

PRESSURE DROP AND HEAT TRANSFER IN THE SODIUM TO AIR HEAT EXCHANGER TUBE BANKS ON ADVANCED SODIUM-COOLED FAST REACTOR

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Abstract

A numerical study was performed to investigate the thermal and hydraulic characteristics and build up design model of the AHX (sodium-to-air heat exchanger) unit of a sodium-cooled fast reactor. Helical-coiled tube banks in the AHX were modeled as porous media and simulated heat and momentum transfer. Two-dimensional flow characteristic appeared at the most region of AHX annulus. Pressure drop and heat transfer coefficient for rectangular, parallelogram and staggered tube banks as the main components of the AHX were evaluated and compared with Zhukauskas empirical correlations.

1. Introduction

The conceptual design of the sodium-cooled fast reactor, KALIMER (Korea Advanced LIquid METal Reactor) has been developed, and various advanced design concepts have been continuously proposed and evaluated to satisfy the Generation IV (Gen IV) technology goals. [Hahn et al. (2007)] On the basis of a stepwise technology development, the Gen IV SFR (hereafter called advanced SFR) system is currently being developed, and its rated power was increased from the conventional 600 MWe to 1,200 MWe to improve plant economics. The design of the advanced SFR system is totally based on a maximum of passive safety features as far as it is technically feasible and economically beneficial. The IRACS (Intermediate Reactor Auxiliary Cooling System) is normally operated in a forced flow condition, while the PDRC (Passive Decay heat Removal Circuit) system relies exclusively on a natural convection heat transfer, i.e. natural circulation on the sodium-side and natural draft on the air-side. Hence independent and diverse decay heat removal (DHR) systems such as PDRC and IRACS have been employed to provide a highly reliable heat removal function from the primary heat transport system to the atmosphere during both scheduled and emergency. Two kinds of DHR systems are proposed which have different heat exchanging components and functional diversity to avoid common mode failures and to achieve a high reliability target of 10^{-7} failure per demand. [Kim et al. (2008)] One of the most important components for the design of both DHR systems is the sodium-to-air heat exchanger (hereafter called AHX). This is because the convection resistance of the AHX shell-side air flow path is the dominant factor for determining the overall heat removal performance from the reactor core to the atmosphere, i.e. final heat sink. Hence, sufficient design considerations for the AHX to reasonably predict the pressure drop and heat transfer coefficient from the sodium tube to the air flow are required to enhance the reliability of the DHR system design.

Zhukauskas et al. (1968) greatly contributed to the thermal and flow resistance design on the circular tube banks related to the present work. They provided the experimental data for the

tube bank arrangements, i.e. aligned and staggered, configuration, number of rows and Reynolds and Prandtl numbers. Some research on the modified tube banks has been conducted, however their work is limited in comparison to the range of the present work. It is not justified to use their empirical correlations and apply them to the present range of tube bank arrangements.

Mittal and Balaroglu (1997) conducted a numerical simulation flow past a cylinder at low $Re < 10^3$. Ku (1995), Chen (2005) and Razavi et al. (2008) also reported drag coefficients for a cylinder at a low Reynolds number. Several numerical simulations have been performed to predict very high Reynolds number flows ($Re > 10^6$) around a smooth circular cylinder due to the complexity of the flow. Gaitanaris (2006) applied a 3D large eddy simulation with wall model as well as an unstructured mesh. Mittal (2005) performed detailed studies for $100 < Re < 1 \times 10^7$ using a 2D URANS with a standard high Reynolds number $k-\epsilon$ model. Many works on flow across a circular cylinder have been performed and compared with the empirical correlations of Hilpert (1933), Whitaker (1977), and others.

Heat exchangers of many shapes and sizes have been approached using one or two equations. The one equation model assumes that the solid and fluid are at the same temperature, or local thermal equilibrium, using one equation for the solid and fluid phase temperature. The two-equation model includes an equation for each phase, treating the solid and fluid phase separately. Due to the complexity of the two-equation model, Ravian (1985), Vafar and Kim (1989) and Haji-Sheikh et al. (2006) used the one equation model in analyzing convection heat transfer in porous media.

In this study, fundamental features of the pressure drop and heat transfer performance of the air flow system across tube banks of the AHX were investigated by using a multi-dimensional CFD analysis. The numerical methods regarding the pressure drop and heat transfer calculations for this type of air flow system were also evaluated by comparing conventional correlations after checking the effectiveness of the numerical methods for the flow across a cylinder. Based on the evaluation results, design guides for a reliable and safe heat transfer performance of the AHX with helical-coiled tube banks are recommended, and specific design directions for the tube arrangement to avoid a local flow separation effect around the tube and AHX annulus are also discussed.

2. System description of the advanced SFR

The heat transport system of the advanced SFR was designed to be a pool-type system to enhance system safety through slow system transients, where primary sodium is contained in a reactor vessel. The heat transport system is composed of a primary heat transport system (PHTS), an intermediate heat transport system (IHTS), a steam generating system (SGS) and a residual heat removal system (RHRS) as shown in Figure 1. During normal power operation mode, heat generated from the reactor core is transferred to the SG through the PHTS and IHTS via Intermediate Heat Exchangers (IHXs) as shown in Figure 2.

With regard to the design of the AHX, its component diversity is mainly achieved by two different designs of the heat exchanging tube bundle, which are helical-coil type and compensated straight tube type. In the PDRC design, the AHX has drum type tube bundles with a total of ten cylindrical rows of un-finned tubes, a central inlet header arranged at the top of the tube bundle with a separated expansion vessel for the PDRC sodium loop, and an outlet common header below the tube bundle. The FDHX (forced-draft sodium-to-air heat exchanger) for the active IRACS system consists of four pass serpentine tube bundles with an inclined tube arrangement, and the shell-side air flow is assumed as a cross flow across the tube bank. The tube bank is attached to an upper inlet and a lower outlet header. The headers and tube bundle are mounted in a support structure that provides support for the tubes allowing for thermal expansion. Functional diversity can also be achieved by different operating conditions of the two groups of AHXs during standby condition in the reactor power operation mode. Consequently, different air-cooling mechanism of the AHXs and FDHXs induce different heat losses and cool-down performance in the AHX and FDHX.

Table 1 Geometry and operation conditions of AHX (sodium-to-air heat exchanger) used in the present work

Parameter	Symbol	Unit	Data
Heat capacity	Q	MW	8.112
Pressure drop of AHX annulus	ΔP	kPa	190
Frontal velocity	u	m/s	6.1
Inlet temperature of air	T_{in}	K	313.15
Outlet temperature of air	T_{ex}	K	551.18
Inlet temperature of sodium	$T_{s,in}$	K	776.31
Outlet temperature of sodium	$T_{s,ex}$	K	623.07
Height of annulus	H	m	5.91
Outer radius of annulus	r_o	m	1.835
Inner radius of annulus	r_i	m	1.0
Tube diameter	D	m	0.054
Longitudinal tube pitch	S_L	m	0.054
Transverse tube pitch	S_T	m	0.054
Interior angle of tube bank	θ	deg	90, 79.5
Number of tube row	N_r	-	71.6
Number of tube step	N_s	-	10
Volume of annulus	V_{core}	m ³	43.95
Surface area of annulus	A_{core}	m ²	105.3
Hydraulic diameter of annulus	$D_{h,core} (= 4V_{core} / A_{core})$	m	1.67
Number of tubes	N_t	-	196
Total length of tubes	L_t	m	6429.5
Surface area of tubes	A_t	m ²	1090.7

Hydraulic diameter of tube	$D_{h,t} (= 4V_{air} / A_t)$	m	0.1072
Compactness	A_t / V_{core}	m ² /m ³	24.82
Porosity	α	-	0.66
Permeability	K	m ²	3.63×10^{-6}
Quadratic resistance coefficient	β	kg/m ⁴	0.864

3. Numerical Simulation Method

Figure 2 and Table 1 are the configuration of tube banks and typical operation conditions in the AHX annulus of the present study. The tube banks are consisted of a total of ten steps (radial direction of AHX annulus, r direction of **Figure 2 (a)**) and on average of 71.6 rows (vertical direction of AHX annulus, y direction of **Figure 2 (a)**) as shown in **Figure 3 (a)**. The reference configuration is the parallelogram tube banks ($\theta = 79.5^\circ$) as shown in **Figure 3 (b)**, and it is compared with the rectangle ($\theta = 90^\circ$) and staggered tube banks as shown in **Figure 3 (c)**.

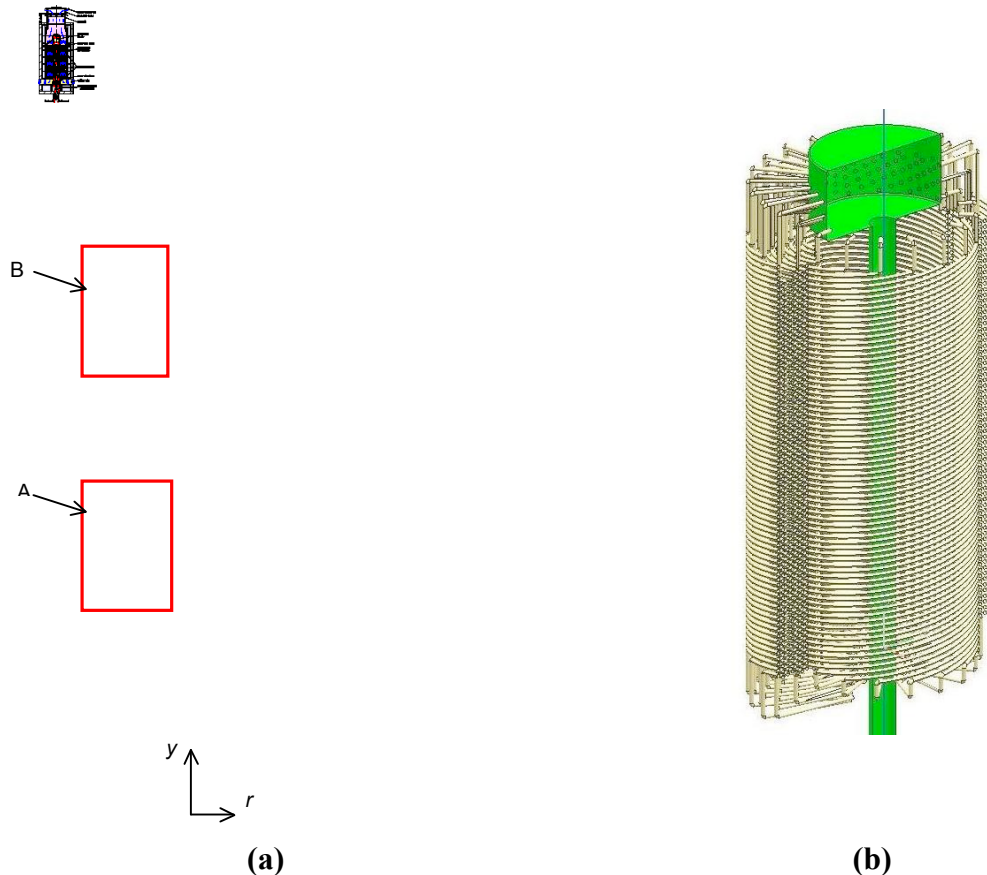


Figure 2 Schematics of the AHXs (sodium-to-air heat exchangers) (a) cross view, (b) tube bank annulus.

The heat transfer coefficient of the tube banks can be obtained as follows.

$$\text{Nu}_D = \frac{hD}{k} = C_1 C_2 \text{Re}_D^m \text{Pr}^{0.36} \left(\frac{\text{Pr}}{\text{Pr}_s} \right)^{1/4} \quad (1)$$

$$1000 < \text{Re}_D < 2 \times 10^6, 0.7 < \text{Pr} < 500 \quad (2)$$

The coefficient C_1 , C_2 and m are the correction factor and exponent of Re number for the number of tube rows and the alignment of tube bank respectively. The properties are defined at the film temperature, and the Pr_s means the Prandtl number at tube wall temperature. The Reynolds number is based on the maximum flow velocity, $V_{\max}$ at the minimum cross section.

$$\text{Re}_D = \frac{\rho V_{\max} D}{\mu} \quad (3)$$

The pressure drop across the tube bank can be expressed as below.

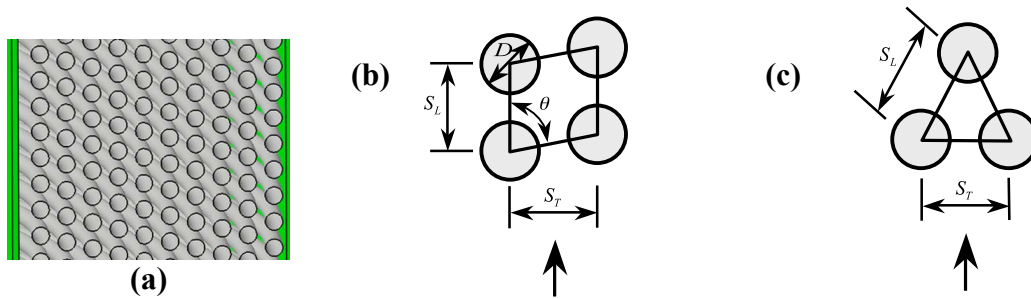


Figure 3 Tube bank (a) cross view of the present AHX, (b) rectangle and parallelogram (c) staggered tube banks.

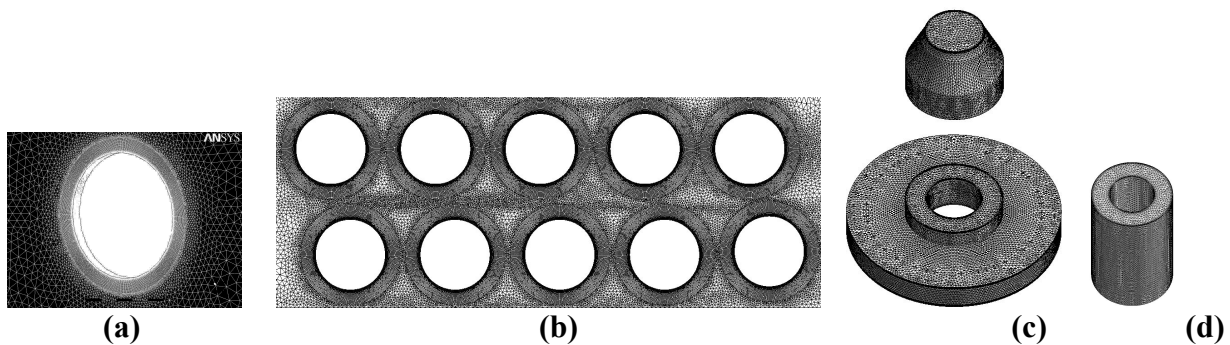


Figure 4 Grids used in the present numerical simulation (a) flow across a cylinder across, (b) flow through the tube banks, (c) inlet and outlet of the AHX, (d) AHX annulus.

$$\Delta P = N_L \mathcal{K} f \left(\rho V_{\max}^2 / 2 \right) \quad (4)$$

Where N_L , χ and f are the number of tube rows, correction and friction coefficient respectively. The correction coefficient can be obtained from the table as a function of the number of tube rows and the tube bank alignment.

The AHX is made up of multiple tube banks. Sodium flows from the upper to the lower storage tanks. Air flows into the lower part, is heated and exhausts at the upper part. The heat transfer mechanism of the airside and tubes inside of the AHX is the natural convection from the viewpoint of the AHX system while the forced convection is from the viewpoint of the tube banks.

The CFX 10 (2006) was used in the present simulation. The preliminary numerical simulation was performed to search for the best simulation conditions for the flow across a cylinder under the operating conditions. After comparing the empirical correlations of heat transfer and pressure drop with numerical results, the best turbulent model was the $k-\omega$ SST. The number of surface elements was increased to 6.21×10^5 , attaining an asymptotically converging value that agreed reasonably with the experimental data. The first grid near the wall was $y^+ \sim 1$ to apply the law of wall. **Figure 4 (a)** and **(b)** show grids in the present numerical simulation of a single tube and tube bank respectively.

The AHX was classified as the inlet and outlet parts without tube bank and the AHX annulus with tube bank. The AHX annulus was approximated as the porous media because of its geometric complexity. Porosity is the ratio of air to total volume, $\alpha = 1 - V_t/V_{core}$, is 0.66. Permeability is part of the proportionality constant in Darcy's law that relates the discharge (flow rate) and fluid physical properties (e.g. viscosity), to a pressure gradient applied to the porous media. The permeability and volumetric heat generation was defined as below.

$$\hat{u} = -\frac{K}{\mu} \frac{\partial P}{\partial y} \quad (5)$$

$$\dot{q} = \frac{\dot{m} c_p (T_{ex} - T_{in})}{V_t} \quad (6)$$

$$\beta = \frac{\mu}{\hat{u} K} \quad (7)$$

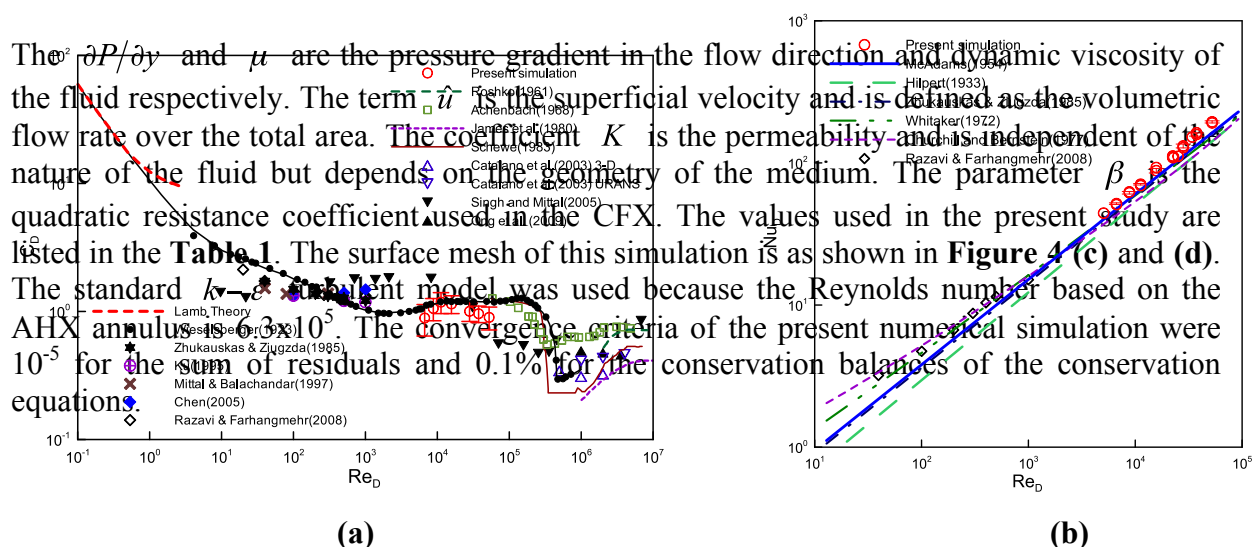
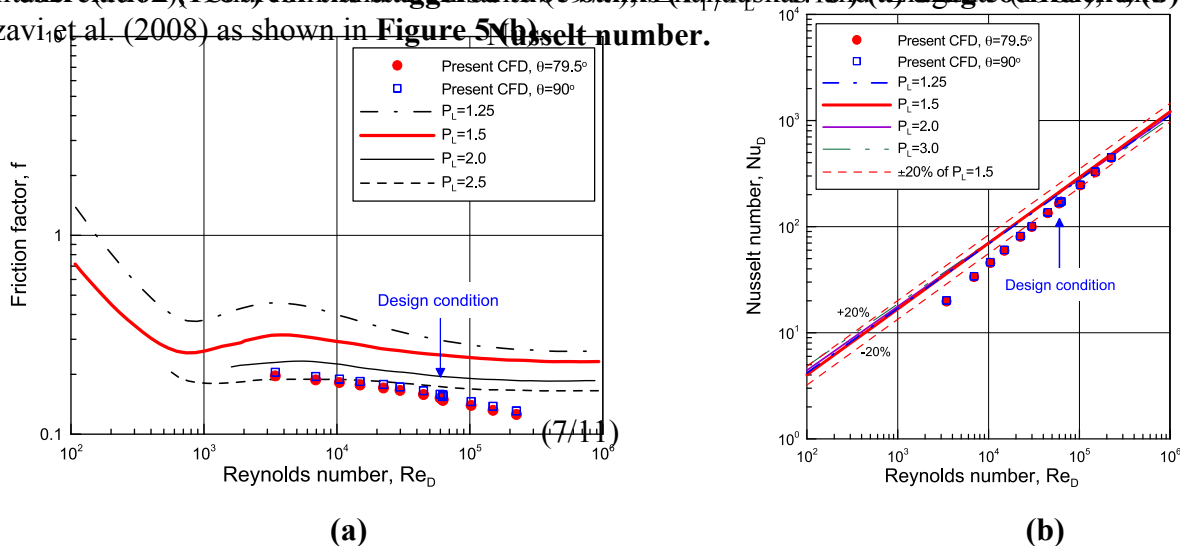


Figure 5 Comparison of the present numerical simulation with the previous works for the forced convection on a single tube (a) drag coefficient, (b) Nusselt number.

4. Results and discussion

4.1 Flow across a circular cylinder

The validity of the present numerical simulation is performed for the flow across a circular cylinder. The drag coefficient is compared with the previous experimental results of Wieselsberger (1923), Roshko (1961), Achenbach (1968), James et al. (1980), Schewe (1983), Zhukauskas (1985), Ku (1995), Mittal and Balachandrar (1997), Catalano et al. (2003), Chen (2005), Singh and Mittal (2005), and Razavi et al. (2008). The present result is 20% lower than the experiment of Wieselsberger (1923) at the operation condition ($Re \sim 5 \times 10^4$) as shown in Figure 5(a). The vortex generated behind the circular tube in the present numerical simulation was irregular and the drag coefficient varied. This work averaged ten sets of numerical data for different iterations. The error bars of the figure show the deviation of the numerical results. The result of numerical simulation for the heat transfer coefficient readily agreed with the experimental results of Zhukauskas (1985), Zhukauskas and Ziygda (1985), and Razavi et al. (2008) as shown in Figure 5(b).



4.2 Pressure drop and heat transfer of tube banks

Figure 6 compared the numerical result of the drag coefficient and Nusselt number for the parallelogram tube banks with Zhukauskas' experimental correlation (1968) for $P_T = S_T/D = 1.5$ and $P_T/P_L = 1.0$. The pressure drop and heat transfer coefficient of the rectangular tube banks and parallelogram tube banks were within 2~3%. Both pressure drop and heat transfer of the rectangular tube banks were greater than those of the parallelogram tube banks. This is because the minimum cross-section area of the rectangular tube banks for the flow is smaller than parallelogram tube banks. However the present numerical simulation underestimated the pressure drop by about 40% less than Zhukauskas' empirical correlation for the rectangular tube banks. The Nusselt number was also underestimated and showed bigger deviation in the low Reynolds number range. The reason is that the vortexes behind the tubes are isolated from the airflow between the tubes. The airflow between the tubes is restricted to meet the new heat transfer surface. The tube arrangement should be taken care of in the design of the AHX.

Figure 7 compares the pressure drop and heat transfer results obtained by the numerical simulation on the staggered tube banks with Zhukauskas' correlation for $P_T/P_L = 1.15$. The trend of the results was similar to those of the rectangular tube banks. However the deviation between the empirical correlation and numerical simulation of the staggered tube banks was smaller than those of the rectangular tube banks. The effect of the longitudinal tube pitch ratio ($P_L = S_L/D$) was small in the staggered tube banks. The Nusselt number of the numerical simulation agreed with the correlation within 20%.

4.3 Thermal and hydraulic characteristic of AHX

The uniformity of the airflow inside tube bank is important in the heat transfer performance of the AHX. The AHX inlet and outlet shapes of **Figure 2 (a)** 'A' and 'B' influence the uniformity of the flow. In the present work, the heat transfer of the rectangular edge, 45° chamber and shape of the curved side were reviewed and compared. The surface area density of the AHX tube bank is high, and it approximated as porous media. The inlet of the AHX is a perpendicular corner, where a strong vortex is generated right behind the perpendicular corner. A vortex is also formed behind the perpendicular edge of the outlet. In the case of the inlet with the 45° chamber and round shape, the size of the vortex at the inlet significantly decreases and air flows smoothly along the shapes. It can be found that the 1:2 inclination chamber significantly mitigates the vortex to the perpendicular corner at the outlet.

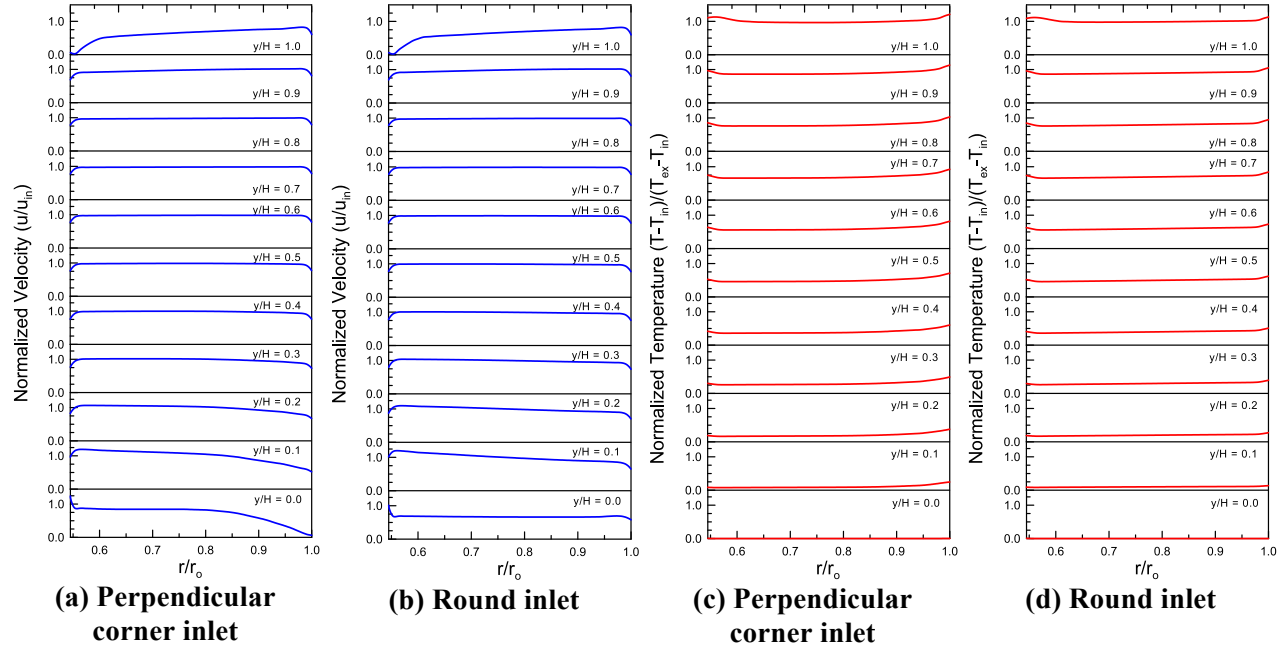


Figure 8 Velocity distribution of air at the AHX for the perpendicular corner inlet configuration.

Figure 8 (a) and (c) show the velocity and temperature distributions along the radial direction r for 10 sections in vertical direction y of the AHX annulus for the perpendicular corner shape. The velocity and temperature were normalized to the inlet velocity and inlet/outlet temperatures respectively. At the bottom part ($y/H < 0.3$), the air velocity is higher in the AHX annulus inside than the outside, while the air temperature is higher in the AHX annulus outside than the inside. This is because the inlet airflow is driven to the bottom of the AHX inner wall from the outside. The velocity distribution changes, in the inlet region, $0 < y/H < 0.3$, and the entrance effect exists. The velocity is the same as the inlet velocity within 5% in the region $0.22 < y/H < 0.95$. Therefore, it can be treated as a two-dimensional flow in the region. The temperature distribution was almost linear along the flow direction. **Figure 8 (b) and (d)** show the velocity and temperature distributions when the inlet shapes of the AHX are round. The distributions of the perpendicular corner in $0 < y/H < 0.3$ are improved to a two-dimensional flow. If the inlet shape of the AHX is designed as the shape proposed in the present work, the thermal load of the AHX will be more uniformly distributed and the safety of the AHX would be improved.

5. Conclusions

As a result of performing a numerical simulation for the heat flow characteristic of the AHX (sodium-to-air heat exchanger) and tube banks that are the main cooling devices of a sodium cooled reactor, the following conclusions were reached in this study:

When the AHX to the tube banks of five rows were simplified and compared, the parallelogram tube banks having 79.5° in the present design of the AHX showed similar thermal and hydraulic characteristics to the rectangular tube banks that have 90° .

The pressure drop and heat transfer of the staggered tube banks were higher than the parallelogram tube banks by 80% and 40% respectively. The pressure drop prediction for the parallelogram and staggered tube banks were lower than Zhukauskas' correlation by 40% and 25% respectively and the heat transfer coefficients were within 20% of the correlations.

When the AHX was modelled as porous media and an approximate calculation was performed, it showed two-dimensional thermal characteristics at the inlet area, $0 < y/H < 0.3$. When the inlet shape was changed to a chamber and round, the size of the inlet area is reduced and the uniformity of flow was enhanced.

An empirical verification of the results of numerical simulation will be required in the future.

6. Acknowledgements

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