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Analysis of thermal mixing in circle shaped body inserted inclined channel

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ABSTRACT

In this study, thermal mixing (*TM*) phenomena in a rectangle channel with adiabatic circle shaped body are investigated experimentally. Two parallel jets in different temperatures are located in the channel which has a circular exit hole to supply continuity of mass. Experiments are carried out for different inclination angle (0°, 30°, 60° and 90°) of the channel. Also, effects of ratio of flow rate, jets diameters, and temperature difference between hot and cold jets were analyzed. A circle shaped passive element with low thermal conductivity is located into channel to control thermal mixing. Thermal mixing index is calculated from measured temperatures. Experimental results showed that thermal mixing of fluid is effected from geometric parameters, drastically. It is found that *TM* is function of the temperature difference of inlet jets.

1. Introduction

Homogenous mixing of fluids in different temperature is commonly encountered subject in many fields of industry and engineering applications such as nuclear engineering, mechanical engineering and environmental science. Temperature fluctuations occur in the mixing region of cold and hot fluid. These fluctuations may cause fatigue with high cycles and subsequently cracks in surrounding surfaces. For instance, in 1992, extensive cracking was found in a control rod guide tube that had been removed from the core of the UK Prototype Fast Reactor (PFR). High-cycle thermal fatigue was found to be the cause of the cracks in the connecting pipe and the middle-stage heat exchange shell at the Tsuruga-2 PWR (Japan) in 1999. These industrial cases revealed that controlling of thermal mixing in industrial systems is a challenging subject that needs to be solved. Using jets to increase mixing efficiency is one common method that is used recently.

The three-dimensional flow and mixing characteristics of multiple and multi-confined turbulent round opposing jets in a novel inliner mixer are examined numerically using the standard k–ε turbulence model by Wang and Mujumdar [1]. In their study, they indicated that multiple opposing jets can achieve better mixing than single opposing jets. Wang et al. [2], made a numerical investigation to examine mixing efficiency of opposing jets in a confined channel using CFD code FLUENT. Some new design approaches to improve mixing effectiveness under laminar conditions in two-dimensional configurations were studied for different operating conditions and geometric configurations. Dissimilar inlet momenta with equal/unequal slot width, addition of baffles in the exit channel, effect of baffle height and multiple baffles are the parameters which used in the study. Also pressure loss due to the baffles was examined. Consequently, it is found that dissimilar inlet momenta and unequal slot width could significantly improve the mixing effectiveness; this improvement depended strongly on the operating conditions and geometric configurations. Addition of baffles in the exit channel enhances mixing effectiveness. The pressure loss was found to depend strongly on the mixer geometry and operating conditions. In their numerical work Wang et al. [3] studied the mixing characteristics and flow field of two-dimensional laminar confined opposing streams using air and water as the working fluids. The study carried out for temperature-dependent fluid properties for various temperature differences of the opposing jets, different operating conditions and different geometries. Numerical results show that the effects of temperature-dependent fluid properties on the mixing characteristics are dependent strongly on the magnitude of temperature difference, flow conditions and geometric configurations. Chang et al. [4] made a numerical analyzes to investigate the thermal mixing efficiency in Y-shaped channel. They solved two dimensional incompressible,

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Nomenclature

d	diameter (mm)
$\dot{m}$	flow rate (kg/s)
MI	mixing index
n	number of jet
PE	passive element
S_t	standard deviation of fluid temperature
t	time (s)
T	temperature of fluid (°C)
TM	thermal mixing
W_{MI}	uncertainty of mixing index
ΔT	temperature difference between hot and cold fluid (°C)
ϕ	inclination angle of the channel (°)
Subscript	
h	hot flow
c	cold flow
avg	average

steady state equations using Lattice Boltzmann method. They inserted different types of passive element to improve thermal mixing efficiency. It is demonstrated that the enhanced mixing efficiency is result of an increased intersection angle between the velocity vector and the temperature gradient within the channel. Durve et al. [5] studied a numerical model to understand physics of thermal mixing phenomena based on some former fundamental studies. In the study they investigate the capabilities of various temperature fluctuation models to predict the temperature fluctuation intensities in a triple jet flow. The model which developed was in good agreement with former studies and predicts the mean velocity and temperature field with considerable accuracy. Durve et al. [6] studied hydrodynamic characteristics of flow field of single, two parallel and three parallel jet flows, numerically. In the study, effects of jet spacing, merge-combine point of jets and velocity ratio on flow field were investigated. Also flow field of single jet, two jet and three jet cases were compared with each other. Based on the results, it is found that the merge point of the jet flow is influenced from spacing between the jets and the jet exit conditions such as turbulent intensity. Also velocity ratio of jets plays important role on the location of merge and combining point of the jets. Naik-Nimbalkar et al. [7] made an experimental study and numerical investigation of thermal mixing on single/twin jets in cross flow. Experiments were carried out for different velocity ratios and 15 °C temperature difference between main pipe and jets fluid. Based on findings, prediction of mean temperature, magnitude of the temperature fluctuations, characteristic frequencies of temperature fluctuations, attenuation of the temperature fluctuations in the boundary layer near the pipe wall, effect of multiple jets entering in the cross flow and effect of space between twin jets was investigated. Consequently, authors assert that numerical predictions are in good agreement with experimental measurements. Jung and Yoo [8], carried out Large Eddy Simulations (LES) of the triple jet flow for two different sub-grid scale models. The sub-grid scale models used were Smagorinsky–Lilly model and the k–l model. The simulations were aimed to investigate the effect of inlet thermal intensities (T_{rms}/T) on thermal mixing. The predictions were validated by comparing with the available experimental data. It was concluded that LES predicted faster decay of mean temperature along the axis of the

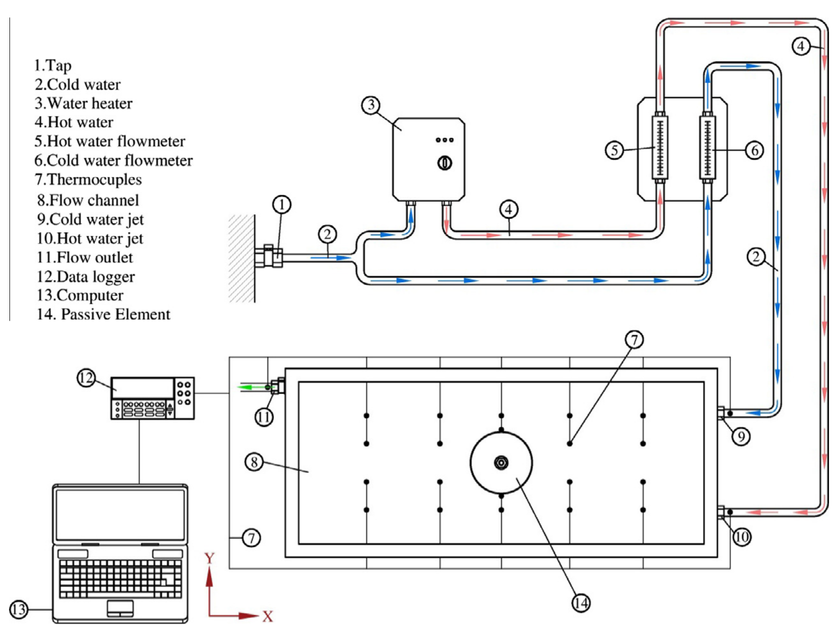


Fig. 1. Diagram of experimental Set-Up.

central jet. The inlet values of thermal intensity and the sub-grid scale models had no effect on the solution. The discrepancies between the predicted and the experimental results can be attributed to the large grid size and time step. The effect of the grid size and the time step size are also evident in the profiles of mean and RMS temperature presented by the authors. Chandran et al. [9], made a numerical study to investigate thermal striping in Prototype Fast Breeder Reactor (PFBR). Ten-jet water model which consist from seven hot and three cold jets was used in the study. Turbulence modeling has been done using Reynolds stress model (RSM). Five different hot jet to cold jet velocity ratios were used for analysis. The cold jet Reynolds number was taking constant while hot one has been varied. Based on analysis, the maximum temperature fluctuation was observed when hot and cold jet velocities are equal and the lattice plate is found to be better for thermal striping than core cover plate. Turki [10] used square cylinder to make a control mechanics for flow mixing numerically. Computations are carried out for different Reynolds numbers. In the study, the affect of splitter plate length and its location, the drag and lift coefficients are analyzed. It is found that, the presence of the splitter plate significantly reduces the lift and drag fluctuations, also changing of location of the splitter plate affect the flow. Naik-Nimbalkar et al. [11] made thermal mixing experiments on T-junctions with water. Velocity and temperature fields were measured using hot film anemometer (HFA). Three dimensional steady state computational fluid dynamics (CFD) simulations have been carried out to predict the velocity and temperature fields. The predicted velocity and temperature fields were in good agreement with the experimental measurements. Suyambazhahan et al. [12], carried out a numerical investigation of flow and thermal oscillation characteristic in buoyant twin jets. In the study, effects of jet nozzle spacing, jet inlet temperature and jet width are investigated. It was found that even in the turbulent forced convection regime, buoyancy has considerable influence on the jet flow oscillation characteristics. It also has significant effects on recirculation zones and merging point between jets. In the interacting shear layers, the frequency of oscillations decreases and amplitude increases, with nozzle spacing. Nishimura et al. [13] performed an experimental and numerical investigation on thermal mixing of three vertical jets in different temperatures. Low Reynolds number turbulent stress and heat flux equation models (LRSFM) and standard two-equation k–ε turbulence models were used to solve the numerical part of the study. The authors showed that the LRSFM could predict the profiles of mean quantities of flow with significant accuracy while the k–ε model was under-predicted

Table 1
Experimental condition for the channel.

Inclination angle of the channel (°)	Hot jet flow rate (kg/s)	Cold jet flow rate (kg/s)	Temperature difference (°C)	Jet diameter (mm)
0, 30, 60, and 90	0.05	0.033, 0.042, and 0.05	15, 20	5, 10

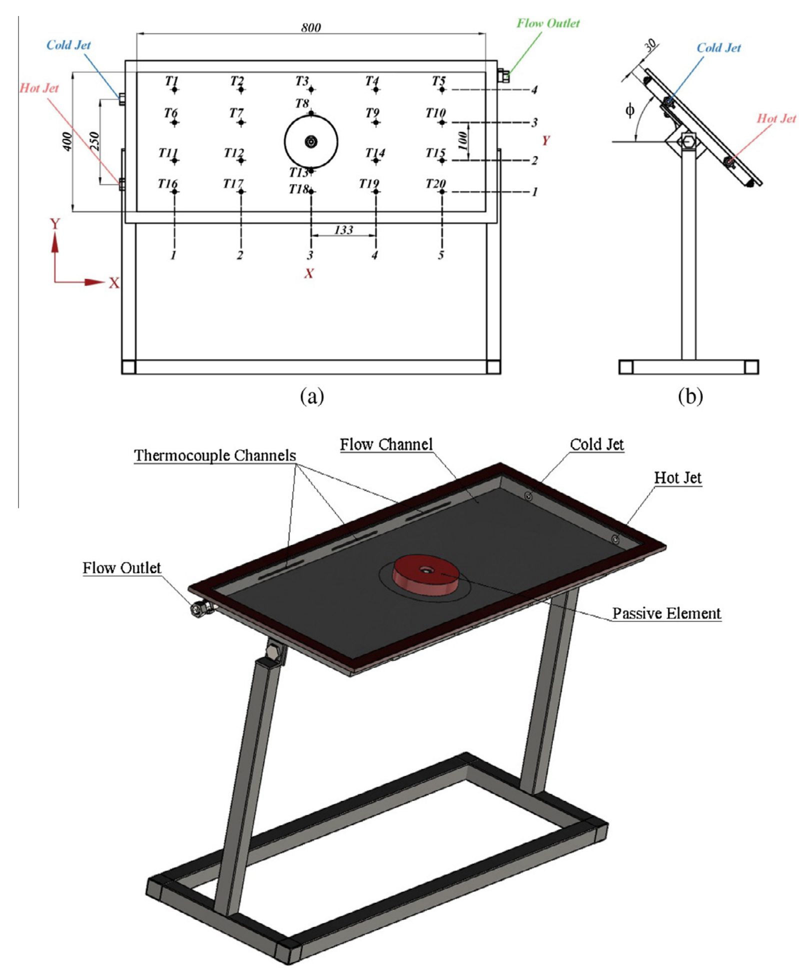


Fig. 2. (a) Dimensions experimental Set-Up and location of thermocouples, (b) side view and (c) model of experimental Set-Up.

the mixing profile. They also observed the periodic oscillation of jet contributes to the thermal mixing process. Cao et al. [14], prepared a numerical investigation on flow and temperature fluctuation phenomena of triple-jet model using large eddy simulation (LES). Numerical data validated with experimental results of Nishimura et al. [13]. The flow field observed in simulation showed that many vortices are closely related with the temperature fluctuation phenomenon. Also authors revealed that amplitudes of temperature fluctuation are different in flow field, while the frequency of temperature fluctuation remains constant at all monitoring position. With the increasing of Reynolds number, the mixing of hot and cold flows is delayed and convective mixing region is enlarged. Chandran et al. [15] performed a numerical investigation of thermal striping phenomena of a two jet water model. In the model the jets impinge on a lattice plate. This is 1/5 scale model of Prototype Fast Breeder Reactor (PFBR). Reynolds stress turbulence model used to evaluate the temperature fluctuations near the plate. Simulations were carried out for different velocity ratios of hot and cold jets and different location of lattice plate. Also numerical data were validated with available experimental results. Consequently, jets with unity velocity ratio showed maximum temperature fluctuations. Cold jet dominated and hot jet dominated flows demonstrated high and low temperature fluctuations, respectively. Kok et al. [16] made a numerical investigation in a body inserted channel to analyze thermal mixing behavior. Two parallel jets which have different temperatures were used to supply the flow to the channel. Bodies in different aspect ratios were located for different cases into the channel with the aim of increase thermal mixing efficiency inside the channel. The study carried out for different Reynolds numbers of jets. Consequently, it is found that mixing performance increases in the first half of the channel (x/L = 0–0.5) with increasing of Re number and chosen parameters can be used as control mechanism of thermal mixing in the channel. Varol et al. [17] made an experimental study on thermal mixing phenomena in a square body inserted inclined narrow channel. Two parallel jets in different temperatures were used to supply flow into the channel and water used as working fluid. Flow ratio of flow rate, inclination angle of the channel, temperature

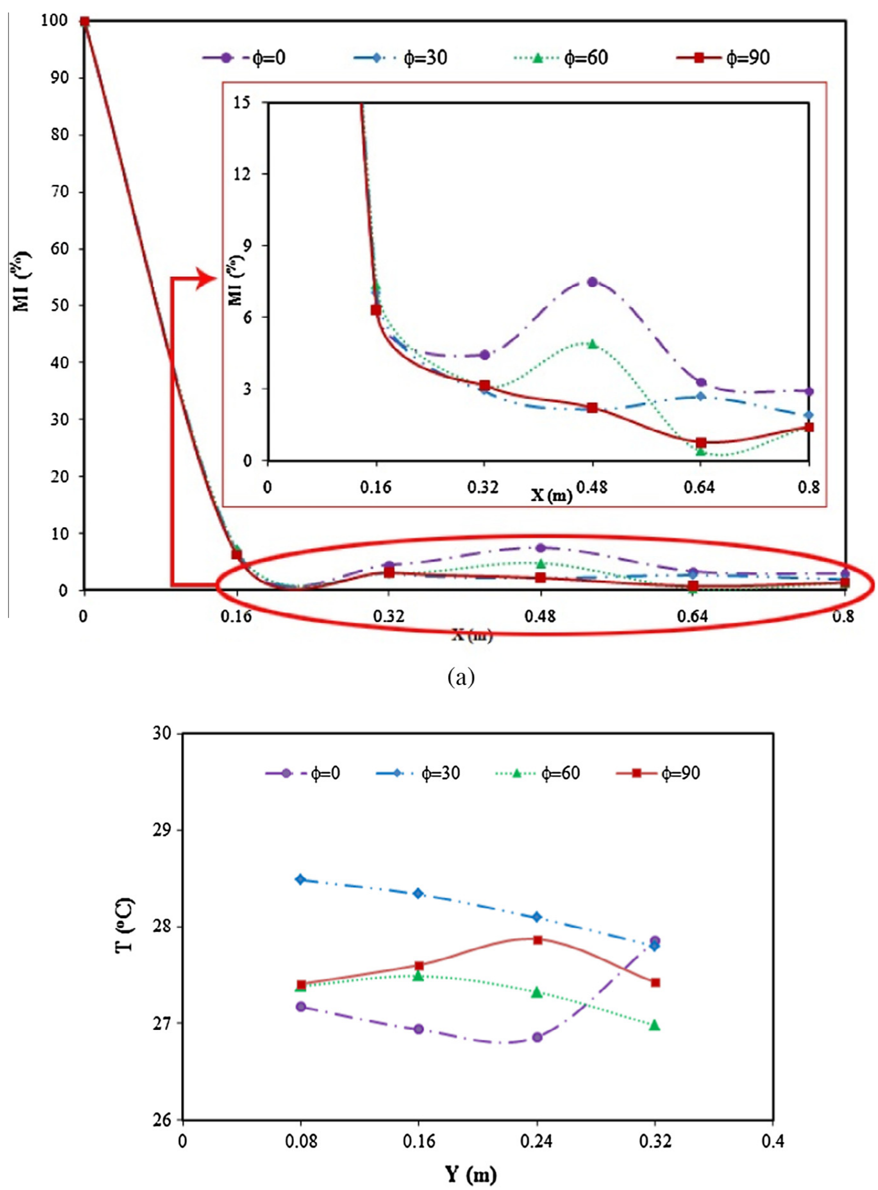


Fig. 3. For $\dot{m}_h = 0.05$ kg/s, $\dot{m}_c = 0.042$ kg/s, ΔT = 15 °C and D = 5 mm jet inlet, (a) variation of mixing index with X coordinate, and (b) temperature variation along station 5.

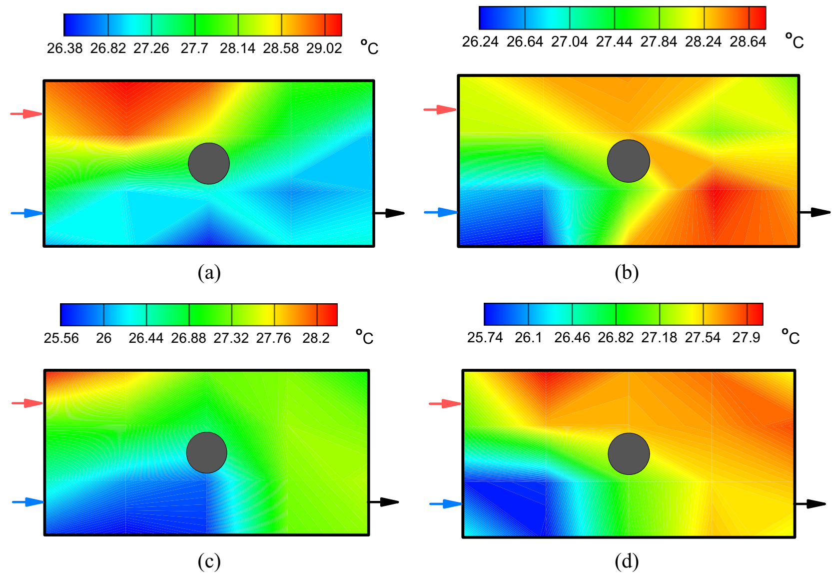


Fig. 4. Temperature field for $\dot{m}_h = 0.05$ kg/s, $\dot{m}_c = 0.042$ kg/s, $\Delta T = 15$ °C and $D = 5$ mm jet inlet, (a) $\phi = 0$, (b) $\phi = 30$, (c) $\phi = 60$, and (d) $\phi = 90$.

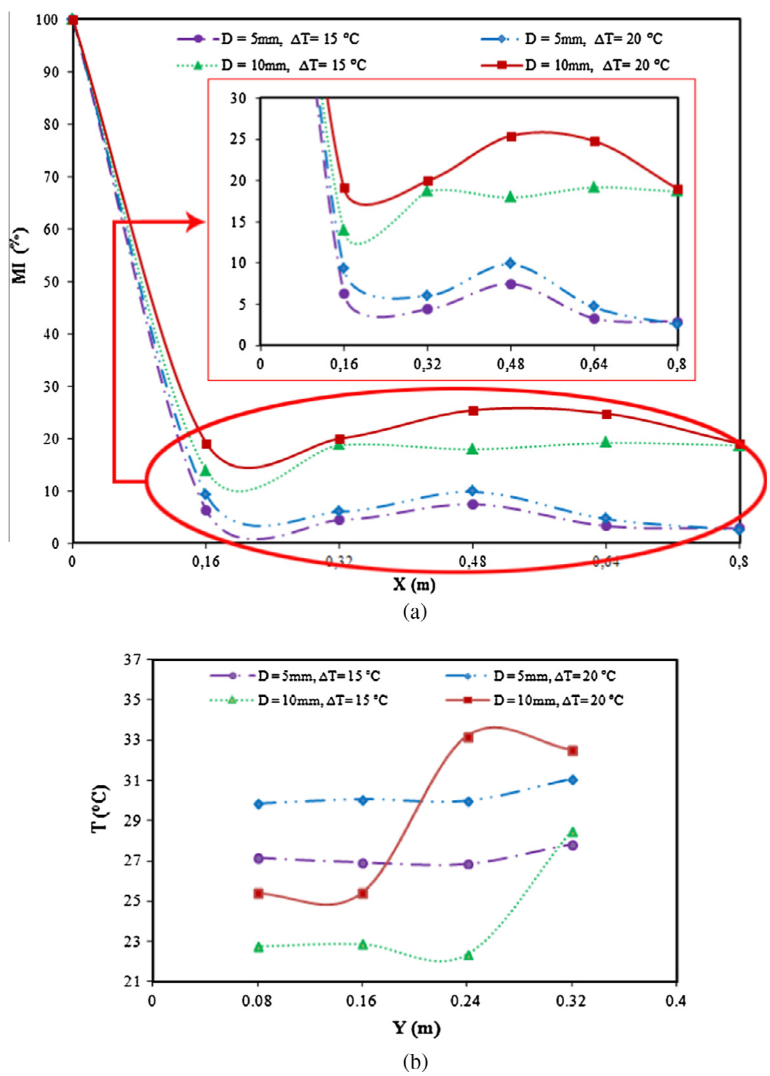


Fig. 5. For $\dot{m}_h = 0.05$ kg/s, $\dot{m}_c = 0.042$ kg/s and $\phi = 0$, (a) variation of mixing index with X coordinate, and (b) temperature variation along station 5.

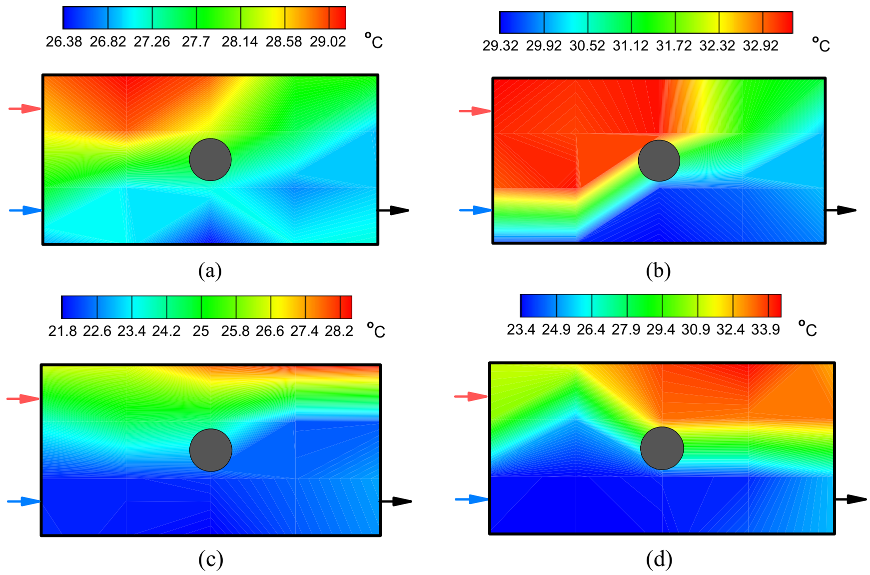


Fig. 6. Temperature field for $\dot{m}_h = 0.05$ kg/s, $\dot{m}_c = 0.042$ kg/s and $\phi = 0$.

difference value between jets and adiabatic square body were used as experimental parameters. It is found that inclination angle of the channel has important effect on the thermal mixing efficiency in the channel. Also, other chosen parameters can be used to control mixing. An experimental investigation was carried out by Kok et al. [18] to analyze the thermal mixing phenomena in a rectangular cross-section narrow channel. Inclination angle of the channel, ratio of flow rate of inlet fluid, temperature difference between hot and cold jets and jet inlets diameters were used to control thermal mixing phenomena in the channel. Also an artificial neural network (ANN) model was built with limited number of experimental measurements for a forward model. The obtained results from ANN model indicated that estimations can predict the output parameters without carry out any experiment.

As seen from above literature survey, most of investigations on thermal mixing have been focused on T-junctions and thermal mixing with parallel jets inside a cavity. The main aim of this work is to understand the phenomena of flow and thermal mixing performance in a narrow channel with circle shaped passive element inserted. Based on authors' knowledge, there is no such an experimental work.

2. Experimental Set-Up

A schematic view of the experimental Set-Up is presented in Fig. 1. It is composed of tap, water heater, flowmeters, the channel, data logger and computer as seen from the figure. In the experiment, a circular two jets inlet is obtained. Water was used as working fluid. Each jet has different flow temperature. The temperatures were measured by T-type thermocouples which located 100 mm distance in vertical and 133 mm distance in horizontal directions as given in Fig. 2(a). The measuring points are named as $T1$, $T2$, $T3, \ldots, T20$. We called station for each four thermocouples in a column (y-direction). Thus, there are 5 stations in the experiment from left to right. Fig. 2(c) also shows the model of experimental Set-Up. The circular body that inserted into the channel is adiabatic, same thickness with the channel and 150 mm diameter. The channel can be inclined for four positions such as 0°, 30°, 60° and 90° as seen from the side view in Fig. 2(b). The experiments were carried out for 5 mm and 10 mm jet nozzle diameters, 15 °C and 20 °C temperature differences. Other experimental conditions can be seen from Table 1. The length of the pipe is chosen to allow the flow to be fully developed at the exit. Hot water is supplied into the system using an electrical heater. Flowrates of hot and cold jets are measured by two different water flowmeters. The inlet and outlet temperatures of water were also measured at the inlet and outlet pipe. The channel was insulated using rock wool at 100 mm thickness.

3. Definition of mixing index

During the mixed fluid flow both along channel and channel outlet, better thermal mixing of cold and hot fluid is expected to be achieved. To quantify the thermal mixing efficiency of such a mixer under different operating conditions, a mixing index suggested by Wang and Mujumdar [1]. This MI is used as a measure of the closeness of the temperature profile to the mean temperature at any axial station in the channel:

$$MI = \frac{S_t}{\Delta T} \times 100 \tag{1}$$

where S_t based on the average temperature, is the standard deviation of temperature at any axial station in the channel

$$S_t = \sqrt{\left(\frac{\sum_{i=1}^{n}(T_h - T_{avg})^2}{(n-1)}\right)} \tag{2}$$

$$T_{avg} = \left(\frac{1}{n}\right)\sum_{i=1}^{n} T_h \tag{3}$$

T_h is hot inlet fluid temperature and T_{avg} is the average temperature at the corresponding axial station. Physically, this mixing index is a measure of how well the average temperature at any specific axial station represents the set of temperatures that it comes from. $MI = 0$ shows complete mixing, which corresponds to a perfectly flat temperature profile.

4. Uncertainty analysis

Uncertainty analysis is needed to prove the accuracy of the experiments. An uncertainty analysis was performed using Eq. (4). In the present study, the temperatures were measured by

T-type thermocouples. The total uncertainties of the measurements are calculated as ±1.59% for the temperatures.

$$w_F = \left[w_{F1}^2 + w_{F2}^2\right]^{1/2} \quad (4)$$

5. Results and discussion

In this study, thermal mixing phenomena in a rectangular cross-section channel are investigated, experimentally. Two parallel water jets which in different temperatures were used to supply flow into the channel. Inclination angle of the channel, jet diameter, temperature difference between hot and cold jet, ratio of inlet flow rates and passive element were used as studied parameters.

5.1. Effect of inclination angle

Effect of inclination angle of the channel on thermal mixing performance is one main parameter in this study. Fig. 3(a) and (b) shows variation of mixing index along the channel (x coordinate) and temperature variation along the station 5, respectively. For this case, hot water flow rate $\dot{m}_h = 0.05$ kg/s, cold water flow rate $\dot{m}_c = 0.042$ kg/s, temperature difference between cold and hot jet $\Delta T = 15$ °C and inlet jet diameters $D = 5$ mm were chosen as experimental parameters. As seen in Fig 3(a), the lowest values of mixing index are observed in the vertical position of the channel ($\phi = 90$), $\phi = 30$ and $\phi = 60$ is following, and the highest values are plotted in the horizontal position ($\phi = 0$) of the channel. This means that thermal mixing performance gets better with the increasing value of inclination angle of the channel for chosen parameter. In this parts, the flow is not hydrodynamically developed. Fig. 3(b) present effects of inclination angle on temperature distribution on the station 5 which is the closest station to the exit part of the channel. As seen from the figure temperature values present almost liner trend in $\phi = 30$ and $\phi = 60$. A fluctuating shape of temperature is observed in the vertical and horizontal position of the channel. These values mean thermal mixing performance is better at

$\phi = 30$ and $\phi = 60$ position of the channel near exit. Fig. 4 illustrates temperature field of the channel with same experimental parameters of Fig. 3 for different inclination angle of the channel. In the figure hot jet represents with red, cold jet with blue and flow outlet hole with black arrows. These values of temperature distribution were obtained in the steady state of condition of the flow. As seen from the figure, the average temperature field divides the channel diagonally in the horizontal position of the channel and cold water cumulates at bottom-right and hot one in left-up part. In all other positions of the channel, the cold water sets around cold

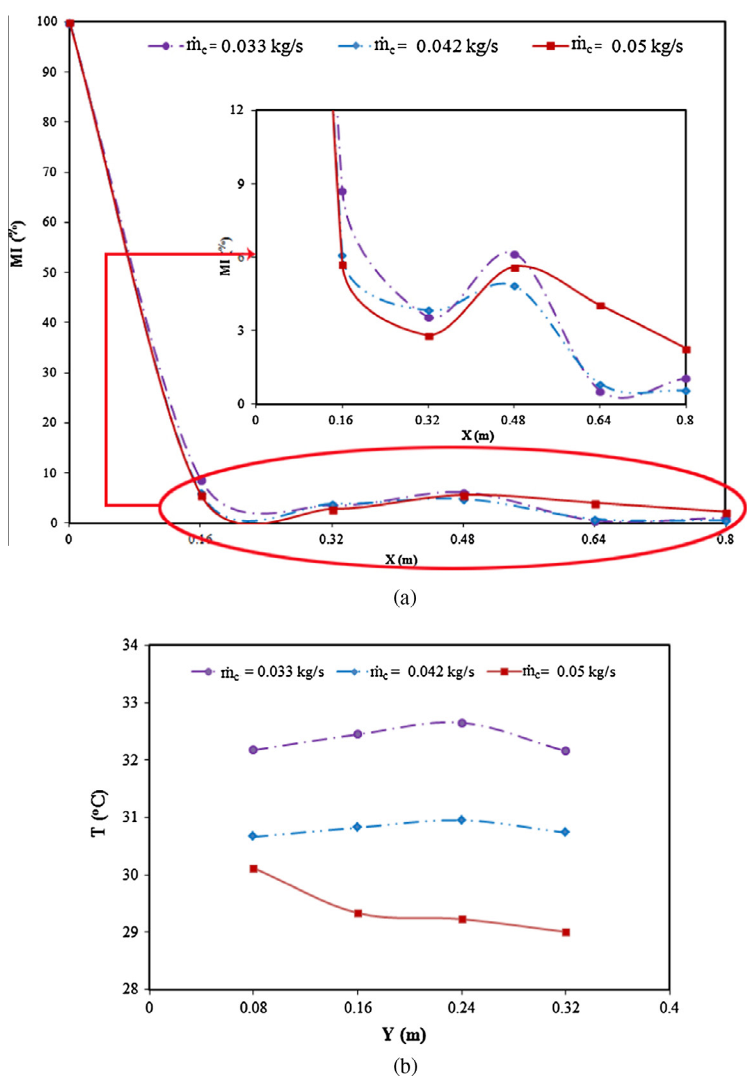


Fig. 7. For $\dot{m}_h = 0.05$ kg/s, $\Delta T = 20$ °C, $\phi = 60$ and $D = 5$ mm jet inlet, (a) variation of mixing index with X coordinate, and (b) temperature variation along station 5.

jet region. The best thermal mixing performance in the second half (x = 0.4–0.8 m) of the channel is observed at ϕ = 60.

5.2. Effect of temperature difference and jet diameter

Fig. 5 illustrates effects of temperature difference and jet diameters on thermal mixing performance for $\dot{m}_h = 0.05$ kg/s, $\dot{m}_c = 0.042$ kg/s and ϕ = 0. Fig. 5(a) gives variation of *MI* along the channel. As seen from the figure better thermal mixing efficiency is achieved at lower (D = 5 mm) diameter of the jets. Also mixing performance shifts to better with decreasing of temperature difference between hot and cold jet. Fig. 5(b) shows variations of temperature along the station 5, which the nearest station to the exit part of the channel. Temperature values which presented in the figure are nearly constant at D = 5 mm jet diameter. This mean better thermal mixing efficiency is achieved at lower jet diameter. Comparing effect of inlet jets temperature difference (ΔT) on thermal mixing performance, ΔT = 20 °C values present wavy trend and ΔT = 15 °C values are almost constant along the station. Based on these findings it is concluded that better thermal mixing efficiency is obtained at ΔT = 15 °C. Temperature field for different jet diameters and ΔT are given at Fig. 6 for $\dot{m}_h = 0.05$ kg/s, $\dot{m}_c = 0.042$ kg/s and ϕ = 0. As seen from the figure, domination of average temperature to the channel is higher at lower ΔT values (Fig. 6(a) and (c)). This mean thermal mixing performance is better at lower (ΔT = 15 °C) temperature difference between jets.

5.3. Effect of ratio of flow rate

Fig. 7 illustrates effects of ratio of flow rate on thermal mixing performance. Variation of mixing index along the channel is given at Fig. 7(a) for $\dot{m}_h = 0.05$ kg/s, ΔT = 20 °C, ϕ = 60 and D = 5 mm. *MI* shows almost same trend along the channel for all values of $\dot{m}_c$, just $\dot{m}_c$ = 0.05 kg/s leaves the other values around x = 0.5 m. The best thermal mixing performance is observed in the first half (x = 0–0.5 m) of the channel at the case that both hot and cold jets flow rate are equal ($\dot{m}_c$ = 0.05 kg/s), in the exit part of the channel the best thermal mixing performance is plotted for $\dot{m}_c$ = 0.042 kg/s. Temperature variation along station 5 is shown in Fig. 7(b) with same parameters of Fig. 7(a). Temperature values are approximately constant along the station for all cases of flow rate. The physical meaning of this is that there is no important effect of ratio of flow rate in the exit part of the channel. Fig. 8(a)–(c) shows temperature field for $\dot{m}_c = 0.033$ kg/s, $\dot{m}_c = 0.042$ kg/s and $\dot{m}_c = 0.05$ kg/s, respectively with same parameter of Fig. 7. In all cases of flow rates, cold water cumulates around cold jet. Hot water dominates upward and exit part of the channel at $\dot{m}_c = 0.033$ kg/s and $\dot{m}_c = 0.05$ kg/s cases. Average temperature collects at the second part of the channel at $\dot{m}_c = 0.042$ kg/s case. As indicated in the previous figure the best thermal mixing performance is observed at $\dot{m}_c = 0.042$ kg/s in the second half of the channel. Based on these findings, it can be concluded that ratio of flow rate effect thermal mixing performance and it can be use as control parameter.

5.4. Effect of passive element

One of the main parameter of this study is to see the effect of circle shaped adiabatic passive element on thermal mixing performance. Fig. 9 shows effect of circle shaped body that inserted into the channel on thermal mixing for $\dot{m}_h = 0.05$ kg/s, $\dot{m}_c = 0.033$ kg/s, ΔT = 15 °C, ϕ = 30 and D = 5 mm. Fig. 9(a) displays variations of *MI* with x coordinate. Effect of passive element (*PE*) that inserted into the channel appear quite clear in the figure. In the case that no *PE* inserted, *MI* shows a fluctuating trend along the channel. But *MI* values of *PE* inserted case are more uniform and its values are also lower. This shows that inserting of a circular shaped passive element into the channel effects thermal mixing performance positively. Fig. 9(b) exhibits temperature variation along the station 5 with same experimental parameters of the previous figure. Temperature values of *PE* inserted case shows a minor change along the station while in no *PE* inserted case shows nearly 1 °C decrease along the station. This means *PE* has an increasing effect on thermal mixing performance in the flow outlet region of the channel. Impinging flow onto passive element increases the kinetic energy of the flow and it affects the mixing parameters. Fig. 10 demonstrates temperature field with same experimental parameters of Fig 9. In this figure, channel without passive element and passive element located channel is compared from the temperature distribution point of view. Fig. 10(b) shows that flow temperature is increased at the outlet of the channel due to presence of passive element. The flow with higher temperature is accumulated at the right of the passive element toward to outlet.

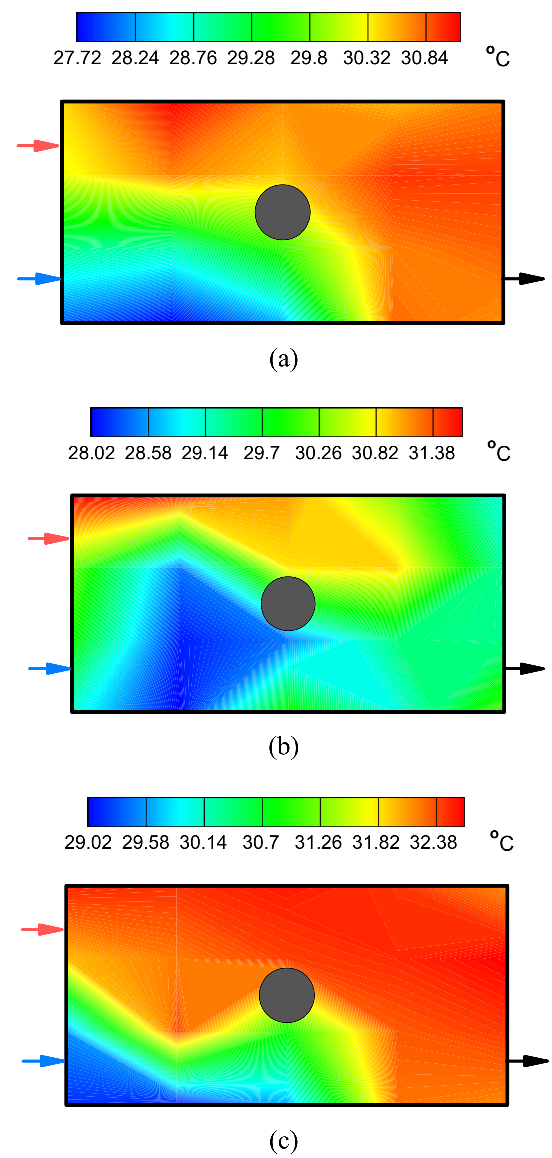
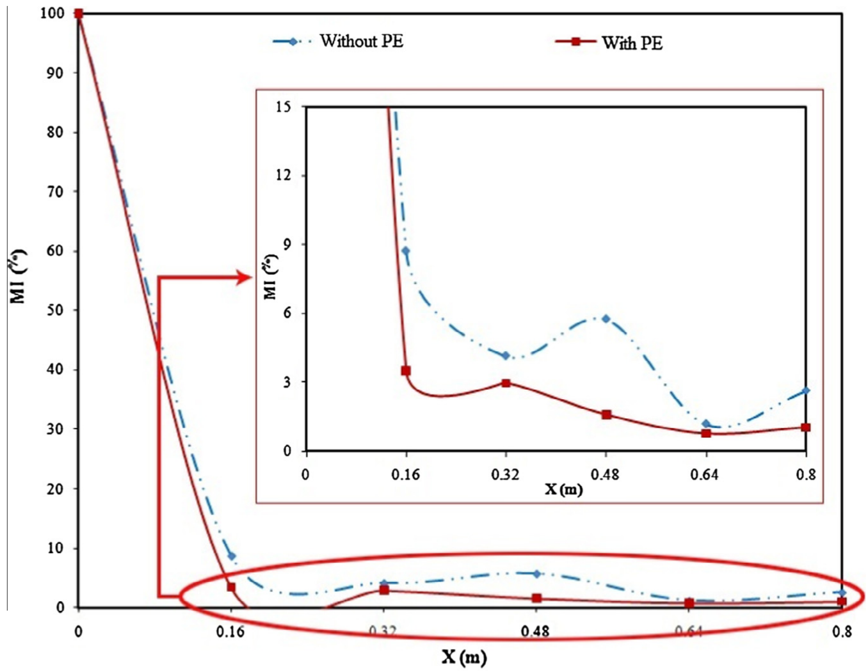
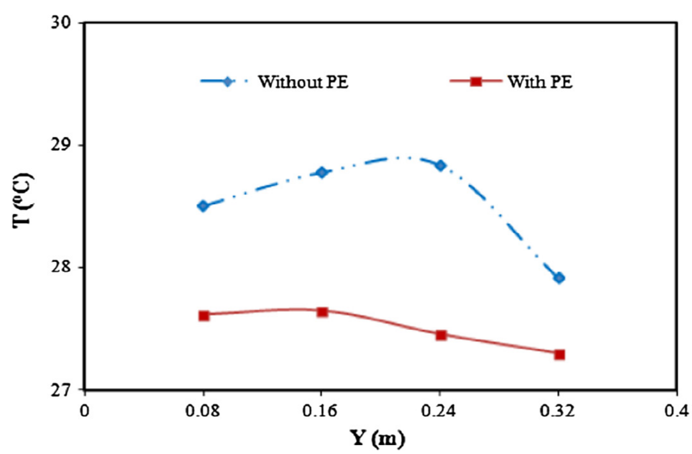


Fig. 8. Temperature field for $\dot{m}_h = 0.05$ kg/s, ΔT = 20 °C, ϕ = 60 and D = 5 mm jet inlet, (a) $\dot{m}_c = 0.033$ kg/s, (b) $\dot{m}_c = 0.042$ kg/s, and (c) $\dot{m}_c = 0.05$ kg/s.

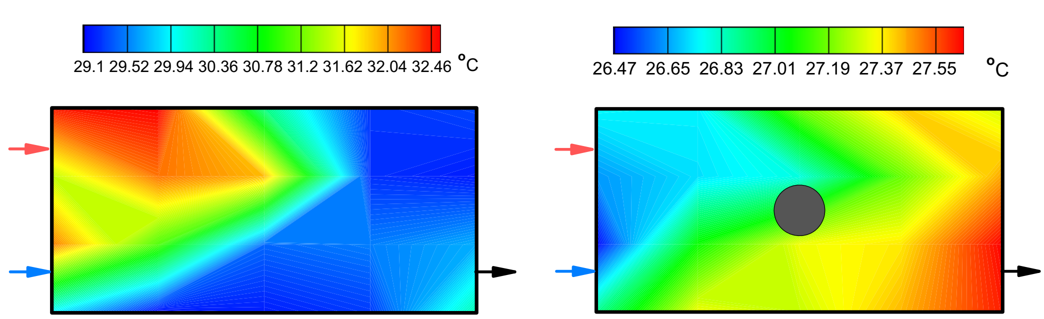


(a)



(b)

Fig. 9. For $\dot{m}_h = 0.05$ kg/s, $\dot{m}_c = 0.033$ kg/s, $\Delta T = 15$ °C, $\phi = 30$ and $D = 5$ mm jet inlet, (a) variation of mixing index with X coordinate, and (b) temperature variation along station 5.



(a) (b)

Fig. 10. Temperature field for $\dot{m}_h = 0.05$ kg/s, $\dot{m}_c = 0.033$ kg/s, $\Delta T = 15$ °C, $\phi = 30$ and $D = 5$ mm jet inlet, (a) no passive element inserted channel, and (b) passive element inserted.

6. Conclusions

Present study was focused on effect of inclination angle of the channel, jet diameter, temperature difference between hot and cold jet, ratio of inlet flow rates and passive element on thermal mixing performance in a rectangular cross-section channel. The main findings from the studied parameters can be listed as follows,

- Thermal mixing performance increases with the increasing of inclination angle of the channel.
- Better thermal mixing performance is formed at $\phi = 30°$ and $\phi = 60°$ and decreasing of jets diameter enhances the thermal mixing performance.
- Thermal mixing efficiency is better for lower value of temperature difference namely, $\Delta T = 15$ °C.
- In the first half of the channel (x = 0–0.5 m) better thermal mixing performance is observed when flow rate of inlet jets are equal.
- The best mixing efficiency is found in the exit part of the channel for $\dot{m}_h = 0.05$ kg/s, $\dot{m}_c = 0.042$ kg/s.
- Effects of inserting PE into the channel make positive impact on thermal mixing performance.

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