DESIGN AND IMPLEMENTING A HEAT PIPE EXPPERIMANTAL SYSTEM FOR RESIDENTIAL HEATING

Zeina A. Al-Saadi Dr. Mikdam M. saleh

University of Baghdad-College of Eng.
Department of Nuclear Engineering

ABSTRACT:

The design and construction of an instrumented heat pipe H.P for domestic heating that uses water as a working fluid was undertaken to investigate experimentally the performance of the H.P under various operating conditions of: Power levels, water inventories and angle of inclinations. A theoretical model to predict the temperature of the condenser surface (the temperature at which heat is rejected) and system pressure at steady state conditions was developed and used to compare these parameters with the experimental findings. The model utilizes the total heat supplied to the evaporator to predict system pressure and condenser temperature. The theoretical model is suitable for vertical H.P (i.e. θ =0) and its predictions of condenser surface temperature is within ±16% and of system pressure is within ±21%. An acceptable H.P design may have a condenser heat flux of 1.16 k W/m^2 with a corresponding system pressure of 1500 kPa.

الخلاصة

تم تصميم و بناء منظومة أنبوب حراري للتدفئة المنزلية التي يستخدم الماء كمائع تشغيلي. أخذنا بنظر الاعتبار عند البحث التجريبي لأداء الأنبوب الحراري التغيرات في العوامل التشغيلية التالية:مختلف مستويات القدرة الداخلة إلى المسخن ومستويات مختلفة للمائع داخل المنظومة(Inventory) إضافة إلى زوايا ميل مختلفة (Inclination Angle). النموذج النظري الذي تم تطويره يتمكن من حساب المعدل لدرجة الحرارة السطحية للمكثف (t_c و ضغط المنظومة(P)في حالة الاستقرار.وتمت مقارنة النتائج العملية مع الحسابات النظرية ووجد أن النموذج النظري يستخدم الحرارة الكلية المسلطة على المسخن لتوقع ضغط المنظومة ودرجة حرارة المكثف ويعتبر هذا النموذج مناسب للأنبوب الحراري الوضع العمودي (θ= 0°) والنموذج الرياضي يعطي قيم

بحدود (±15%) لدرجة الحرارة السطحية للمكثف و (±21%) لضغط المنظومة. نستنتج أن التصميم العملـي المقبـول لمنظومـة التدفئـة المنزليـة يجـب أن يكـون ضـمن حـدود كثافـة فـيض حـراري(1.16kw/m^2) خلال ضغط منظومة (1500 kpa) وزاوية ميل تتراوح بين(0−30 ◦).

KEY WORDS:
Thermosyphon Design, Natural Convection, Heat Pipe

INTRODUCTION

A heat pipe (H.P) is a heat transfer device, which has high effective thermal conductivity. Heat pipes are evacuated vessels, typically circular in cross section, which are back-filled with a small quantity of working fluid. They are totally passive and are used to transfer heat from a heat source to a heat sink with minimal temperature gradients **(Qian and Chen 1999)**. Some applications of heat pipes are in cooling electronic devices, heat recovery from exhaust ventilation, application in cooling a computer processor **(Shunji and Suzuki 2000)**, Geological applications **(Torrance 1979)** and **(Chi 1976)** , Electronics cooling **(Marquet. and Solecki 1989)**, nuclear thermonic space power concept using rod control and heat pipes **(Dunn and . Reay 1976).** Most of the published research dealt with the theory and performance of wick type heat pipe, since most of its applications are in zero gravity environment (cooling of re-entry vehicles). To the authors knowledge there has been no experimental or theoretical work done on heat pipes specifically intended for residential heating in the open literature.

The wickless H.P to be considered in this work is like a thermosyphon. The working fluid of choice was water because of its availability and low cost and also because one can use cast iron pipe for the H.P.This choice of material allows high-pressure operation of the H.P (up to 40 bars). Of course we can use copper for construction materials because of its compatibility, corrosion wise with water.

The concept of using a H.P for domestic heating stem from the idea of providing very clean heating without resorting to more expensive electrical heating. And also avoid using any moving parts. The condenser must be cooled by natural convection driven by density differences only. Fins are to be used to dissipate the rejected heat from the condenser in order to lower the thermal resistance across the condenser.

There is no application specifically tailored for residential heating using a heat pipe in the open literature. What prompt this work is the convenience of the H.P in providing large heat flux at small temperature gradient coupled with the attractive feature of using conventional and available fuel like gasoline to heat residential homes.

The present work will attempt to address the following goals:

1. Design and implement an experimental H.P rig with a finned condenser cooled by natural convection. To study its performance under a variety of operating conditions of input power to the evaporator, water inventory and angle of inclination.
2. Develop a theoretical model to predict the operation characteristics of the H.P: mainly system pressure and condenser temperature, from a single parameter, namely the total heat supplied to the evaporator.
3. Compare the theoretical predictions with experimental results.

EXPERIMENTAL APPARATUS

The rig used for this purpose was designed and built here and is shown schematically in **Fig.A** consists of a cast iron pipe of length (2.5m) which is divided into an evaporator, condenser and adiabatic zone. Carbon steel was chosen because of its compatibilty corrosion with water. Detail description of these components along with its associated instrumentation is given below.

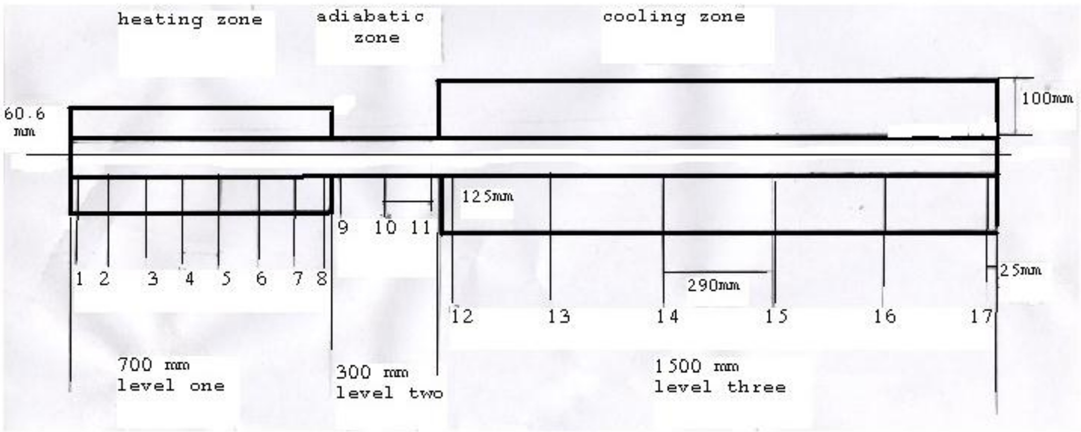


Fig.A: Experimental Rig of the H.P (Showing Thermocouple Locations)

The Evaporator:

The evaporator has a length of (0.7m) and is heated electrically by eight co-axial type heaters of (750 watt) each. The heaters were fixed to the outside surface of the evaporator at (45) degree intervals. The length of each is equal to the length of the evaporator. In order to reduce heat losses to the outside and to insure uniform heat flux along the evaporator, a heat shield in the form of a cast iron pipe of length (0.7m) and thickness (0.003m), was installed concentrically around the evaporator pipe. The heat shield is then wrapped with (0.03m) of glass wool insulator to further reduce the heat losses. In order to measure the average temperature of the evaporator, nine thermocouples were imbedded on the outside surface of the evaporator in (0.002m) deep groves. These groves were filled with lead-zinc alloy to insure good thermal contact. Three thermocouples were attached to the outside surface of the insulator to get the average temperature of that surface. All thermocouples were of (0.2mm)-asbestos sheathed Alumel-Chromel (type K).

The Condenser:

The condenser has a length of (1.5m) and was cooled by longitudinal fins that reject heat with the environment through natural convection. The fins were welded to the out side surface of the condenser. The length of each is equal to the length of the condenser with width (0.1m) and thickness (0.002m). In order to measure the average temperature of the condenser; six thermocouples were imbedded on the outside surface of the condenser. **Table.1** shows the dimensions of the fin.

The Adiabatic Zone:

The segment of the rig between the finned condenser and the evaporator is normally referred to as the adiabatic zone. The temperature profile is monitored by three thermocouples type K. **Table.2** shows the heat pipe specifications.

Table .1 fins Specifications

1	length	L=1.5 m
2	width	l=0.1 m
3	thickness	t_{fin}=0.002 m
4	numbers of fins	8

Table .2 Heat pipe specification

1	Length of evaporator zone	L_e = 700mm
2	Length of adiabatic zone	L_a = 300mm
3	Length of condenser zone	L_c=1500mm
4	Inner diameter of the pipe	di= 26.6mm
5	Outer diameter of the pipe	do = 33mm
6	Thermal conductivity of the pipe	kiron = 73
7	Thermal conductivity of the insulator	kins =.0206

Purging and Filling the apparatus:

The heat pipe is filled with water through tube A in **Fig.B** to a given level, say to the top of the evaporator or to (-5) cm from the top. Valves 1and 2 are now open. Closing valve 2 and turning two heaters on, the water inside the evaporator is allowed to boil at atmospheric pressure and the vapor leave the system through valve 1 and will carry

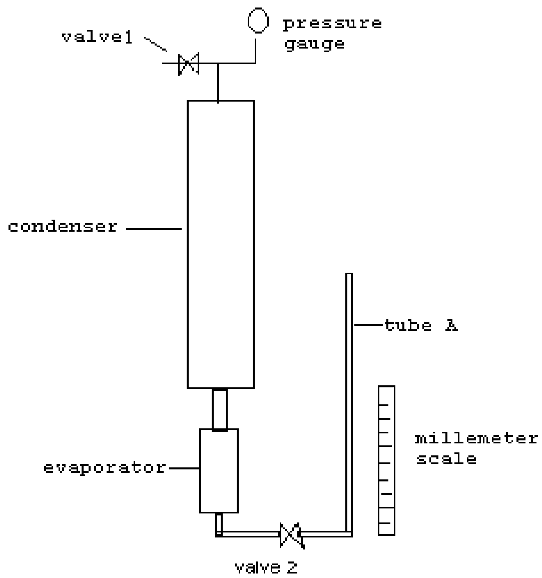


Fig.B: Schematic Diagrams Showing Purging and Filling the H.P with Water with it the entrapped air. The process of purging the system out of air is continued for one hour. Then valve 2 is opened and distilled water is added to bring the water to the required level then valve 2 is closed. It is to be noted that valve 1 and 2 are of the ball type.

THEORETICAL MODEL

A theoretical steady state model that describes the performance of a heat pipe (H.P) under a variety of operating conditions is developed below. The heat pipe has a diameter of (d_i=0.0266m,d_o=0.033m) and is (2.5 m) long with an evaporator that is driven by a constant heat flux of length (0.7m) and longitudinally finned condenser that reject heat with the environment through natural conviction. The length of the condenser is (1.5m). Between the evaporator and condenser; there is an adiabatic zone (i.e. insulated zone) of length (0.3m). Full description of the heat pipe system with the instrumentations was given earlier. The variable operating conditions include different input heat flux, fluid charge (water) and inclination of the heat pipe with the vertical.

The net heat input (Q) to the evaporator is given by:

$$Q = Q_t - Q_l \qquad (1)$$

Where Q_t is given by:

$$Q_t = V*I \qquad (2)$$

$$Q_l = \frac{2\pi K_{ins.} \; Le}{\ln(ro/ri)} \Delta t_{ins}$$

In steady state operation, Q is balanced exactly by the heat rejected by the condenser Q_c. Thus:

$$Q = Q_c = VI - \frac{2\pi Kins. \; Le}{\ln(ro/ri)} \Delta tins \qquad (3)$$

The energy transfer from condenser to ambient is given by:

$$Qc = h_c A_c (t_c - t_\infty) \qquad (4)$$

Where Qc = heat flow through the condenser and Ac= the effective surface area of the finned condenser $A_c = \pi d_o L_c E \eta_s$. Here **E** is the area enhancement factor, **(Dunn and Reay 1976)**:

$$E = \frac{\text{Area of condenser surface with fins}}{\text{Area of condenser surface without fins}}$$

and η_s is the surface efficiency

$$\eta_s = 1 - \frac{A_f}{A_{oc}} (1 - \eta_F) \qquad (5)$$

Where η_f is the fins efficiency

$$\eta_f = \frac{\tanh(mL)}{}$$

(mL)

Where:

$$mL = l^{3/2}\sqrt{\frac{2\,h_o^*}{K_f\, l\, t_{fin}}}$$

The heat rejected by the condenser to the ambient, may by written as:

$$Q_c = A_c h_c (t_c - t_\infty) + A_c \sigma (t_c^4 - t_\infty^4) \qquad (6)$$

Eq. (5) may be put in a more convenient form by defining an effective heat transfer coefficient h_o^* given by **(Zeina 2005)**:

$$h_o^* = (h_c + \sigma(t_c^3 + t_c^2 t_\infty + t_c t_\infty^2 + t_\infty^3))$$

Thus **eq. (5)** may read now:

$$Q_c = A_c h_c^* (t_c - t_\infty) \qquad (7)$$

And **eq.(3)** may simplify to:

$$A_c h_c^*(t_c - t_\infty) = VI - \frac{2\pi\, Kins.\ \ Le}{\ln(r_o/r_i)} \Delta t_{ins} \qquad (8)$$

The natural convection heat transfer coefficient hc may be calculated from the following correlation, El-Wakil M. M., (1988), which is suitable be for constant heat flux and for the range $2*10^{13} < Gr^* Pr < 10^{16}$, Incorporeal, F.P., and Dewitt D.P., (1990).

$$Nu = \frac{h_c L_c}{k} = (0.17)\,(Gr^* pr)^{1/4} \qquad (9)$$

All physical properties for air and are evaluated at film temperature = **($t_c + t_\infty$)/2**
The pressure **p**, of the heat pipe may now be evaluated by back calculating t_g (refer to **Fig.C**)and using the steam tables to find p=p(t_g).

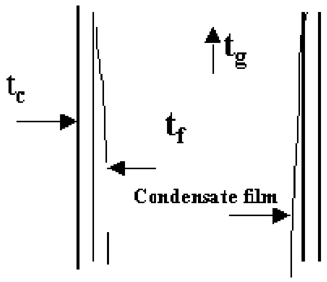


Fig.C: Condenser section showing the definitions of t_c , t_f and t_g.

Using the concept of thermal resistance across the film condensation ,condensate film and wall of the H.P, one easily evaluates t_g as:

$$t_g = t_f + \frac{Q_c}{} \qquad (10)$$

$A_i\, h_1$

Where:

$$t_f = t_c + (Q_c / A_i)\left[\left(\frac{r_i}{k_s}\ln\frac{r_o}{r_i} + \frac{r_i}{k_w}\ln\frac{r_i+\delta_w}{r_i}\right)\right] \qquad (11)$$

δ_w= average condensate film thickness, **(Incorporeal and Dewitt 1990).**

$$\delta_w = \frac{4}{5}\left[\frac{4\nu_w\, k\,(t_g - t_f)}{g\, Lc^4\, h_{fg}\,(\rho_w - \rho_g)}\right]^{1/4} Lc^{5/4}$$

h_1=the average film condensation heat transfer coefficient and is given by **(Holman 1997):**

$$h_1 = 0.943\left[\frac{k_f^3\, g\, h_{fg}\,(\rho_w - \rho_g)}{\mu_w\, Lc\,(t_g - t_i)}\right]^{1/4}$$

All physical properties for condensate (μ_w, ρ_w, ν_w, h_{fg}) are evaluated at the average condensate temperature $(t_i + t_g)/2$

Where t_i may readily be written as:

$$t_i = t_c + (Q_c / A_i)\left(\frac{r_i}{k_s}\ln\frac{r_i+\delta_s}{r_i}\right)$$

Equation (9) may now be solved iteratively to determine t_g and hence the pressure of the H.P.

The performance of the H.P may be conveniently characterized by the following efficiency η and thermal resistance, R of the H.P:

$$\eta = \frac{Q_c}{Q_t} \qquad (12)$$

$$R = \frac{t_e - t_c}{q_c''} = \frac{(t_e - t_c)}{Q_c} Ai \qquad (13)$$

RESULTS AND DISCUSSIONS

Experimental results that relates the basic parameters of the H.P (average temperature of condensers surface, average temperature of the evaporator surface and system pressure) under various operating conditions (inventory, net power input to the evaporator and angle of inclination of the H.P with the vertical) are presented and discussed along with performance indices represented by the thermal resistance of the heat pipe and its efficiency. A comparison of both the experimental pressure and condensers temperature with theoretical predictions were presented and discussed.

Transient Behavior of the Heat Pipe:

Fig.1 and **Fig.2** show the transient behavior of a typical run for the evaporator and condenser respectively. The familiar rise to the eventual steady state temperatures is shown. The rise time is around 40-50 minutes. Thus in 2-2.5 hours, the steady state of the system is reached, and hence the steady state measurements may be recorded.

Steady State Behavior of the H.P:

According to the transient time of the H.P discussed previously, all steady state temperatures and pressures were recorded after two hours in half-hour interval and steady state is assume to be reached when successive readings were within 1c$^{\circ}$.

Heat Pipe Behavior:

The steady state behavior of the H.P components were taken and include the temperature profiles of the condenser, adiabatic zone and the evaporator respectively. For each component, the average value was computed. The behavior of the H.P was investigated experimentally in relation to: water inventory, net power supplied to the evaporator and angle of inclination of the H.P with the vertical. The results and relevant discussion are given below:

Heat Pipe Behavior with Water Inventory:

When the evaporator is filled to the top, the inventory is said to be 100%. **Fig.3** shows the average temperature of the evaporator (t_e) in relation to percent inventory for different net powers. It is noted that (t_e) is approximately constant (the maximum scatter in the experimental points is ≤ 8% and the constancy is within ± 4%. This behavior is to be expected since, every thing being equal, as the inventory is reduced, the condensation process continues to be of the film type, while the evaporation process changes from totally nucleate boiling to a mixture of nucleate boiling in the submerged depth plus sub-cooled and nucleate boiling in the non sub-cooled boiling occurs as the condensate film descend through the top of the evaporator at a temperature which is lower than saturation. The change in the boiling regime as inventory is reduced does not change the evaporation process significantly since both sub cooled boiling is just as effective as a heat transfer mechanism as that of nucleate boiling. Thus as inventory is reduced, the mechanisms remain essentially the same and subsequently, the temperature (t_c) remains constant. The scatter of experimental points reflects the fact that each experimental point is the result of a given net power and ambient environmental temperature that is slightly different from the next point. These differences, which constitute an inadequacy of the experimental rig, are indicated on **Fig.3** and are unfortunately unavoidable. **Fig.4** shows the average condenser temperature in relation to inventory for different net power to the evaporator. Again, one notes the constancy of the average condensers temperature as the inventory is changed (the mean standard deviation of the points is ≤ 6%). The deviation from constant (t_c) is with in ≤ 5%). The same explanation given earlier for the constancy of (t_e) holds here.

Fig.5 shows the average temperature for the adiabatic zone as a function of inventory and for different net evaporator's power.

Fig.6 shows system pressure for the H.P as a function of inventory for different net evaporator's power. The pressure of the H.P is directly related to (t_c) and the heat flux, q_c'' leaving through the condenser, thus for constant (t_c) and q_c'', the H.P pressure should have a given value. **Fig.6** shows that the pressure is not constant with inventory, as one would expect if (t_c) and q_c'' were kept constant. But as explained earlier and as indicated on the captions in **Fig.6** both, the net power q_c and (t_∞) vary slightly from one inventory point to the next. As indicated in chapter four (t_g)(and hence the system pressure) is related to (δ_w) and (t_∞). But (δ_w) is directly related to q_c and therefore the shape of the pressure curves indicated in **Fig.6** reflects the combined variations in q_c and (t_∞) as indicated in captions.

Fig.7 shows the thermal efficiency of transferring power through the H.P as a function of inventory, for different net power to the evaporator. Due to slight variation in main voltage, the input power during these measurements fluctuated by few percent and the ambient temperature change by $\pm 5C^\circ$. Since both of these parameters affect the actual heat loss from the evaporator, and subsequently they affect the surface efficiency as defined in **eq. (10).** In split of this fluctuation, the efficiency is almost constant with inventory at all power levels. The maximum standard deviation is $\pm$ 3.8% while the best-fit straight lines have slopes ≤ 0.06.

It is to be noted that the efficiency is very high at all powers but being the highest at 99% for an input power 2.25kW. The efficiency is still very high at 96% for an input power of 3kW.

Behavior with Angle of Inclination:

Before attempting to present and discuss the results of the H.P, it is instructive to describe what happens inside the H.P. As the H.P is tilted away from the vertical, the condenser area available for film condensation decreases while that available to drop wise condensation increases.

Since heat transfer coefficient in drop wise condensation is nearly four times the corresponding film wise condensation. And that there is temperature drop across film condensate, while no such drop exists over the region where drop wise condensation is occurring, then it follows, with every things being equal, that condensation occurs at lower temperature as the angle of inclination is increased. This indeed is the case as shown in **Fig.6.** It is known that the condenser controls the pressure of the H.P and since the dominant heat transfer occurs in the region of drop wise condensation where there is no temperature drop across a film then we expect that as the angle of tilt charges. One sees no charge in the surface temperature in that region and this temperature controls the pressure of the H.P. Thus one would expect that with every things being equal, the pressure of the H.P remains constant with tilt angle. This deduction is proved correct in **Fig.9** where the pressure of the heat pipe is shown to be content.

In the evaporator region the picture is more complicated. As the angle of inclination is increased, boiling changes from nucleate boil in the liquid film region plus in the submerged region at angle zero to nucleate boiling in the wetted region plus film boiling in the non wetted or dry region. As the angle of inclination is further increased (i.e. as the heat pipe approaches the horizontal position) then plug flow boiling dominates the entire region of the evaporator since plug flow takes over. Thus one would expect the rate transfer mechanism to decrease initially, then to recover and start to increase. Consequently, the evaporator's temperature (t_g) is expected to rise and eventually to reach a maximum, then declines. This behavior is observed in the experimental H.P as shown in **Fig.10.**

The efficiency of the H.P as a function of inclination angles is shown in **Fig.11.** The efficiency of transferring power actually slightly improves initially (from 96% at 30°angles) then it declines rapidly to below (97% at 60°angles). Thus the heat pipe is very efficient at an inclination angle between (0-30°).

Behavior with Heat Flux (or Net Power):

Fig.12 shows the thermal resistance for the H.P as a function of the heat flux through the condenser at 100% inventory. It is shown that the value of thermal resistance is reasonably small up to a condenser heat flux of (1.158 kW/m^2) and the resistance then begins to increase sharply with heat flux due to changing the boiling mechanism from nucleate to partial film which is accompanied by sharp increase in (te) and subsequent increase in thermal resistance. Thus a conservative upper limit for normal operation of the heat pipe is at (1.2 kW/m^2).

From **Fig.12** one may consider the last point, which corresponds to heat flux (1.48 kW/m^2) to represent a point very close to the critical flux since further increase in heat flux resulted in heaters burn up. These values of thermal resistances could by drastically reduced (by ten folds for example) if the electric heaters were welded directly to the surface of the boiler instead of being attached, by metal band as in the current experiment. Of course in actual practice gas or kerosene fuels will heat these residential H.P, and then the actual thermal resistance will be many folds lower than obtained in the present experiment.

Fig.13 shows the system pressure (which is solely controlled by the condenser) as a function of heat flux through the condenser at (100%) inventory. The pressure increases with heat flux. We note that the slope of the curve also increases with the heat flux. A reasonable pressure of 1.2 to1.4 Mpa may be chosen as working pressure for a domestic H.P.

In summary, the design heat flux through the condenser should be ($\leq$ 1.158 kW/m^2).

Comparison with the Theoretical Model:

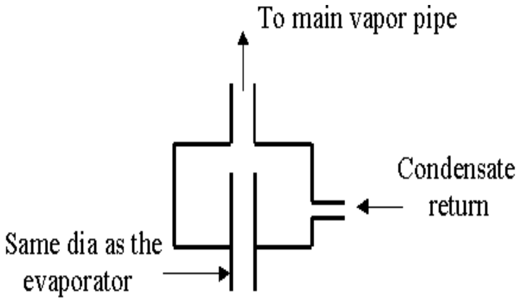
Fig.14 and **Fig.15** depicts the comparison between the actual (tc) measurements and the corresponding theoretical counter part as a function of inventory percent. It is to be noted that the complicated shape traced by the theoretical model is only a reflection of the combined variation in net power (rejected by the condenser) and (t_{∞}). These variations as mentioned earlier, are beyond the experimenter's control. The maximum standard deviation is ($\leq \pm$ 16%). The model tends more often to over estimate the value of (t_c) but non the less the theoretical prediction may by regarded as adequate for design purposes.

Fig.16 shows the comparison between experimental system pressure and the corresponding theoretical predictions as a function of inventory. It is observed that the theoretical model always underestimate system pressure. Fortunately, the maximum standard deviation is 21% and is not considered very large.

Design of a Residential H.P:

The heat flux of (1.1589 kW/m^2) with the corresponding system pressure of 1500kpa (at 100 % inventory) will be taken as a basis for any design, such as this one. Thus all piping and units should withstand 1.5x design pressure or 2250kpa. No instrumentation is necessary. The closed system should be purged after installation, then filled with distilled water to 50% inventory and then sealed. Actual implementation of the H.P to heat a residence (composes, say of three spaces with each space requiring 4kW heating) necessitate the following components:

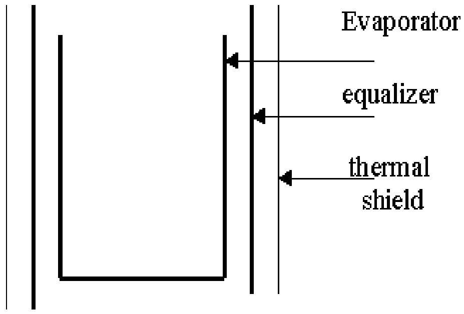
1. *Collector and film flow generator*: This unit which connects to the



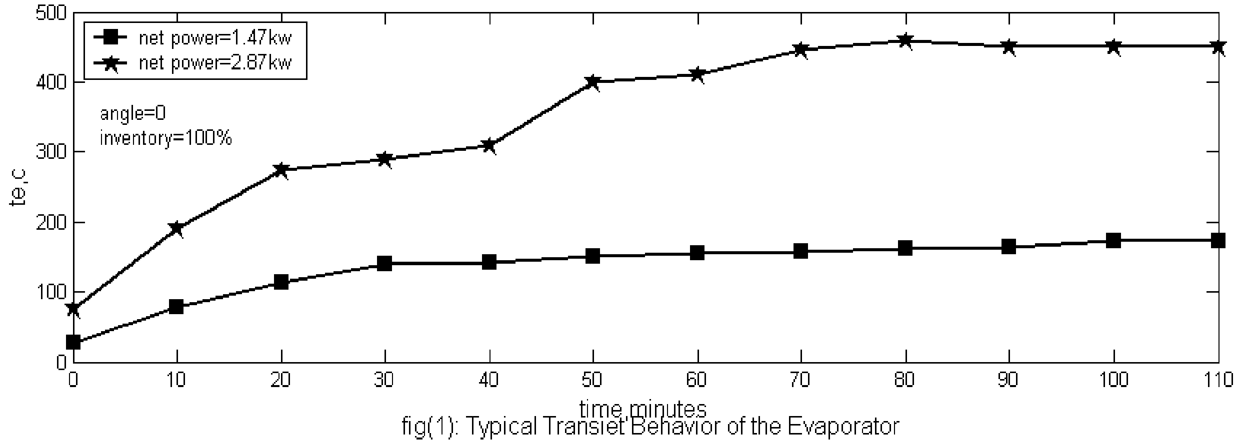
Sketch of the Collector and Film Flow Generator

evaporator simply receives the condensate as a stream and convert it to film flow down the evaporator. This unit should be welded on top of the evaporator and is shown below schematically for one such design.

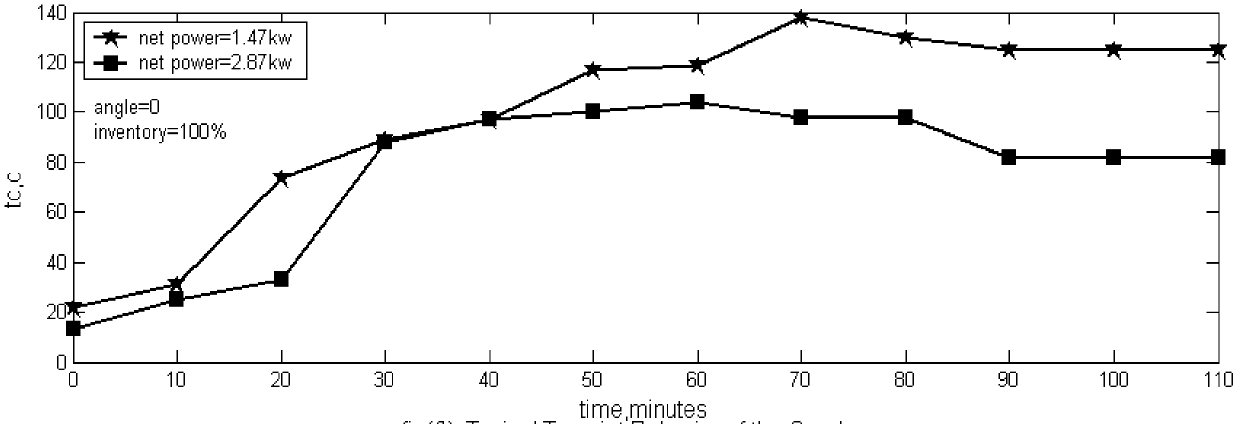
2. *Vapor supply pipe:* This pipe (same diameter as the evaporator pipe) is attached to the top of the collector and supply steam to all the condensers, one for each space. Since in our example we have three condensers, then the branched pipes (pipe to each condenser) have a pipe diameter $1/\sqrt{3}$ =0.577 times the diameter of the main branch. The main branch should be designed for 12 kW and since the experimental H.P (with di of 26.6mm) operated normally up to 1.2 kW/m^2 (i.e. at 3kW total power), then it follows that the main branch should have a di of $26.6\sqrt{12/3}$=53.2mm. one important consideration is that this pipe should have a slope that is ≥ 0.2 to facilitate the return of any condensate to the evaporator.
3. *The condenser*: This is a standard finned tube condenser with inlet manifold at the top and condensate out let at the bottom. It is sized on 1.12 kW/m^2. A twenty five percent increase in area is added for conservative design. This gives a total of $(4/1.12)*1.25=6\ m^2$ as the required surface area for each condenser. It should be noted here that all exposed piping inside the residence should be considered part of the condenser unit.
4. *Condense pipe*: The main condensate pipe receives the condensate from the individual condensers. The diameter of the main pipe should be sized to give water velocity of 1m/minute. This requires a diameter of ≥21.7mm. The branch pipes required have a diameter of $21.7\sqrt{1/3}$ mm. All condensate pipes should have a slope ≥ 0.2.
5. *The evaporator*: This is located outside the residence, and has the main branch should be designed for 12kW and since the experimental H.P (with di of 26.6mm) operated normally up to 1.2 $kW/\ m^2$.(i.e. at 3kW total power), then it follows that the main branch should have a di of $26.6\sqrt{12/3}$= 53.2mm. For gas or kerosene heating, a tube of approximate diameter that is greater than the burner should enclose the boiler. This tube acts as a flux equalizer: to distribute the input heat uniformly along the cylindrical evaporator. A thermal shield is required to enclose the flux equalizer in order to reduce heat losses and improve the thermal efficiency. A sketch is shown below.



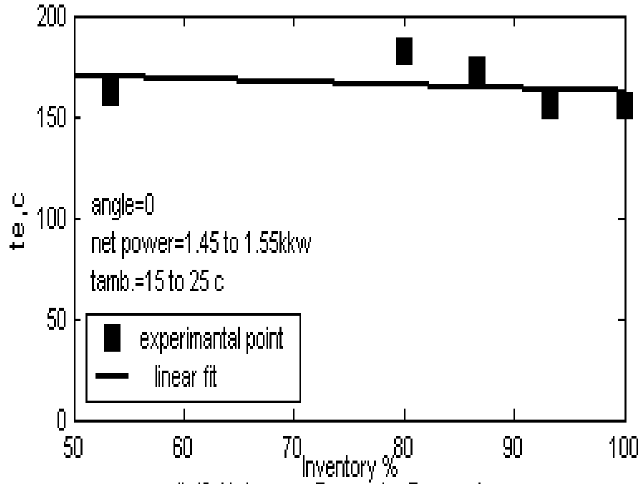
6. The parts of the pipes that are located out side the residence should be insulated.
7. A pressure safety device, such ruptured disk should be attached, very close to the evaporator and is placed out side the building. It is vented to the outdoors. The setting for the ruptured disk should be 1.5 system pressures (~ 1.8mpa).



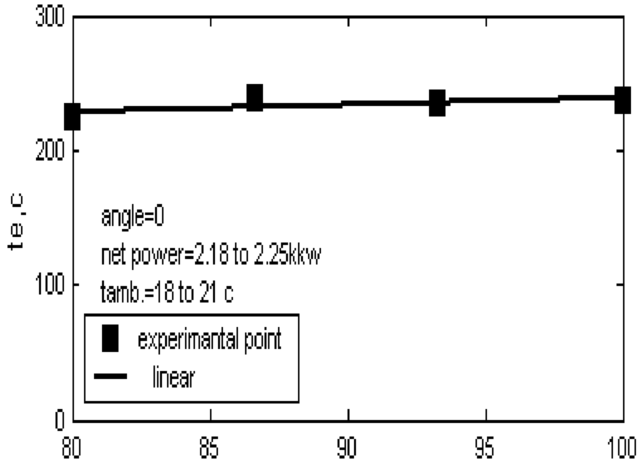
fig(1): Typical Transiet Behavior of the Evaporator



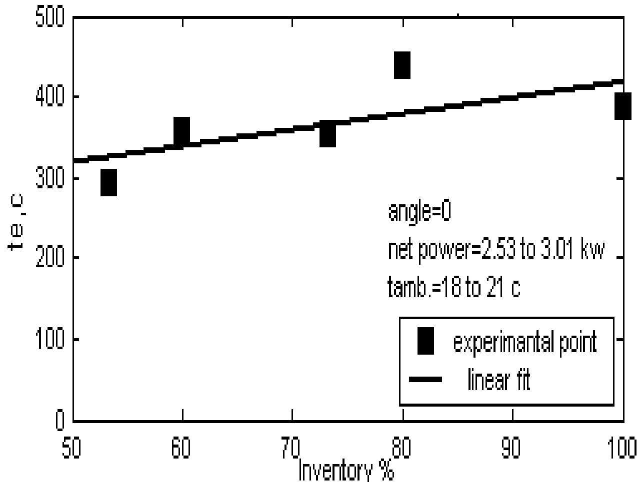
fig(2): Typical Transiet Behavior of the Condenser



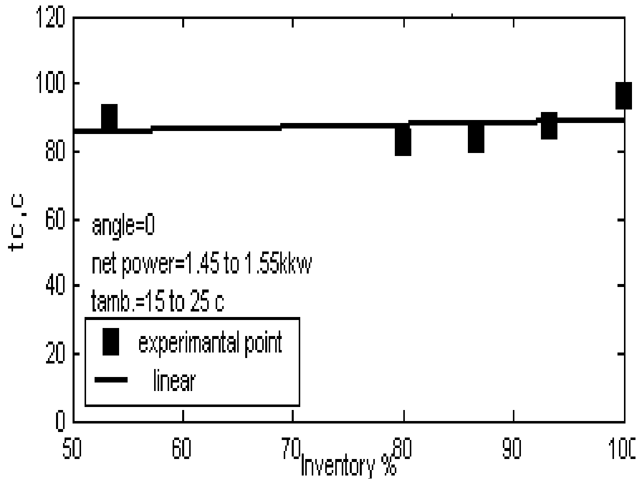
fig(3:A):Average Evaporator Temperature With Different Water Inventory



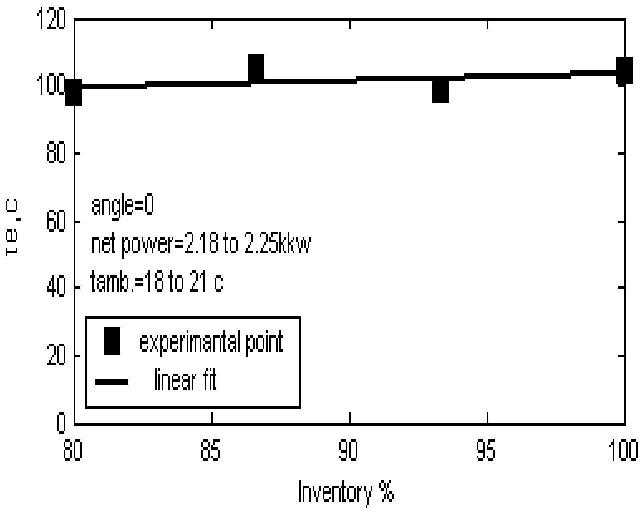
fig(3:B):Average Evaporator Temperature With Different Water Inventory



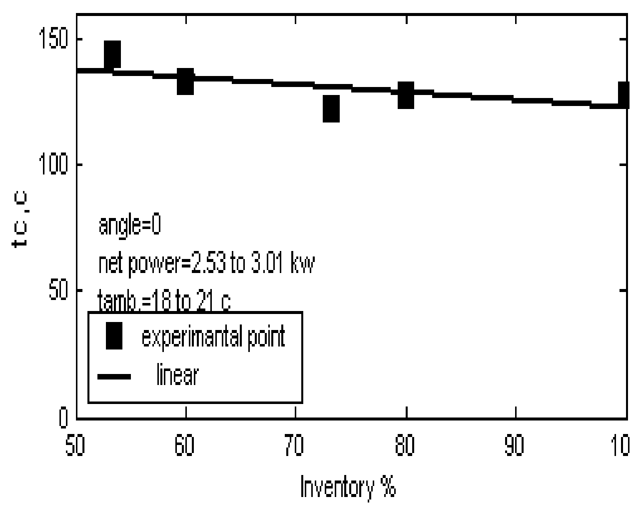
fig(3:C):Average Evaporator Temperature With Different Water Inventory



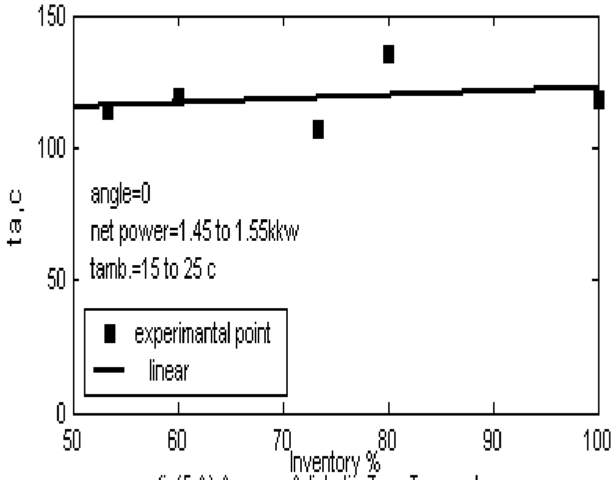
fig(4:A):Average Condenser Temperature With Different Water Inventory



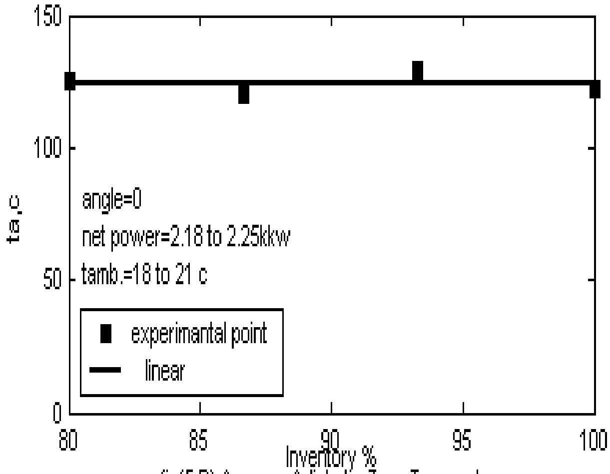
fig($:B):Average Condenser Temperature With Different Water Inventory



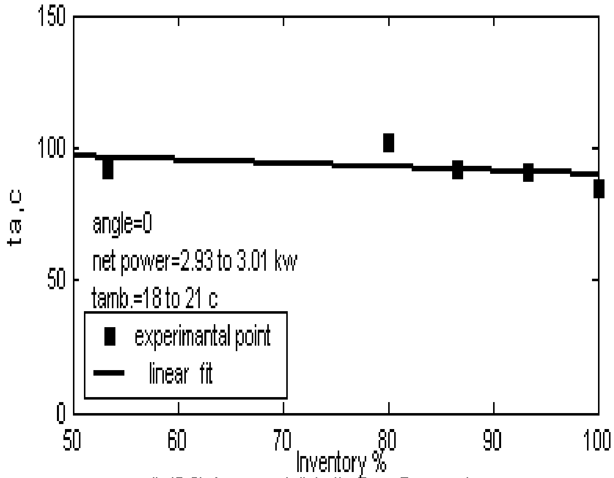
fig(4:C):Average Condenser Temperature With Different Water Inventory



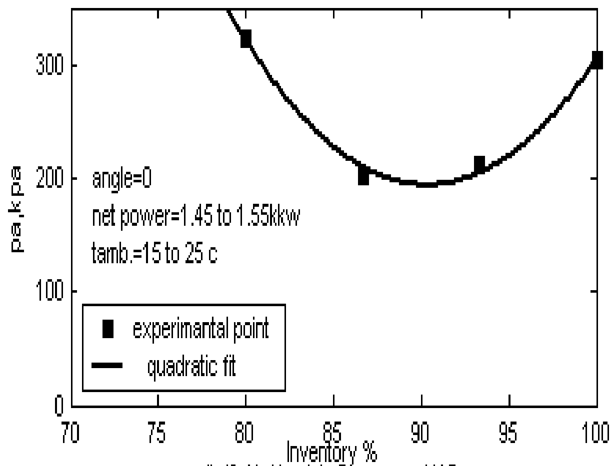
fig(5:A):Average Adiabatic Zone Temperature
With Different Water Inventory



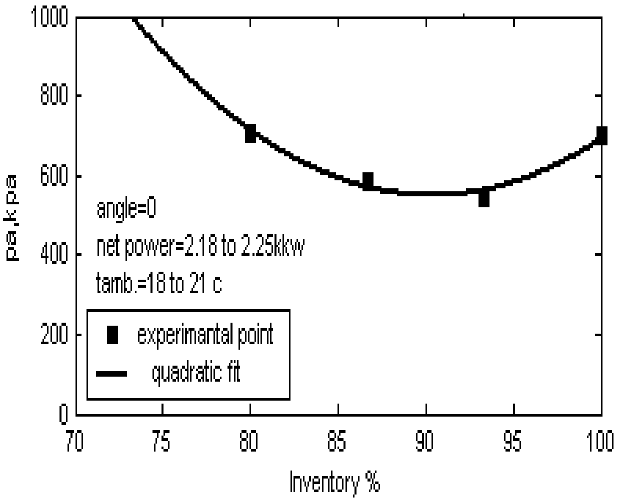
fig(5:B):Average Adiabatic Zone Temperature
With Different Water Inventory



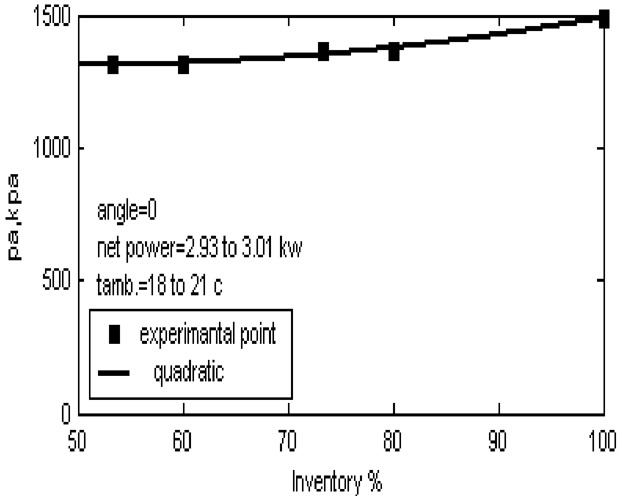
fig(5:C):Average Adiabatic Zone Temperature
With Different Water Inventory



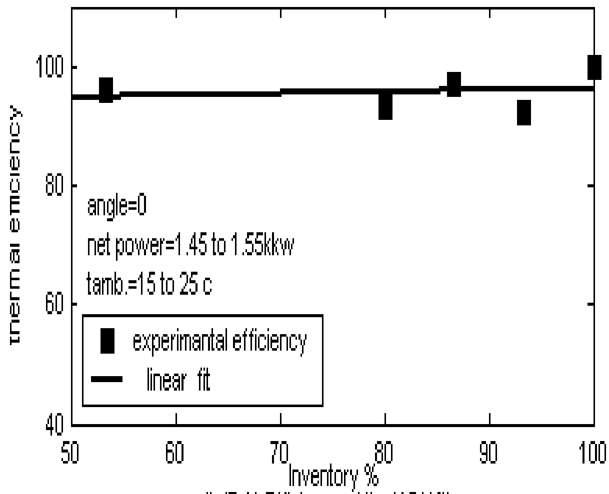
fig(6:A):Absolute Pressure of H.P
With Different Water Inventory



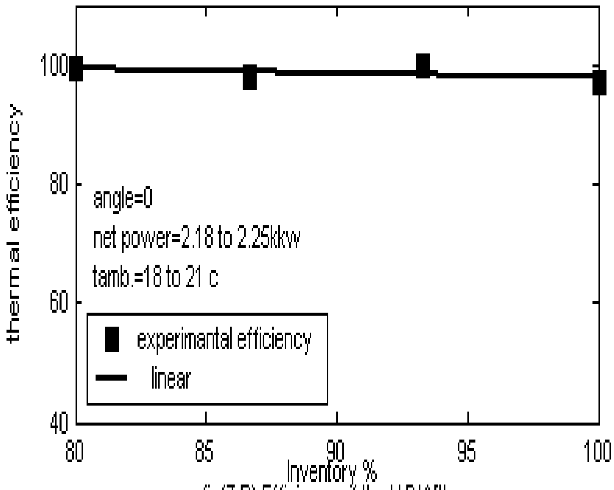
fig(6:B):Absolute Pressure of H.P
With Different Water Inventory



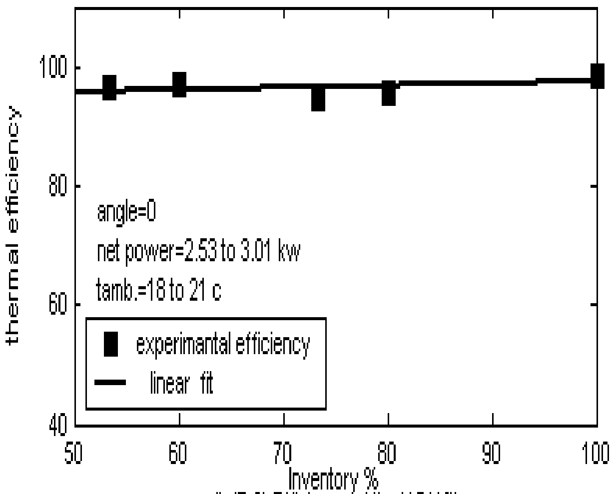
fig(6:C):Absolute Pressure of H.P
With Different Water Inventory



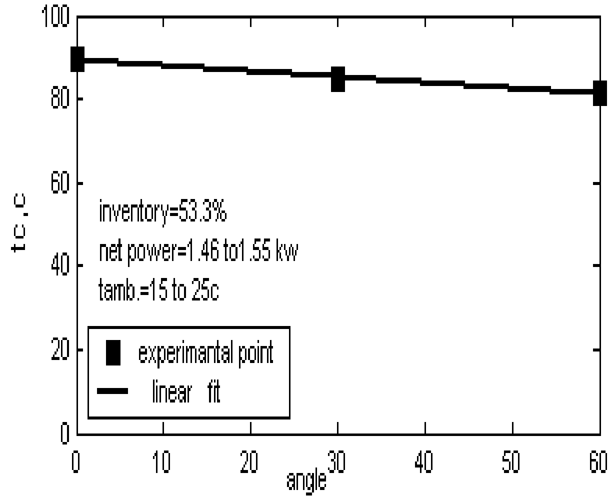
fig(7:A):Efficiency of the H.P With Different Water Inventory



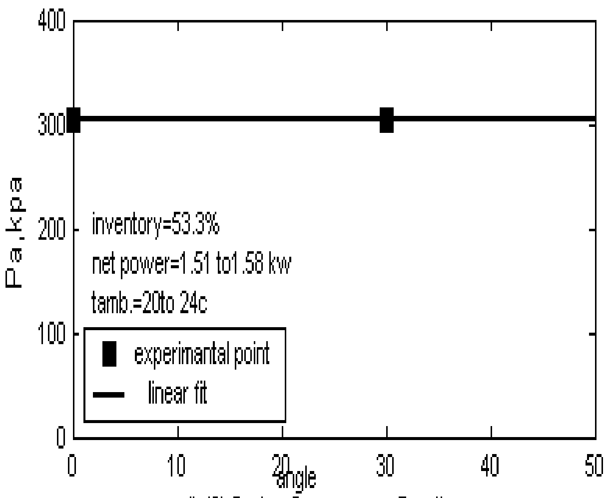
fig(7:B):Efficiency of the H.P With



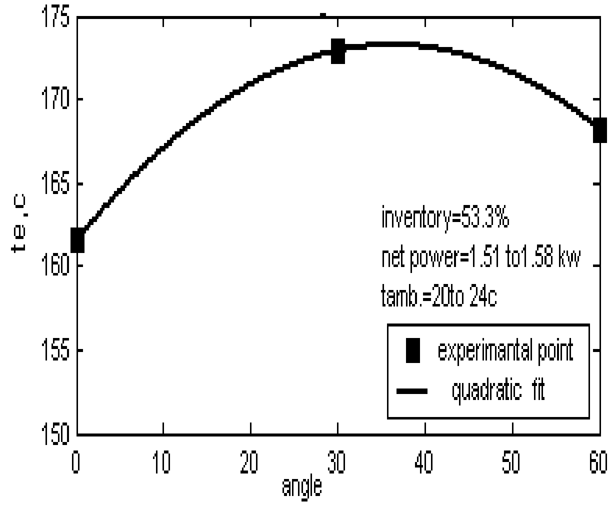
fig(7:C):Efficiency of the H.P With Different Water Inventory



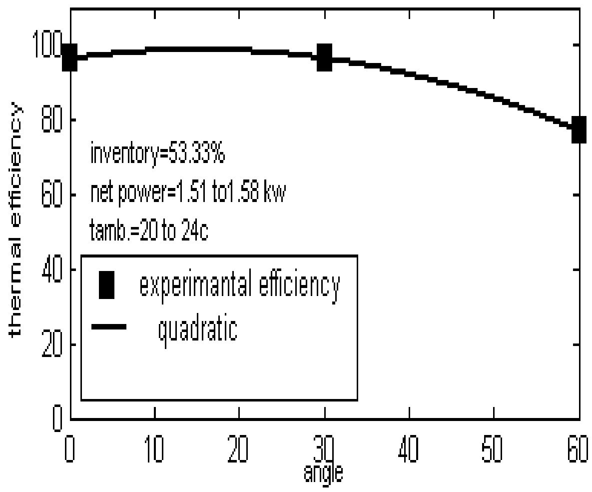
fig(8):Average Condenser Temperature For Different Angle of Inclination



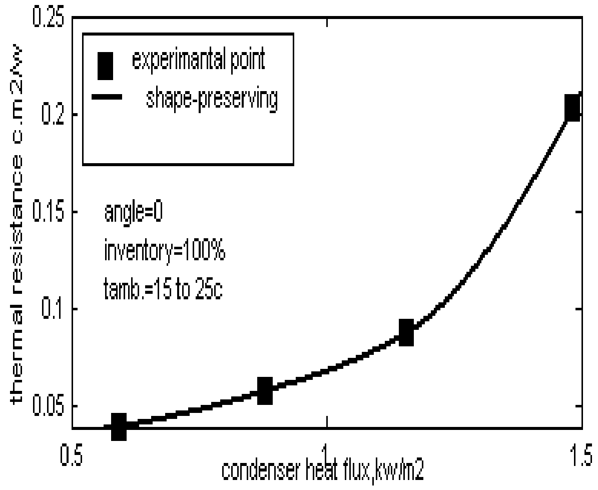
fig(9):System Pressure as aFunction of Inclination Angle



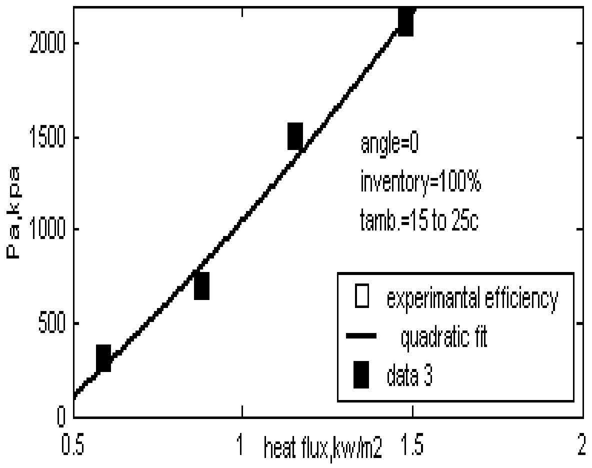
fig(10):Average Evaporator Temperature For Different Angle of Inclination



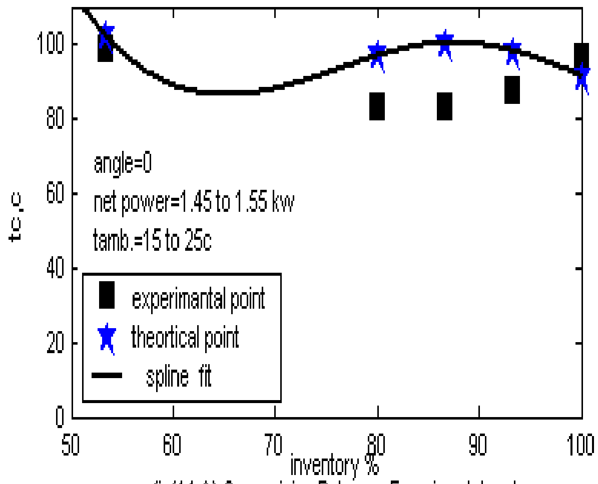
fig(11):Efficiency Relation to Angle of Inclination



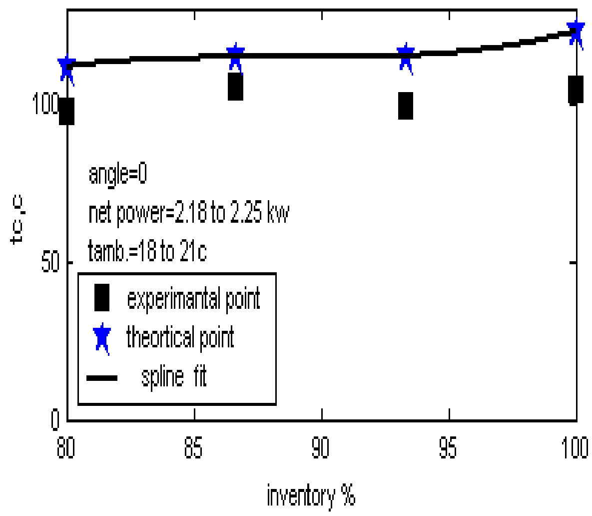
fig(12):Thermal Resistance of the H.P
With Condenser Heat Flux



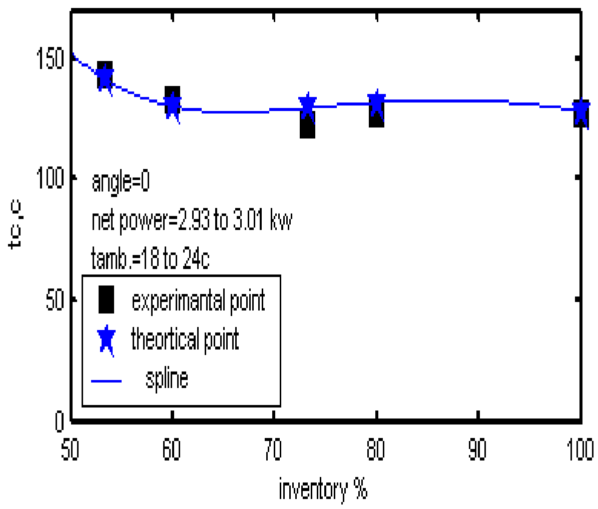
fig(13):Efficiency of the H.P With
Different Water Inventory



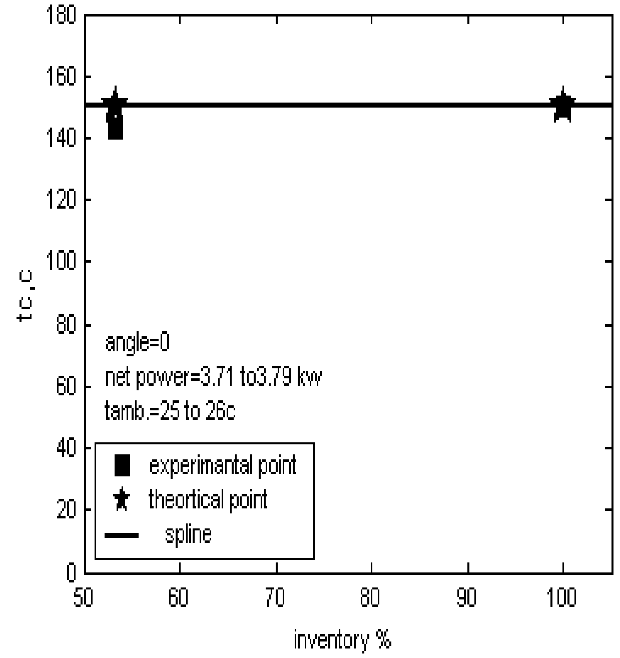
fig(14:A):Comparision Between Experimantal and
Theortical Average Condenser Temperature



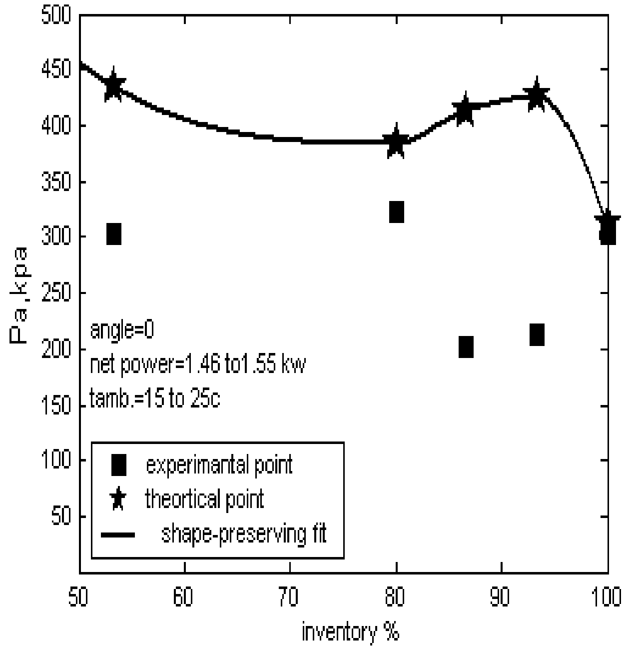
fig(14:B):Comparision Between Experimantal and
Theortical Average Condenser Temperature



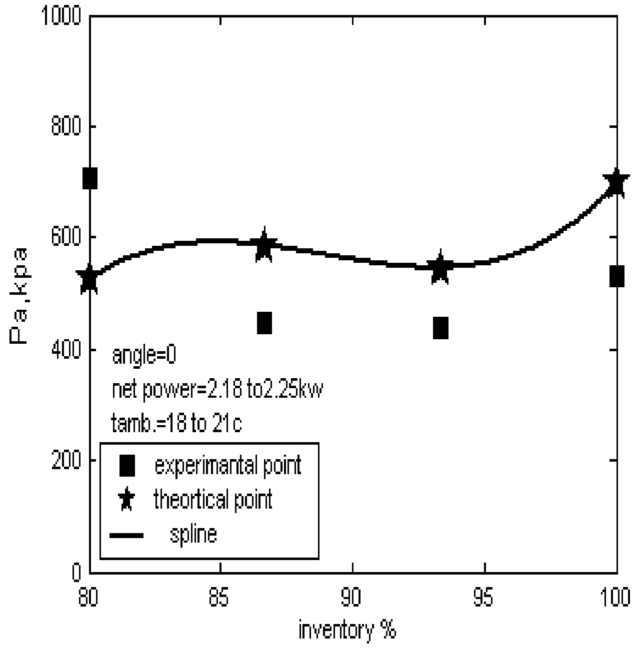
fig(15):Comparision Between Experimantal and
Theortical Average Condenser Temperature



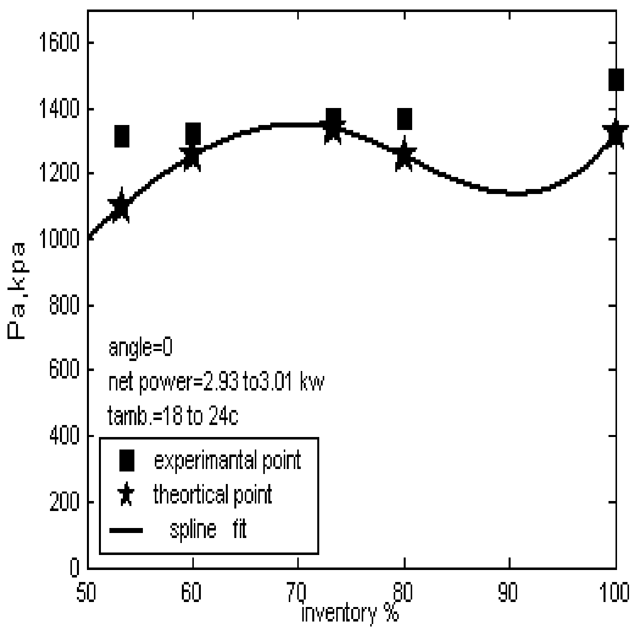
fig(15:B):Comparision Between Experimantal and Theortical Average Condenser Temperature



fig(16:A):Comparision Between Experimantal and Theortical Absolute Pressure



fig(16:B):Comparision Between Experimantal and Theortical Absolute Pressure



fig(16:C):Comparision Between Experimantal and Theortical Absolute Pressure

CONCLUSIONS:

1. For a given (vertically oriented) H.P design, the important operating parameters of (p, t_c ,q_c'') are found to be independent of total inventory. Practical consideration, however, favor an inventory close to 50 %. Since lower inventory may cause unwarranted high evaporator's temperature during start up while higher inventory may reduce the part of the evaporator where boiling film flow takes place.
2. A conservative design flux for the condenser unit (in vertically oriented H.P) is 1.2 kW/m^2. This flux is nearly 30 % lower than the critical flux for the H.P. This design flux, which controls the system pressure, maintains a system pressure of 1500 kpa.
3. For an inclined H.P, the performance (as reflected in efficiency) is slightly improved with angle of inclination equal to 30°. The pressure of the system is totally independent of inclination angle.
4. The theoretical model developed to predict the condensers temperature and system pressure for a given net power input is accurate to ± 16% in predicting t_c and with ±21% in predicting system pressure. This accuracy is adequate for design purposes.

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NOMENCLATURE:

Symbol	Meaning	Unit
A	area	m^2
C_P	specific heat capacity	kJ/kg.k
d	pipe diameter	m
E	enhancement factor	---------
g	gravitation acceleration=9.8	m/s^2
h	heat transfer coefficient	$W/m^{2} \cdot {}^{0}c$
K	thermal conductivity	$W/m \cdot {}^{0}c$
l	Fin length	m
L	pipe length	m
Nu	Nusselt number=hd/k	---------
p	power	W
P	pressure	kPa
q"	the heat flux to the evaporator	$kJ/m^2.s$
Q	heat flow	w
r	radius of pipe	m
s	thickness of fins	m
t	temperature	${}^{0}c$
V	voltage	volt
I	current	ampere
Pr	prandtl number = $\mu\ C_P /k$	-------
Gr	Grashof number = $g\ \beta\ (t_s - t_\infty) L_c^3 /\nu^2$	--------
Gr^*	modified Grashof number = Nu Gr	--------
h_{fg}	Heat of vaporization	J/kg

Greek Symbol:

Symbol	Meaning	Unit
β	Volume expansion coefficient for steam	1/k
μ	dynamic viscosity	$N.s/m^2$
ρ	density	kg/m^3
ν	kinematics viscosity	m^2/s
δ	Stefan-Boltzman constant $=5.669*10^{-8}$	$W/m^2 .k^4$
∞	ambient	--------
θ	angle	degree
Δ	change (t_2-t_1)	---------
η	efficiency	---------
φ	Heat flux	W
δ	average liquid film thickness	---------

Subscripts:

Symbol	Meaning
a	adiabatic zone
c	condenser
e	evaporator
f	fluid
fin	Fins surface area
g	vapor
ins	insulator
i	inner
l	losses

o	outer
s	cast iron
t	total
uf	un finned
w	water
x	local