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Bypass line assisted start-up of a loop heat pipe with a flat evaporator †

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Abstract

Loop heat pipes often experience start-up problems especially under low thermal loads. A bypass line was installed between the evaporator and the liquid reservoir to alleviate the difficulties associated with start-up of a loop heat pipe with flat evaporator. The evaporator and condenser had dimensions of 40 mm (W) by 50 mm (L). The wall and tube materials were stainless steel and the working fluid was methanol. Axial grooves were provided in the flat evaporator to serve as vapor passages. The inner diameters of liquid and vapor transport lines were 2 mm and 4 mm, respectively, and the length of the two lines was 0.5 m each. The thermal load range was up to 130 W for horizontal alignment with the condenser temperature of 10°C. The experimental results showed that the minimum thermal load for start-up was lowered by 37% when the bypass line was employed.

Keywords: Loop heat pipe; Flat evaporator; Start-up; Bypass line

1. Introduction

Loop heat pipe (hereinafter denoted by LHP) has been used as a thermal control device in many engineering applications including space vehicles, since it was initially developed in early 1970's [1]. Various configurations and features of LHPs are well described in the literature [1-10]. Owing to its prominent advantages over conventional heat pipes, LHP has the potential to be used widely as efficient cooling devices for high-power electronic components in the near future. Despite its excellent heat transport capability however, certain technological difficulties have been involved in the fabrication and operation of LHP. One prevalent difficulty is that LHP may experience start-up problems, especially under low thermal loads.

A few papers reflect efforts to mitigate the most prevalent drawbacks associated with LHP. Maidanik and Fershtater [1, 2] suggested employing an auxiliary heater as an active temperature control device of the compensation chamber to maintain the temperature of LHP quasi-stable against a certain range of heat-load variation. Muraoka et al. [3] investigated bubble formation in the liquid core and its effect on the dry-out failure, using a specific LHP having a porous element in the condenser, and suggested conditions to predict the dry-out. Mo et al. [4, 5] employed an electro-hydrodynamic technique to induce an additional pressure difference in an effort to reduce the start-up time and thus to enhance the thermal performance of LHP.

Previous studies include those designed to elucidate the physical phenomena associated with LHP. Zhao and Liao [6] conducted a theoretical and experimental study on the LHP evaporator to predict the critical heat flux and the heat transfer coefficient. Khrustalev

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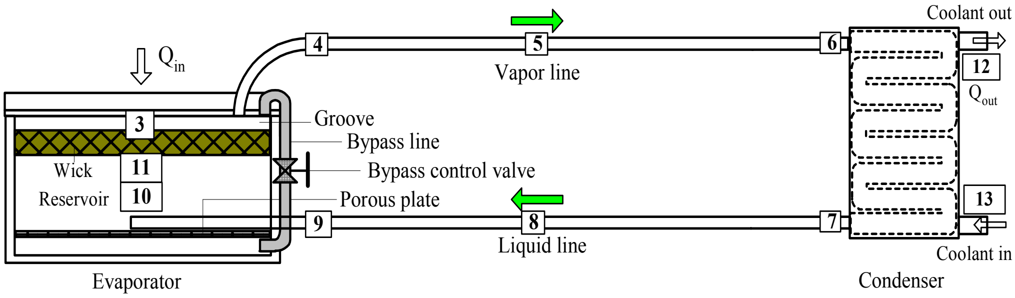


1, 2: Evaporator wall (Evp. Wall), 3: Evaporator Vapor (Evp. Vap), 4: Evaporator outlet (Evp. Out), 5: Vapor line (Vap. L), 6: Condenser inlet (Cond. In), 7: Condenser outlet (Cond. Out), 8: Liquid line (Liq. L), 9: Evaporator inlet (Evp. In), 10: Liquid Reservoir (Liq. Res.), 11: Wick inlet (Wick. In), 12: Coolant inlet (Cool. In), 13: Coolant outlet (Cool. Out)

Fig. 1. Schematic of the LHP fabricated in this study and location of thermocouples.

and Faghri [7, 8] distinguished low and high heat fluxes according to the location where the liquid-vapor interface formed and calculated the interface locations numerically.

The physical mechanism associated with the start-up of an LHP is very complicated since it is affected by several factors and conditions. Zhang et al. [9] investigated a few typical patterns that LHP might experience during the start-up under various working conditions and explained some of these in relation with a reverse flow of the working fluid according to the inclination angle. A successful start-up depends on how fast a temperature difference, and thus a pressure difference required, can be produced between the liquid in the liquid reservoir (sometimes called ‘compensation chamber’) and the vapor generated in the evaporator. The start–up time may be reduced with a higher heat flux since the necessary temperature difference can be generated faster [1]. In this case however, a higher possibility of dry-out exists. A start-up failure may occur since the corresponding heat flux results in less temperature difference between the liquid reservoir and the vapor than the minimum value required for a successful start-up. For low heat-flux values the vapor generated prior to the start-up may move to the liquid reservoir and the accompanying heat may cause vaporization of the liquid in the reservoir, which will eventually cause start-up failure.

An effort was made in this study to improve, without external power consumption, the start-up characteristics of an LHP by employing a bypass line between the vapor channel in the evaporator and the liquid reservoir. Analysis and discussion of the experimental results are presented for cases with the new auxiliary loop.

2. Experimental setup and procedure

The experimental apparatus used in this study was fabricated by modifying Boo and Chung’s LHP [10] by attaching an auxiliary bypass line between the evaporator vapor channel and the liquid reservoir. Fig. 1 shows a schematic of the LHP with thermocouple locations. The container and tubing of the LHP were made of stainless steel and the working fluid was methanol. The evaporator was of a flat rectangular shape suitable for local temperature control or cooling of electronic components. The outer dimensions of the evaporator were 40 mm (W) × 50 mm (L) × 30 mm (H) with an embedded liquid reservoir. The heating area in the evaporator was 35 mm × 35 mm and nine axial grooves with trapezoidal cross-section were provided to serve as vapor passages. The heater block used as a heat source consisted of three cartridge-type resistance heaters. The capillary structure inside the evaporator was made of polypropylene with a nominal pore size of 0.5 microns and a thickness of 5 mm. The nominal pore size indicates that 93% of the particles of the designated size cannot pass through the wick. Table 1 shows the specifications of the polypropylene wick used in this study. The condenser had a planar dimension of 40 by 50 mm in which ten meandering channels were machined for circulation of the coolant. Inner diameters of the liquid and vapor lines were 2.0 mm and 4.0 mm, respectively, and the length of the two lines was 0.5 m each.

The bypass line connected the vapor outlet of the evaporator and the liquid reservoir, and had the same diameter as the vapor line. A metering valve was installed in the middle of the bypass line to control

Table 1. Specifications of the wick in the LHP in this study.

Item	Description/Values
Material	Polypropylene
Pore size	0.5 μm
Porosity	0.5 - 0.6
Melting temperature	160 - 170°C at 1 atm
Limit temperatures	90 - 120°C at 4.5 atm
	48 - 65°C at 18 atm
Thermal conductivity	0.2 W/m·°C

flow rate of the bypassed vapor. A porous plate was inserted between the inlet for the bypassed vapor and the liquid in the liquid reservoir to minimize disturbance of the reservoir liquid and to distribute bypassed vapor over the liquid interface.

The liquid fill was 36.5 ml corresponding to a 50% fill charge ratio based on the literature [10, 11]. The thermocouples installed in the experimental apparatus were T-type and the diameters were 0.254 mm (AWG 30 gage). Four thermocouples were attached to measure the temperatures at the evaporator wall (No. 1 and 2) and the walls of the vapor and liquid transport lines (No. 5 and 6 in Fig. 1). The other thermocouples were probe type and were installed at the locations designated by the numbers in Fig. 1. Every thermocouple was calibrated between 0 and 100°C before reading and had an uncertainty of ±0.5°C in measurement. The whole LHP system was insulated with ceramic wool to minimize interaction with the environment. The thermal load was controlled by a voltage regulator and measured by a watt-meter with a maximum error of 0.5% of the full scale. The temperature and flow rate of the coolant was controlled by an isothermal bath. During the experiment, the coolant flow rate was maintained with 5 gph (1.05 ml/s), monitored by a rotameter with a maximum error of 4% of the full scale (10 gph). Based on the energy balance between the input thermal load and the heat recovered by the coolant in the condenser, the heat loss was estimated to be less than 10% of the input thermal load. The whole set of temperature data was acquired by a data acquisition system that took readings every 2 seconds.

All experiments in this study were conducted with a horizontally aligned evaporator and condenser since focus of this study was on the function of the bypass line. The performance of the LHP during start-up period as well as once a steady state was achieved was investigated from very low thermal loads to the maximum of 130 W, where the evaporator wall temperature reached 110℃. To protect the polypropylene wick and the working fluid from permanent deformation or deterioration, the maximum allowable operating temperature of the LHP was set to 110℃. No dry-out failures were observed up to these values of thermal load and temperature.

One of the important performance indices, the thermal resistance, R_{th}, is defined by the following equation in this study.

$$R_{th} = \frac{T_w - T_{cool.in}}{Q_{in}} \tag{1}$$

where T_w denotes the average temperature at the evaporator wall.

3. Results and discussion

At each operating condition the LHP was tested in two different modes to investigate the effect of the bypass line. Normal operating mode (NOM) denotes the mode with the bypass valve closed, and Bypass operating mode (BOM) denotes the mode with the bypass valve open. From the experimental data presented in Boo and Chung [10], the minimum thermal load for the LHP without bypass line was 10 W. Based on the experiment in this study, it was determined that the LHP in NOM was impossible to start up for thermal loads lower than 8 W. Therefore, careful experiments were conducted for thermal loads between 5 and 7 W to distinguish start-up characteristics between NOM and BOM.

Fig. 2 summarizes temperature response of the LHP with an initial thermal load of 6 W (heat flux of 0.49 W/cm^2) in BOM. It was assumed that the start-up of the LHP was achieved if the vapor temperature at the condenser inlet (No. 6 in Fig. 1, 'Cond. In' in Fig. 2) increased following the vapor temperature at the evaporator outlet (No. 4 in Fig. 1, 'Evp. Out' in Fig. 2), and if the liquid temperature at the evaporator inlet (No. 9 in Fig. 1, 'Evp. In' in Fig. 2) decreased following the temperature at the condenser outlet (No.7 in Fig. 1, 'Cond. Out' in Fig. 2), so that the circulation of the working fluid was ensured. Startup was achieved in this mode after about 232 min since the vapor temperature at the condenser inlet was 47˚C while that at the evaporator outlet was 56˚C. Steady state was reached after about 280 min when every temperature exhibited a stable value. The test continued with lower thermal loads to identify the minimum

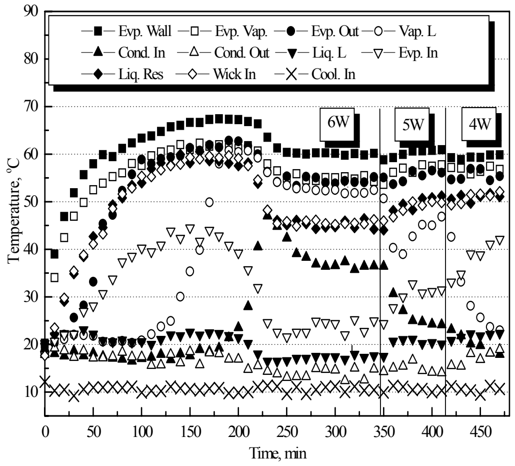


Fig. 2. Minimum start-up thermal load in Bypass Operating Mode (BOM).

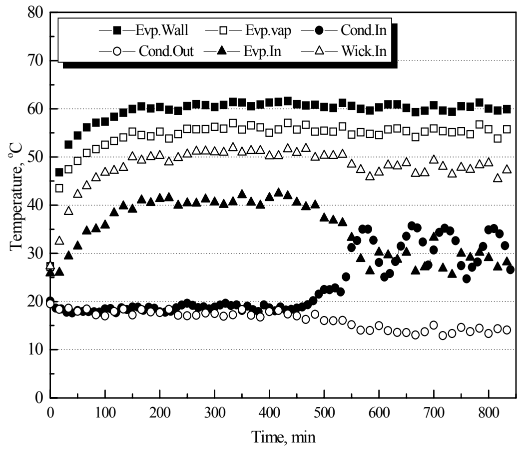


Fig. 3. Start-up of the LHP in Bypass Operating Mode for thermal load of 5 W.

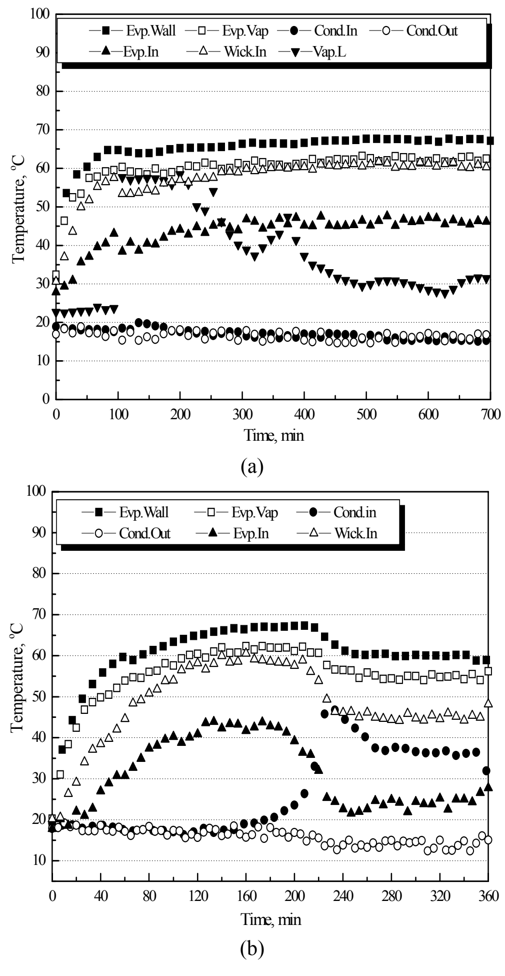


Fig. 4. Start-up characteristics of the LHP in NOM and BOM for thermal load 6 W: (a) NOM, (b) BOM.

possible value for start-up. After thermal load was lowered to 5 W (0.41 W/cm^2) at 343 min, a little variation was observed in temperatures even though stable operation was maintained. During the steady state, more than a 10˚C decrease at the condenser inlet temperature (‘Cond. In’) and less than a 10˚C increase at the evaporator inlet (‘Evp. In’) were observed. After the thermal load was lowered to 4 W (0.32 W/cm^2) at 413 min, however, the vapor temperature at the condenser inlet decreased by 7˚C, the liquid temperature at the evaporator inlet increased by 10˚C, and eventually the LHP ceased operation.

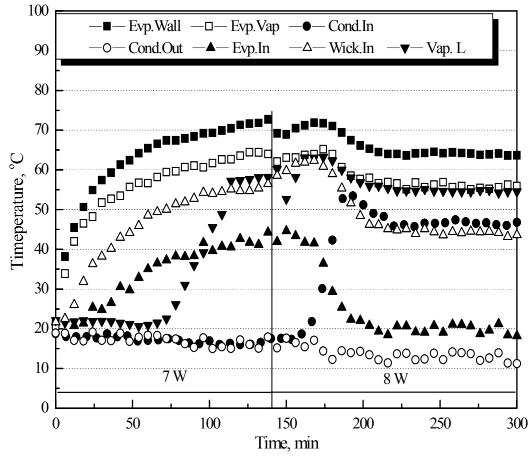
Fig. 3 represents the start-up process for BOM with a thermal load of 5 W, which corresponded to the minimum start-up thermal load with the bypass valve open. The time required to attain start-up was about 470 min as observed through the temperature response of the LHP. Then, the vapor temperature at the condenser inlet (‘Cond. In’) increased but fluctuated between a maximum of 37˚C and a minimum of 24˚C with about a 100 min period. During the same period the liquid temperature at the evaporator inlet (‘Evp. In’) varied between 24˚C and 30˚C. This is the typical case of very low thermal load and it looked as if start-up had been achieved and failed periodically. With very low heat transfer, a considerable time would have been required to generate vapor at the liquid-vapor interface.

Fig. 4 compares the LHP performance for NOM and BOM under thermal load of 6 W. For NOM in (a), the start-up was not achieved until 700 min. The vapor temperature at the condenser inlet stayed low around 16˚C and never increased. On the other hand, the liquid temperature at the evaporator inlet slowly increased to 47˚C and never decreased. For BOM in (b) however, start-up was achieved after 230 min. The vapor temperature at the condenser inlet increased to 48˚C and the liquid temperature at the evaporator inlet decreased to 22˚C.

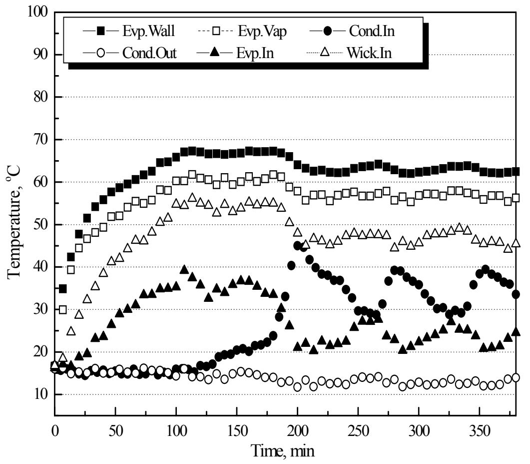
Fig. 5 represents the start-up characteristics of the LHP under NOM and BOM with an input thermal load of 7 W (0.57 W/cm^2). For normal operating mode (Fig. 5(a)), the start-up was not achieved until 140 min when the vapor temperature at the condenser inlet ('Cond. In') was maintained around 17˚C and the liquid temperature at the evaporator inlet ('Evp. In') increased to 42℃. After the thermal load was increased to 8 W at 140 min, a noticeable temperature change began to occur in about 30 min. It was obvious that start-up was achieved in 47min after thermal load was increased to 8 W when the temperature at the condenser inlet ('Cond. In') increased to 52℃ and that at the evaporator inlet ('Evp. In') decreased to 25℃. Steady state was achieved after 30 more minutes (experimental time of 220 min in Fig. 4(a)), and the two temperature values, at 'Cond. In' and 'Evp. In', exhibited 47℃ and 20˚C, respectively.

For a start-up with 7-W thermal load in BOM (Fig. 5(b)), the vapor temperature at the condenser inlet ('Cond. In') began to increase after 120 min and reached 45℃ at 201 min, while the liquid temperature at the evaporator inlet ('Evp. In') decreased to 20℃. Therefore it might be stated that the start-up was achieved after 201 min. After the achievement of start-up however, the vapor temperature at the condenser inlet began to decrease while the liquid temperature at the evaporator inlet began to increase until they reached the minimum and the maximum values, respectively, at 258 min exhibiting a trend for failure of start-up. Then, the two temperatures reversed changing trends and proceeded to obtain another successful start-up at 270 min. The period between the success and failure of start-up was shorter for higher thermal loads: 89 min for a 5-W thermal load (see Fig. 3) and 84 min for a 7-W thermal load (see Fig. 5). The time required to achieve the start-up also decreased as the thermal load increased: 450 min for 5 W, 180 min for 6 W, and 120 min for 7 W.

The typical reason for start-up failure of LHPs under low thermal loads is that the vapor generated in the evaporator may not provide sufficient vapor pressure to drive the whole working fluid in the loop to



(a)



(b)

Fig. 5. Start-up characteristics of the LHP in NOM and BOM for thermal load of 7 W: (a) NOM, (b) BOM.

circulate in the favorable direction (clockwise, in Fig. 1). With lack of this driving pressure, primarily due to low heat transfer from the groove through the evaporator shell, the vapor generated at the wick-groove interface may not escape easily the evaporator and move toward condenser section along the vapor line. At the same time, the liquid temperature inside the wick may increase with almost the same rate with the liquid near the wick-groove interface. The temperature increase of the liquid in the wick and the reservoir may cause vapor bubble generation inside the wick and hinder successful start-ups. From the experimental observation in this study it is presumed that, in BOM, a lower vapor pressure than NOM has induced favorable working-fluid circulation through the bypass line. The auxiliary bypass loop provided lower flow resistance to the fluid flow than the main loop path, and the bypassed vapor, of which the thermal mass was very small, condensed readily by heat transfer to adjacent structures, surroundings or to the liquid reservoir through its flow path. Even a small amount of working-fluid flow in favorable direction through the wick or the liquid reservoir driven by the bypassed vapor contributed to activating the whole loop.

Fig. 6 compares the minimum thermal resistance values of the LHP in NOM and BOM for thermal loads between 5 and 7 W (heat flux between 0.41 and 0.57 W/cm^2). The minimum thermal resistance reduced by 4%, 15%, and 13% for 5-W, 6-W, and 7-W thermal loads, respectively. The minimum required heat flux for the LHP for BOM was lower than that of NOM by 37%. That is, the bypass line was useful for achieving a start-up at a lower thermal load, which would have been impossible otherwise.

Further experiments were conducted to determine how the bypass line would affect thermal performance of LHPs under high thermal loads. Fig. 7 summarizes the performance of the LHP for the thermal loads from 10 W to 130 W, which corresponds to the maximum thermal load for an imposed upper limit of 110℃ at the evaporator wall. For thermal loads higher or equal to 50 W, LHP in BOM resulted in almost the same values (within 5% of relative values) of thermal resistance with NOM. Exceptions were observed for 10-W thermal load, where BOM exhibited lower thermal resistance by about 10%, and for 30-W thermal load, where BOM resulted in a little higher thermal resistance than NOM by 0.5℃/W. From the experimen tal results, it was obvious that the bypass loop did not deteriorate the LHP performance under high thermal loads.

If the vapor through the bypass line was transferred to the liquid reservoir before it was completely condensed, a successful start-up would have not been achieved due to bubble blockage effect to the liquid flow and also due to insufficient pressure gradient across the wick. It is presumed, therefore, the bypass loop functioned as an auxiliary condenser loop when the LHP was operating under normal to high thermal loads.

4. Conclusions

The following can be stated based on the experiments in this study:

(1) With a bypass line in an LHP, the minimum thermal load, or heat flux, required for a start-up was able to be reduced to some extent. For the specific LHP in this study, the reduction was up to 37% from the LHP without a bypass line.

(2) The thermal resistance of the LHP with a bypass line can be reduced by 15% from the LHP without a bypass line, for very low thermal loads.

(3) Once start-up is achieved for the LHP with a bypass line opened (in BOM) under low thermal loads, periodic performance variation might exist with the specific period varying with the applied heat flux.

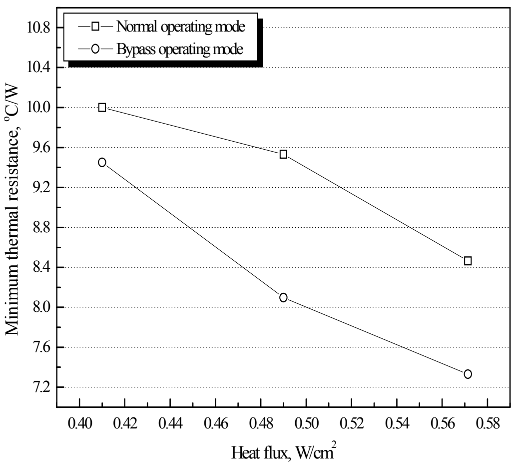


Fig. 6. Comparison of the minimum thermal resistance against heat flux in NOM and BOM under low thermal loads (5 to 7 W).

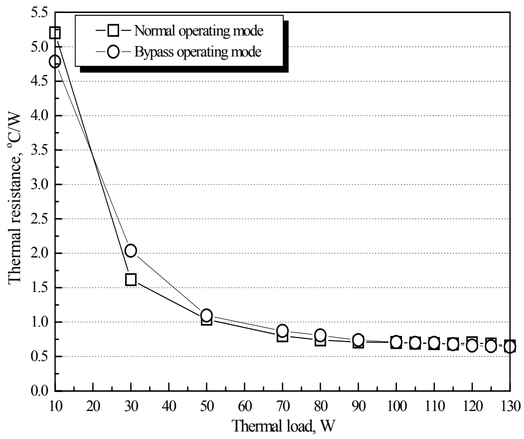


Fig. 7. Thermal resistance of the LHP in BOM and NOM for high thermal loads between 10 and 130 W.

(4) Under normal to high thermal loads, steady-state performance of the LHP in BOM exhibited only minor difference from that in NOM. Therefore, a bypass line can be employed in an LHP as a start-up aid for low thermal loads.

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