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Thermal Performance Of A Flat Plate Oscillating Heat Pipe As A Thermal Spreader With Centered Heating

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Abstract- WSNs use autonomous sensors spread across an area of interest to detect various occurrences. These sensor networks needed extensive planning, building, and deployment to meet real-time sensing and monitoring needs. These nodes have microprocessors, transceivers, power, memory, and wireless modules. Sensors organize, combine, send, receive, and process massive amounts of data. This means they must efficiently use memory, CPU power, and, most critically, energy to enhance longevity and productivity. . Clustering helps wireless sensor networks last longer (WSNs). It needs clustering sensor nodes and selecting "cluster heads" (CHs) for each cluster. This paper presents an improved clustering algorithm Improved Data Aggregation Clustering (IDAC) that will minimize energy and improve network lifetime.

Keywords- Clustering, Data aggregation, Energy and WSN.

I. INTRODUCTION

As a new member of the heat pipe family, oscillating heat pipe (OHP), also known as pulsating heat pipe (PHP), was first proposed by Akachi in 1990 [6].

It is generally composed of a serpentine capillary tube with wickless structure, namely, tubular oscillating heat pipe (TOHP), which is vacuumized internally and partially filled with working fluid. As another typical configuration of the OHPs, flat-plate oscillating heat pipe (FPOHP) usually exists as a metal plate with an embedded closed-loop meandering capillary channel. Compared with the TOHP, the FPOHP is more suitable to combine with the flat electronic chip/component [7].

Whether for TOHP or FPOHP, the working fluid is distributed randomly as a series of vapor-liquid slugs in the capillary tube/channel because of the prominent role of surface tension. Once the OHP is partially heated and cooled simultaneously, a rapid evaporation of liquid induces increasing local pressure in the heated section (namely, the evaporator), meanwhile, condensation of vapor induces a low local pressure in the cooled section section to the cold one under the action of “bubble pumping” [8]. Especially, because of uneven pressure distribution and local heat transfer caused by the randomly distributed vapor and liquid [9], the movement of fluid in the OHP exhibits unique oscillatory characteristics, e.g., the self-sustained oscillation and even circulation [10]. Because of the unique operating principle and wickless structure, the OHPs possess several advantages over the traditional heat pipes with wick structure, such as simple structure, low cost and high flexibility [11].

Accordingly, during the last two decades, many researches have been conducted to investigate the fluid flow and heat transfer in the OHPs, which indicated that the self-sustained oscillatory fluid flow behaviors displayed various patterns under different conditions [11], such as local oscillation, bulk oscillation, and oscillatory circulation. Especially, these flow patterns are closely related to the thermal performance of the OHPs. In addition, the operating performance of OHPs can be influenced by both structural and operational parameters, such as heat load [12], filling ratio [13], turn number [14],

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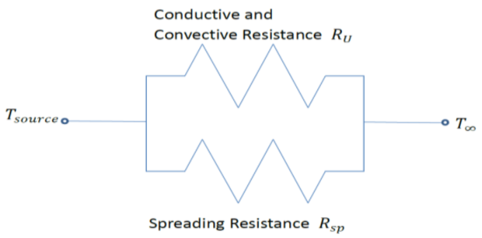
inclination angle (i.e., the gravity) [15] and physical properties of working fluid [16], etc.

FPOHP is designed and fabricated for electronic thermal management with multiple heat sources and sinks, which has two mirror symmetric tandem branches of capillary serpentine square channel with a hydraulic diameter of 2 mm. Additionally, a corresponding experimental study is performed to test the overall start-up characteristics and quasi-steady thermal performance of the FPOHP with arrangement of "uniform heating by multiple heat sources at center and air cooling at both ends". The thermo-hydrodynamic characteristics of the FPOHP under different inclination angles and heat loads are examined and analyzed. Furthermore, the thermal performance deviation between two symmetric tandem sections of the FPOHP is compared and discussed.

II. LITERATURE REVIEW

With varied degrees of success, a variety of models have been tried to anticipate how an oscillating heat pipe (OHP) works. In an effort to update and summarise the problems still affecting our understanding of oscillating heat pipes, Zhang and Fahgri [12] cited nearly 50 studies, many of which attempted mathematical models. These models typically have two characteristics: Few of them are useful from an industrial standpoint due to their complexity as they attempt to forecast the behavior or physics inside the heat pipe. In order to overcome gravitational effect issues, the OHP typically needs to have a lot of turns. In order for surface tension to prevail over gravitational forces, the diameter must meet a specific critical value, which is determined by the Bond number. The observed thermal resistance typically ranges from 0.03 K/W to 0.08 K/W. To achieve this, a straightforward resistance network is suggested, as shown in Fig.1.



Basic representation of the thermal resistance network in Figure 2.1.

There are numerous helpful methods to determine conductive and convective resistances based on geometry, cooling fluid flow patterns, etc. Although it has typically been extensively documented, thermal spreading is a phenomenon that is difficult to model. On rectangular plates with centred heat sources and non-uniform aspect ratios, Ellison [13] was able to precisely solve the problem. The FPOHP that Ellison [13] studied is comparable to an FPOHP with centralised heating. The distinction is that for the FPOHP, a constant heat conductivity is presumpted.
It is presummated that a planar heat source of Q with a width x and a length y is taken into consideration, much like the issue mentioned by Ellison [13]. It has a, b, and t dimensions in x, y, and z, and is centred at z=0 in the xy plane. Using a heat transfer coefficient h and radiative heat transfer defined by the equation h = hc + hr, Newtonian cooling is taken into account.

Thermal spreading happens when heat is transferred to the surface at z=t from a smaller, uniformly distributed heat source. The spreading resistance Rsp is identified as a fraction of the internal conductive flow from the source. The remainder, denoted as Ru, is quantified and is revealed to be the result of adding outside Newtonian cooling to internal one-dimensional conductive resistance. With a constant thermal conductivity, the energy equation regulating the temperature distribution of the FPOHP is as follows:

A FPOHP can be assumed to have adiabatic surfaces at the plate edges based on the first set of boundary requirements. With the exception of the heat source Qvol, the second set makes the assumption that the plane at z=0 is a well-insulated, constant temperature (adiabatic) plane.

Where is a known volumetric heat source in three dimensions called Qvol. The heat sink at the top of the setup is where all heat is removed.For an FPOHP in a wind tunnel with well-insulated surfaces other than the heat-removal surface, this is likewise a valid assumption. All calculated temperatures are temperature rises because it is assumed that the ambient temperature is zero. A double Fourier series in the xy plane with z-dependent Fourier coefficients, or I(z), is used to describe the volumetric heat source, Qvol. The arrangement of the temperature distribution is made to resemble that of the heat source, i.e.

$$\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}=-\frac{Q_{vol}}{k}$$

It has boundary conditions of:

$$k\frac{\partial T}{\partial x}=0\ at\ x=0,a;\quad k\frac{\partial T}{\partial y}=0\ at\ y=0,b$$

$$k\frac{\partial T}{\partial z}=0\ at\ z=0;\quad k\frac{\partial T}{\partial z}=-hT\ at\ z=t$$

The derived source Fourier coefficients for adiabatic edge boundary conditions are displayed below. These are simplified by employing the Dirac delta function (z) such that Q(x,y,z)=q(x,y) and assuming a homogeneous heat flux q(x,y) at the z=0 plate surface (z). This leads to

$$Q_{vol}(x,y,z)=\sum_{l=0}^{\infty}\sum_{m=0}^{\infty}\varepsilon_l\varepsilon_m\,\phi_{lm}(z)\cos\left(\frac{l\pi x}{a}\right)\cos\left(\frac{m\pi y}{b}\right)$$

The partial differential is substituted with the Fourier expansions for Qvol and T.

III. METHOD WITH APPLICATION

$$\overline{Nu_D}=\frac{\bar{h}D}{k}=C\,\overline{Re}_D^{\,n}Pr^{1/3}$$

Internal spreading and resistance are particularly adequately handled by Ellison's [13] model. It still does not adequately solve an anticipated heat removal, though. Still required is the expected heat loss from the Newtonian cooled side. A Nusselt correlation [14], which is the ratio of the convection to the conduction of a specific boundary layer, controls it generally.

$$Re_D=\frac{VD}{\nu_{air}}$$

where D is the hydraulic diameter in m, and h is the average convection coefficient in W/m2 K. Pr is the dimensionless Prandtl number, ReD is the dimensionless Reynold's number related to the flow conditions, and C and n are constants based on the Reynold's number and selected from the relevant table. The definition of the Reynold's number [14] iswhere air is the kinematic viscosity of the air in m2/s, D is the hydraulic diameter in m, and V is the velocity in m/s.

A pinned finned aligned heat sink is used in this type. For this, the Nusselt correlations for flow over a bank of tubes and the Reynolds number are utilised. It is housed in aa well-insulated, closed duct with a known entering velocity and beginning temperature. These equations are combined with the Ellison solution to get temperature predictions for a known heat input, Q. Using Ellison's resistance formula. Although [13] appears difficult, it is a relatively straightforward method for calculating the heat evacuated from the heat sink. The resistance and its associated values are computed using a set of Mfiles created in MatLab.

There are a few important first impressions from the Ellison Solution. A conduction term (k), a convective heat coefficient (h), and the related geometry are required to determine a resistance. Geometry is well recognised. The impact of h and k is still not as clearly recognised intuitively, though.

IV. EXPERIMENTAL SETUP AND METHOD

The goal of this study is to ascertain whether a flat-plate oscillating heat pipe (FPOHP) may be used as a thermal spreader. The experimental set-up is built around the FPOHP's design. A size of 6.14 in (156 mm) long by 4.1 in (104.1 mm) wide is selected based on the literature research, a similar earlier experimental setup [11], and stock components that are readily available. It's made from copper plate that is 0.25 in (6.35 mm) thick. Acetone of the HPLC (high performance liquid chromatography) grade is selected as the working fluid. The maximum diameter at which a working fluid can form the crucial meniscus is determined by the Bond number. You can determine the working fluid channel's maximum diameter from.

$$D_{max}=\sqrt{\frac{Bo_{crit}\sigma}{g(\rho_{liq}-\rho_{vap})}}$$

where Bocrit is the critical Bond number, denotes surface tension in N/m, g denotes gravity's acceleration in m/s2, liq denotes liquid density in kg/m3, and vap denotes vapour density in kg/m3, and g denotes the acceleration of gravity in m/s2. Rectangular channels measuring 1 mm by 1 mm are selected, and the crucial Bond number of 1.84 is acknowledged. The ideal number of turns is several. The implementation of 34 turns is done for this dimension and design. To ensure a proper brazing area, there are 2 mm margins all around. Two fill ports each measuring 0.065 in (1.651 mm) in diameter have been drilled into the channel at each ends. It is brazed shut and covered with stock copper plate that is 0.03125 in (0.79375 mm) thick. It is then filled to a filling ratio of 85 0.5 with HPLC grade acetone. Dimensioned drawing of the heat pipe is shown in Figure 4.1. All measurements are given in millimetres. The 1 mm-deep channels and the fill ports, which are located dead centre on either end, are not visible.

A consistent volumetric heat source is required, according to the model. A 0.0625 in (1.5875 mm) hole has been cut dead centre into a copper plate that measures 1 in (0.0254 m) x 1 in (0.0254 m) x 0.25 in (0.00635 m). Using Omega therm 201 thermal epoxy, this hole is filled with an Omega type T thermocouple for accurate temperature measurement. Four 250 Watt/10 chip Barry Industries resistors are connected in series to this. Just below and in the middle of the FPOHP, these are secured with the aid of two clamps.

Cool Innovations' model 4-614111U stock copper heat sink is selected. It has the following measurements: 4.1 in (0.1041 m) in width, 6.14 in (0.156 m) in length, and 1.1 in (0.0279 m) in height. It has 1350 copper pins and, according to reports, varies from 0.14 to 0.07 °C/W based on air speeds between 1 and 3 m/s. That matches up well with a pin fin aligned array's predicted thermal resistance from [14]. Thermal paste (OmegaTherm 201) is used to attach it, and high temperature padding insulation is used to hold it in place. It is then gently but firmly fastened. Acrylic polycarbonate (Lexan) sheets are purchased from Interstate Plastics and connected to the required sizes for the wind pipe and heat sink assembly.

There are fifteen type T Omega thermocouples fitted (maximum 0.5 0C). To monitor temperature distribution, nine are placed directly on the heat sink, with a positioning variation of 1 mm. To determine the temperature and make sure that heat is being removed, two are placed before the heat sink and two after. These are averaged on each side, accordingly. To determine the maximum temperature for calculation, one is situated at the centre of the heat source on the bottom. A chip resistor (Barry Industries 250W/ 10 chip resistor) is also right on top of one to preserve avoidance of unintended burnout, defined as 1000C. Fig. 3.3 depicts the arrangement of the Omega type T thermocouples on the heat sink's surface.

V. PROCEDURE

The assembly is evaluated at speeds ranging from 1 m/s to 3 m/s, as determined by

Q = m aircp,irT (3.2).

$$Q = \dot{m}_{air} c_p \Delta T \quad (3.2)$$

$$\dot{m} = \rho_{air} A_{cross} V_\infty (3.3)$$

where V is the average incoming air velocity in m/s, Q is the heat input in watts, m air is the mass flow rate of the air in kg/s, T is the change in temperature in °C, airis is the air density in kg/m3, Across is the cross sectional area of the duct in m2, J is the input voltage in volts, and I is the input current in amperes. In order to determine the velocity for the initial input, a minimum heat input of 10 watts was used to create a temperature differential.

By manually regulating the input power on the VARIAC transformer power source for the blower, the velocity was varied until the required temperature difference was attained. The type T thermocouples' values were averaged before and after the heat sink and heat pipe installation to confirm the incoming and exit temperatures. The heat input is adjusted at these speeds in increments of 25 Watts, ranging from 25 Watts to a maximum of 225 Watts.Each time the heat input was increased, a quasi-steady state was obtained.For each increase in heat input, this took ten to fifteen minutes.Temperatures were noted for future research.The temperature distribution and thermal spreading were checked using the type T thermocouples on top of the heat sink.These were then added together to get an average temperature for the top surface of the model.The thermocouples on the bottom are used to verify the heat source's temperature and to establish a maximum temperature for later calculation and plotting (with the chip resistor arrangement).

VI. RESULTS AND DISCUSSION

Both new data and older data are being used to support this concept. The model's validity was initially checked using data from a prior experiment, Fritz Laun [16]. Data on a central heat source with a cooling block attached was collected. Two pieces of data exist. 10 cm by 10 cm and.031 m thick describe the cooling block and heat pipe arrangement. The heater is 3 cm by 3 cm for the first pair and 4 cm by 4 cm for the second. While the second set lacks a copper plate to promote thermal spreading, the first set does. The power output is changed from 250 Watts to 350 Watts while the cooling bath is kept at 20 0C. The second set is adjustable between 550 and 600 Watts.

$$R = Te - Tc\ /Q$$

Te denotes the temperature of the evaporator in 0°C, while Tc denotes the temperature of the condenser in 0°C. A heat transfer coefficient is also required for the thermal spreading model that is used. Newton's Law of Cooling [14] provides experimental evidence that support this:

$$h = Q\ /A\Delta T$$

where A is the area of the heat transfer surface in m2, Q is the heat input in Watts, h is the heat transfer coefficient in W/m2 K, and T is the temperature difference in 0C. Table 1 shows the calculated data. The degree of agreement between the two is excellent, as can be observed.

The experimental thermal resistance was measured and the analytical thermal resistance was estimated. A greater K value could be used to get better agreement. For an order of analysis estimate, it was presummated that k=1000 W/m k. Calculated and experimental thermal resistances are shown in Table

Table 1 Calculated and Experimental Thermal Resistances

	Q (Watts)	A(m2)	ΔT (°C)	h (W/m² K)	Rexperimental (k/w)	Rcalculated (k/w)
Copper spacer	250	0.01	45	555	0.18	0.19608
	300	0.01	50	600	0.166666667	0.18257
	350	0.01	60	583.3	0.171428571	0.18733
Nospace r	500	0.01	40	1250	0.08	0.090897
	550	0.01	45	1222.2	0.081818182	0.092716
	600	0.01	50	1200	0.083333333	0.094231

Data from a second experiment [11] is also utilised to validate the model. In this instance, a 3.0 dimensional FPOHP made from 2.67 mm thick alloy 101 copper plate and two layers of 1.02 mm x 1.02 mm channels is selected. It is filled with HPLC-grade acetone at a ratio of 0.8:0.1. Through the use of brazing, a 0.254 mm sheet was used to cover each side. In order to match the heat sink (Cool Innovations model 4-414111U) with a base 2.79 mm thick, the heat pipe itself is 0.1041 m by 0.1041 m in size. The pin fin array contains 900 pins. It is positioned horizontally to the ground inside a 1.8 m long Lexan wind tunnel. With a 3 cm by 3 cm heater, the heat sink is centrally heated for a thermal spreading design. For wind speeds of 1 m/s, 2 m/s, and 3 m/s, a blower connected to the wind tunnel is controlled by a variac. A hot wire anemometer is used to measure the velocity (Extech). The data was gathered using a computer, a DAQ setup, and a thermocouple setup. Up until the maximum of 230 watts (at 3 m/s), power is supplied by a DC power source in increments of 10 watts. A To confirm the effectiveness of the experiment, a copper plate with approximately comparable dimensions to the heat pipe is utilised as the control. T entering is maintained at 21 °C +/- 0.5 °C. Calculating experimental thermal resistance from

$$R = \frac{Tmax - T\infty /\ Q}{}$$

where T is the ambient temperature, Q is the input heat in Watts, and Tmax is the maximum time-averaged temperature in 0C. The experimental and predicted thermal resistances for this experiment are shown in Tables 4.2 and 4.3. It is important to note that the agreement in this case is low and that it is hypothesised that the temperatures reported are in fact from the FPOHP bottom surface and the heat sink's surface, which is where the model's predictions are supposed to be made. Although this might result in different ways to interpret the results, the study being compared to is solely concerned with the thermal performance of the heat pipe itself in such a configuration.

Table 2 Compares Estimated and Experimental Resistances for Heat Pipes

ResistanceTableHeatPipe						
V∞	Qin (Watts)	A (m²)	ΔT(C)	h (W/m² K)	Rexperiment	R calculated
1 m/s	50	0.1348	28	78.9493	0.56	0.1856
	100	0.1348	51	78.9493	0.51	0.1856
	140	0.1348	79	78.9493	0.564285714	0.1856
2 m/s	50	0.1348	21	109.0507	0.42	0.1368
	100	0.1348	40	109.0507	0.4	0.1368
	150	0.1348	60	109.0507	0.4	0.1368
	200	0.1348	79	109.0507	0.395	0.1368
3 m/s	50	0.1348	17	131.7306	0.34	0.1148
	100	0.1348	35	131.7306	0.35	0.1148
	150	0.1348	50	131.7306	0.333333333	0.1148
	200	0.1348	68	131.7306	0.34	0.1148
	230	0.1348	82	131.7306	0.356521739	0.1148

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Table 3 Copper Substrate Experimental vs. Calculated Resistance Values

Resistance Table Copper							
V∞	Qin (Watts)	$A(m^2)$	ΔT (°C)	h (W/m² K)	Rexperiment	R calculated
1 m/s	50	0.1348	28	78.9493	0.56	0.2374
	100	0.1348	50	78.9493	0.5	0.2374
	140	0.1348	79	78.9493	0.564285714	0.2374
2 m/s	50	0.1348	20	109.0507	0.4	0.1955
	100	0.1348	42	109.0507	0.42	0.1955
	150	0.1348	65	109.0507	0.433333333	0.1955
	180	0.1348	78	109.0507	0.433333333	0.1955
3 m/s	50	0.1348	19	131.7306	0.38	0.1761
	100	0.1348	38	131.7306	0.38	0.1761
	150	0.1348	56	131.7306	0.373333333	0.1761
	200	0.1348	79	131.7306	0.395	0.1761

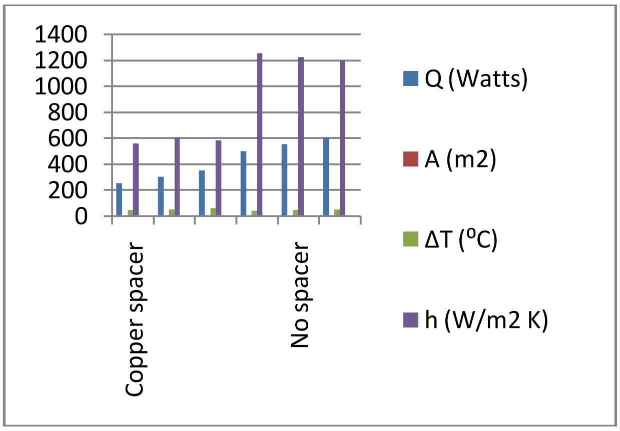


Figure .1 Experimental lyobtained values from [10].Used with permission.

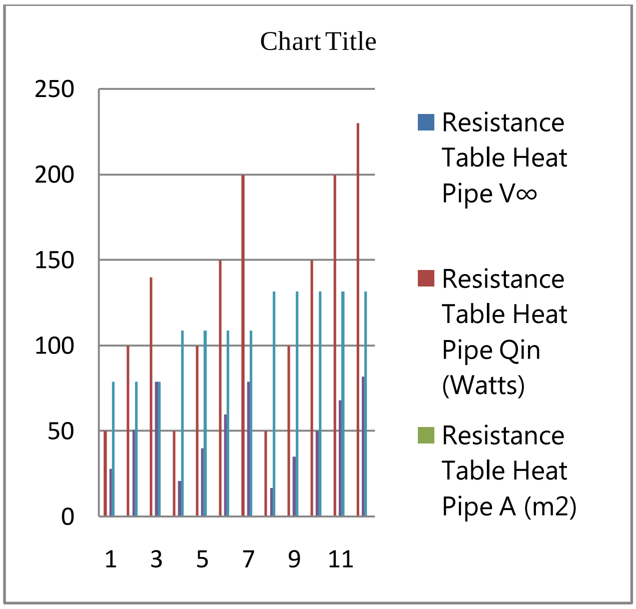
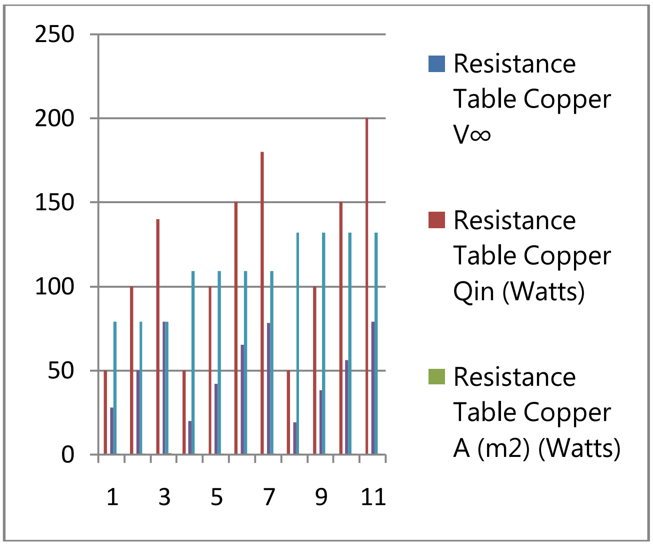


Figure 2 Calculated and predicted surface temperatures for the heat pipe at various air speeds using the model described in chapter 2.



It's also important to note that Figures 5.2 and 5.3 closely follow Figure 5.1's trend. This demonstrates the model's validity despite variations in methods for analysing the data. An experiment is conducted for this study using the setup mentioned in the chapter.

VII. CONCLUSIONS

There has been a review of the literature on mathematical models for thermal spreading.Many were largely axial layouts but contained significant information on how oscillating heat pipes worked. The configuration of thermal spreading was only of interest to one. Three sets of data were investigated using an analytical method resembling that used to determine the spreading resistance as described in one of the publications studied [13]. The third experiment was created specifically to test this paradigm, while the first two were existing trials. One experiment used a 3D FPOHP square design with a setup identical to the one described in this study. Unlike the other two, it did not provide a good level of agreement with the model. The predicted values of the highest surface temperature were shown for this experiment, and while they were lower than the actual values, there was generally excellent and what looked to be straight linear agreement with the patterns and data supplied.

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