INTERNATIONAL JOURNAL OF ADVANCED STUDIES IN COMPUTER SCIENCE AND ENGINEERING
IJASCSE VOLUME 11 ISSUE 2, 2022
02/28/2022

Experimental Study on Cooling of an Electronic Component in a Vertical Duct by Free and Forced Convection

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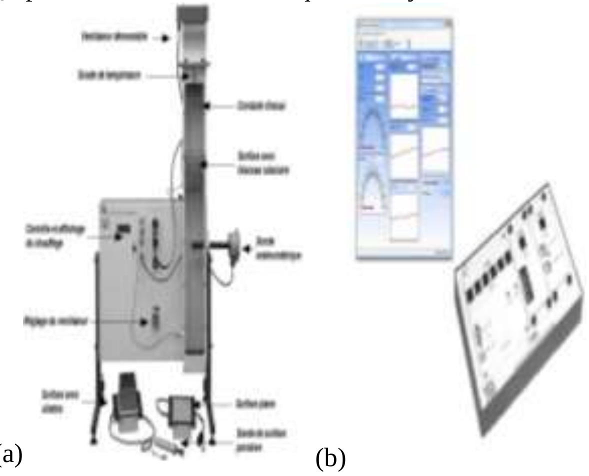
***Abstract*–The purpose of this study is to compare experimentally three electronic components : A Flat Plate, A Pinned Surface – similar to a tubular heat exchanger, A Finned Surface – similar to the fins on air-cooled engines or electrical heat sinks, in both heat transfer modes (free and forced) from the point of view Nusselt Number for forced and free convection by using TecQuipment's free and forced convection apparatus, by imposing a constant thermal flux of different values while calculating the cooling efficiency indicators as: the convective heat exchange coefficient (h), the number of Nusselt (Nu), and the average temperature Tm.**

This experimental study showed that the forced convection beam plate, is the best point of view cool it as is illustrated in the chapter of calculation.

Keywords: mixed; natural; forced convection; electronic component; cooling; Nusselt number.

I. INTRODUCTION

TecQuipment’s free and forced convection apparatus allows to perform heat transfer experiments for three different surfaces, both free (or natural) and forced convection. For automatic acquisition of the results of your experiments and to save time, this device can be operated with TecQuipment’s universal data acquisition system, VDAS®.



(a) (b)

Figure 1. Presentation of the free and forced convection (a) device TD1005 (b) The data acquisition system (interface and software).

Heat dissipation problems have a great importance in the field of the electronics industry. the study and control of these problems is essential to ensure guarantees of operation and reliability. Excessive heat generation requires good and sufficient heat removal, otherwise the service life of electronic components will be affected by this excessive heat generation. Bouttout [1]

Much of the numerical and experimental work has been done by researchers and scientists regarding convection in its three forms, among these works: Icoz and Jaluria [2] who made a numerical simulation of natural convection in two dimensions, in an open rectangular channel and containing two identical heat sources. The results indicate that the dimensions of the channel and the presence of openings have considerable effects on the flow of the fluid. Adel and Rachid [3] have numerically studied the mixed convection cooling of electronic components mounted in a horizontal rectangular channel. Manca et al. [4] Have made an experimental study of the effect of the distance between a discreetly heated plate and another placed parallel to the first on convection. Bazylak et al. [5] performed an estimated numerical analysis of heat transfer due to sources on the lower wall of a horizontal enclosure. They found that the optimum rates of heat transfer and the onset of thermal instability depend on the length and spacing of the sources and the aspect ratio of the enclosure. Gunes [6] Made an analytical study of natural convection in a vertical channel containing heat sources, he drew analytical expressions describing the variations of variable fields in stationary regime, in two and in three dimensions. He found that for small Grashof numbers, these expressions are in excellent agreement with numerical solutions throughout the computational field. the variation in the Nusselt number was obtained by the author. Calgani et al. [7] experimentally and numerically studied the transfer of heat in natural convection in square enclosures heated from below and cooled from the side walls. Numerical and experimental studies show a conductive transfer for $Ra \leq 10^4$ (Rayleigh number), while the local Nusselt number Nu is evaluated on the surface of the heat source and exhibits a symmetrical appearance near the heat sources. Furukawa and Yang [8] have developed a numerical method to know the thermal behavior of a fluid flowing into two parallel planes where there are heat generating blocks (heat sources).Bhowmik and Tou [9] carried out experiments to study single-phase transient heat transfer in forced convection. The experimental device comprises four (4) heat sources mounted in a rectangular vertical channel. Experimental results indicate that heat transfer is strongly related to the number of sources and the Reynolds number. Fu and Tong[10] have made a numerical simulation of forced convection, studying the influence of an oscillating cylinder on heat transfer in a number of heat sources subjected to flow in a horizontal channel. The results show that heat transfer increases with increasing Reynolds number and improves remarkably for a large oscillation of the cylinder.

Timothy and Vafai [11] conducted a detailed investigation into forced convection cooling of a set of heat sources on the lower wall of a channel. The results of this investigation show that the shape and material of the source have considerable effects on the characteristics of flow and heat transfer. Young and Vafai [12] have done a numerical investigation of forced convection for a compressible fluid in a channel with heated obstacles mounted on its lower wall. They studied the effects of Reynolds number, height, width and spacing of obstacles as well as their thermal conductivity. The results show that all these parameters have remarkable influences on the variation of the average Nusselt number of the components of the velocity and temperature distribution within the fluid. Mohamed [13] has made an experimental investigation in order to know the characteristics of air cooling in a cooling device. The results indicate that the average heat transfer coefficient increases slightly with the increase in the temperature of the module device. Korichi et al. [14] have made a numerical study of transient laminar forced convection, in a channel in the presence of a cylinder of square section. The average Nusselt number increases with the increase in the Reynolds number. Papanicolaou and Jaluria [15] made a numerical simulation of the transient mixed convection from laminar stationary regime to periodic regime in an aerated two-dimensional cavity and subjected to local heating. The results show that as soon as the critical Grashof number is exceeded, an unstable situation arises. Bhowmik et al.[16] performed steady-state experiments to study the mixed convection heat transfer of four electronic heating elements placed in a vertical rectangular channel. The effects of heat flow, flow, geometric parameters and the number of heating elements were examined. Experimental results indicate that heat transfer is strongly related to Reynolds number. Icoz and Jaluria [17] Have developed a methodology for the design and optimization of cooling systems for electronic equipment. This investigation shows that we can use the results that make it possible to achieve adequate and optimal geometries in order to have the best cooling device for electronic components.

Wang and Jaluria [18] have numerically studied the transfer of conjugated heat in a three-dimensional rectangular pipe with two heat sources as part of the cooling of electronic equipment. The results show that the Reynolds number, the spatial arrangement of heat sources and the ratio of thermal conductivities have considerable effects on the improvement of heat transfer.

A numerical study was presented by Islam et al.[19] on the heat transfer by mixed convection in laminar stationary regime, and in particular, at the entrance of the horizontal annular part of two coaxial cylinders. Studies reveal that increasing the Rayleigh number improves heat transfer, that the average Nusselt number increases with the aspect ratio and with the Prandtl number.

Chen et al. [20] have made a combination that consists of experimental visualization and temperature measurement, to be able to study the possible stabilization and elimination of instability due to buoyancy force, in mixed convection, in a horizontal pipe. This was done by placing a heated plate at the top of this pipe.

Lin and Chen [21] Have made a numerical study of thermal instability in a mixed convection flow on horizontal and inclined plates. The results show that increasing the inclination stabilizes the thermal instability and has no pronounced effects on the Nusselt number and increases the value of the critical Grashof number.

Bousedra et al. [22] experimentally studied mixed convection in a laminar flow of water in the inlet region of a semicircular pipe with ascending and descending inclinations. The results reveal that for ascending inclinations, the Nusselt number and wall temperature increase with the Grashof number. For descending inclinations, the Reynolds number has a very important effect on the average Nusselt number.

Yoo [23] presented a numerical study of the mixed convection of air flow, between two concentric cylinders maintained at constant and then different temperatures. Forced flow is induced by the cold outer cylinder that rotates slowly with a constant angular velocity. The circulation of fluid in the direction of rotation of the cylinders decreases by increasing the Rayleigh number. The overall heat transfer to the wall is rapidly reduced when the critical value of the Reynolds number at transition is reached. Kim et al. [24] have numerically studied the characteristics of mixed convection flow and heat transfer in a channel with heat sources attached to a canal wall. The results found of the local Nusselt number along the surfaces of the sources and the temperature distribution and the density of the heat flux on the surface of the plates, indicate that it can be said that overly simplifying assumptions are not appropriate to simulate the cooling of electronic equipment.

Abid et al. [25] carried out an experimental study on the spatio-temporal intermittency of cylindrical flow in a horizontal conduit in mixed laminar convection, measuring the temperature gradient on the wall. The results show that for high flow values there appears an intermittent phenomenon for which the temperature gradient varies over time with large amplitudes depending on the position of the section studied and the velocity of the fluid. This is the phenomenon that the authors have tried to characterize, examining it as a transition from the laminar regime to the turbulent regime. Leong et al. [26] have numerically studied the heat transfer resulting from the mixed convection of a bottom of an open cavity heated and subjected to an external air current. Chang and shian [27] conducted a numerical investigation with the intention of investigating the effects of a horizontal partition on mixed convection heat transfer characteristics with pulsating flow, in an open channel. Habchi and Acharya [28] have made a digital investigation of the mixed convection of air in a vertical channel, containing an obstacle on one of its supposedly heated walls, while the other is considered adiabatic or heated as well (two cases). The results indicate that the average Nusselt number upstream and at the obstacle level increases as Ri decreases. Behind this obstacle, the Nusselt number decreases as the Grashof number increases. For both cases, the average Nusselt numbers are smaller than those in a smooth duct. Banarjee et al [29] studied the passive cooling by natural convection of two semiconductors arranged horizontally on the lower surface of a square enclosure whose walls are thermally insulated. They concluded that there is a specific length of the component that produces a maximum constant temperature on each component. The aim of our study is to Compare free and forced convection for different surfaces and to make a Comparison also of heat transfer surface efficiency and Nusselt Number.

II. HEAT EXCHANGE SURFACES

The TecQuipment bench is supplied with three different heat exchange surfaces allowing to compare their performance according to their characteristics. Each heat exchange surface is mounted simply and quickly in a square opening placed at the rear of the pipe. A rubber seal placed around each surface ensures sealing to prevent parasitic heat transfer by convection when the studied surface is in place.

II.1 The Flat Plate

The flat plate is a simple plate made of flat aluminum. This surface is unique compared to the other two surfaces in the sense that it is completely in alignment with the inner wall of the pipe, without any element placed in the airflow passing through the pipe.

Figure 2 . The Flat Plate

II.2 The Surface with Fins

This is the most well-known surface, used in particular as a heat sink in most electrical and electronic circuits. It is also used for combustion engines or air-cooled compressors. It effectively increases the exchange surface ensuring better heat transfer with the surrounding air both to cool and to recover heat. This surface is very useful for demonstrations in free convection both vertically (the fins being placed vertically) and horizontally (with fins placed horizontally). The holes on the side of the pipe allow the portable temperature probe to be introduced to measure the temperature (and therefore heat) distribution along a fin.

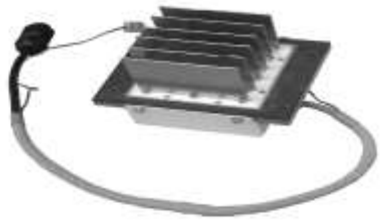


Figure 3. The surface with fins

II.3 The Surface with Tubular Beams

This type of surface is most common in the case of cross-current heat exchangers, where one of the fluids flows into the hollow tubes perpendicular to the second fluid that passes around the tubes. Part of the energy of the hot fluid is transmitted through the surface of the tubes to the cold fluid. Thus, as in the case of surfaces with fins, the exchange surface is effectively increased favoring the transfer of heat with the surrounding fluid (with air in the case of this device). The holes on the side of the pipe allow the portable temperature probe to be introduced to measure the temperature (and therefore heat) distribution along a fin.

Figure 4. The surface with a tubular beams

III.CHARACTERIZATION OF THE HEAT TRANSFER

III.1. Heat transfer coefficient (convective) (h_c)

The heat transfer coefficient is the ability of one material to transfer heat to another material. (These values show that heat transfers to air are better in forced convection than in free convection).

$$h_c = \frac{Q}{A_s.T_\infty} \quad \textbf{(1)}$$

With Tm, the logarithmic mean temperature and(Q)˙the heat flow between the surface and the air.

$$T_\infty = \frac{T_{out} - T_{in}}{\ln\frac{T_s - T_{in}}{T_s - T_{out}}} \quad (2)$$

III.2. Nusselt number (Nu)

The Nusselt number applies to heat transfer. It is a dimensionless number representing the ratio of total heat transfer to conduction transfer.

$$\mathrm{Nu} = \frac{\mathrm{h_c.l}}{\mathrm{k_{air}}} \qquad (3)$$

With L, the characteristic length of the surface in contact with moving air (for a flat plate, it is simply the length of the plate).

III.3. Heat flow transferred to the air

In the case of a perfect theoretical application, the heat flow transferred to the air can be determined from:

$$\dot{Q} \ = \dot{m}.C_{pc}.\Delta T \qquad (4)$$

However, the air velocity in the pipe is not uniform in a section. There is a velocity profile and thus a variation in the mass flow of air m˙(see Figure 8).

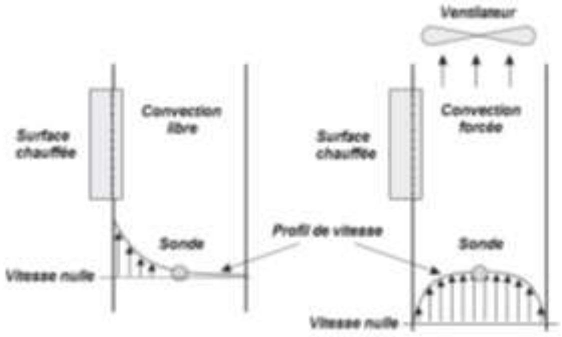


Figure 5. Velocity Profile

TecQuipment has designed this equipment to have minimal heat loss - the insulating materials used minimize conduction heat loss and the bare metal surface helps minimize radiation loss. This means that the losses to the ambient environment will be low and that the electrical power absorbed at the surface can be considered almost equivalent to the heat applied to the air in the duct, which means that:

$$\mathrm{W} \approx \dot{\mathrm{Q}} \qquad (5)$$

III.4. Downstream Air Temperature (T_{OUT})

Due to the location of the heat source on the side, the air temperature in the duct downstream of the heated surface may not be identical in the duct section, giving a temperature profile through the duct and a variation of T_{OUT} (see figure Temperature profile).

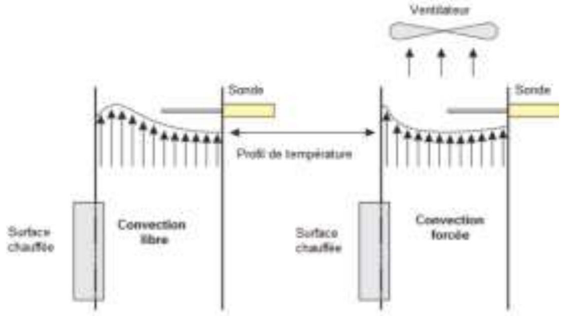


Figure 6. Temperature profile

IV. Experimental approach

In this part, six different experimental works have been done in order to know the best geometry of electronic component from point of view cooling. It is an optimization based on a comparative study in the case of natural and forced convection for different heat fluxes ($\dot{Q}$=10W, $\dot{Q}$=20W , $\dot{Q}$=30W). Histograms are shown to facilitate the reading of the results obtained for each experimental work.

IV.1. Natural convection

In most applications a "cold source" cools a critical component such as the cylinder head of a heat engine or an electronic component. So, to make a simple comparison of surfaces, we apply different constant flow values to the flat plate, plate with fins and plate with tubular beams while measuring the surface temperature. The surface that reaches the lowest surface temperature will be the most efficient at transferring heat to the air.

We calculated the Nusselt number in natural convection for different heat flux values for each moment (from 0 s to 240 s). We notice that the Nusselt and the heat transfer coefficient decrease over time, which is interpreted by component cooling.

Case 1- • A Flat Plate

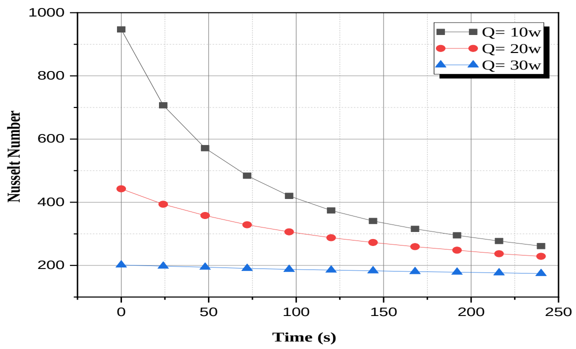


Figure 7. Evolution of Nusselt number in natural convection at different times of Flat surface for different heat fluxes ($\dot{Q}$=10W, $\dot{Q}$=20W , $\dot{Q}$=30W).

Case 2- A Pinned Surface

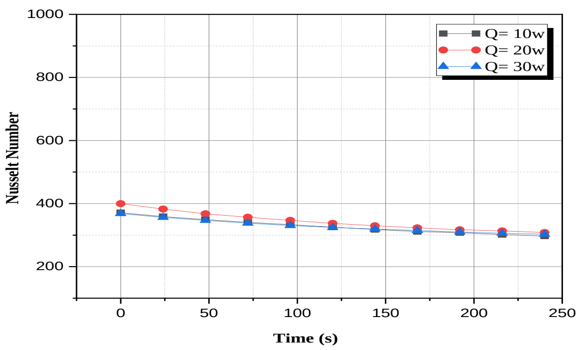


Figure 8. Evolution of Nusselt number in natural convection at different times of pinned surface for different heat fluxes ($\dot{Q}$=10W, $\dot{Q}$=20W , $\dot{Q}$=30W).

Case 3- A Finned Surface

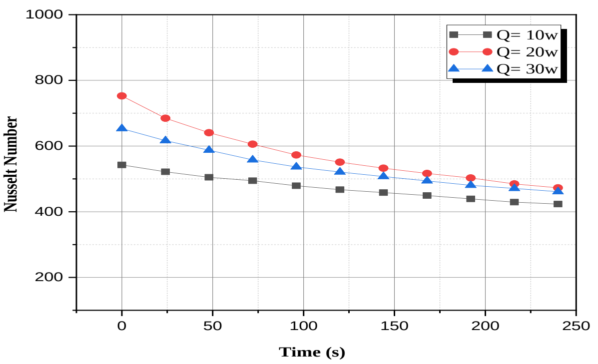


Figure 9. Evolution of Nusselt number in natural convection at different times of finned surface for different heat fluxes ($\dot{Q}$=10W, $\dot{Q}$=20W , $\dot{Q}$=30W).

Result analysis

We calculated the Nusselt number in natural convection for the case of a flat plate, a finned plate and a beam plate at different heat flux values for each moment (from 0 s to 240 s). We notice that the Nusselt and the heat transfer coefficient decrease over time, which is interpreted by component cooling.

IV.2. Forced convection

In natural convection, the naturally created low convective currents limit the rate of heat transfer. As the input power increases, the surface temperature and convective currents also increase. However, this restriction prevents the dissipation of more power at a lower surface temperature for any given surface the artificial increase in airflow through the surface (using a fan) therefore helps to increase the rate of heat transfer, obtaining a reduced surface temperature for any given input power.

Case 1- • A Flat Plate

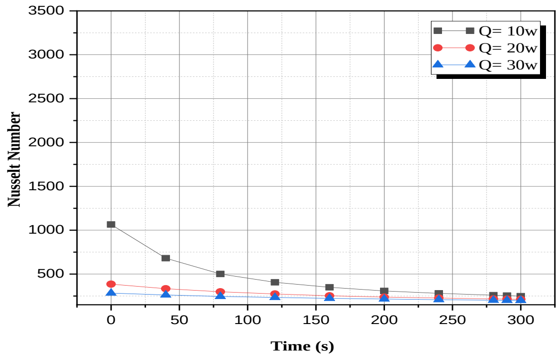


Figure 10. Evolution of Nusselt number in forced convection at different times of Flat surface for different heat fluxes ($\dot{Q}$=10W, $\dot{Q}$=20W , $\dot{Q}$=30W).

Case 2- A Pinned Surface

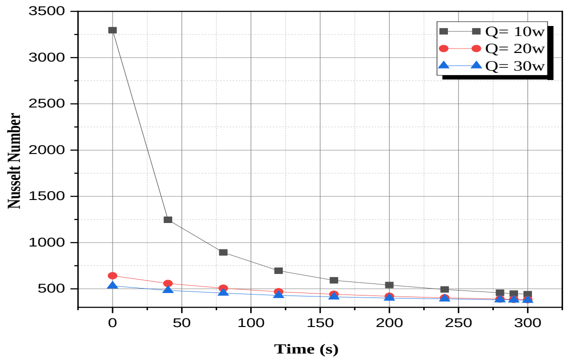


Figure 11. Evolution of Nusselt number in forced convection at different times of plate with Pins for different heat fluxes ($\dot{Q}$=10W, $\dot{Q}$=20W , $\dot{Q}$=30W).

Case 3- A Finned Surface

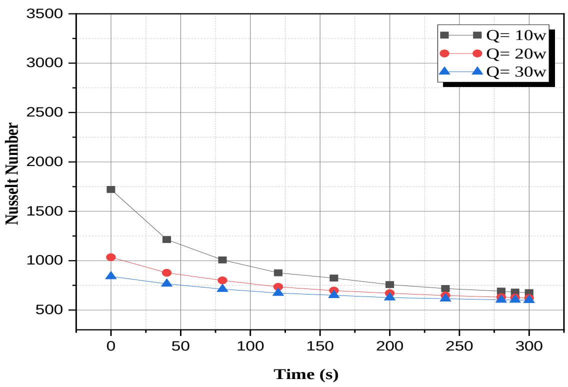


Figure 12. Evolution of Nusselt number in forced convection at different times of finned surface for different heat fluxes ($\dot{Q}$=10W, $\dot{Q}$=20W , $\dot{Q}$=30W).

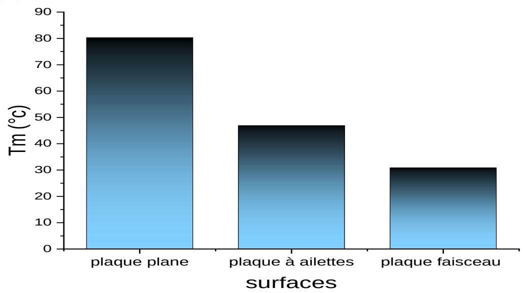
Result analysis

We calculated the Nusselt number in forced convection for the case of a flat plate, a finned plate and a beam plate at different heat flux values for each moment (from 0 s to 300 s). We notice that the Nusselt and the heat transfer coefficient drop sharply over time, which is interpreted by component cooling.

IV.3. Overall comparison between the three components

The results again show that the Surface Plate has less chance of transferring its heat to the air even in forced convection. All average surface temperatures have decreased compared to results obtained with natural convection, showing that forced convection helps improve heat transfer. Please note that the Beam Surface shows the best improvement because it has the coldest surface in forced convection.

(a)



(b)

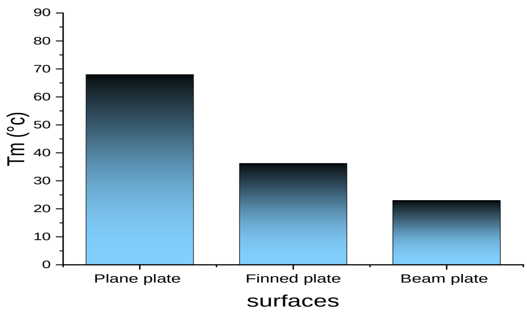


Figure 13. Evolution of average temperatures for (a) Free convection (b) forced convection

Conclusion

This experimental study that was done in mechanical laboratory at the University of Djelfa that treats a problem of heat transfer and precisely a problem of mixed convection. Indeed this experimental work was a comparative study concerning the ability of the electronic components already mentioned (flat plate, finned plate, beam plate) to cool down and this for the two modes of convection (natural and forced) by taking each time a different value of heat flux. The results obtained were in terms of Nusselt number and heat exchange coefficient (cooling efficiency indicators). The graphs and illustrated histograms show that the forced mode in the case of beam plate is the best from the point of view ability to cool.

The cooling is considered as a major solution in industrial field especially in the domain of electronics but the problem is always the energy consumption. For that we will try to work in this subject i.e. we will perform and protect our electronic material at a low cost.

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