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Effect of V-Shape Twisted Jaw Turbulators on Thermal Performance of Tube heat exchanger: An Experimental Study

Abstract- *The main purpose of the present investigation is enhancing heat transfer rate in a tube heat exchanger by using V-shape twisted Jaws. The air is used as a working fluid and pumped through the test section with different values of Reynolds number (6000 - 19500), while the heat flux has been selected as a constant boundary condition around the tube section. In this study, two type of twisted jaw turbulators are used with two twisted ratio (TR= 2 & 4) as well as, the effect of using different numbers of turbulators (N= 6, 8 and 10) inside test section with equal distances between pieces are studied. The results indicated that, using augmentations with TR=2 gives better heat transfer rate and thermal performance factor comparing with the other case TR=4. The maximum rate of heat transfer is achieved in case of N=10 by an increased 160.29% for TR=2 and 102% for TR=4 comparing with plain tube case. In addition, results show that the values of thermal performance factor exceed the unity and shows uptrend behavior with rising numbers of turbulators indicating to feasibility of using these turbulators practically.*

Keywords- *Tube heat exchanger, Twisted Jaws turbulators, Thermal performance factor.*

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1. Introduction

The Numerous industrial applications depend on heat exchangers, and their economy and efficiency are affected by performance of these devices [1]. Hence, high performance is required for achieving efficiency criteria without high cost [2]. Many ways were used for enhancing the performance like manipulating in heat exchangers' size or using nanotechnology, but the simplest way to achieve this purpose is using turbulators. This way which is called passive technique depends on creating disturbance in working fluid flow that reduces the thermal boundary layer near the wall. However, inserting turbulators leads to significant pressure drop and hence extra pumping power will be required [3]. Therefore, low pressure drop and other factors should be taken into account when selecting turbulator. Twisted tapes are considered frequent turbulators which are used in passive technique because they are simple in manufacturing, stable in performance and easy in installation [4]. Given the importance of this technique, many researchers put their efforts for studying many types of augmentations. Patil et al. [5] used continuous corrugated twisted tapes with three twisted ratios (8.33, 9.79 & 10.42) and different values of Reynolds number (4000 – 9500). The results show that using corrugated twisted tape offers increasing in heat transfer efficiency between 18% and 52 %. Pongjet Promvonge [6] used wire coils insertion in conjunction with twisted tapes inside a circular tube. The results indicats that the wire coils with twisted tapes give a double increasing in heat transfer compared with using wire coil or twisted tape alone. Dhamane et al. [7] studied experimentally the effect of helical strips with regularly spaced cut passages. Three different helix angles (300, 450 and 600) were used in this investigation. The researchers revile that using helical strips show rising in the heat transfer rate up to 20% depending on Reynolds number values. While, increasing in Reynolds number values lead to decreasing in efficiency enhancement. Altaie et al. [8] studied numerically enhancing thermal performance and heat transfer in circular tube by inserting open rings with square cross section. Numerical results showed that internal ribs lead to enhancing in the heat transfer rate and give high performance factor for turbulent air flow. Shinde [9] investigated experimentally using screw tapes for improving heat transfer in double pipe heat exchanger. These screw tapes were made from different materials (including aluminum and M.S. materials). The researchers

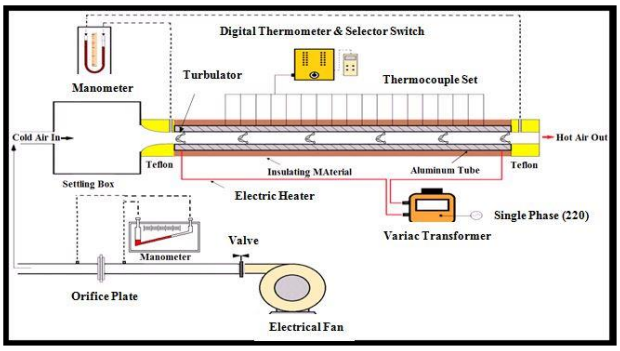
demonstrated that using aluminum screw tape offers higher heat transfer coefficient by 20 % than M.S. screw tape. Chokphoemphun et al. [10] inserted different numbers of twisted tapes (single, double, triple, and quadruple) inside tube heat exchanger and studied their effect on heat transfer. The authors reviled that inserting quadruple tapes give maximum heat transfer rate. Using V-shaped hexagonal conical-rings inserts were studied numerically by Sripattanapipat et al. [11] for enhancing heat transfer in heat exchanger. These inserts were obtained by cutting both symmetric plane of the cone-tip of hexagonal conical rings at 30^0, 45^0 and 60^0 to reduce the loss of pressure. It was found from the numerical results that these inserts provide higher heat transfer rate than the typical conical rings insert and the plane tube case. Man et al. [12] studied using alternation of clockwise and counterclockwise twisted tape (acct) in addition of using typical tape inside tube pipe heat exchanger. The researchers used different lengths of tapes (quarter, half, three quarters and full length) in performing experiments and they found that inserting (acct) with full length gives maximum enhancement with performance evaluation equal 1.42. Moreover, the (acct) case offer better heat transfer rate than using typical tape. Suri et al. [13] investigated experimentally the effect of inserting number of square perforated twisted tapes. The perforation width ratio for the square holes in tapes varied from 0.083 to 0.333, and the twisting ratio varied also from 2 to 3.5. It was found from the experimental results that maximum improvement can be achieved by using perforation width ratio 0.25 and twisting ratio 2.5.

The main aim of present work is investigate the heat transfer, frication flow and thermal performance factor in tube heat exchanger by inserting V-shape twisted jaws inside it. The effect of twisted ratio (2 & 4) and number of turbulators (6, 8, and 10) on heat transfer characteristics are examined in a range of Reynolds number (6000 - 19500).

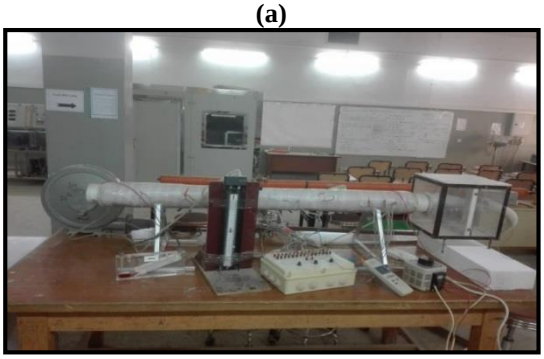
2. Experimental Section

I. Apparatus Description

The facilities which are used to perform the experiments are shown schematically & photographically in Figure 1a & 1b. An aluminum tube with length 1350 mm and inner diameter 45 mm is used as a test section. A uniform constant heat flux is applied around the tube by using an electrical wire, and a multimeter with variac transformer are used to specify the heat flux value. The test section is covered by two different layers of insulations (rubber and gypsum) to avoid radial heat losses. Moreover, the ends of tube are insulated as well by Teflon material to reduce heat losses from tube ends. The temperature distribution along test section wall is achieved by fixing eighteen thermocouples type K along tube wall as shown in Table 1. Furthermore, four thermocouples are inserted in the inlet and outlet of the test tube for measuring bulk air temperature. All the thermocouples are connected to selector switch which is connected to digital thermometer in order to measure all the system temperatures. The pressure drop along the test tube is measured by using U-tube manometer. Figure 2 shows the V-shape twisted jaw which is made from aluminum tape with thickness of 1 mm. Where, two twisted ratios (TR=2 and 4) are selected to be used in experiments as shown in Figure 2a and 2b.



(a)



(b)

Figure 1: (a) Schematic diagram of the using rig, (b) photograph of the using rig.

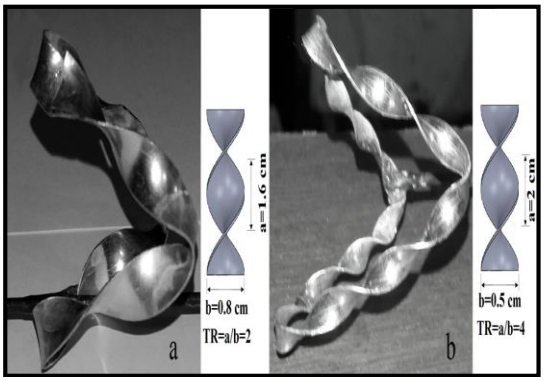


Figure 2: Details of V-shape twisted jaw.

Table 1: Thermocouples positions

No.	Position of Thermocouple (cm)
1	1
2	2
3	3
4	5
5	7
6	9
7	13
8	17
9	23
10	29
11	38
12	47
13	57
14	67
15	77
16	90
17	104
18	118

II. Experimental procedure

Firstly, the electric fan is switched on to pump air through system. Then, the value of flow rate is specified by adjusting the control valve with help of inclined manometer and orifice plate. After that, the required constant heat flux is adjusted. All the initial values of flow velocity, temperatures and pressure are measured. After that, the system is left for about 2.5-3 hours for achieving steady state condition, and each 15 min, the temperature distribution reading is recorded. Finally, after reaching steady state, the values of U-tube manometer reading (pressure drop) and temperatures are measured again. This procedure is repeated with two cases of TR and different numbers (N=6, 8 &10) of turbulators.

III .Study Plan

The main purpose of present investigation is enhancing heat transfer in heat exchanger therefore; the following points are adopted as a study plan:

1-Studying the effect of varying velocity flow by using 10 values of Reynolds number (Re starts from 6000 to 19500 with increment 1500 in each value).

2-Studying the effect of twisted ratio (TR=2 &4).

3-Studying the effect of inserting different numbers of turbulators in tube test section where 6, 8 and 10 pieces were selected in order to be distributed along the tube test section during performing experiments as shown in Figure 3.

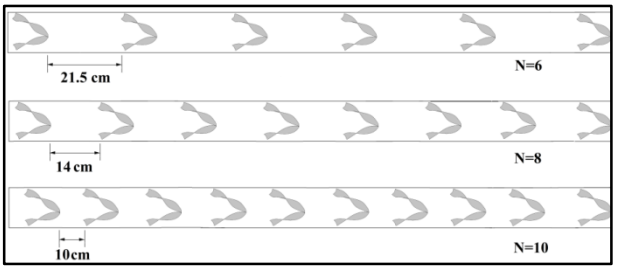


Figure 3: Inserting turbulators with different values of N

IV. Error analysis

The reliability of experimental facility can be achieved by determining the uncertainties of experimental data. The uncertainties of both Nusselt number and friction factor data can be expressed as follows [14].

- Nusselt number:

$$\left(\frac{E_{Nu}}{Nu}\right)^2 = \left[\left(\frac{E_v}{V}\right)^2 + \left(\frac{E_I}{I}\right)^2 + \left(\frac{E_{D_h}}{D_h}\right)^2 + \left(\frac{E_{\Delta T_s}}{\Delta T_s}\right)^2 + \left(\frac{E_{A_s}}{A_s}\right)^2 + \left(\frac{E_{\Delta T_{oi}}}{\Delta T_{oi}}\right)^2\right] \quad (1)$$

E_{Nu} =0.0367.

- Friction factor:

$$\left(\frac{E_f}{f}\right)^2 = \left[\left(\frac{E_{\Delta p}}{\Delta p}\right)^2 + \left(\frac{E_{Re}}{Re}\right)^2 + \left(\frac{E_{D_h}}{D_h}\right)^2\right] \quad (2)$$

E_f =0.33.

3. Mathematical Formulation

In this work, air is used as a working fluid and pumped through insulated tube under constant heat flux. In steady state case, the convective heat transfer rate can be written as follows:

$$Q_{air} = (V * I)_{in} - Q_{lo} = Q_{conv.} \quad (3)$$

$$Q_{air} = \dot{m}_a * C_{p_a}\left(T_{b,out} - T_{b,in}\right) \quad (4)$$

$$Q_{conv.} = \tilde{h} * A_{i,s} * \left(\tilde{T}_{ave.w} - T_b\right) \quad (5)$$

Where T_b is a bulk air temperature which calculated by equation (6):

$$T_b = T_{b,out} + T_{b,in} /2 \quad (6)$$

And $A_{i,s}$ is the inner surface area equal to πDL, and $\tilde{T}_{ave.w}$ is the average value of the outer tube surface temperature which calculated from 18 point of T_w lined along test tube and can be defined as:

$$\tilde{T}_{ave.w} = \sum_{i=1}^{18} T_{wi} \Big/ 18 \quad (7)$$

The average Nusselt number (Nu) was calculated as follow:

$$\widetilde{Nu} = \frac{\tilde{h} * D}{k_a} \quad (8)$$

The Reynolds number can be calculated by equation (9):

$$Re = \frac{u * D}{\nu_a} \quad (9)$$

The Friction factor is given by equation (10):

$$f = \frac{\Delta P}{(L/D) * (\rho u^2/2)} \quad (10)$$

In which, ΔP is the pressure drop across length of the test tube (L) , D is the inner diameter of tube test section and u^2 is the square of average air velocity. To evaluate the heat transfer enhancement, the thermal performance factor (η) was used for this purpose as shown in follow equation [15]:

$$\eta = \frac{Nu_{t\,j}/Nu_p}{\left(f_{tj}/f_p\right)^{1/3}} \quad (11)$$

All air thermo-physical properties (C_{p_a},k_a, ν_a and ρ) are estimated at bulk air temperature (T_b) from equation (6).

4. Results and Discussion

The average Nusselt number and friction factor values of plain tube case are validated with those from empirical correlations of Gnielinski [11] and Blasius [12] as shown in Figures 4 & 5. The present results show acceptable deviation within ±7 and ± 2.3 comparing with Gnielinski and Blasius correlations.

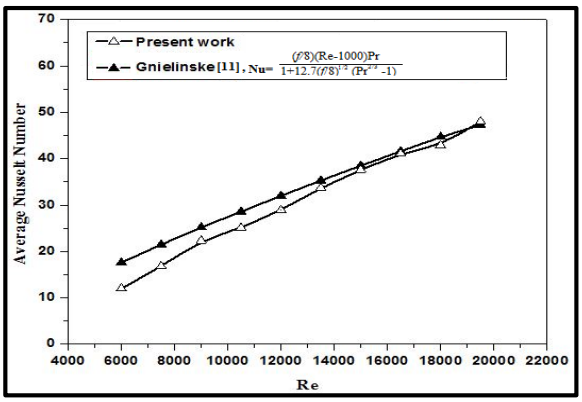


Figure 4: Validation of heat transfer rate for plain tube case.

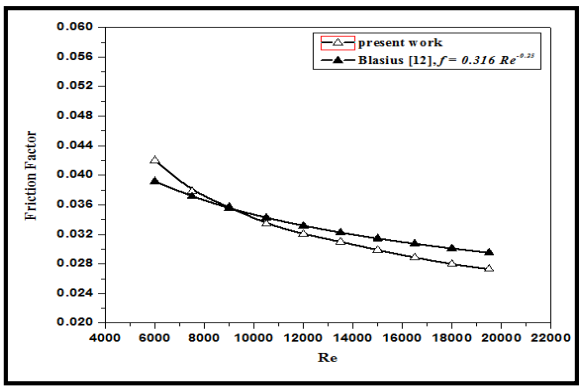


Figure 5: Validation of friction factor for plain tube case.

The results of average Nusselt number against Reynolds number for the two cases of TR and different numbers of turbulators are depicted in Figures 6 & 7. It is shown that Nusselt number values show increasing behavior with increasing Re value due to increasing vortex flow near the tube wall which leads to increment heat convection. In addition, inserting additional numbers of turbulators increases heat transfer compare with plain tube case by about of 73.74%, 112.37% and 160.29% for TR=2 and 47.5%, 73.66% and 102% for TR=4 for N=6, 8 and 10 respectively. This behavior can be attributed to that turbulence intensity will be increased as inserting more turbulators leading to reduce thermal boundary layer near the wall, hence, increasing in heat transfer rates. Moreover, it can be seen from these figures that TR =2 offers maximum heat transfer rate than TR=4 by about 55% because of the smaller TR make the swirl intensity stronger than the higher TR.

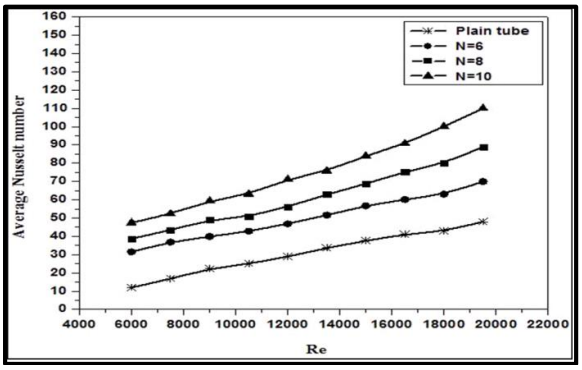


Figure 6: Variation of average Nusselt number agains (Re) for TR=2 and different numbers of turbulators.

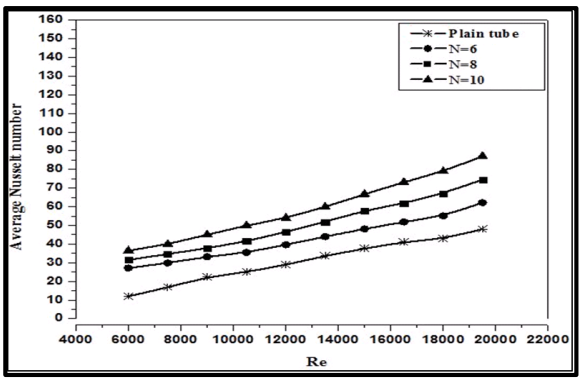


Figure 7: Variation of average Nusselt number against (Re) for TR=4 and Different numbers of turbulators.

Figures 8 & 9 show the relationship between friction factor (*f*) and Reynolds number values for the two TR cases and different N values. It is noted that considerable rising in friction factor with increasing number of turbulators' pieces due to increasing vortex flow intensity and surface area. In addition, TR=2 provides friction factor higher than those in TR=4 that is because of TR=2 provides larger surface area which provides flow resistance and reversing flow higher than those from another TR.

Thermal performance factor of inserting turbulators of TR= 2 & 4 are depicted in figures (10 & 11). It is important to find this given factor because of its importance in evaluating the feasibility of using turbulators in practical applications. It can be observed that using turbulators increases thermal performance factor more than unity indicating to that increasing friction loses due to inserting turbulators is less than the improving in heat transfer rate. In addition, the values of this factor decrease with rising Reynolds number. On the other hand, Thermal performance factor values show uptrend with adding additional pieces of turbulators.

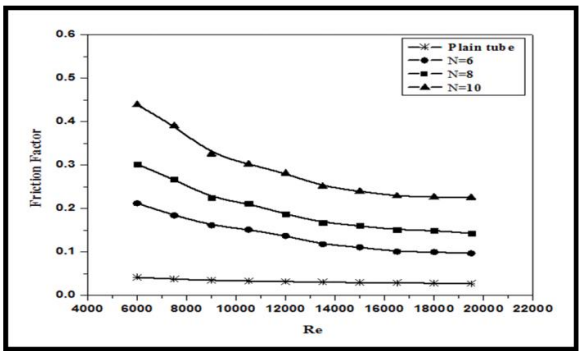


Figure 8: Variation of (f) against (Re) for TR=2 and different numbers of turbulators.

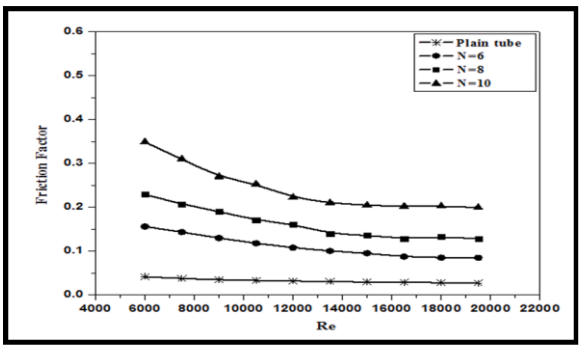


Figure 9: Variation of (f) against (Re) for TR=4 and different numbers of turbulators.

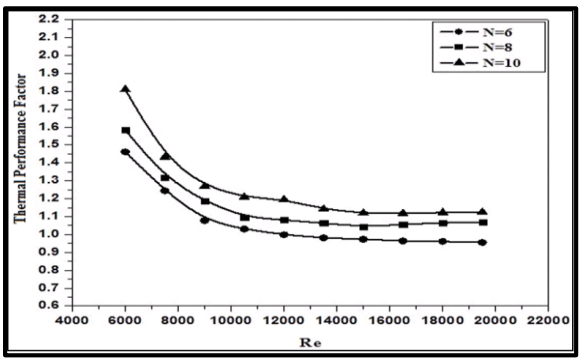


Figure 10: Variation of thermal performance factor against (Re) for TR=2 and different numbers of turbulators.

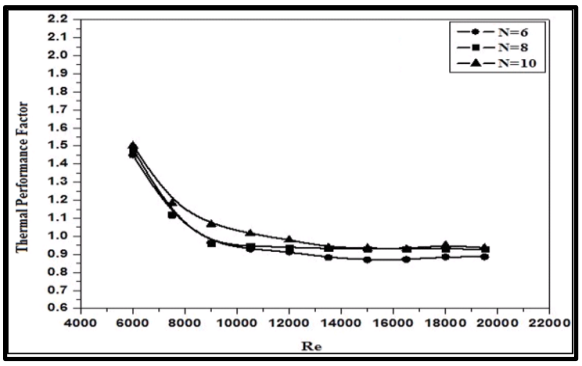


Figure 11: Variation of thermal performance factor against (Re) for TR=4 and different numbers of turbulators

The experimental results are correlated in empirical correlations that cover plain tube case as well as the two cases of TR.

For plain case:

$$Nu = 0.009Re^{0.87}Pr^{0.4} \quad (12)$$

$$f = 0.43Re^{-0.275} \quad (13)$$

For TR cases:

$$Nu = 0.055Re^{0.758}Pr^{0.4}\mathrm{N}^{-0.0212}TR^{-0.0352} \quad (14)$$

$$f = 1.99Re^{-0.479}\mathrm{N}^{1.429}TR^{-0.828} \quad (15)$$

5. Conclusions

Depending on the experimental results, the following conclusions can be written:

1- Reynolds number has positive relationship with heat transfer rate, while it has reverse relationship with friction factor.
2- Using different numbers of present turbulators increases Nusselt number values considerably (N=10 provides maximum enhancement by 160.29% for TR=2 and 102% for TR=4.
3- Twisted ratio TR=2 improves heat transfer rate more than TR=4 by around 55%.
4- Thermal performance factor values of both TR cases exceed unity indicating to the usefulness of these turbulators in practical applications.

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7. Nomenclature

$A_{i,s}$ = Inner Surface Area of tube Test Section (m^2).
a = Ratio of Twist Length in 180° rotation (m).
b = Tape Width (m).
Cp_a = Specific Heat of Air (J/kg.K).
D = Hydraulic Diameter (m).
E = Uncertainty Interval.
f = Friction Factor.
f_p = Friction Factor of Plain tube Case.
f_{tj} = Friction Factor of Inserting Turbulators.
$\tilde{h}$ = Average Convection Heat Transfer Coefficient ($W/m^2.K$).
I = Current (A).
k_a = Thermal Conductivity of Working Fluid (W/m.K).
L = Length of Tube Test Section (m).
$\dot{m}_a$ = Air Mass Flow Rate (kg/s).
N = Number of Turbulators pieces.
Nu_p = Nusselt Number of Plain tube Case.
$Nu_{t,j}$ = Nusselt Number of inserting Turbulators.
Pr = Prandtl Number.
TR = Twisted Ratio (a/b).
Q_{lo} = Heat losses (W).
$Q_{conv.}$ = Convective Heat Transfer from the Test Section (W).
Re = Reynolds Number.
$T_{b,in}$ = Temperature of Air at the Test Section Entrance (° K).
$T_{b,out}$ = Temperature of Air at the Test Section Exit (° K).
T_b = Bulk Air Temperature, (°K).

$\tilde{T}_{ave.w}$ = Average Surface Temperature of test Section (^{o} K)
T_{wi} = Local surface Temperature of test Section (^{o}K).
u = Velocity of Working Fluid (m/s).
V = Voltage, Volt.
ΔP = Pressure Drop (N/m^2).
η =Thermal performance factor.
ρ = Density (kg/m3).
ν_a = Kinematic Viscosity (m2/s).
ΔT_s = Surface to bulk air temperature.
ΔT_{io} =Average outer and inner lagging surface temperature.

Author (s) biography

Akram H. Abed is a lecturer at Electro-Mechanical Engineering, Energy Department, Renewable Energies Branch, University of Technology – Baghdad, Iraq, since July 2014. His main research interests are heat transfer, fluid mechanics and solar energy. He has published more than 10 articles in journal and conference proceedings at international and local levels.

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