Numerical Investigation of Heat Transfer Enhancement by Plain And Curved Winglet Type Vertex Generators

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Abstract— Numerical simulations were carried out to investigate the performance of plane and curved rectangular winglet type vortex generators (VGs). Effects of shape of the VGs on heat transfer enhancement were evaluated using dimensionless numbers - j/j0, f/f0 and h =(j/j0)/(f/f0). The results showed that curved winglet type VGs have better heat transfer enhancement than plain winglet type in both laminar and turbulent flow regions. The flow resistance is also lower in case of curved winglet type than corresponding plane winglet VGs. The best results for heat transfer enhancement is obtained at high Reynolds number values (Re > 10000) by using VGs. The processes in solving the simulation consist of modeling and meshing the basic geometry of rectangular channel with VGs using the package ANSYS ICEM CFD 14.0.Then the boundary condition will be set according to the experimental data available from the literature. Finally result has been examined in CFD-Post. This work presents a numerically study on the mean Nusselt number, friction factor and heat enhancement characteristics in a rectangular channel having a pair of winglet type VGs under uniform heat flux of 416.67 W/m2. The results indicate the advantages of using curved winglet VGs for heat transfer enhancement.

Keywords: Heat transfer enhancement, Vertex generators, Winglet types, Rectangular channel, Numerical investigation, CFD, Flow simulation.

I. INTRODUCTION

Computational Fluid Dynamics (CFD) is a useful tool in solving and analyzing problems that involve fluid flows and heat transfer to the fluid. As a kind of passive heat-transfer enhancing devices, vortex generators (VGs) have been widely investigated to improve the convective heat transfer coefficient (usually the air-side) of plate fin or finned tube type heat exchangers. The basic principle of VGs is to induce secondary flow, particularly longitudinal vortices (LVs), which could disturb the thermal boundary layer developed along the wall and ensure the proper mixing of air throughout the channel by means of large-scale turbulence [1]. Among the various types of VGs, wings and winglets have attracted extensive attention since these VGs could be easily be mounted on the channel walls or fins and could effectively generate longitudinal vortices for high enhancement of convective heat transfer. However, the heat transfer enhancement (HTE) by LVs is usually accompanied with the increase of flow resistance. Experimental research by Feibig et al. [2] showed the average heat transfer in laminar channel-flow was enhanced by more than 50% and the corresponding increase of drag coefficient was up to 45% by delta and rectangular wings and winglets. Further experiment with double rows of delta winglets in transitional channel flow by Tiggelbeck et al. [3] showed that the ratio of HTE and drag increase was larger for higher Reynolds numbers. Feibig [4] also pointed out that the winglets are more effective than wings, but winglet form is of minor importance. Recently, Tian et al. [5] performed three dimensional simulations on wavy fin-and-tube heat exchanger with punched delta winglets in staggered and in-line arrangements and their results showed that each delta winglet generates a downstream main vortex and a corner vortex. For Re = 3000, compared with the wavy fin, the Colburn j-factor and friction f-factor of the wavy fin with delta winglets in staggered and in-line arrays are increased by13.1%, 7.0% and 15.4%, 10.5%, respectively. Chu et al. [6] numerically investigated the three row fin-and-oval-tube heat exchanger with delta winglets for Re = 500-2500. They reported that, compared with the baseline case without LVGs, the average Nu with LVGs was increased by 13.6-32.9% and the corresponding pressure drop was increased by 29.2-40.6%, respectively.

A. NOMENCLATURE

A_c	cross sectional area of air channel (m^2)
A_i	heat transfer area of each small element on copperplate (m^2)
A_p	heat transfer area of copper plate in tested channel (m^2)
b	width of vortex generator (mm)
CRWP	curved rectangular winglet pair
CTWP	curved trapezoidal winglet pair
HTE	heat transfer enhancement
C_p	specific heat (J /kg °C)
D	hydraulic diameter of the air channel (m)
f	Darcy friction factor
f_0	Darcy friction factor of smooth channel (i.e. without VG)
h	height of VG-trailing edge (mm)
h_c	convective heat transfer coefficient (W /m^2°C)
j	Colburn factor
j_0	Colburn factor of smooth channel (i.e. without VG)
l	length of vortex generator (mm)
L	length of tested channel along air flow direction (m)
LVG	longitudinal vortex generator
Nu	Nusselt number
p	pressure (Pa)
P	electric power (W)
Pr	Prandtl number
Q	heat transfer rate (W)
Re	Reynolds number
S_1	front edge pitch of a pair of vortex generators (m)
S_2	distance of vortex generator pair downstream the in let of the test section (m)
T	temperature (K)
U	velocity (m/s)
VG	vortex generator
K	thermal conductivity

Greek letters

α inclination angle of VG (º)
β attack angle (º)
r density (kg m_3)
DP pressure drop (Pa)
µ dynamic viscosity (Pa s)
h thermal enhancement factor

Subscripts

a air
c cross section or convective
e effective
E expanded
i number of thermocouple or element
in inlet
m average
out outlet
w wall
x coordinate along flow direction of the tested air channel (m)

The above experimental and numerical results show that pressure drop penalty is comparative with the heat transfer enhancement caused by the LVGs. Under some conditions, the increase of pressure drop can be even 2-4 times higher than the heat transfer enhancement by LVGs [7-9], which weakens the advantages of LVGs. Chen et al. [10] pointed out that the form drag of the LVGs is predominant for the pressure drop, and the LVs the many additional pressure drop of the flow. Minet al. [11] developed a modified rectangular LVG obtained by cutting off the four corners of a rectangular wing. Their experimental results of this LVG mounted in rectangular channel suggested that the modified rectangular wing pairs (MRWPs) have better flow and heat transfer characteristics than those of rectangular wing pair (RWP). In recent work, a new kind of vortex generator-curved trapezoidal winglet (CTW) [12] was developed and experimental results indicated its advantages of heat transfer enhancement and low flow resistance due to the streamlined configuration. Under some small attack angles, the curved trapezoidal winglet pair (CTWP) presents thermo hydraulic performance with (j/j0)/(f/f0)as high as 1.6.Habchi et al. [13] numerically investigated the performance of trapezoidal wing with excavation at the bottom. The results showed that the excavation really reduced the flow resistance, but on the other side, the cavity also reduces the contact surface between the heated wall and the vortex generator and thus reduces the conduction heat flux through the vortex generator; as a result, convective heat transfer between the vortex generator and the surrounding fluid is decreased. Therefore, the size of the cavity should be optimized to maximize the effect of heat transfer enhancement and flow resistance reduction Numerical study by Biswas and Chattopadhyay [14] on delta wing with punched hole in base wall showed that heat transfer enhancement and friction factor (f × Re) at the exit are both relatively lower than those of the case without any punched hole. To address the heat transfer enhancement in recirculation zone as well as flow drag reduction, the present paper attempt to numerically investigate the plane rectangular winglets as well as recently developed curved rectangular winglets and CFD simulation were performed to examine the effect of these kinds of VGs on air-side heat transfer enhancement and flow resistance in channel flow. Then, the average convective heat transfer coefficient was measured and dimensionless numbers j/j0, f/f0 and heat enhancement performance factor h=(j/j0)/(f/f0)were used for performance evaluation. The effect of the shape of winglet pair on the performance of these VGs were then evaluated.

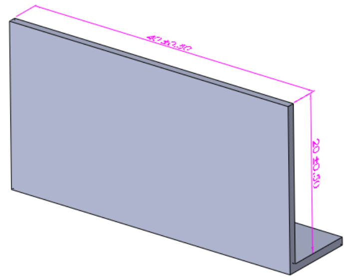

Fig. 1: Rectangular Winglet (RW)

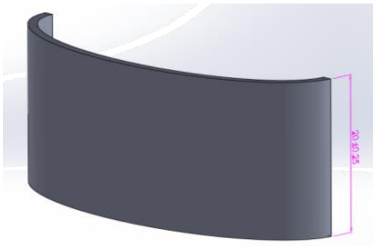

Fig. 2: Curved Rectangular Winglet (CRW)

II. NUMERICAL SIMULATION

A. Physical Model

The numerical simulations were carried out using FLUENT V6 Software that uses the finite-volume method to solve the governing equations. Geometry was created in CATIA Design tools for air flowing through an electrically heated rectangular channel with copper plate at bottom of 1000mm×300mm and the dimension of the channel is 1000mm×240mm×30mm. Meshing has been created in ICEM CFD 14.0 with tetrahedral shapes (Fig. 2). In this study Reynolds number varies from 750 to 21000.

B. Numerical Method

For turbulent, steady and incompressible air flow with constant properties .We follow the three-dimensional equations of continuity, momentum and energy, in the fluid region.

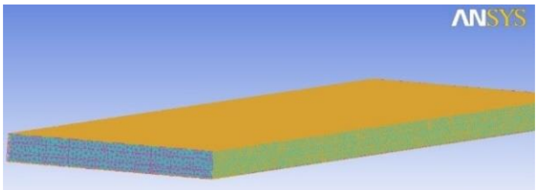

Fig. 3: Meshing in ICEM CFD 14.0

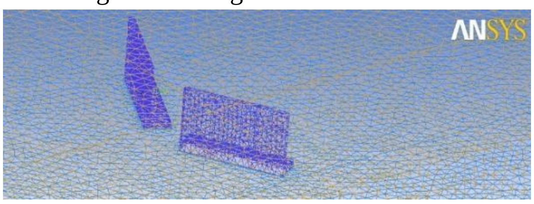

Fig. 4: Meshing of RWP

These equations are below:

Continuity equation:

$$\frac{\partial p}{\partial t} + \nabla.(rV) = 0 \ \ldots\ldots.\ (1)$$

Momentum equation:

$$\frac{\partial}{\partial x_i}(\rho u_i u_j) = \frac{\partial}{\partial x_i}\left(\mu \frac{\partial u_j}{\partial x_i}\right) - \frac{\partial p}{\partial x_j}. \quad \ldots\ldots.\ (2)$$

Energy equation:

$$\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_i}\left(\frac{k}{C_p}\frac{\partial u_j}{\partial x_i}\right). \quad \ldots\ldots\ldots.\ (3)$$

Table. 1: Properties of air at 25^{o}C

Properties	value
Specific heat capacity, C_p	1006 J/kg K
Density,	ρ 1.225 kg/m^3
Thermal conductivity, k	0.0242W/m K
	1.7894 x10^{-5} kg/m s

Velocity and pressure linkage was solved by SIMPLE algorithm. For validating the accuracy of numerical solutions, the grid independent test has been performed for the physical model. The tetrahedral grid is highly concentrated near the wall regions and also near the VGs

Table. 2: Nodes and Element in geometry are below.

Geometry type	Nodes	Elements
Smooth	159803	1427375
RWP	179461	1945697
CRWP	183820	1975908

Table 2 shows that CRWP)have maximum nodes and element in comparison of smooth and RWP. In addition, a convergence criterion of $10^{-7}$was used for energy and 10^{-3} for the mass conservation of the calculated parameters. The air inlet temperature was specified as 293 K and three assumptions were made in model: (1) the uniform heat flux was along the length of rectangular channel. (2)Wall of the channel will be perfectly insulated. (3) Steady and incompressible flow. In fluent, inlet was taken as velocity-inlet and outlet was taken as pressure-outlet.

C. Data Reduction.

Three important parameters were considered-friction factor, Nusselt number and thermal performance, which determined the friction loss, heat transfer rate and the effectiveness of heat transfer enhancement in the rectangular channel, respectively.

The friction factor (*f*) is investigated from pressure drop, DP across the length of rectangular channel (L) using the following equation:

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f = 2DpD / (LU^{2}r) ….. (4)

The Nusselt number is defined as

$$Nu_L = \frac{hL}{k} = \frac{convective heat\,transfer}{conductive heat\,trasnfer} \ ..\ (5)$$

The Nusselt number and the Reynolds number were based on the average of the channel wall temperature and the outlet temperature, the pressure drop across the test section, and the air flow velocity were measured for heat transfer of the heated wall with different kind of VGs. The average Nusselt numbers and friction factors were obtained and all fluids properties were found at the overall bulk mean temperature.

Thermal performance factor was given by:

$h = (j/j_0)/(f/f_0)$ …(6)

Where $j_0$and j and f and f_0 were the Colburn factor and friction factors for the smooth channel and channel with VGs respectively.

III. Results and Discussion

A. Validation of setup

The CFD numerical result of the smooth channel without any VG been validated with the experimental data as shown in Figures3.1 and 3.2. These results are within ±9% deviation for heat transfer (Nu) and ±3% the friction factor (*f*) with each-other. In low Reynolds number the deviation become small in experimental and CFD results but when Reynolds number become more then these deviation slightly higher in experimental and CFD results, respectively.

B. Heat Transfer

Effect of the VGs of RWP & CRWP types on the heat transfer rate is presented in Figure- 3.3.The results for the dimensionless number (*j/j$_0$*) of channel with RWP & CRWP Vs Reynolds number is shown. All the Reynolds numbers used due to the induction of high reverse flows and disruption of boundary layers. We clearly seen that as the Reynolds number goes on increasing, the heat transfer coefficient also goes on increasing or Colburn factor.

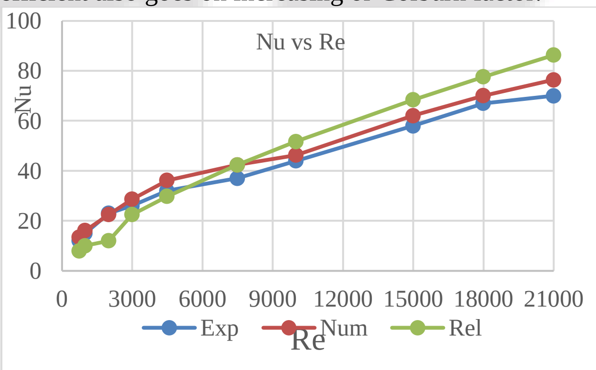

Fig. 5: Nusselts number Vs Reynolds number

We clearly seen that as the Reynolds number goes on increasing, the heat transfer coefficient also goes on increasing or Colburn factor. The channel with RWP & CRWP increase the heat transfer rate by average 60% & 58% respectively than the smooth channel.

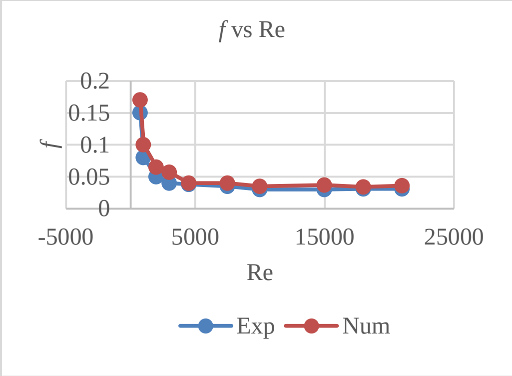

Fig. 6: Friction factor Vs Reynolds number

C. Friction Factor

The variation of the pressure drop is presented (eq.4) in terms of friction as Figure 3.4.It shows the friction factor Vs the Reynolds number, for RWP & CRWP in rectangular channel. It is seen that friction factor decreases with an increases in Reynolds number. It was found that the pressure drop for the RWP & CRWP was average 59% & 51% respectively. The curved shape winglet pair have lower friction in comparison with the plain type winglet pair.

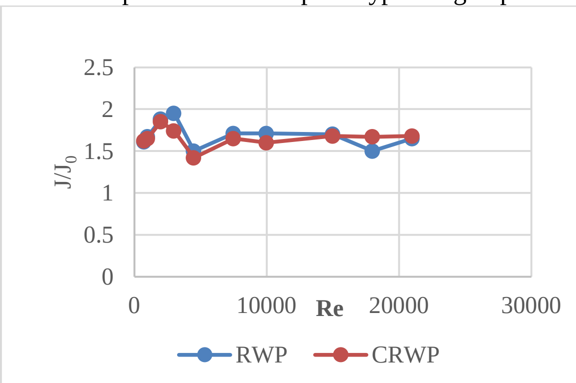

Fig. 7: *(j/j_0)*Vs Reynolds number

D. Thermal Performance Factor

From Figure 3.5, it hasbeen observed that the thermal performance factor is high for CRWP in comparison with RWP. It was also observed that the thermal enhancement factor is increase as the Reynolds number increases. The maximum enhancement factor was observed at Reynolds number upper limit consideration I,e Re = 21000for the present study.

Fig. 8: *(f/f_0)* Vs Reynolds number

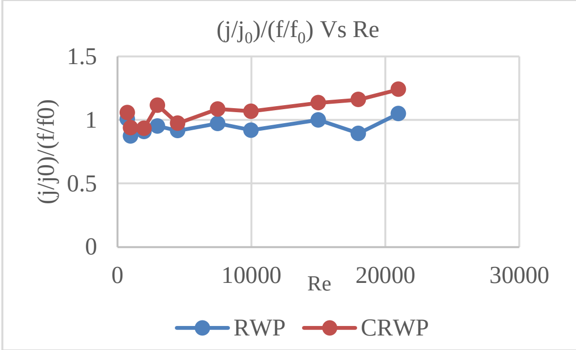

Fig. 9: *(j/j_0)/(f/f_0)* Vs Reynolds number

The figure 4.1 to 4.3 shows the temperature distribution for RWP at different flow zones I,e laminar, transient and turbulent zone and figure 4.4 to 4.6 shows the pressure distribution for the same. The figure 4.7 to 4.9 shows the temperature distribution for CRWP at different flow zones and figure 4.10 to 4.12 shows the pressure distribution for CRWP.

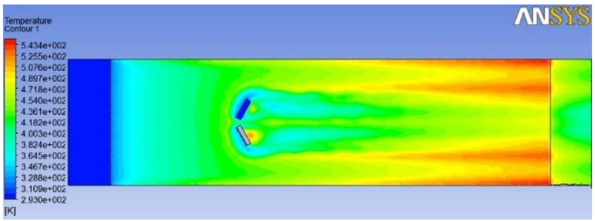

Fig. 10: Temperature distribution at velocity 0.165 m/s

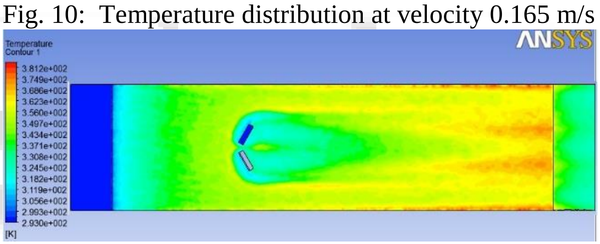

Fig. 11: Temperature distribution at velocity 0.662 m/s

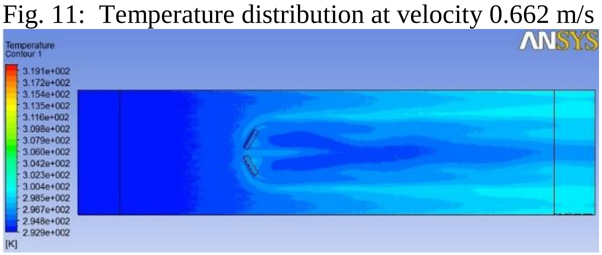

Fig. 12: Temperature distribution at velocity 3.31 m/s

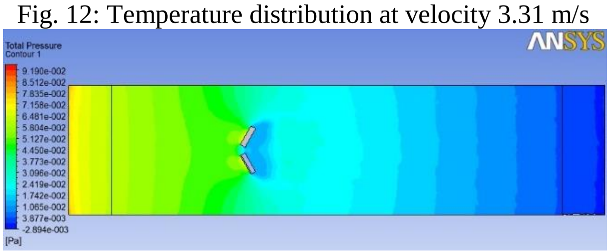

Fig. 13: Pressure distribution at velocity 0.165 m/s

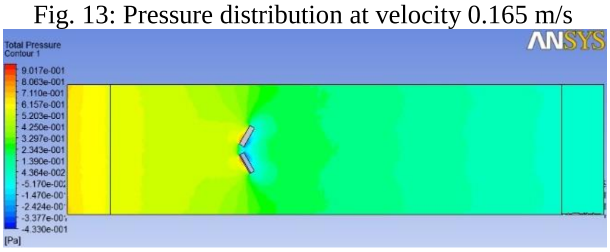

Fig. 14: Pressure distribution at velocity 0.662 m/s

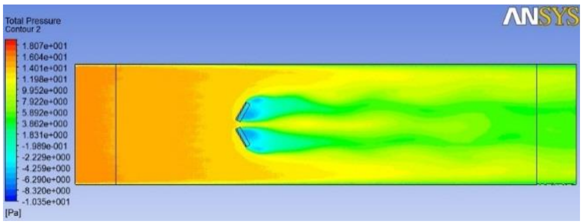

Fig. 15: Pressure distribution at velocity 3.31 m/s

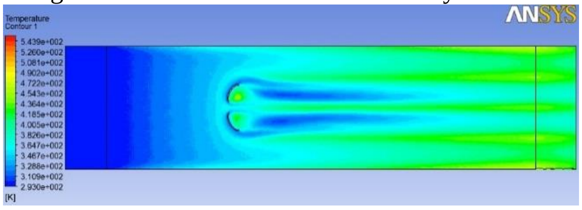

Fig. 16: Temperature distribution at velocity 0.165 m/s

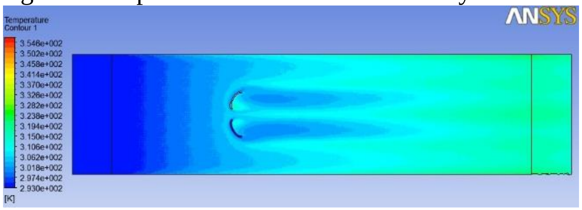

Fig. 17: Temperature distribution at velocity 0.662 m/s

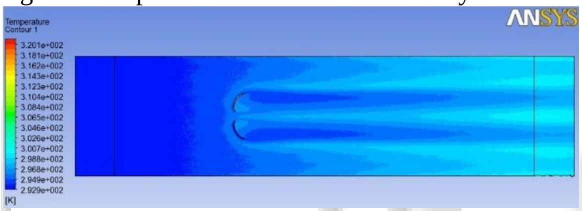

Fig. 18: Temperature distribution at velocity 3.31 m/s

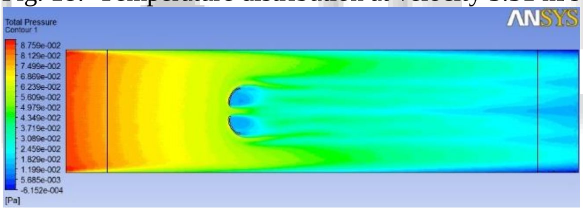

Fig. 19: Pressure distribution at velocity 0.165 m/s

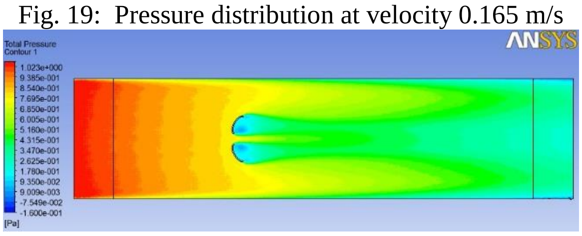

Fig. 20: Pressure distribution at velocity 0.332 m/s

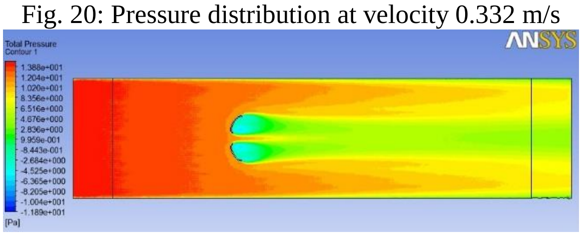

Fig. 21: Pressure distribution at velocity 3.31 m/s

IV. CONCLUSION

When the winglet type of VGs (RWP & CRWP) are fitted in a rectangular channel, the effect on the heat transfer (Nu) or Colburn factor, friction factor (f) and thermal performance factor (h) have been investigated numerically by using ANSYS-14 software. The following conclusions are below:

We clearly seen that as the Reynolds number goes on increasing, the heat transfer coefficient also goes on increasing .The RWP in rectangular channel increase heat transfer by average 60% and CRWP is by average 58% more than smooth channel.

Pressure drop for the RWP configuration are 59% more than smooth channel and for the CRWP configuration is 51% more than the smooth channel.

It has been observed that the thermal enhancement factor tends to decreases at low values of Reynolds number and it increases at high values of Reynolds number.

Overall it is concluded that the use of winglet type VGs enhance the heat transfer and the curved type of winglet pair are more effective in heat enhancement than plain winglet type VGs.

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