runaway. Similarly, the spacing between cells cannot be reduced as it causes more pressure drop. Therefore, a lower and an upper limit for both ζ_{cc} and \bar{W}_f exists below and beyond which it cannot be reduced and increased respectively.

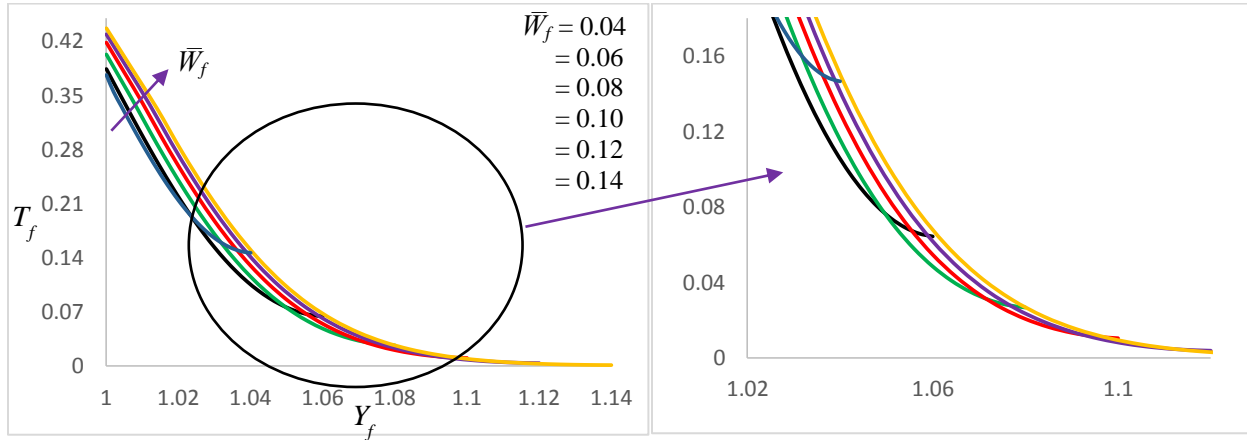


Figure 5. Fluid temperature distribution in the downstream for changing spacing between the cells

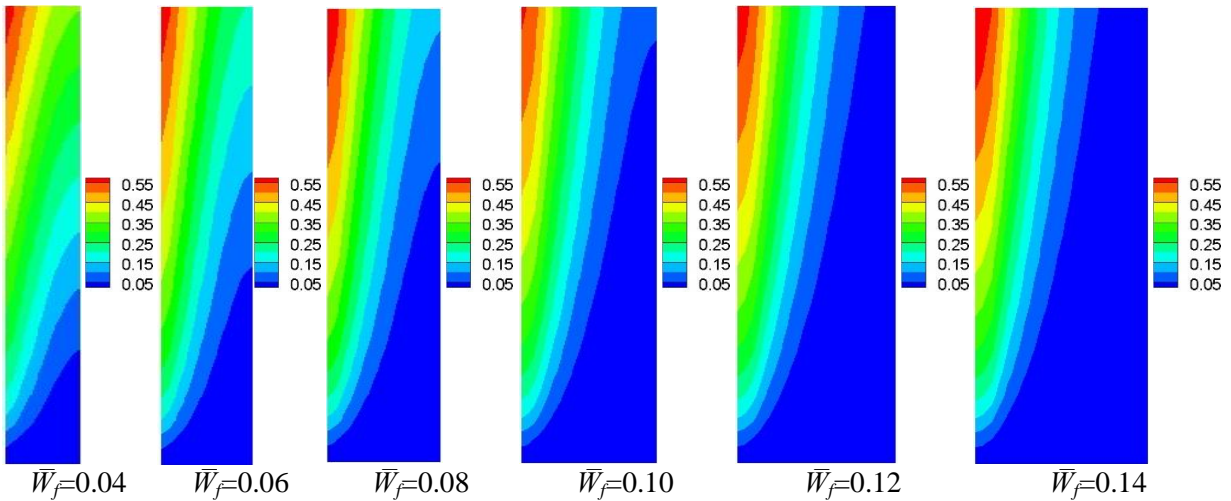


Figure 6. Temperature contours of coolant for different spacing between cells

The variation in temperature distribution of coolant at $X=0.5$ along the transverse direction with increase in \bar{W}_f is shown in Figure 5. The behavior of temperature distribution in coolant with change in spacing is similar to the effect of spacing shown earlier in Figure 5. For the sake of brevity, figures are not added at different ζ_{cc} and Re . The temperature of coolant is least at $\bar{W}_f=0.04$ and increases with increasing \bar{W}_f , however the temperature distribution is monotonous at all spacings. It is noticeable that at lower spacing temperature gradient is steeper and reduces with increasing spacing. Maximum temperature of coolant is at the trailing edge of the cell and coolant interface and it spreads with spacing as shown in Figure 6. At $\bar{W}_f=0.04$, the temperature is more uniformly carried by the coolant due to its increased velocity hence the

temperature gradient is less. At higher spacings due to decrease in velocity the temperature distribution is mainly concentrated near the surface of cell in stream wise direction and therefore steeper temperature gradient. The coolant temperature cannot be decreased by merely decreasing the spacing as it causes more pressure drop. Whereas the coolant temperature increases up to a spacing of 0.10 and then remains nearly same on further increase in spacing.

3.2. Effect of Re and spacing on local friction coefficient and Nusselt number

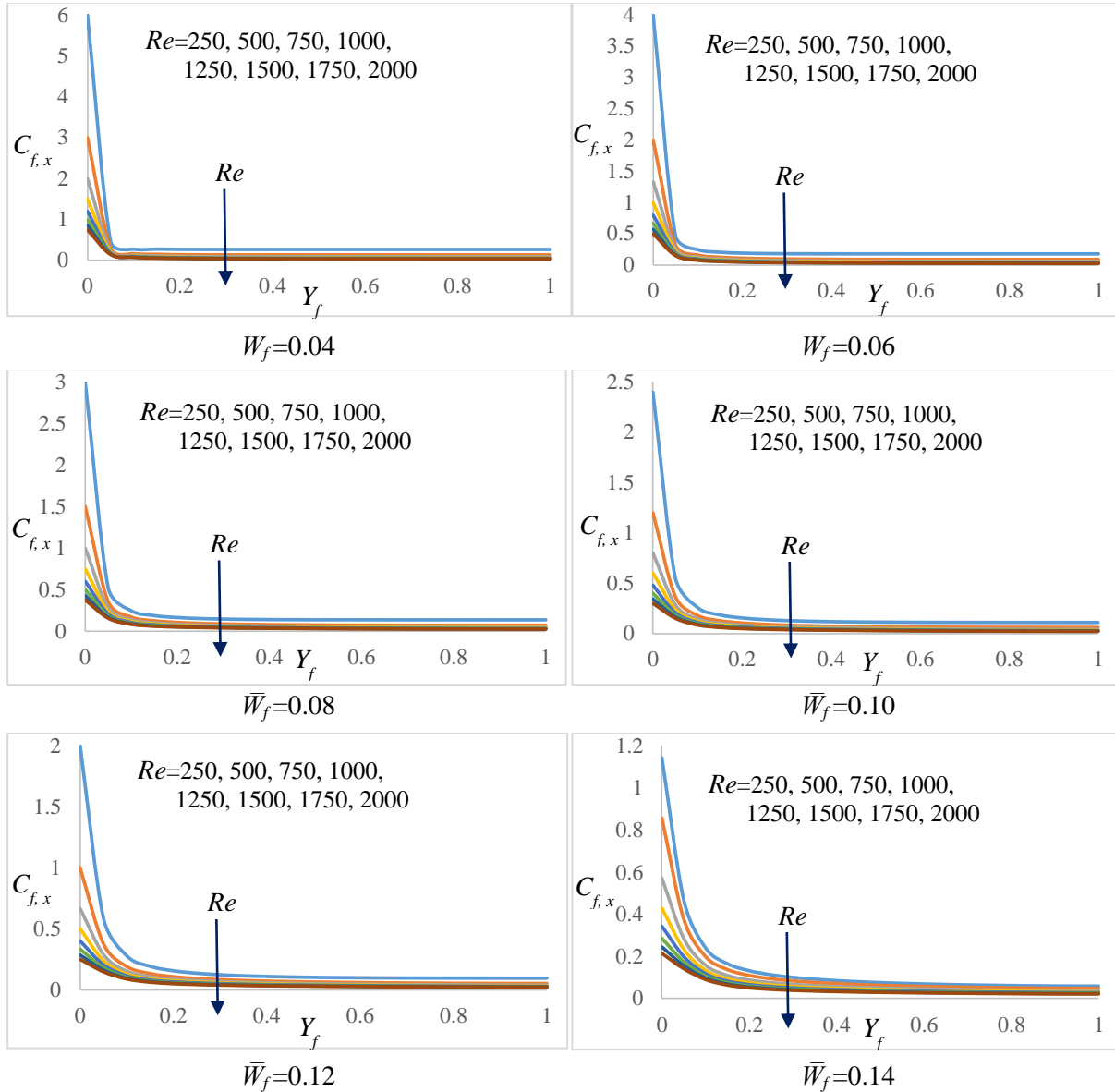


Figure 7. Local friction coefficient ($C_{f,x}$) at different Re and spacing (\bar{W}_f) between cells

Figure 7 depicts the variation in local friction coefficient ($C_{f,x}$) at different Re from 250 to 2000 and when spacing between cells is increased from 0.04 to 0.14. At all spacings the $C_{f,x}$ is highest for Re at 250 and gradually reduces upto $Re=750$. Upon further increases in $Re>750$, the $C_{f,x}$ remains more or less same and this is true at all spacings considered. The low coolant velocity at $Re\leq 750$ causes rapid development of hydrodynamic boundary layer close to the leading edge of cell hence increased $C_{f,x}$ and there on remains

constant near the leading edge. With increase in spacing, the $C_{f,x}$ also reduces for all Re and remains constant slightly away from the leading edge.

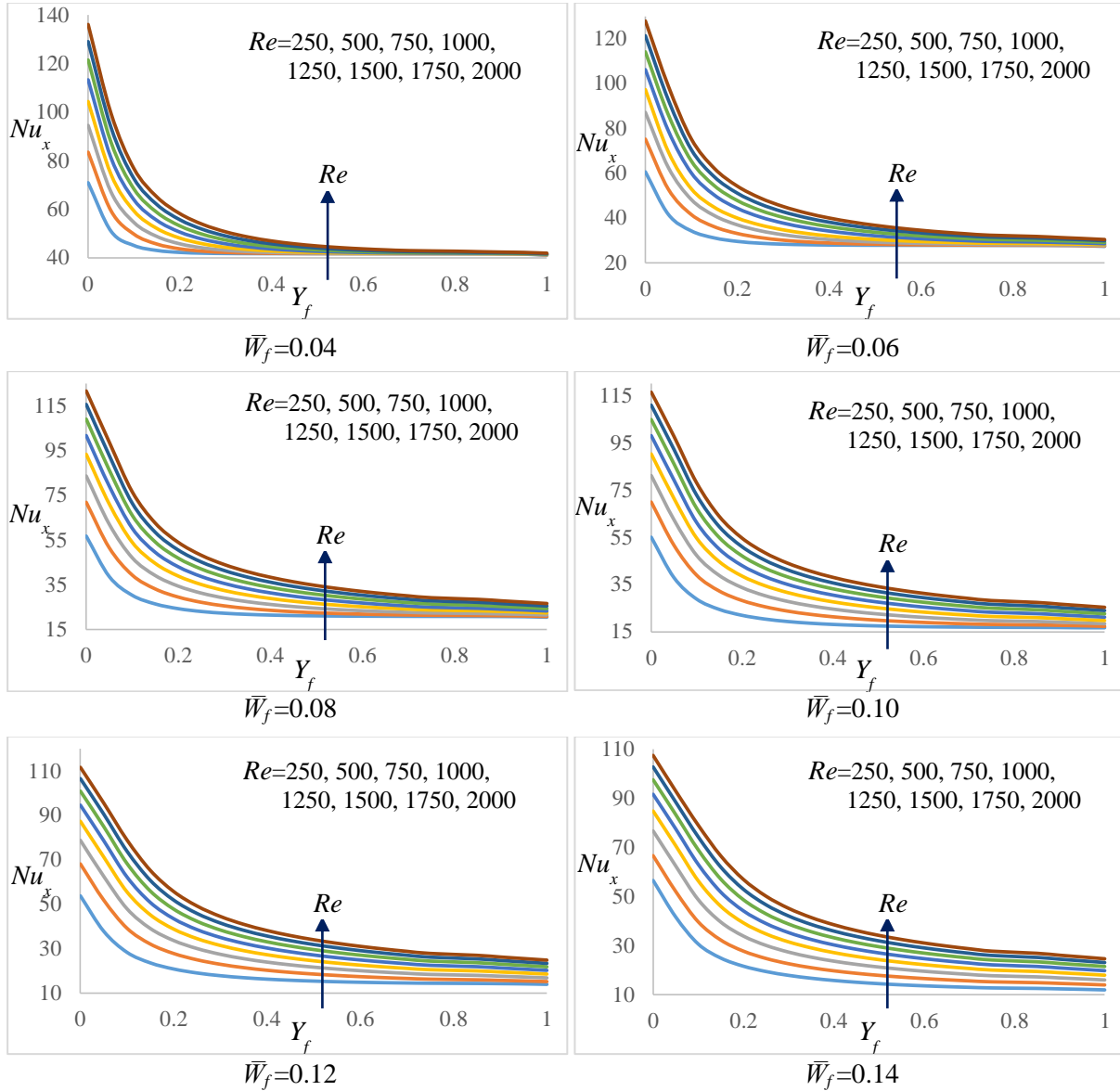


Figure 8. Local Nusselt number (Nu_x) at different Re and spacing (\bar{W}_f) between cells

In Figure 8 the variation of local Nusselt number (Nu_x) with increase in spacing between cells and coolant flow Re is shown. Similar to the effect of Re and spacing on $C_{f,x}$ shown in Figure 7, Nu_x also increases with increase in Re and spacing. At low Re i.e. <750 , Nu_x is highest at the leading edge where the fluid comes in contact with the cell surface. Nu_x keeps on reducing along the stream wise direction and becomes constant close to the leading edge. For higher Re and spacings Nu_x smoothly reduces and becomes nearly constant near the trailing edge of cell. Similar trends of Nu_x for increase in Re is reported by Moharana et al. in [30].

4. Conclusion

In this present work, a realistic conjugate heat transfer analysis of parametric effect on thermal and fluid flow behavior of battery thermal management system is carried. The effect of few important working parameters like volumetric heat generation (\bar{S}_q), conduction –convection parameter (ζ_{cc}), Reynolds number (Re), Aspect ratio (Ar), spacing between the cells (\bar{W}_f) are studied in this work. Important observations made from detailed analysis are following:

- The conjugate condition adopted indicates the proper selection of coolant flowing over the battery which can significantly regulate its temperature. The conduction –convection parameter (ζ_{cc}) arrived in this study dictates the selection of coolant by giving a ratio of thermal conductivity.
- With increase in spacing between the cells the temperature distribution along the width increases up to a certain upper limit as the flow behaves like external flow and the thermal boundary layers do not mix.
- Simultaneous increase of ζ_{cc} along with \bar{W}_f causes temperature reduction significantly due to increase of thermal conductivity of fluid.
- The Nusselt number is highest at maximum Re and also when \bar{W}_f is minimum as the flow mean velocity rises.
- Nusselt number decreases with increase in \bar{W}_f and remains unchanged at higher \bar{W}_f due to thermal boundary layer being fully developed.
- Friction coefficient is highest at low Re and \bar{W}_f . At maximum Re , friction coefficient is same for all \bar{W}_f .

5. Acknowledgments

The authors extend their appreciation to the Deanship of Scientific Research at King Khalid University, Abha, Saudi Arabia for funding this work through the Research Group under grant number (R.G.P.1/213/41)

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Submitted: 06.12.2019.

Revised: 29.08.2020.

Accepted: 02.09.2020.