Eng & Tech. Journal, Vol.30, No.13, 2012

Heat Rejection from Horizontal Tube through Heat Valve

Ali D.Salman
Electromechanical Engineering Department, University of Technology/ Baghdad
Email: aligaphory@yahoo.com

Received on:2 /11/2011 & Accepted on: 5/4/2012

ABSTRACT

This study presents experimental work to construct a non- conventional thermosyphon where it is consists of three parts, which include the evaporator, adiabatic and condenser sections all of these parts are arranged radially in parallel to construct the heat valve (HV). Heat valve used to evacuate heat from horizontal copper electrical heater tube by working fluid which is(distilled water, methanol, ethanol) with different filling ratio 66.6% and 83.3%. Working fluid evacuate heat from copper evaporator towards condenser where a change in phase mechanism make insure to increase the heat performance of HV. Thin stainless steel mesh grid coated evaporator to translate heat to the working fluid at the case of high evaporation of working fluid, while a very thin of aluminum fins are fixed to the condenser surface to extended it and construct the finned heat valve(FHV) with a high level thermal performance where heat reject in short time and make insure to condensate the working fluid's vapor and that lead to low level of temperature in evaporator. The results refer to the high thermal performance of FHV where a low temperature in evaporator with low heat storage because the short distance between condenser and evaporator with a high ratio between area of them .Result refer to the effect of working fluid type with simple effect in filling ratio.

Keywords: Heat valve, Thermosyphon, Finned Tube.

طرح الحرارة من أنبوب أفقي خلال صمام حراري

الخلاصة

من خلال هذه الدراسة العملية تم بناء ثرموسايفون غير تقليدي حيث المناطق الثلاث الأساسية (مبخر، أديباتي، المكثف) مرتبه بشكل قطري على التوازي وليس على التوالي كما في الثيرموسايفون التقليدي لتكوين الصمام الحراري. استخدم الصمام لإزالة الحرارة من مسخن كهربائي بهيئة أنبوب نحاسي أفقي وباستخدام كل من الايثانول، الميثانول والماء المقطر كمائع عامل وبنسب ملء مختلفة (66.6%، 83.3%)، حيث إن المائع العامل يعمل على نقل الحرارة من المبخر (الأنبوب النحاسي)

عن طريق الحمل ثنائي الطور نتيجة التغيير ألطوري للمائع العامل والذي يضمن كفاءة حرارية عالية للصمام الحراري . كما تم استخدام شبكه معدنية في غاية الدقة ومصنعه من الفولاذ تغطي المبخر لتوصيل الحرارة للمائع العامل في حال نسب التبخر العالي للمائع العامل، ولرفع الأداء الحراري للصمام الحراري تمت زعنفة المكثف بزعانف رقيقة السمك مصنعه من الألمنيوم لزيادة ألمساحه السطحية للمكثف من جهة الهواء ومن ثم تأمين طرح الحرارة بشكل سريع وضمان تكثيف المائع العامل لضمان تبريد أسرع للمبخر.

أشارت النتائج إلى أن استخدام الصمام الحراري المزعنف والذي تكون فيه المسافة بين المبخر والمكثف قريبة جدا ومتقابلة إشعاعيا كما أن مساحة المكثف المزعنف اكبر بكثير من المبخر مما أدى إلى طرح الحرارة بشكل سريع (تقليل الحرارة ألمخزونه) والذي أدى بدوره إلى خفض في درجة حرارة المبخر أي أن أداءه الحراري عالي، كما أشارت الدراسة إلى تأثير ملحوظ في تغيير نوع المائع العامل للصمام الحراري المزعنف مع تأثير بسيط لنسبة الملء.

INTRODUCTION

Augmentation of heat rejection from engineering components such as aerospace and automobile vehicles, gas-cooled, nuclear reactors, cooling of electronic component for digital computers, the instrumentation of modern aircraft and others. The cooling of electronic equipment continues to be an active branch of heat transfer investigation. Though new developments in electronics enhance the performance of electronic devices, these developments often mean the downsizing of the devices, which increases the heat flux generated by the electronic components. The dissipation of this flux inside the device can lead to such thermal problems as overheating, which can reduce the devices' performance levels as well as their lifespan. To avoid such problems, it is advisable to design effective cooling systems able to evacuate the considerable heat generated.

Finned tube is a conventional method to reject heat from condensers, radiators , electrical heater tube electrical parts where a wide range are used, beside the heat that reject from the heat pipe and thermosyphon finned condensers. The efficiency of fins and surfaces are important for the thermal performance of finned heat exchangers, sometimes the increase in fin's area or numbers of it are not useful. therefore the optimum number of fins in design must be compute.

A conventional thermosyphon or two-phase closed thermosyphon. It is a device that can transfer heat from equipments and, or be used anywhere that requires an even distribution of temperature. The thermosyphon consists of three parts, which include the evaporator, adiabatic and condenser sections, all these parts are arranged in series to construct the thermosyphon fig.1. The evaporator section of the thermosyphon is the part which is always at the bottom because the condensate returning to this section is assisted by gravity[1]. The operating process begins at the evaporator section which is filled with working fluid. The working fluid is a saturated liquid, which is heated by a heat source such as a hot bath or electrical heating element. The saturated liquid then changes to vapour and moves up to the condenser section. After that, the vapour in the condenser section transfers the heat to a heat sink such as cooled water. As a result, the vapour condenses to a liquid and flows down to the evaporator section [2].

In view of the performing processes, thermosyphon heat exchangers have significant advantages over general heat exchangers, since they do not require any external energy. Moreover, they have a high rate of effectiveness with less maintenance problems, as they contain nonmoving parts. With appropriate working fluid, they are able to operate even when the temperature difference between the heat source and the heat sink is very small. However, it has some operational limits such as entrainment, vapour pressure and boiling limit. Such limitations dramatically affect the thermal performance of the thermosyphon. Therewere many studies that attempted to investigate heat transfer characteristics of it in order to improve thermosyphon performance .

Many studies attempt to improve the thermosyphon thermal performance by change the cross section area or used new working fluid with different filling ratio, others used nano particles in working fluid or used the magnetic field, others used metal working fluid, etc [3,4,5]. but all these studies are used the classical structure of heat pipe or thermosyphon fig.1, except D.Astrain [6] where his thermosyphon consists of a prismatic and hermetically closed chamber, with a fluid

In this experimental study, we investigated the possibility of using a thermosyphon principles to make a heat valve (HV) ,then development it to finned heat valve(FHV). Conventional thermosyphon consists of three parts, which include the evaporator, adiabatic and condenser sections. In this study a real different construction for non conventional thermosyphon are built. A radial successional area are constructed, evaporator on the core is coated with stainless steel mesh grid to ensure to transfer heat from evaporator to the low level working fluid in the work space where a second area or adiabatic area presented, then third area where the surface of condenser shown Fig.2. Heat transfer always from the core to outside therefore we call the device heat valve (HV). To make this device more effective it must to increase thermal efficiency of condenser surface and we do that by extended its surface by added a thin aluminum fins, where a finned heat valve (FHV) are constructed with high thermal performance and low heat storage.

EXPERIMENTAL SET UP

Rig set-up

The rig that use to test the models HT,FHT,HV and FHV fig.3 consist from air duct construct from the Perspex with dimension (w=0.206,h=0.1,L=0.54)meter and coated with cork insulation, air flow through duct by centrifugal fan. A.C.V stabilizer are used with voltage regulator to supply a stable electricity and many measurement instruments are used like Digital voltmeter and ammeter to calculate the heat generation also a digital anemometer used to calculate the velocity through the duct then to find the flow rate of air. A four digit display thermometer with a data logger connected to the computer to record the temperature using type-k thermocouple as show later. In two type models HV and FHV a calibrated Borden gauge used to measure the pressure in working zoon (adiabatic zoon). This gauge jointed to the Three way valve, where that valve connect to the models by capillary tube as shown in fig.2 and fig.3. Three cubic foot per minute(3CFM), two stage vacuum pressure used to evacuate the non-condensable gases. All accurate dimensions are measured by digital microscope with 400X.

Models set-up

Many probational models are constructed to make a comparison between the conventional and non-conventional heat rejection devices:

HT model set-up

Simply heater tube consist from electrical heater with 96 ohm resister and 0.3mm in diameter . The heater insulated electrically by shell that made from ceramic then insert to copper tube with 0.745 cm in nominal diameter. Two ends of heater are locked by ceramic lock then the two ends of heater are connected to the power supply control of the rig. many thermocouple are welded to the HT surface using electronic solder with aluminum soldering flux(Amino ethyl ethanolamine).

HV model set-up

A non-conventional thermosyphon constructed where the three main parts that included the evaporator, adiabatic and condenser sections are arranged radially fig.2. HT that described in above present the evaporator's part to the heat valve (HV). HT coated with mesh grid of stainless steel with 0.08mm in diameter and approximately 51 wire per cm intensity. The mesh grid immersed in working fluid to transfer heat all the time fig.2 side section. Copper shell with 23.7 mm in diameter covered the meshed evaporator then flooded by working fluid to construct the heat valve (HV) where the heat transfer in one direction from the core to the outside. Copper capillary tube welded from its top end to the surface of evaporator (HT) ,while other side end welded to the surface of condenser fig.4, thermocouple probe immersed in this capillary tube and contact to the evaporator surface by connection cream to measure evaporator temperature. Other probes are welded to the surface of condenser in different locations. Calibrated Borden Gauge with three way gate valve connected to the adiabatic zoon in HV by capillary tube .

FHT model set- up

Finned heater tube fig.5 consist from a HT that finned by rectangular aluminum fins to get better heat transfer. Fin dimension (5 *5*0.02) cm, these fins are fixed to the HT by welded it using electronic solder with aluminum soldering flux(Aminoethylethanolamine). Total number of fins are fourteen (14 fin) distributed along the effective length of HT (21 cm). Many thermocouples probes are fixed at the FHT surface and on the fins surfaces in different location.

FHV model set-up

To increase the HV thermal performance by extended the area of its condenser, where extended surface added by fixed thin aluminum fins on its condenser and construct the finned heat valve (FHV) fig.2 and fig.5. Same procedure are used to fixed the fins on condenser's surface. As in the HV the heat transfer radialy in one direction from the core to outside, and many probes are welded to the extended surface (fins) near the root and the tip of fins.

Experimental Procedure

All the models (HT,FHT,HV and FHV) are tested by insert the model in main duct of rig and applied a different voltage (different heat generation) at a constant value of air flow rate. Temperature, voltage applieds, alternative current and others are recorded for each case at transient time arriving to the steady state as shown:

Heater tube (HT)

For A different voltage applied (30,40,50,60and70 volt), many thermocouple probes are welded to the surface of the heater tube while others probes are insulated thermally to measure the heater temperature.

Finned heater tube(FHT):

same procedure for HT is applied with a wider in voltage range (30,40,50,60,70,80,90) beside the temperatures that measured for the root and the tip of fins. Many thermocouple probes welded on fin surface without Thermally insulation to get surface temperature while other probes insulated from air side to record the fin temperature.

Heat valve (HV):

A different voltage applied (30,40,50,60,70,80,90) and different working fluid (distilled water, methanol and ethanol) are used with filling ratio 66.6% and 83.3%. For each case temperature recorded for evaporator, condenser and condenser's surface also the corresponding pressure for each case are recorded.

Finned heat valve(FHV):

Like same procedure in HV was applied at each case beside the temperature that recorded in the root and tip of fins. Each test at each working fluid with different filling ratio and different voltage applied took long time therefore at this model only one filling ratio 83.3% was tested with the different working fluid and voltage applied depended on the good result of this ratio than from the other.

THEORETICAL CONSIDERATION

In this study, it was assumed that there was no heat losses directly from tested models FHV,HV,FHT,HT to the ambient because they were completely insulated. Therefore heat that generated by the electrical heater transferred and stored or transferred completely to the air flow through the duct.

Heat that generated in electrical heater can be calculated from the following equation:

$$Q = V * I \qquad \ldots(1)$$

And to calculate the heat rejected from outside surface of tested model:

$$Q=VI=\overline{h} * A_s * (T_S - T_a) \qquad ..(2)$$

Therefore to evaluate the average value of $\overline{h}$ for heat that transfer from HV or HT surface to air side

$$\overline{h} = \frac{Q}{A_s * (T_S - T_a)} \qquad ..(3)$$

T_S = the avarge temperature of out side surface of HV or HT
A_s = outside surface area of HT or HV(m^2), $T_a = air\ temperature\ (C)$

For the models FHT, FHV and to calculate average heat transfer coefficient, fin efficiency and surface effectiveness as shown below :
To calculate the heat transfer coefficient at air side finned tube :
Fin performance: fin efficiency φ is defined as the ratio of the actual heat transfer from the fin to the heat would be transferred if the entire fin were at its root or base temperature [7].
And to estimate the rectangular fin efficiency as annular fin having same area as the plate fin [8,9]

$$\varphi = \frac{\int h(t - t_a)dA}{\int h(t_r - t_a)dA} \qquad ..(4)$$

Where φ = fin efficiency t_r = root fin temperature (C)
t= tip fin temperature (C) t_a = surrounding temperature (C)

The heat transfer rate from a finned surface , such as a tube ,which includes both finned or secondary area A_f and unfinned or prime area A_p is given by:

$$Q = (h_p A_p + n\varphi\, h_f\, A_f)(t_r - t_a) \qquad \ldots.(5)$$

$n = number\ of\ fins$
Net Area of the rectangular fin equal to the double side rectangular area minus pipe section area (double) plus the circumference area of fin.
$h_p = tube\ heat\ transfer\ coefficient\ \left(\frac{w}{m^2.C}\right)$,
$h_f = fin\ heat\ transfer\ coefficient\ \left(\frac{w}{m^2.C}\right)$

Many references Assuming that the heat transfer coefficients for the finned and prime surfaces are equal[7,8], a surface efficiency φ_s can be derived for use in Equation (5): Then surface effectiveness can be derived :

$$\varphi_s = 1 - \left(\frac{nA_f}{A}\right)(1-\varphi) \qquad \ldots(6)$$

Simple form of equation. 5 can be reform with avarge value of heat transfer coeffitiont as shown

$$Q = \varphi_s \, \overline{h} \, A_{tpf}(t_r - t_a) \qquad \ldots.(7)$$

$$where \quad A_{tpf} = nA_f + A_p \qquad \ldots(8)$$

$$\overline{h} = \frac{I * V}{\varphi_s \, A_{tpf} \, (t_r - t_a)} \qquad \ldots(9)$$

From the assumption above the fin efficiency can be estimation as [7,10]

$$\varphi = \frac{t - t_a}{t_r - t_a} \qquad ..(10)$$

RESULTS AND DISCUSSION

At the heater tube(HT) fig.6 the temperature of its body value always increase with time during the transient region at different voltage applied (diffrent heat generation), while the steady state region was disappeared because there is a little heat evacuated from the body of copper heater tube, where there is no enough external area to exchange heat that generated in heater to the air, and that lead to more heat storage in the HT body with more increase in temperature specially at high value of voltage applied

$$Q_{gen.} = Q_{rej.} + Q_{sto.} \qquad \ldots.(11)$$

With the classic conventional sophisticated, thin aluminum fins were fixed to the external surface of HT to decrease its temperature. Where a finned heater tube(FHT)was construct, result's fig.(7) refers to a short transient duration with long steady state region and clearly decreases in FHT temperature of body comparing with the results of HT fig.6. and that mean :

$$Q_{gen.} = Q_{rej.} \qquad \ldots.(12)$$

With no more heat storage and low level temperature in FHT model Where the main base goal was done. In this conventional model the increase in extended area of heater body help to increase heat that rejected to the air flow through the duct over the tested model. At this model Low level of surface temperature recorded in steady state region, where that mean a low heat storage eq.11, eq.12. According to the results the main differ between HT and FHT present the extended area table.1 where the thermal performance of the model FHT depend on the rate of extended area to the prime area.

For a non conventional model and to evacuate the heat quickly from copper tube, heat valve (HV) and finned heat valve (FHV) are made and used. figures 8(a-g) have been presented the comparison between evaporator temperature for HV and FHV with FHT surface temperature at different heat generation value. while the pressure profile along the time has been compared between HV and FHV.

Heat rejection and temperature profile.

From fig.(8) group, the temperature profile refer to the low thermal performance of HV because the surface area of condenser is not effective. The figures 8(a-g) present real comparison between FHV and FHT. Characteristic results for FHV at different working fluid and heat generation than from FHT and that can be notice in the figures and that refer to the special thermal performance of FHV than from FHT, where the heat rejected quickly in FHV with short transient duration time.

The special results of thermal performance for FHV present the affectivity of its unique design. Where the short distance between evaporator and condenser, and the evaporator covered completely by the condenser along the model and from all directions where that be assuring to transfer heat very quickly, beside there is no chaotic flow behavior between the evaporative and condensate working fluid as in the classic thermosyphon at the high level of heat generation. The area of condenser and evaporator toward adiabatic part (toward each other) along the model and that mean the evaporator length same as the condenser while the condenser inside area it should bigger than the evaporator outside area. Also the special cause for the high thermal performance of FHV is that the evaporator is immersed in the working fluid and that insure to transfer heat by conduction through liquid in low level of heat generation, and at high level of heat generation the phase change are applied to take away the heat. Because of the phase change , caused to absorb a large amount of latent heat.

Profile of temperature difference

In the fig.9 present the temperature difference profile between heater tube and its surface against time for FHT model. The results refer to that the difference in temperature bigger than zero and the difference in temperature proportions with the heat generation amount. Steady state are appeared at the curves except the highest heat generation 84.87 watt. While the maximum average value in temperature difference equal to about 8C.

Figure.(10 a-f) presents temperature difference profile between evaporator and condenser for the HV model at different heat generation and different filling ratio. Fluctuating behaviors of profiles are appeared at the water and ethanol in approximate difference in temperature value about 4C, while the range of this value [-2 to 2]. The model with methanol working fluid had a more stable relation with approximately same difference in temperature and with range [~0 to -4.5] .

At the HV group the fluctuating in difference temperature mean that the chaotic flow behavior formed, because there is no enough saturated liquid , in other words the change in phase from vapour to liquid is not uniform , and that lead to more storage of heat in the copper tube and then increase in temperature where that show clearly in figures.8(a-g). Therefore to evacuate the heat quickly from copper heater tube by condensate the vapour quickly by add extended area to condenser.

Figures.11(a-c) present same relation that in figures.10(a-f) but with the FHV model. Where a different heat generation are applied with only 83.3% filling ratio.

Difference in temperature for the group are more staple and that mean the stability in rate of heat that rejected and uniform change in phase. Fig.11-a with FHV-w-25 model, shown a stability in temperature difference with value range [-0.85 to 5.2 C]. Other model FHV-M-25 had the same behavior in relation with range value [-0.4 to 7.4 C] while the special model FHV-E-25 shown the low in temperature difference value with range [~ -0.53 to ~3.1 C], and that converges between evaporator and condenser temperature refer to the higher thermal performance of the model type FHV-E-25 where the Ethanol used as working fluid.

Pressure behavior

In figures (8 a-g) we can show the pressure behavior for the different model of FHV and HV with different conditions. While fig.(12) presents pressure behavior against evaporator temperature where a different value of heat generation applied. The relation shown that the pressure proportion with heat generation, and at the same test conditions for different models the propriety of working fluid refer to the higher pressure value for methanol, then the mid pressure value present for ethanol , with low level pressure for water. High pressure value refer to quick vaporization of methanol

liquid, and difficultly to condensate its vapour with low latent heat and density than from water, while the low pressure value of water refer to slow vaporization with high latent heat and density. Ethanol properties refer to lower value in latent heat and liquid density than from distilled water and methanol. High vaporization and condensation of Ethanol depend on the low value of latent heat and the lowest vapour viscosity of it.

The pressure profile of models HV and FHV that tested at same condition refer to the effectivity of extended surface that add to the HV to construct FHV. figures (8 a-g) refers to big difference in pressure between HV and FHV. Where the vapour condensate quickly in FHV condenser and return as liquid phase to flooded evaporator and evacuate heat again by applied in the phase change to take away the heat, Therefore there is no much heat storage in evaporator. In HV model the vapour difficultly condensate because the surface of outside condenser it is not effectively enough to reject heat to air flow over it, Therefore we made FHV and that lead to highest heat rejection with low heat storage and that mean low temperate in evaporator.

Fin efficiency and surface effectiveness.

Fin efficiency of FHV bigger than FHT as shown in fig.(13), because the fins area of FHV smaller than of FHT. Also the FHV fin base (circumference) bigger than of FHT, Therefore the temperature of fins area in FHV model approximate to the fin root temperature, and that lead to higher fin efficiency of the model FHV than from FHT. At same reason the surface effectiveness of FHV bigger than of FHT.

Air side heat transfer coefficient($\bar{h}$).

Fig.14 shown that the average heat transfer coefficient in FHT more than in FHV. In equation-7 when we assume that $\bar{h}$ is uniform over the fin and base surface, then the increase in temperature difference for model FHT with low value in surface effectiveness fig.(13) leads to increase in heat transfer coefficient. For model FHV with low difference in temperature and high value in surface effectiveness, low level in heat transfer coefficient are appeared.

Working angle

A specialized design of FHV make it so easy to return the condensate vapour without need to inclined the model body at any angle degree. Therefore the reading that record at zero angle.

CONCLUSIONS

In this experimental work we study the effect of heat generation rate, working fluid type and filling ratio to the thermal performance of the models (HT,FHT,HV and FHV). In the limit condition of work we found that:

1-There is no big effect of the filling ratio.

2-Higher thermal performance appeared for FHV then FHT then HV, while the lowest value for HT model.

3-FHT arrived to the study state in short duration time from than FHV, while HV need more time from the other models.

4- Temperature difference $(T_e - T_c)$ in FHV model is more stable and almost bigger than zero, that mean there is no fluctuating in temperature and the evaporator temperature almost bigger than from condenser , also that refer to the stability of heat rejection .

5- Lower evaporator temperature recorded at the same model and same condition for Ethanol working fluid.

6- Fin efficiency and surface effectiveness of FHV model had a higher value from FHT model.

7- Air side heat transfer coefficient for FHT model higher than the value of FHV model.

8- one of the important thing in the FHV design that there is no limit angle to return Condensate vapour .

REFERENCES

[1]D.A.Reay "Heat Pipes"B&H, Fifth Edition(2006)

[2]P.Amatachaya,W.Srimuang" Comparative heat transfer characteristics of a flat two-phase close thermosyphon(FTPCT)and conventional two-phase closed thermosyphon (CTPCT)" international communications in heat and mass transfer,37(2010)293-298.

[3] Gabriela Huminic, Angel Huminic, Ion Morjan, Florian Dumitrache" Experimental study of the thermal performance of thermosyphon heat pipe using iron oxide nanoparticles" International Journal of Heat and Mass Transfer 54 (2011) 656–661.

[4] Gabriela Huminic, Angel Huminic"Heat transfer characteristics of a two-phase closed thermosyphons using nanofluids" Experimental Thermal and Fluid Science 35 (2011) 550–557.

[5]S.Rittidech,W.Srimuang"Correlation to predict heat transfer Characteristics of a vertical flat thermosyphon (VFT) at normal operating Conditions"International journal of heat and mass transfer,53(2010)5984-5987.

[6]D.Astrain,J.G.Vian,M.Dominguez"Increase of COP in the thermoelectric refrigeration by the optimization of heat dissipation" Applied Thermal Engineering 23 (2003) 2183–2200

[7] ASHRAE Fundamentals Hand book,(2005)

[8] Faye C,McQuiston . Jerald D.,Parker. Jeffrey D,spitler" Heating, Ventilating, and Air Conditioning Analysis and Design"John Wily,2000.

[9]Wilbert F,Stoecker. Jerold W,Jones, "Refregeration and Air Conditioning"McGRAW- HALL ,1985.

[10] Alireza Bahadori . Hari B, Vuthaluru"Predictive tool for estimation of convection heat transfer coefficients and efficiencies for finned tubular section" International Journal of Thermal Science 49(2010) 1477-1483.

Nomenclature

A	area (m^2)
D	Diameter m
E	ethanol
F,f	fin
FHV	finned heat valve
FHT	finned heater tube
FR	filling ratio
HV	heat valve
h	heat transfer coefficient $(W/m^2.C)$
$\bar{h}$	average heat transfer

	coefficient ($W/m^2.C$)
I	alternative current, (Ampere)
M	methanol
P	pressure (pa)
T	temperature (C) ,tube
U	velocity
V	Voltage apply, (volt)
W	water

Dimensionless Nu.

Re	Reynolds number

Subscript

a	air
av	average
cond.	conduction
conv.	convection
d,D	diameter
env	environment
f	fin, fin surface
f,to	total fins surface
i	in
o	out
rej.	rejection
s	surface
sto.	storage
t	tube
tpf	total prime and fins surface
p	prime, prime surface

Greek symbols

φ	fin efficiency
φ_s	surface effectiveness
ρ	density (kg/m^3)

Table (1) Surface area details of HT,FHT,HV and

	D_o (m)	L (m)	Fins number	A_p (m^2)	$A_{f,to}$ (m^2)	$A_{tpf} = A_p + A_{f,to}$ (m^2)	$A_{f,to}/A_{tpf}$	$A_{f,to}/A_p$
HT	0.0079	0.21	14	5.2119e-3	-----	5.2119e-3	---	---
FHT	0.0079	0.21	14	5.1424e-3	14*4.9419e-3	7.4329e-2	0.93	13.45
HV	0.0235	0.21	14	1.5503e-2	-----	1.5503e-2	---	---
FHV	0.0235	0.21	14	1.5297e-2	14*4.1725e-3	7.3712e-2	0.792	3.818

Table (2) code of models work

Model	Working fluid type	Dose (CC)
HT	-	-
FHT	-	-
HV	W, M, E	20, 25
FHV	W, M, E	25
Model-working fluid-dose		

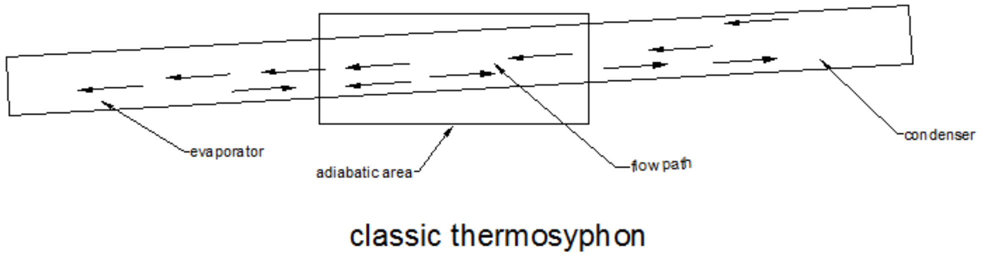


Figure (1) classical thermosyphon .

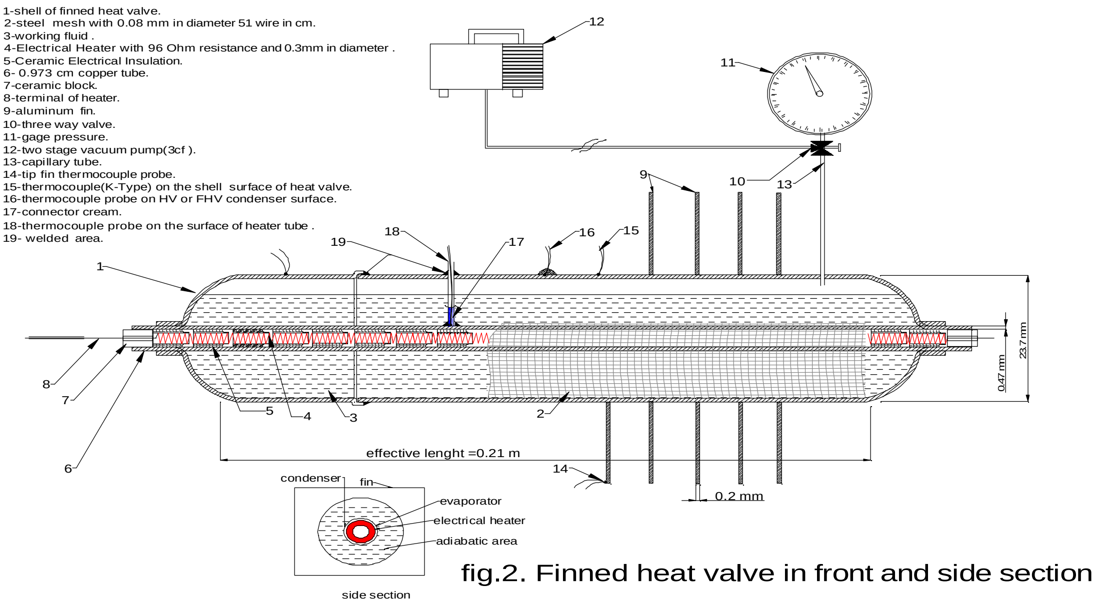


Figure (2) Finned heat valve (front and side sections).

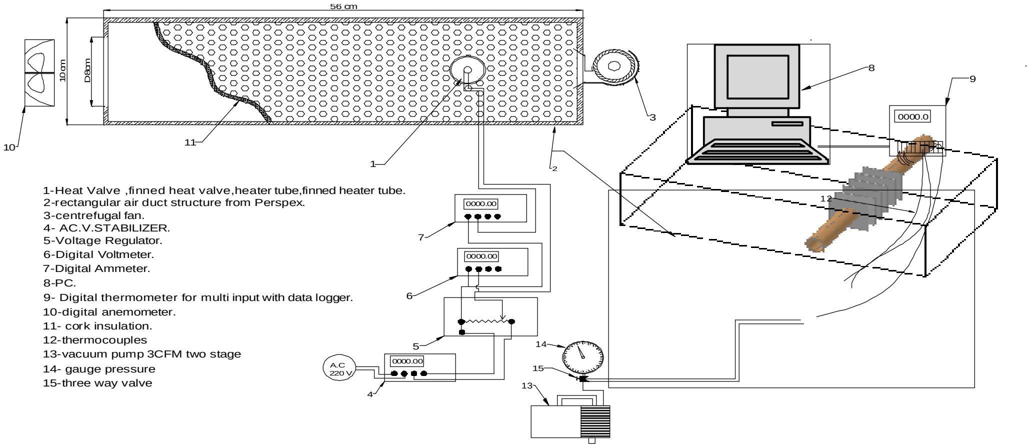


Figure (3) Rig that used to test the models.

Figure (4) heat valve (HV) construction

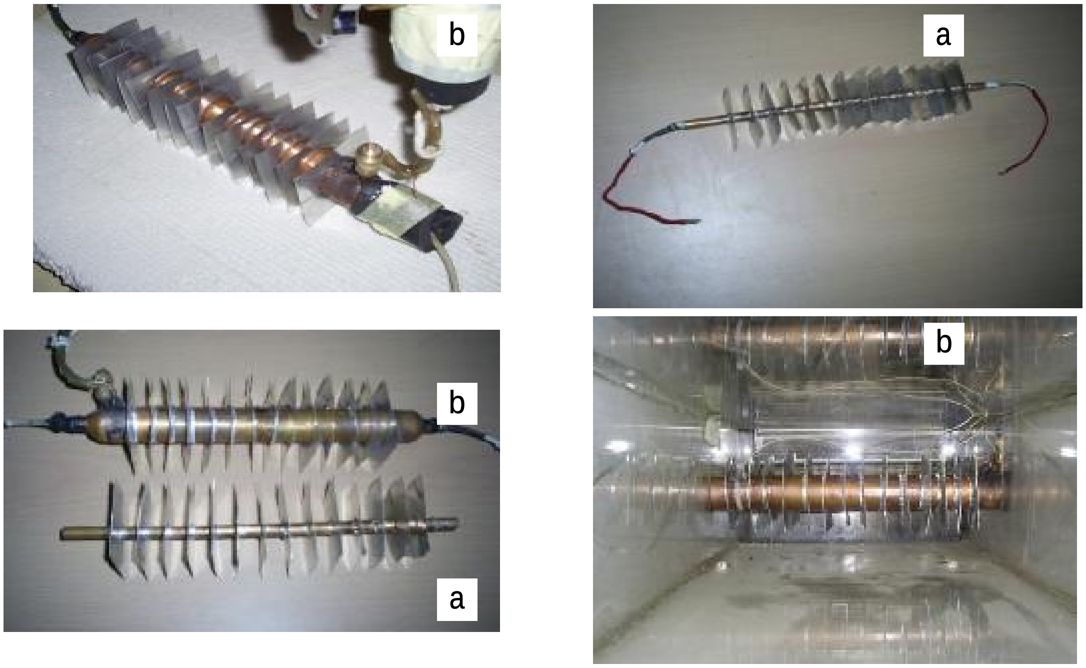


Figure (5) :a-finned heater tube(FHT) , b- finned heat valve(FHV)

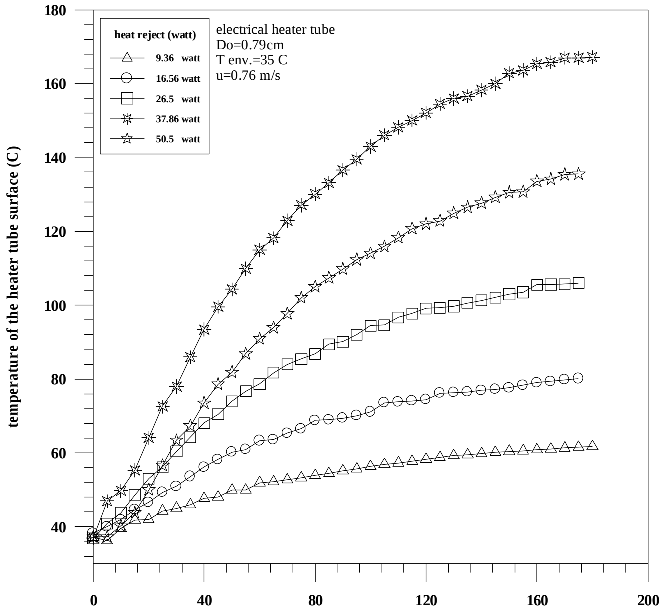


Figure (6) The temperature of the electrical heater tube (HT) surface against time at a different heat rate.

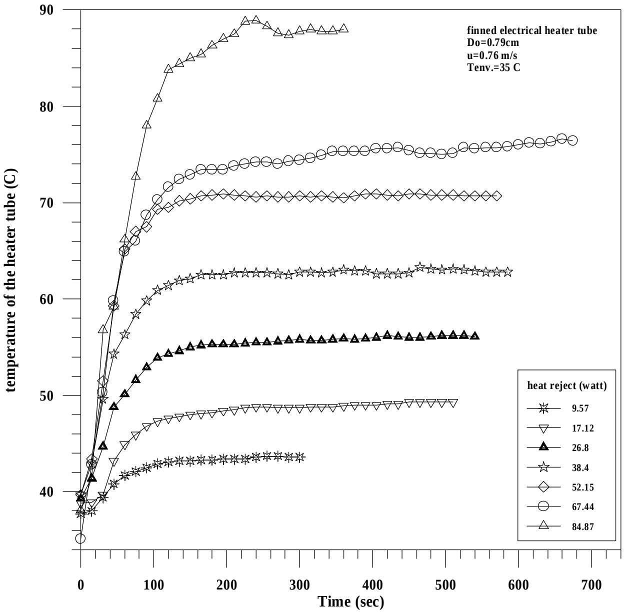
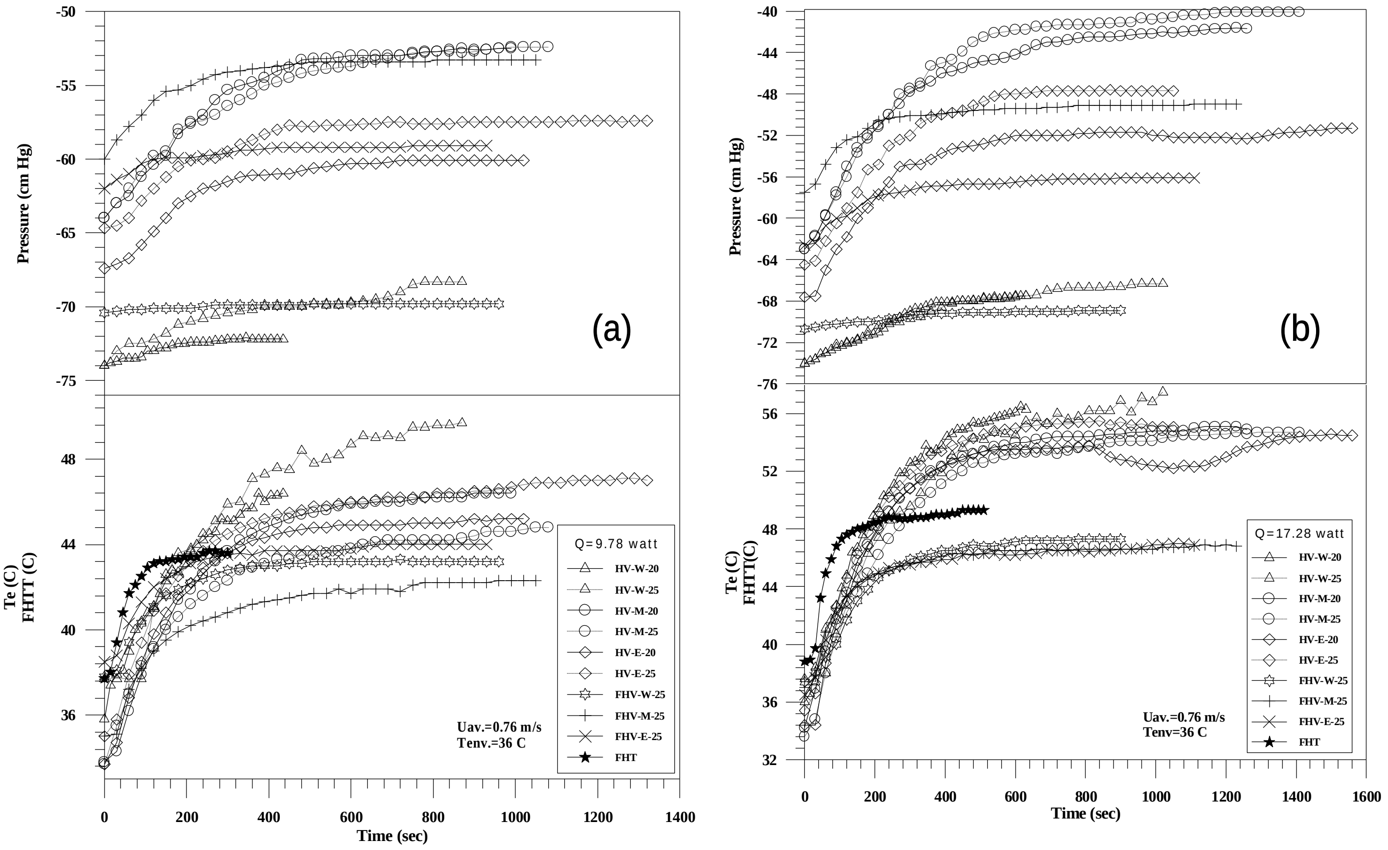
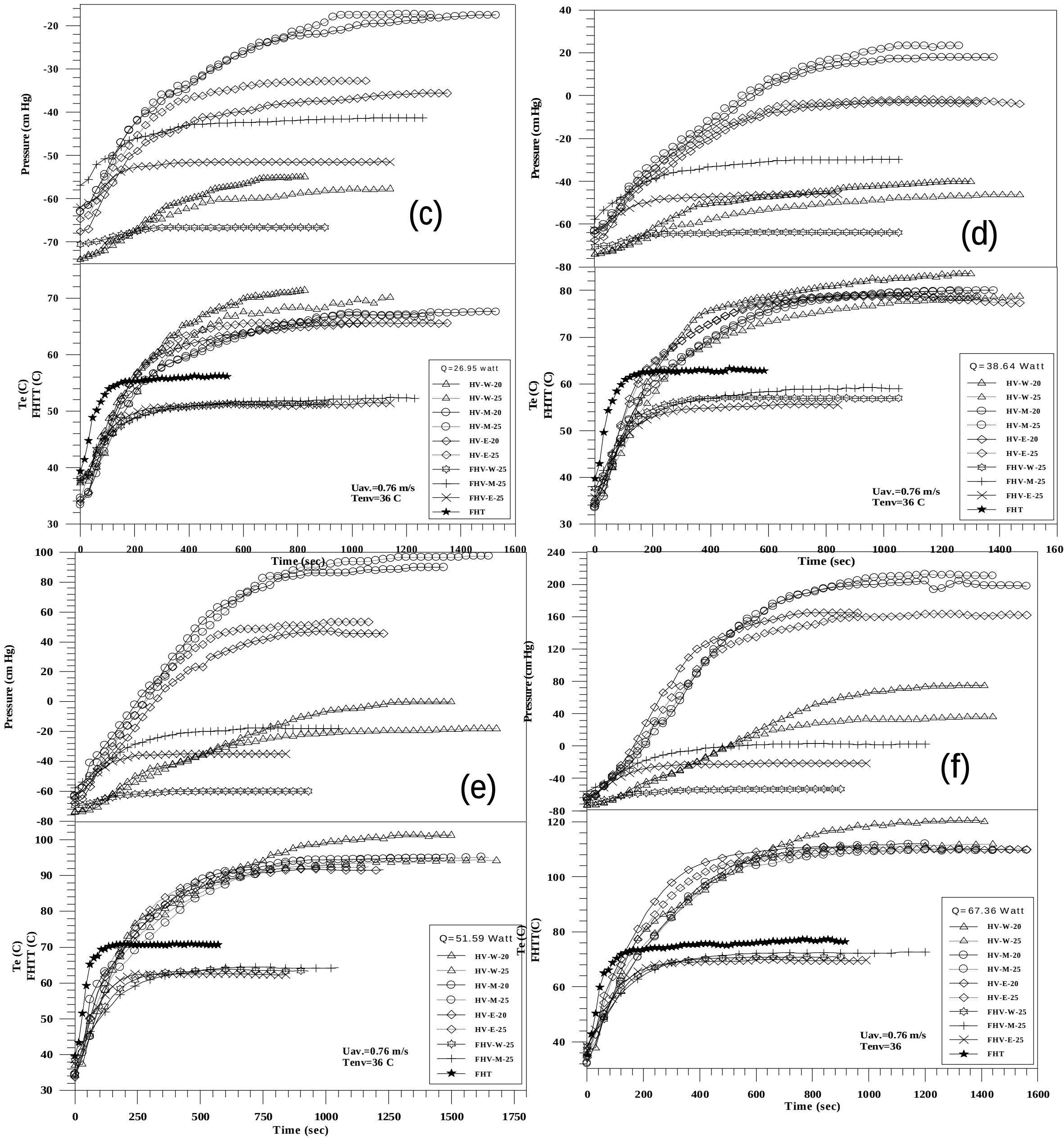


Figure (7) The temperature of the finned electrical heater tube (FHT) surface at a different hear rate.


(a)
Pressure (cm Hg)
-50
-55
-60
-65
-70
-75
Te (C)
FHTT (C)
48
44
40
36
0
200
400
600
800
1000
1200
1400
Time (sec)
Q=9.78 watt
HV-W-20
HV-W-25
HV-M-20
HV-M-25
HV-E-20
HV-E-25
FHV-W-25
FHV-M-25
FHV-E-25
FHT
Uav.=0.76 m/s
Tenv.=36 C
(b)
Pressure (cm Hg)
-40
-44
-48
-52
-56
-60
-64
-68
-72
-76
Te (C)
FHTT(C)
56
52
48
44
40
36
32
0
200
400
600
800
1000
1200
1400
1600
Time (sec)
Q=17.28 watt
HV-W-20
HV-W-25
HV-M-20
HV-M-25
HV-E-20
HV-E-25
FHV-W-25
FHV-M-25
FHV-E-25
FHT
Uav.=0.76 m/s
Tenv=36 C


(c)
Pressure (cm Hg)
Te (C)
FHTT (C)
Time (sec)
Q=26.95 watt
HV-W-20
HV-W-25
HV-M-20
HV-M-25
HV-E-20
HV-E-25
FHV-W-25
FHV-M-25
FHV-E-25
FHT
Uav.=0.76 m/s
Tenv=36 C
(d)
Pressure (cm Hg)
Te (C)
FHTT (C)
Time (sec)
Q=38.64 Watt
HV-W-20
HV-W-25
HV-M-20
HV-M-25
HV-E-20
HV-E-25
FHV-W-25
FHV-M-25
FHV-E-25
FHT
Uav.=0.76 m/s
Tenv=36 C
(e)
Pressure (cm Hg)
Te (C)
FHTT (C)
Time (sec)
Q=51.59 Watt
HV-W-20
HV-W-25
HV-M-20
HV-M-25
HV-E-20
HV-E-25
FHV-W-25
FHV-M-25
FHT
Uav.=0.76 m/s
Tenv=36 C
(f)
Pressure (cm Hg)
Te (C)
FHTT(C)
Time (sec)
Q=67.36 Watt
HV-W-20
HV-W-25
HV-M-20
HV-M-25
HV-E-20
HV-E-25
FHV-W-25
FHV-M-25
FHV-E-25
FHT
Uav.=0.76 m/s
Tenv=36

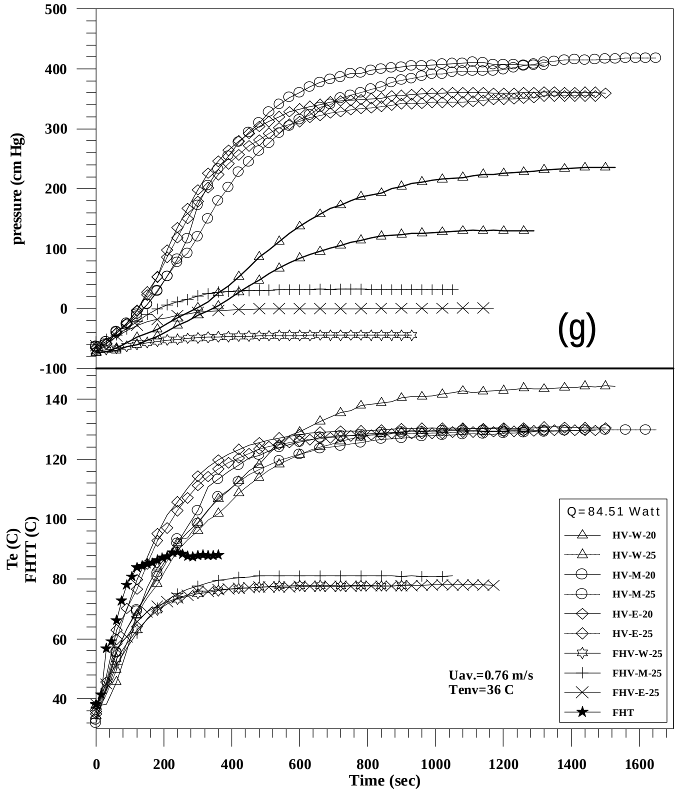


Figure 8(a-g) evaporator, tube temperature and pressure profile for HV,FHV and FHT with different heat rate.

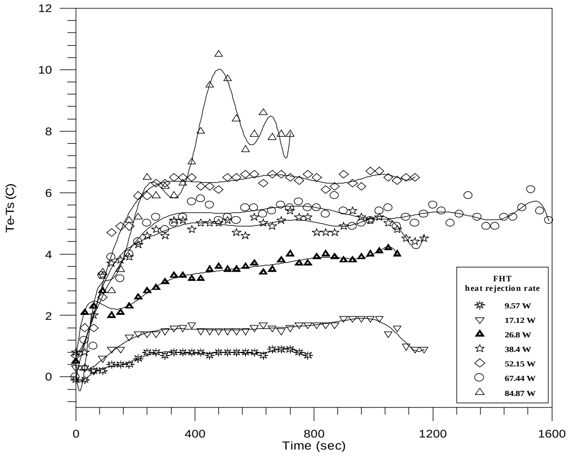
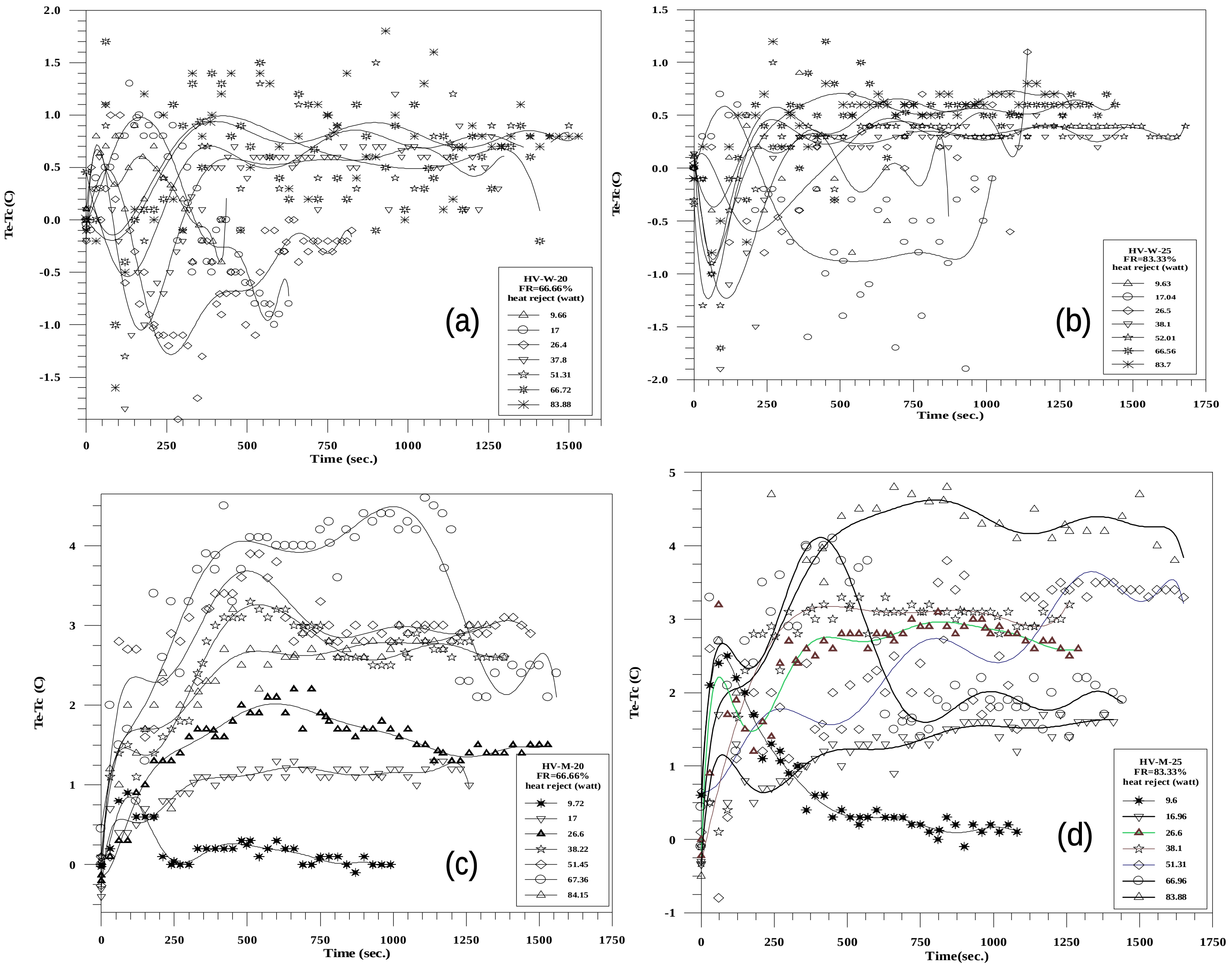


Figure (9) profile of temperature difference between FHT and its surface with different heat rate.


Te-Tc (C)
Time (sec.)
2.0
1.5
1.0
0.5
0.0
-0.5
-1.0
-1.5
0
250
500
750
1000
1250
1500
HV-W-20
FR=66.66%
heat reject (watt)
9.66
17
26.4
37.8
51.31
66.72
83.88
(a)
Te-Tc (C)
Time (sec.)
1.5
1.0
0.5
0.0
-0.5
-1.0
-1.5
-2.0
0
250
500
750
1000
1250
1500
1750
HV-W-25
FR=83.33%
heat reject (watt)
9.63
17.04
26.5
38.1
52.01
66.56
83.7
(b)
Te-Tc (C)
Time (sec.)
4
3
2
1
0
0
250
500
750
1000
1250
1500
1750
HV-M-20
FR=66.66%
heat reject (watt)
9.72
17
26.6
38.22
51.45
67.36
84.15
(c)
Te-Tc (C)
Time(sec.)
5
4
3
2
1
0
-1
0
250
500
750
1000
1250
1500
1750
HV-M-25
FR=83.33%
heat reject (watt)
9.6
16.96
26.6
38.1
51.31
66.96
83.88
(d)

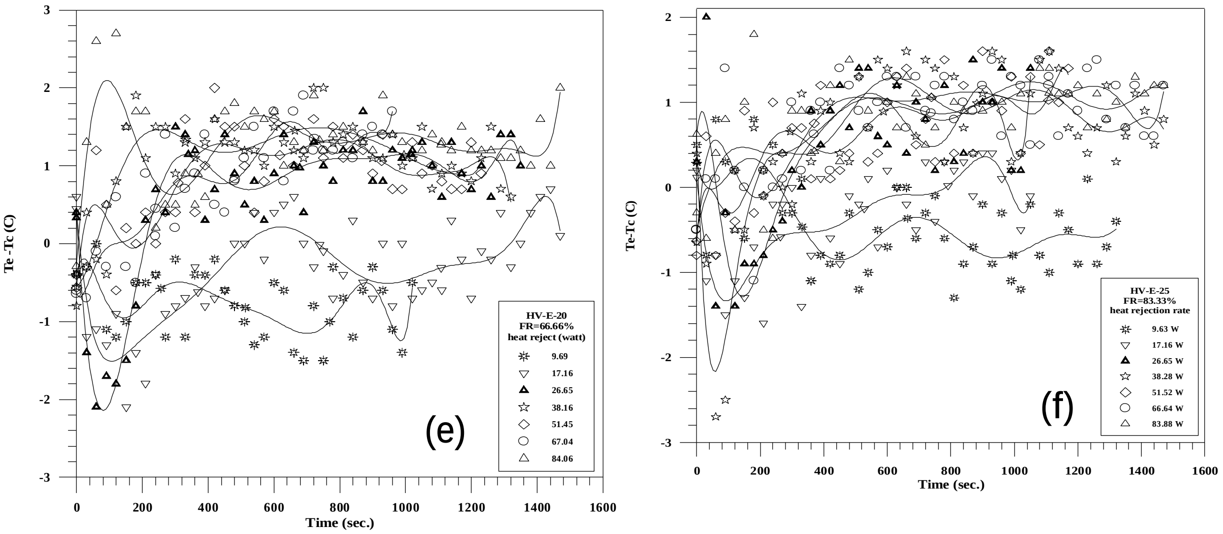


Figure 10(a-f) profile of temperature difference between evaporator and condenser for HV at different heat rate.

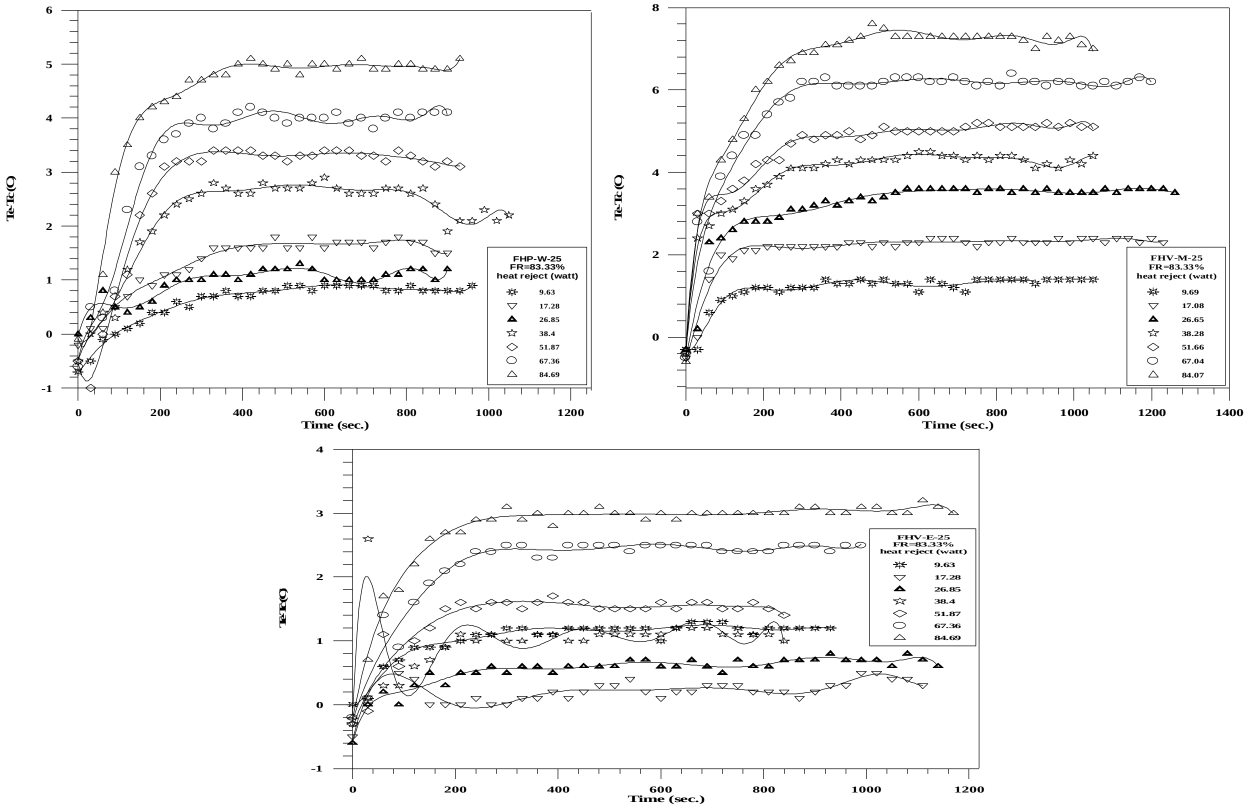


Figure 11 (a-c) profile of temperature difference between evaporator and condenser for FHV at different heat rate.

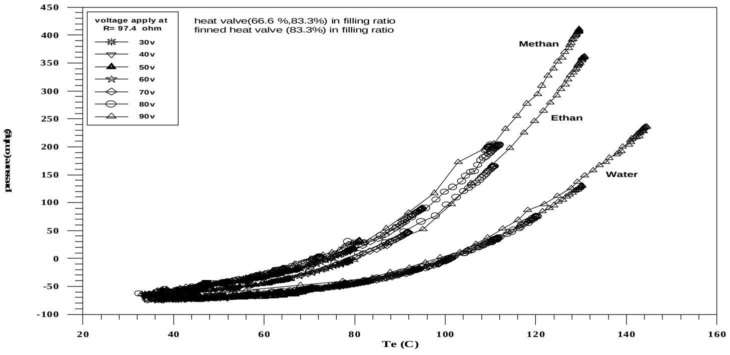


Figure (12) pressure temperature behavior of different working fluid in thermosyphon at HV and FHV.

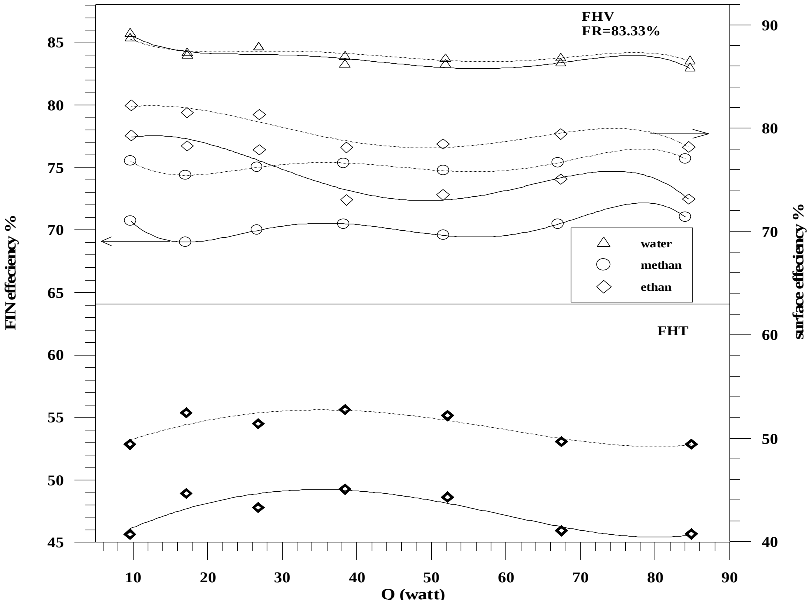


Figure (13) fin and surface efficiency for FHT,FHV at different heat rate.

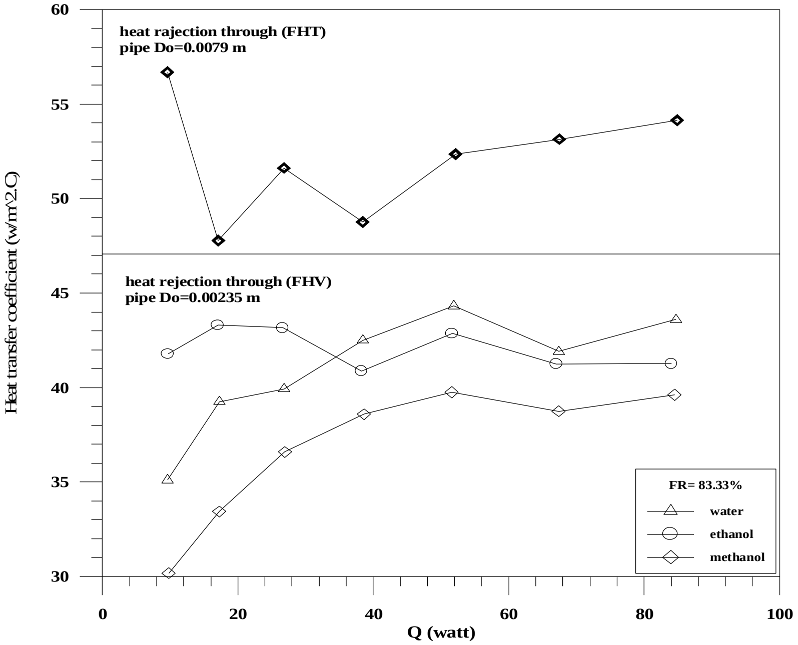


Figure (14) heat transfer coefficient for FHT,FHV at different heat rate .