ISTANBUL TECHNICAL UNIVERSITY INSTITUTE OF SCIENCE AND TECHNOLOGY

THEORETICAL AND EXPERIMENTAL ANALYSIS OF THE DOUBLE WALLED CONCENTRIC HEAT PIPE

M.S.THESIS

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PREFACE

During the last two decades there has appeared a large number of investigations about the heat pipe since the growing in the deficiency of energy has been accelerated at this time.

This thesis is about the double-walled concentric heat pipe which has a difference from conventional heat pipes in the geometrical aspect. It includes developments, theory, design, manufacture of heat pipes and experimentation of a constructed double-walled concentric heat pipe.

I am indebted to Doç.Dr.A.Feridun Özgüç for his invaluable suggestions and contributions. I should also like to thank to Miss. Emine Birinci who typed the manuscript.

NOMENCLATURE

P : Pressure (N/m^2)

σ : Surface Tension (N/m^2)

R : Radius of Curvature (m) or Thermal Resistance (°C/W)

θ : Contact Angle (Degree)

ρ : Density (kg/m^3)

μ : Dynamic Viscosity (kg/m.s)

W : Liquid Velocity (m/s)

K : Permeability (m^2)

A : Area (m^2)

l : Length (m)

ε : Porosity

S : Surface Area (m^2)

Q : Heat Transfer (W)

N : Number of Layers or Grooves

L : Latent Heat (kj/kg)

$\dot{m}$: Mass Flowrate (kg/s)

h : Convection Coefficient (w/m^2 °C)

T : Temperature (°C)

q : Heat Flux (w/m^2)

k : Thermal Conductivity (w/m°C)

u : Axial Vapor Velocity (m/s)

V : Radial Vapor Velocity (m/s)

η : Efficiency

υ : Kinematic Viscosity (1/m)

r : Radius (m)

SUBSCRIPTS

c : Condenser or Capillary

e : Evaporator

a : Adiabatic

l : Liquid

v : Vapor

s : Solid

g : Gravity

i : Inner

o : Outer

i : Inlet or Inertial

e : Exit

w : Wick

r : Radial Coordinate

z : Axial Coordinate

CONTENT

SUMMARY

In this thesis, it has been studied about a heat pipe with a new configuration, namely "double-walled concentric heat pipe".

The double-walled concentric heat pipe consists of two concentric pipes of unequal diameter attached by means of end caps creating an annular vapor space.

It transfer more heat due to increasing in area and increasing in wicking area although its outer sizes are constant. So, it may be used in anywhere, especially for the cases where space is at a premium.

A double-walled concentric heat pipe with 400 mesh SST wick was manufactured and experienced in the laboratory. The results showed that the concentric heat pipe may have better performance than the standart type which was accepted so by applying no heat transfer in or out of the inner pipe of the constructed double-walled concentric heat pipe.

This thesis also includes the theory, design and manufacture of the heat pipes generally.

ÖZET

KONSANTRİK ISI BORUSUNUN TEORİK VE DENEYSEL ANALİZİ

20.yüzyılın sonlarına doğru herşeyin başı olan enerji problemini çözmek için çalışmaların yoğunlaştığı gözlenmektedir. Enerjinin bir yerden bir yere taşınmasında ayrıca önemli bir problemdir. Enerji konusunda geliştirilmiş çok basit bir yöntem bile çok önem kazanmaktadır. Bundan dolayı işlevi açısından bakarsak ısı borusununda değer verilmesi gereken bir yöntem olarak düşünülebilir. Ayrıca uzay çağının başlaması ve elektronik sanayinin gelişmesi bir ısı transferi cihazı olan ısı borusunun önemini artırmıştır. Öyleki, Groover ve arkadaşlarının 1964'teki "çok yüksek ısıl iletkenli bir yapı" adlı makalesinden sonra 2000'den fazla çalışma yapılmış ve 5 kadar kitap yayınlanmıştır bu konuda.

Isı borusunun beş esas işlevi vardır. Bunlar (1) çok yüksek ısıl iletkenliği, (2) Evaporator ve kondenser alanlarının orantısını değiştirerek ısı akısı dönüştürücüsü olarak çalışabilmesi, (3) sabit sıcaklıkta çalışması, (4) değişken iletkenlikte bir cihaz haline getirilebilmesi, (5) tek yönlu çalıştığından ısıl diod veya anahtarı olarak kullanılabilmesidir. Bunlardan tabiki en önemlisi ısıl iletkenliğin çok yüksek olmasıdır. Öyleki 150°C'de suyla çalışan bir ısı borusunun ısıl iletkenliği çok iyi bir iletken malzeme olan bakırın iletkenliğinin yüzlerce hatta binlerce katı olabilmektedir.

Bir ısı borusu, esas itibariyle bir akışkanın faz dönüşümü esnasında en yüksek enerji taşıma prensibinden yararlanır. İçinde belirli bir miktarda çalışma sıvısı bulunduran kapalı bir yapıdır. Akışkan, yapının bir tarafından ısıyı alır, buharlaşır ve hız kazanarak bir başka tarafına gider. Isıyı burada bırakır ve yoğunlaşır. Bu yoğuşmuş akışkanı tekrar eski yerine getirmek için çeşitli yöntemler mevcuttur. Örneğin, çok eskiden beri bilinen termosifonda (bir ısı borusu çeşididir) yoğuşan akışkan yerçekimi kuvvetiyle eski yerine gelir. Diğer yöntemler; kılcal, merkezkaç, ozmotik, manyetik, vs. kuvvetleriyle yapılan yöntemlerdir. En çok kullanılan yöntem kılcal kuvvetlerden yararlanmadır. Bu kılcallığı veren pek çok gözenekli malzeme mevcut olmasına rağmen, endüstriyel mühendisliğin en çok kullandığı ince gözenekli elek telleridir. Çünkü bunlar çok geniş sıcaklık aralıklarında çalışabilmektedir.

Bir ısı borusunun çalışabilmesi, yanı akışkanın sürekli faz dönüşümü yaparak çevrim yapabilmesi için belirli şartlar yerine gelmelidir. Bunlar, ısı borusu çalışma limitleri olarak adlandırılır. Bu limitler, şekil 3.1.1'de gösterilmiştir. Bunlardan en önemlisi, kılcal kuvvetlerin, bir ısı borusunda meydana gelebilecek basınç kayıplarını karşılayacak kadar büyük olması gerektiğidir. Yani,

$$(\Delta P_c)_{max} \geqslant \Delta P_l + \Delta P_v + \Delta P_g$$

olmalıdır. Isı borusunun kapasitesini artırmak için, eşitsizliğin sol tarafını büyültmek yada, sağ tarafını küçültmek gerekir. Kılcallığı artırmak için, daha ince kılcallığı gözenekli fitilden kullanmak gerekir. Ancak, bu seferde fitil içindeki sıvıda basınç kayıpları artacaktır. Böyle durumda optimize yapılması gerekir.

Teorik çalışmalarda, fitil içindeki gözenekli bir malzeme içindeki, sıvıdaki basınç kaybının hesabında, Darcy denklemi kullanılmaktadır. Bu denklem, Fanning denkleminin aksine hız dağılımını lineer kabul eder. Bu varsayım, deneylerlede doğrulanmıştır. Brinkman, Ergun bu denklemi dahada genişletmişlerdir. Ancak, bu denklemler fitilin homojen gözeneklere sahip olduğunu varsayar. Dolayısıyla, buradaki olayı tanımlamada daha çok ampirik ifadeler kullanılmaktadır.

Buhar için yapılan hesaplarda, bilinen Fanning denklemi, yahut ta en genel olarak, Navier-Stokes denklemleri kullanılabilir. Teorik çalışmalar, daha çok buhar üzerinde yapılmıştırlar. Bunun çeşitli sebepleri vardır. En önemlisi; termosifonun normal bir ısı borusundan daha çok ısı taşıması ve termosifonda en büyük basınç kaybının buhar fazında meydana gelmesindendir.

Isı borusu teeorik hesaplarında diğer bir üzerinde durulması gereken husus sıvı buhar arayüzeyindeki olayların tanımlanmasıdır. Değişik fitil yapıları ve geometrileri için, değişik sınır şartları kullanılır.

Isı borusu çok çeşit geometrilerde (üçgen, dikdörtgen, halka, elips, vs.) olmasına rağmen en çok kullanılan şekil silindiriktir. Bundan dolayı silindirik olmayanlarda ısı borusu olarak adlandırılmaktadır.

Isı transferi kaynaktan kuyuya kadar altı çeşit prosesden etkilenir. (1) ısı borusu duvarı boyunca ve fitil-sıvı-buhar arayüzeyi boyunca olan ısı tranferi; (2) buharlaştırıcadaki, sıvı-buhar arayüzeyindeki sıvının buharlaşması; (3) buharın, buharlaştırıcadan yoğuşturucuya borunun orta kısmından gitmesi; (4) yoğuşturucuda, buharın sıvı-buhar arayüzeyinde yoğuşması; (5) sıvı-buhar ara yüzeyinden, fitil ve borudan geçerek radyal yönde ısı transferi; (6) kılcallıkla sıvının yoğuşturucudan buharlaştırıcıya dönmesi.

Isı borusu dizaynında ilk yapılacak iş, çalışma sıcaklığına uygun olarak akışkanı seçmektir. Sonra, uygun boru ve yardımcı elemanlar seçilir. Dizayn ve imalat esnasında dikkat edilecek hususlar; çalışma sıcaklığında uygun akışkan seçilmesi, akışkanın boru malzemesi ve fitil ile uyuşabilir olması, basınç kayıplarını yenecek kapasitesi olan fitilin seçilmesi, içinde yalnızca çalışma akışkanı kalacak şekilde doldurulması ve sızdırmazlığın sağlanmış olması istenir. Ayrıca, imalatının kolay ve ucuz olması istenir.

Daha öncede belirttiğimiz gibi, ısı borusunun en büyük özelliği, ısı akısının çok büyük olmasıdır. Yani,çok küçük alanlardan çok büyük miktarlardaki ısıyı alıp, yine çok küçük alanlardan transfer edebilmesidir. Sanayide, bir cihazın kaplayacağı yer önem kazandığında, cihazı büyütmeden kapasitesinin arttırılması gerekebilir. Bunu yaparken, ucuza ve daha kolay gerçekleştirilmeye çalışılır.

Konsantrik ısı borusuda, böyle bir amaçla A. Faghri [1] tarafından 1988 yılında önerilmiştir. Diğer ısı borularından farkı, yalnızca geometrik gözükmekle birlikte aslında esas farkı; bir tane değil iki tane ısı borusunun yan yana getirilmiş olmasıdır. Akışkan, iki boru arasında bulunmaktadır. Yazar, 1989 yılında, konsantrik termosifon üzerinde deneysel gözlemler yapmıştır. Burada yapılan ısı borusunun farkı ise, dıştaki borunun iç ve içteki borunun dış kısmına fitil ilave edilmesindedir.

Deneysel çalışmada, ikiside 46 cm uzunluğunda ve biri 5 cm iç çapında, diğeri 2.3 cm iç çapında iki adet bakır boru seçilmiştir. Kalın borunun içine ve ince borunun dışına, 400 mesh lik paslanmaz (316 tip) çelik elek sarılmıştır. Paslanmaz çeliğin bu tipi, suyla, 150 °C den sonra uyuşmamaktadır. Ancak, burada sıcaklık 100 °C yi dahi geçmediğinden uyuştuğu kabul edilebilir. Daha sonra, iç kısımlarına diş açılmış, pirinç bir kapakla kapatılmışlardır. Bir doldurma tesisatında standartlara yaklaşık uygun bir şekilde doldurulmuş ve 0.5 cm çapındaki doldurma borusu, bir mengene aracılığıyla, hava girmeyecek şekilde, ezilmiştir. Borunun yoğuşturucu kısımlarına ceket sistemleri yapılmış, buharlaştırıcı kısmına ise, elektrikli rezistanslar monte edilmiştir. İçteki borunun yoğuşturucusu ile buharlaştırıcısını ayırmak için adyabatik kısmına cam yünü sıkıştırdıktan sonra, lastik tapalarla kapatılmış ve bunların etrafı, silikon içerikli bir malzemeyle sızdırmazlığı sağlanmıştır. Daha sonra, ısı borusu hazırlanmış bir test düzenine taşınmıştır. Burada; dıştaki ve içteki ısıtıcılara giren enerjiyi kontrol etmek amacıyla, her biri için birer adet ampermetre, voltmetre ve varyak (ayarlı transformatör) kullanılmıştır. Yoğuşturucu kısmında ise; sabit seviye tankı, debiyi ölçmek amacıyla kalibre edilmiş bir tank, geniş hacimli bir boşaltma tankı ve sabit seviye tankını beslemek amacıyla, küçük bir pompa kullanılmıştır. Sıcaklık dağılımını ölçmek amacıyla Kromel-Alumel termoeleman çiftleri kullanılmıştır. Boru üzerine açılan küçük yuvalara sokulan uçların borudan ayrılmasını önlemek amacıyla yapıştırılmışdır. Sıcaklık okumaları, multimetre aracılığıyle yapılmıştır.

İki ayrı deney yapılmıştır. Birincisinde, her iki ısıtıcıya elektrik verilmiş ve soğutma her iki borudan yapılmıştır. Bu işlemde, 5'er dakika aralıklarla toplam 30 dakika sıcaklık değişimi gözlenmiştir. İkincisinde ise, standart ısı borusuna benzetmek amacıyla, sadece dış borudan ısı verilip alınmış ve yine 5'er dakika aralarla toplam 30 dakika içindeki sıcaklık değişimi gözlenmiştir.

Elde edilen değerlerden her iki deneydede sıcaklık profilinin yaklaşık aynı eğimde olduğu gözlenmiştir. Oysa ki; ikinci deneyde verilen ısı miktarı daha az olduğundan sıcaklık profilinin daha yatay olması beklenir. Buradan çıkarılan sonuç, önerilen ısı borusunun daha kapasiteli çalıştığı olup, teoriyede uymaktadır.

Ayrıca, konsantrik ısı borusunun verimi, yani çekilen ısının verilen ısıya oranı, standart ısı borusu olarak kabul ettiğimizden daha büyük çıkmaktadır. Bu da, konsantrik ısı borusun daha yüksek kapasiteye sahip olduğunun belki bir ölçüde göstergesidir. Tam bir gösterge olarak kabul edilmemesinin sebebi, standart ısı borusunun tek borulu olmasıdır. Gerçekte, tam bir kıyas yapabilmek için, içteki borunun çıkarılıp test edilmesi gerekir.

Sonuç olarak, konsantrik ısı borusu, dar alanlarda ısı borusunu büyütmeden kapasite artırılması gereken yerlerde verimli olarak kullanılabilir. İmalatı zor değildir. Ayrıca, pek fazla bir ek masrafda gerektirmez.

Bu tezde; birinci bölümde, genel olarak ısı borusunun ve tez konusunun bir tanıtımı yapıldıktan sonra, ikinci bölümde ısı boruları ile ilgili gelişmelerden bahsedilmiştir. Isı borusu teorisi, üçüncü bölümün kapsamındadır. Dördüncü bölümde ise, dizayn edilirken dikkat edilmesi gereken hususlar anlatılmıştır. Beşinci bölümde, önerilmiş olan konsantrik ısı borusu ayrıntılı bir şekilde tanıtıldıktan sonra hazırlanan bir deney tesisatından ve yapılan işlemlerden bahsedilmiştir. Son bölüm ise, sonuç bölümüdür.

CH.1. INTRODUCTION

The heat pipe is a heat transmitting device of very high effective thermal conductivity, which is several tens or several hundreds times as large as the thermal conductivity of the same size copper rod. It has received a great deal of attention in recent years.

The principle of heat pipe operation is essentially based on the exchange of latent heat of a pure working fluid associated with its evaporation and condensation in the enclosure. Therefore, it can transport, in its steady state operation, a large amount of thermal energy from one end of the pipe (evaporator section) to the other (Condenser section) even when a small temperature difference is applied to both ends.

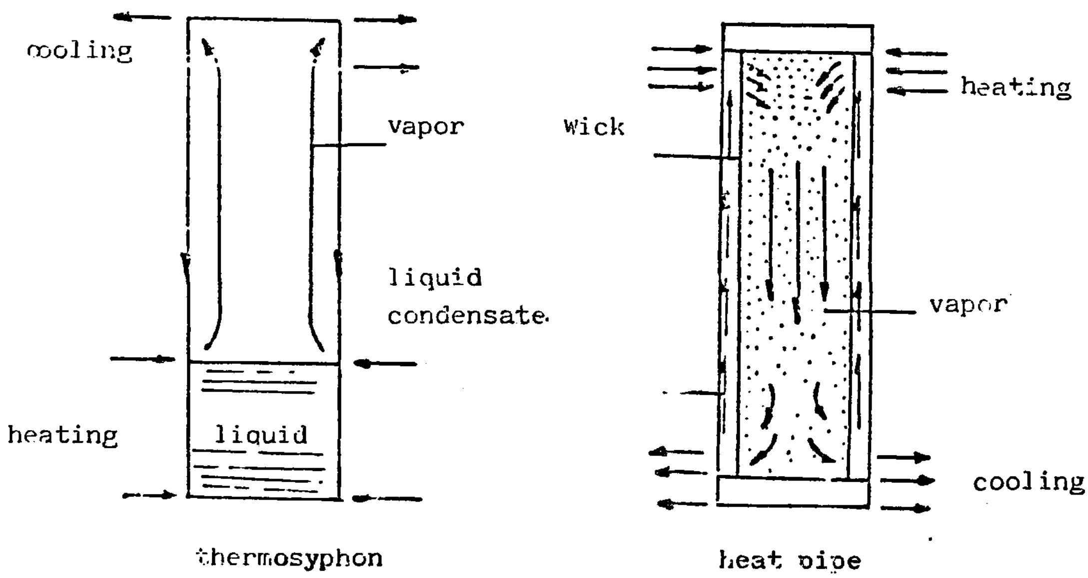


Fig.1.1 Thermosyphon and heat pipe

The heat pipe is similar in some respectsto the thermal syphon. They are shown in Fig.1.1. In thermal syphon which has been used for many years, condensate return is achieved by gravitational force. The heat pipe makes use of several methods of condensate return as shown in Table 1.

TABLE 1 Methods of condensate return.

Gravity	Thermal syphon
Capillary force	Standart heat pipe
Centripetal force	Rotating heat pipe
Electrostatic volume forces	Electrohydrodynamic heat pipe
Magnetic volume forces	Magnetohydrodynamic heat pipe
Osmotic forces	Osmotic heat pipe

The heat pipe is characterized by [2] :

(1) Very high effective thermal conductivity
(2) The ability to act as a thermal flux transformer
(3) An isothermal surface
(4) Variable thermal impedance or VCHP
(5) Thermal diodes and switches

Heat pipe technology had has a remarkable developments in the last years. The applications of heat pipes are expanding in many fields (e.g.for satellite, industrial and home use). In many applications space is at a premium, so any increase in heat transfer for a given pipe size will be important. For this aim, Faghri proposed a new configuration, double-walled concentric, and experienced wickless type (thermosyphon) of heat pipe.

In this thesis, it was aimed to manufacture a heat pipe with a new shape similar to that tested by Faghri , except being wickless, and to evaluate its performance comparing to standart heat pipe. First, some knowledge about the heat pipe was given, and then the double-walled concentric heat pipe was described. Finally, the experiment and its results obtained on a heat pipe constructed as having the introduced configuration. The results show that its performance would be higher than that of the standart heat pipe.

CH.2. DEVELOPMENTS IN HEAT PIPE

Heat pipe design, theory, and applications have developed quickly over the 20 years since the invention of the pipe by Groover et al. [3]. This is evidenced by more than 2000 references, five international conferences, and more than five books on heat pipes.

It has been used in many areas such as in

- space applications
- cooling electronic circuits
- stabilization of the pipe works
- recovery of waste heat
- preheating of the combustion air
- cooling the brake systems of air planes
- temperature control
- solar energy systems
- geothermal energy systems
- storing the energy
- heating inside the vehicle by the exhaust
- thermionic power generation
- preservation of permafrost
- stirling engines
- and many other areas

These come into a number of broad groups, each of which describes o property of the heat pipe. These groups are:

(1) Seperation of heat source and sink

(2) Temperature flattening

(3) Heat flux transformation

(4) Temperature control

(5) Thermal diodes and switches

The heat pipe concept was first put forward by R.S. Gaugler of the General Motor Corporation, Ohio, USA. Then an extensive programme was conducted on heat pipes at Las Alamos Laboratory, New Mexico, Under Groover, and preliminary results were reported in the first publications on heat pipes. It followed in many countries such as England, Italy, USSR, Germany, Japan, Czechoslovakia, etc.

By now the theory of the heat pipe was well developed, based largely on the work of Cotter, and a wide variety of heat pipes were commercially available from a number of companies in the developed countries. Unfortunately there has not been any company in Turkey even availability in these country since 1970.

The most of the theoretical investigations has been done for the wickless type (thermosyphon) heat pipe since analitical solutions desribing the flow are easy to find comparing the standart heat pipes, and it has larger heat transfer capability than a wicked heat pipe.

Investigations have usually been focused on the aim of improving the performance by extending their operating ranges, i.e. operating limits. There is a growing number of studies on the thermosyphons. Fukan, etal.[4]pointed out that the last correlations of the critical heat flux by using new parameters such as height of the Liquid pool and the buble kinetic energy. He concluded that the operating limit of a thermosyphon is not only caused by flooding but is also dependent upon the liquid lumps generated by boiling in the evaporator. Chi, et al. [5] suggested dryout, burnout and flooding limits. Dobran [6] showed that the sonic heat transfer limit is affected by the working fluid, vapor flow area, manner of vapor introduction into the vapor core region of the evaporator, and lengths of the evaporator and the adiabatic regions of the high-temperature heat pipes. J.E.Hart [7] observed some oscillations that reflect quasi-periodic flow as well as a type of periodic chaos. There has been also some fundamental investigations about the effects of geometry and positions of heat pipes or thermosyphons. For example, Lock and Liu [8] did an experiment showing that the length-diameter ratio's effect is greater than that of heated-cooled length ratio at low Rayleigh numbers. The influences of the inclination angles between 5-45° determined as by capillary limit and between 45-90° by both the capillary and flooding limits are the conclusions of Gürses [9] . Thus under the tilting mode, gravity affects considerably the heat pipe performance [10] .

The dealers of heat pipes should also interest with the areaof porous media, especially who works around the capillar wicked heat pipe. A fundamental approach namely a continuum model for the phase change in the porous media in the presence of coupled head flow, fluid flow and species transport processes was presented by Kececıoğlu and Rubinsky [11] . Although the Darcy's equation has been still used in the design of the heat pipe, many authors have been studying about the aparture from this law. Two of them are about the thermal dispersion effects on non-Darcy convection in porous media bay Laı [12] and non-Darcian effects on the wicking limit by Özgüç et al. [13] which are both based on Ergun's model. In the wicked heat pipe many types of porous media has been used. An interesting one is the cornstalk experienced by Kim [14] for the aim at the application of waste heat recovery in the lower temperature range of 40 to 60° C.

Looking at the aspect of heat pipe from the thermodynamical view, we'll be face to a new cycle proposed by Kobayaski [15]. This cycle can be applied for engine and heat rejecting systems as in the case of heat pump. The author called it as heat pipe engine or heat pipe refrigerator. It is composed of two adiabatic and two constant-volume changes in two-phases thermodynamic condition. the author said that a theoretical thermal efficiency of this ideal engine cycle will be inevitably low owing to its intrinsic nature that it is intended to be operated at low temperature ranges with small temperature differences. On the other hand, head rejecting systems utilizing this cycle are expected to be highly efficient.

Although the most practical configuration used in industry is the cylindirical standart heat pipe geometry, of course, various configurations may be designed such as shown in Fig.2.1. Broadly there are there are three classes of closed evaporator-condenser devices. (1) Heat pipe (2) Thermosyphons (3) Heat transfer devices. The capillary heat pipe, two phase thermosyphon, and electro hydrodynamic heat pipe are the kinds of heat pipe, thermosyphon and heat transfer device, respectively [16] . Fig.2.2.

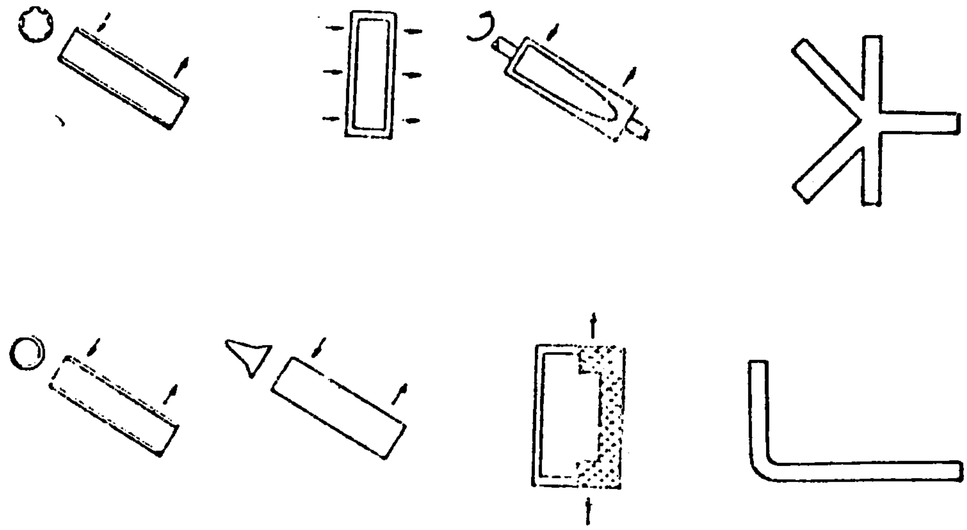


Fig.2.1 Some typical geometries of the heat pipe

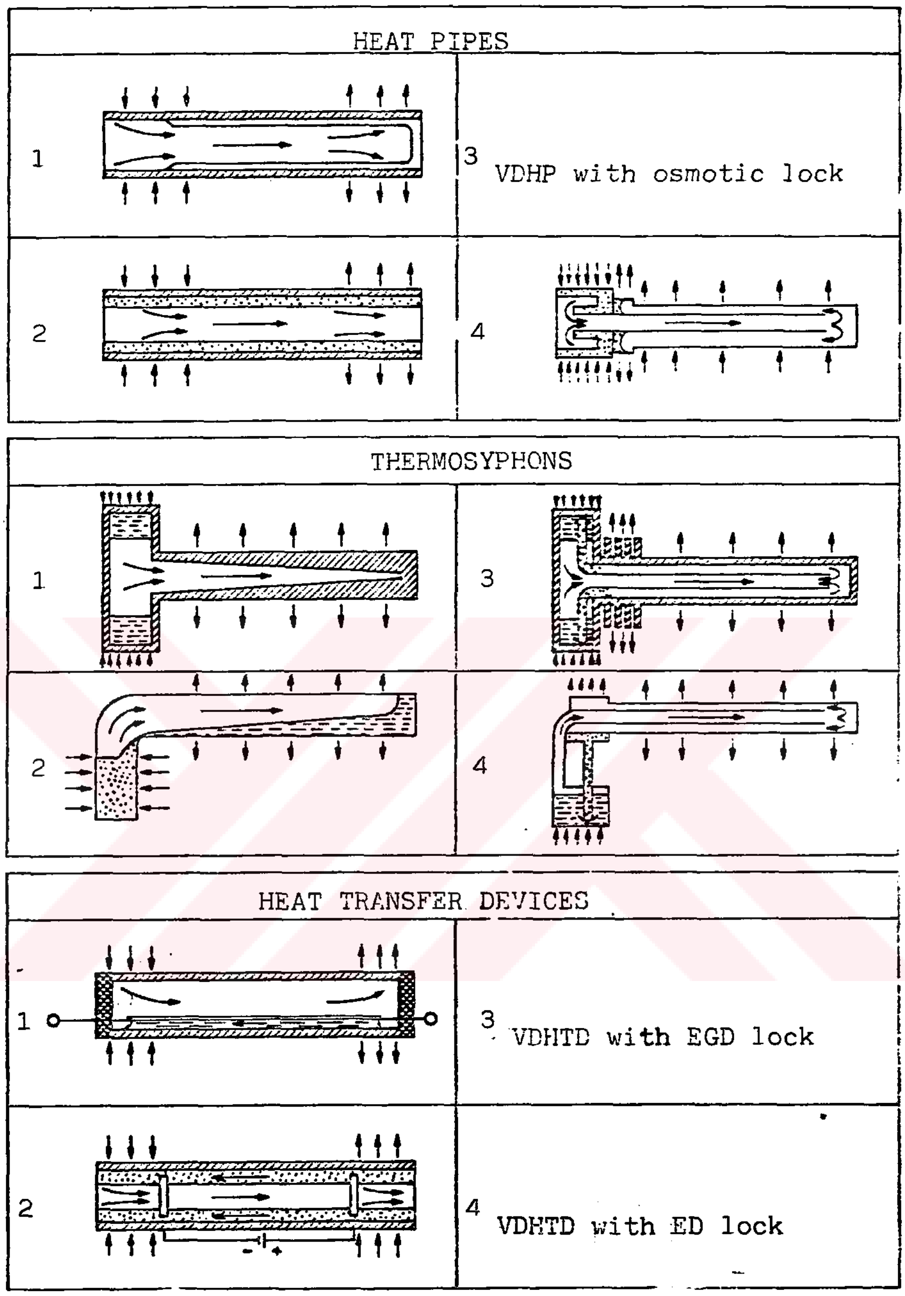


Fig.2.2. Classification of closed evaporator-condenser devices I, Heat pipes:1, osmotic;2, capillary;3, vapor-dynamic with an osmotic hydraulic lock;4, vapor-dynamic with a capillary lock. II, Thermosyphons;1, centrifugal;2, two-phase;3,vapor-dynamic with a centrifugal lock;4, vapor-dynamic with a gas lock. III, Heat transfer devices:1, electrohydrodynamic heat pipe;2, electroosmotic heat pipe;3, vapor-dynamic thermosyphon with an electrohydrodynamic lock;4, vapor-dynamic thermosyphon with an electro-osmotic lock.

CH.3. THEORY OF HEAT PIPES

3.1. INTRODUCTION

The heat pipe is a novel device that allows the transfer of very substantial quantities of heat through small surface areas. In order for the heat pipe to operate the maximum capillary pumping head $(\Delta P_c)_{max}$ must be greater than the total pressure drop in the pipe, i.e. pressure drop in liquid and vapour phase and sometimes gravitational head. Otherwise, it will not operate. Thus,

$$(\Delta P_c)_{max} \geqslant \Delta P_l + \Delta P_v + \Delta P_g \qquad (3.1.1)$$

This is one of the limits known as viscous or capillary limit. There are also some other limits which would be evaluated empirically due to wick or pipe structure. They are shown in Fig.3.1.1.

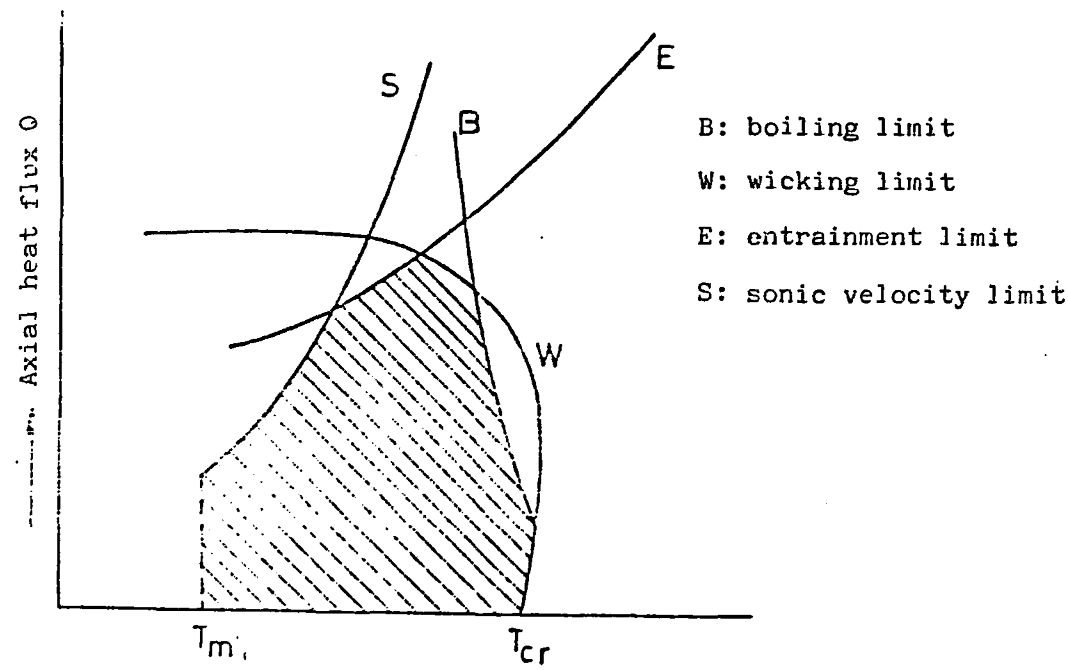


Fig.3.1.1 Heat pipe operating limits

3.2. SURFACE TENSION AND SURFACE ENERGY

Molecules in a liquid equally attract each other in all directions experiencing any resulting force. But, at or near the surface, an unbalance of attraction forces occurs. Because of effect a liquid will tend to take up a shape having minimum area. In this case, a liquid surface behaves like a membrane.

In order to increase the surface area work will need to be done onthe liquid. The energy associated with this work is known as the free surface energy.

When a liquid is in contact with a solid surface mdecules near the solid will experience forces from the solid molecules. Depending on attraction or repulsion liquid solid surface will curve upwards or downwards. When the forces are attractive the liquid is said to "wet" the solid.

The condition for wetting to occur is that the total surface energy is reduced by wetting.

$$\sigma_{sl} + \sigma_{sv} < \sigma_{s}$$

Where the subscripts s,l,v refer to solid, liquid and vapor phases respectively, as shown in Fig.3.2.1.

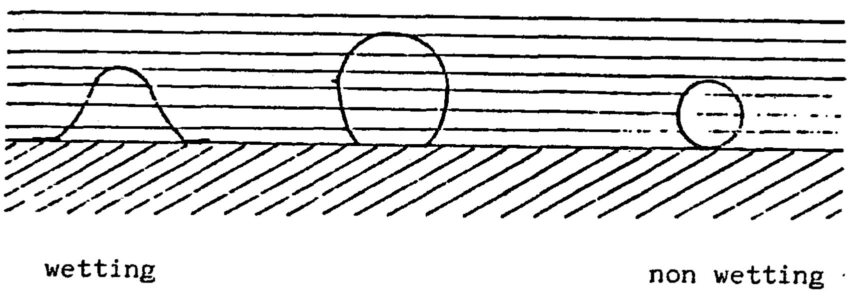
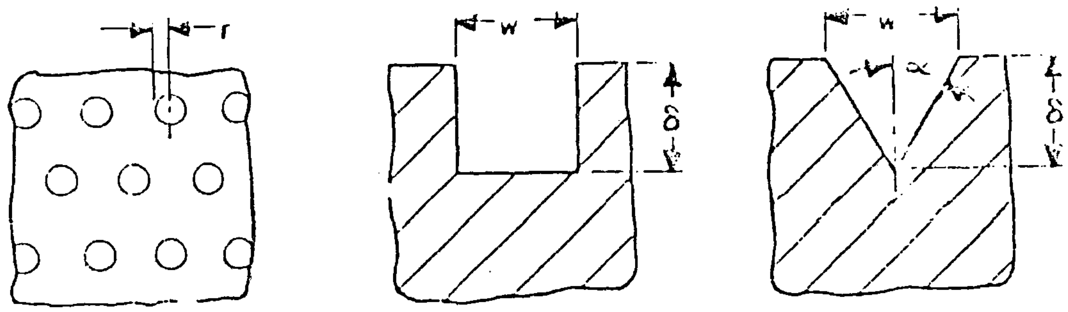


Fig.3.2.1 Wetting and non-wetting contact

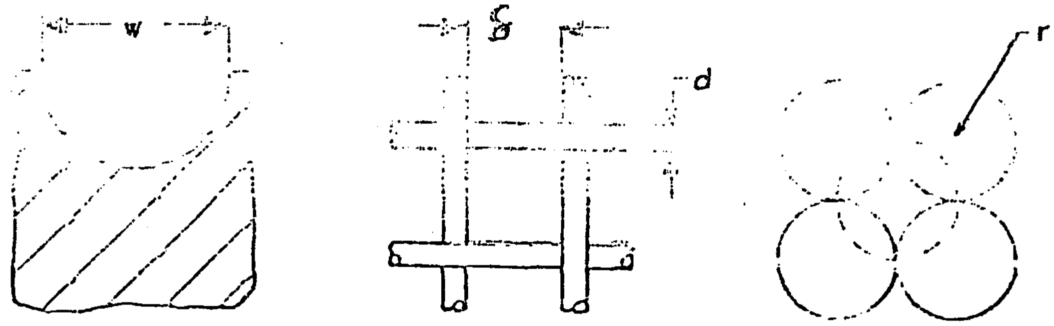
$(\Delta Pc)_{max}$ depends on the working fluid and the shape of liquid channels, i.e.R_c. Various configurations with R_c value of them are shown below.



(1) Cylindirical Pores $R_c = r$

(2) Rectangular Grooves $R_c = w/2$

(3) Triangular Grooves $R_c = w \ \cos\alpha \ /1-\sin\alpha$



(4) Semicircular Grooves $R_c = \frac{1}{2}\frac{\pi \ w}{(\pi-2)}$

(5) Square Mesh $R_c = (\delta+d)/2$

(6) Packed s heres $R_c = 0.41 \ r$

Fig.3.2.2 Cases for which the effective pore radius can be estimated.

Consequently, a pressure difference occur on the curved surface related to surface energy σ_1 and radius of curvature of the surface, such that

$$\Delta P = 2\ \sigma_1/R \tag{3.2.1}$$

If the surface has two radii of curvature at right angles R_1, R_2, then it becomes

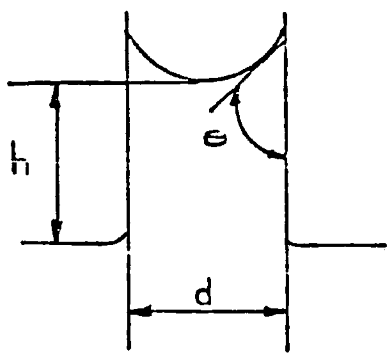
$$\Delta P = \sigma_1\ (1/R_1 + 1/R_2\ ) \tag{3.2.2}$$

That is the basic driving force for standart heat pipes.

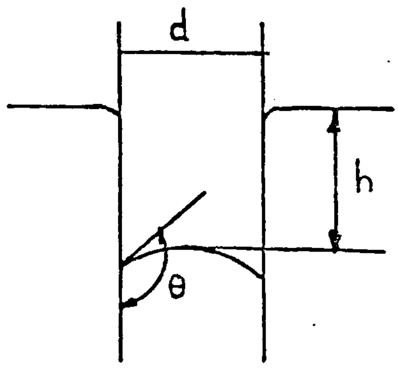
Due to this pressure difference, if a vertical tube, radius r, is placed in a liquid which wets the material of the tube, the liqid will rise in the tube (Fig.3.2.3.) by an amount of

$$h = (2\,\sigma/r\ \text{Cos}\ \theta\ )\ 1/\rho_1\ \ g \tag{3.2.3}$$

Where ρ_1 is the liquid density and θ is the contact angle. It has a maximum value when Cos θ =1.



Wetting fluid



Non-wetting fluid

Fig.3.2.4 Capillary rise in a tube

3.3. PRESSURE LOSSES

3.3.1. PRESSURE LOSSES IN THE LIQUID PHASE ΔP_l

The flow regime in the liquid phase is almost always laminar. Due to complexity of liquid chanuels, a correction factor is used to modify the Hagen-Poiseuille equation. Three principal capillary geometries are;

(1) Porous wicks

(2) Open grooves

Porous Wicks. In order to calculate ΔP_l, Darcy's law including a correction K to take account of pore size, distribution and the tortuosity. Darcy's "law" is written as

$$\Delta P_l = \mu_l \ w_l / K \tag{3.3.1}$$

or

$$\Delta P_l = \mu_l \ l_{eff} \dot{m} / \rho_l \ K \ A \tag{3.3.2}$$

where $l_{eff} = l_a + (l_e + l_c)/2$

K : the wick permeability (which is easily measured)

A : the wick cross-sectional area

But Darcy's law can not describe completely the behaviour of the liquid motion, especially for the cases of high permeability, [17] and Re >1

For high permeability, using Bringman extended Darcy model derived from Navier-Stokes equations for reverse flows, C.Y.liu et al. [17] analysed the flow to account the frictional losses for cylindirical heat pipe.

$$\mu_1 \left(\frac{d^2 w}{dr^2} + \frac{1}{r}\frac{dw}{dr}\right) = \frac{dp_1}{dz} + \frac{\mu_1}{K} w \tag{3.3.3}$$

The other equation given based on the Darcy Brinkman Ergun model takes also the inertial terms into account.

$$0 = -\frac{dp_1}{dz} - \frac{\mu_1}{K} w_1 - \frac{\rho_1}{K_i} |w_1| w + \frac{\mu_1}{\varepsilon} \nabla^2 w \tag{3.3.4}$$

Where

$K = \varepsilon^3 / (1-\varepsilon)\, g_1\, S_w^2$, $K_i = \varepsilon^3 / (1-\varepsilon)\, g_2\, S_w$

S_w : specific surface area of the wick

g_1, g_2 : two empirical constants

ε : volume of voids/volume of body

But, in the case of standart heat pipe, these two terms, frictional and inertia, can be easily neglected.

Axially groove wick: Axially grooved heat pipes are at present among the most widespread HP modifications including gravity-assisted pipes characterized by high technological effectiveness, reliability of parameters, high heat transfer properties. For this type, pressure drop in the liquid is given by:

$$\Delta P_1 = 8\, \mu_1\, Q\, l \,/\, \pi\, r^4\, N\, \rho_1\, L \tag{3.3.5}$$

where N is the number of grooves and

r_e = 2 Flow area/wetted perimeter.

(some wick sections are shown in Fig.3.3.1.)

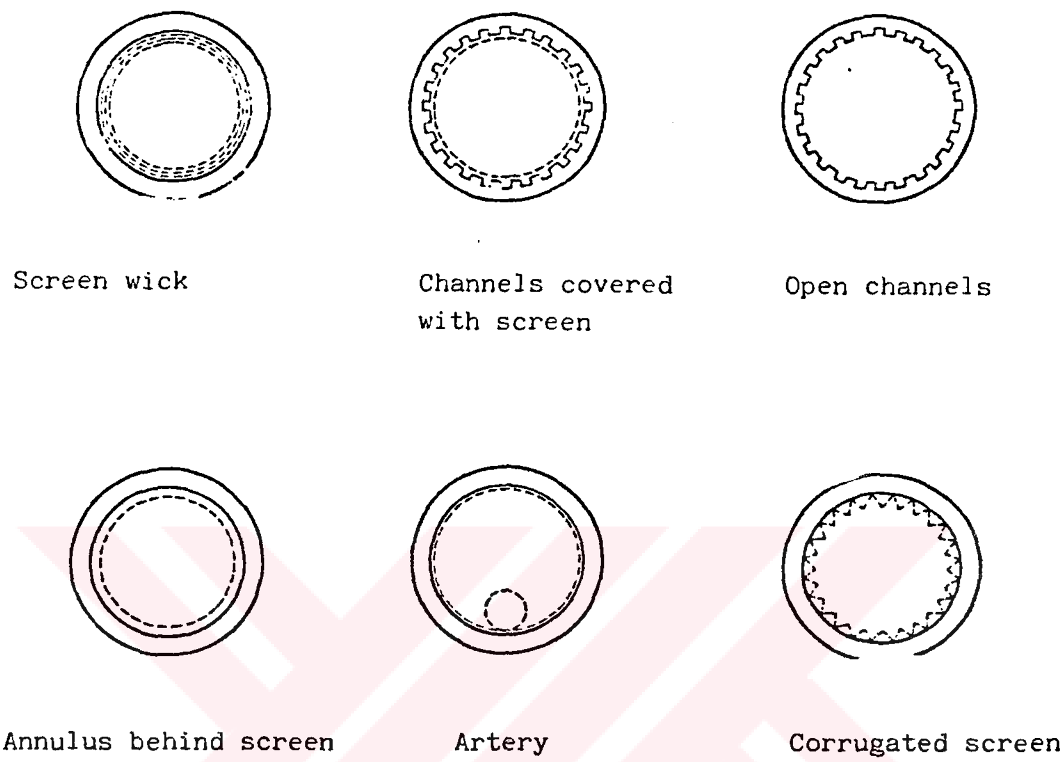


Fig.3.3.1 Some wick sections

3.3.2. PRESSURE LOSSES IN VAPOR PHASE

The total vapor pressure difference is the sum of the pressure drops in the three regions, namely the evaporator drop ΔP_{ve}, the adiabatic section drop ΔP_{va}, and the pressure drop in the condensing in the condensing region ΔP_{vc}. There has been some theories due to compressibility or whether is one dimensional or two dimensional.

In the simple one dimensional theory (in compressible flow), the flow velocity is assumed to be small compared to the velocity of sound C, i.e. $V/C < 0.3$. This is true except in the case of high temperature heat pipes or during start up. This theory suggest that the inertial term will have the opposite in the condenser region and should cancel out that of the evaporator. The total pressure drop in the vapour phase will be due entirely to the viscous terms. In the adiabatic section the pressure difference will contain only the viscous terms which will be given either by the Hagen-Poiseuille equation or the Fanning equation.

Hence the total pressure drop for laminar flow and with full pressure recovery

$$\Delta P_v = 8\ \mu_v\ \dot{m}\ [(l_e + l_c)/2 + l_a]\ /\rho\ r_v^4 \tag{3.3.6}$$

This is however broadly correct. A considerable number of papers have been published giving a more precise treatment of the problem. For example, one of them by cotter who made an earliest theoretical treatment obtained a pressure recovery value of

$$\Delta P_{vc} = +\ 4\ \dot{m}^2\ /\pi^2\ 8\ \rho_v\ r_v^4 \tag{3.3.8}$$

His full expression is

$$\Delta P_v = -\ (1-4/\pi^2)\ \dot{m}^2/8\ \mu_v\ \ r_v^4 -\ \ 8\ \mu_v\ \dot{m}\ l_a\ /\pi\ \rho_v\ \ r_v^4 \tag{3.3.9}$$

Another case, two dimensionality, shows that, especially in the condenser region, the temperature and the pressure are not constant across the section in practical heat pipes. The most important result about this problem is the possibility of reverse flow at high evaporation or condensation rates, as in the case of radial Reynold number, $R_r < -2.3$. [18].

The effect of compressibility of the vapor is important in high temperature liquid metal heat pipes and during start-up.

3.4. HEAT TRANSFER IN HEAT PIPES

Heat can both enter and leave the heat pipe by conduction from a heat source/sink, by convection, or by thermal radiation. The temperature drops in a heat pipe can be represented by thermal resistances and an equivalent circuit as shown in Fig.3.4.1.

In order to provide an indication of the relative magnitude of the various thermal resistances Table (3.1.) lists some approximate values/cm^2 of a water heat pipe

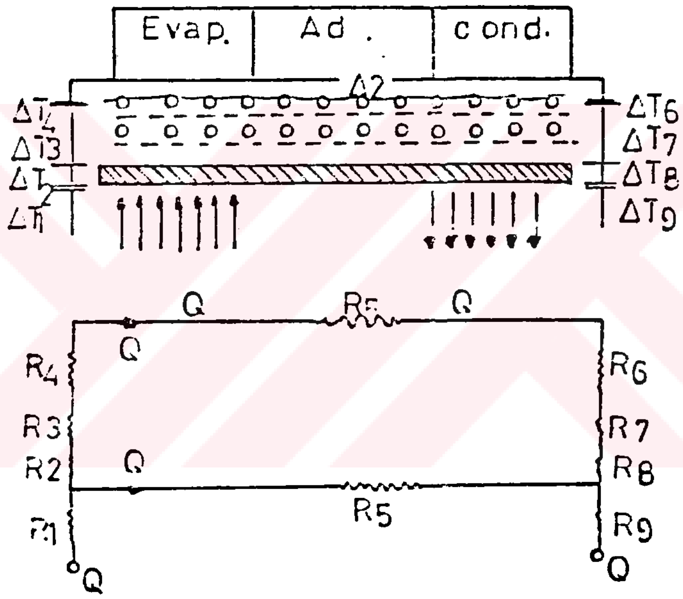


Fig.3.3.2 Temperature drops and equivalent thermal resistances in a heat pipe

Notes: Resistance=Temperature difference/Total heat flux T_1 to T_9, R_1 to R_9, and R_s are defined in Fig.3.4.1. A_e and A_c the evaporator and condenser wall areas he and h_c heat transfer coefficients at the evaporator and condenser outer surfaces

q_e and q_c heat flux through the evaporator and condenser walls

k thermal conductivity of the heat pipe wall

p_v vapour pressure

L latent heat

T absolute temperature of vapour

d wick thickness

Table 3.1 Equivalent thermal resistances and its values for water heat pipe.

Term	Defining Relation	Thermal Resistance	Comment	W/°C
1	$Q = h_e A_e \Delta T_1$	$R_1 = 1/h_e A_e$		10^3–10
2	Plane geometry $q_e = k\Delta T_2/t$ Cylindirical geometry $q_e = 2\pi K \Delta T_2 / \log_e \frac{r_2}{r_1}$	$R_2 = \frac{\Delta T_2}{A_e q_e}$		10^{-1}
3	Eq. 3.4.3 or 3.4.4.	$R_3 = \frac{d}{K_w A_e}$	true for liquid metals	10
4	$\Delta T_s = \frac{L^2 P_v}{(2\pi RT)^{1/2} RT^2}$	$R_4 = \frac{RT(2\pi RT)^{1/2}}{L^2 P_v A_e}$	usually neclected	10^{-5}
5	$\Delta T_s = \frac{RT^2 \Delta P_v}{L P_v}$	$R_5 = \frac{RT^2 \Delta R}{Q L P_v}$	//	10^{-8}
6	$q_c = \frac{\Delta T_6 L^2 P_v}{(2\pi RT)^{1/2} RT^2}$	$R_6 = \frac{RT^2 (2\pi RT)^{1/2}}{L^2 P_v A_c}$	//	10^{-5}
7	$q = \frac{\Delta T L^2 P_v}{(2\pi RT_s)^{1/2}} \times \frac{1}{RT_s^2}$	$R_7 = \frac{d}{K_w A_c}$		10
8	Plane geometry $q_c = \frac{K \Delta T_2}{t}$ Cylindirical geometry $q_c = \frac{2\pi K \Delta T_2}{\log_e r_2/r_1}$	$R_8 = \frac{\Delta T_8}{A_c q_c}$	for thin walled cylinders $\log_e \frac{r_2}{r_1} \simeq \frac{d}{r}$	10^{-1}
9	$Q = h_c A_c \Delta T_g$	$R_9 = \frac{1}{h_c A_c}$		10^3–10

3.4.1. HEAT TRANSFER IN THE EVAPORATOR REGION

In this region the heat will be transported to the liquid surface by conduction and natural convection mechanisms in the case of low heat flux. As the flux is increased convective heat transfer will also increase because of the buble formation. We can increase it until burnout which is a limit of the heat pipe occurs.

We will not discuss the case of unwicked surfaces here.

Boiling from wicked surfaces. The effect of the wick is to further complicate the boiling process since the wick provides additional nucleation and significantly modifies the movement of the liquid and vapor phases. There is now considerable literature on boiling from wicked surfaces and many experiments have been done about this phenomena. As a result, the two models of boiling mechanism are described by the authors as

(1) A layer of liquid filled wick adjacent to the heated surface with conduction across this layer and liquid vaporisation at the edge of the layer. In this model liquid most be drawn into the surface adjacent by capillary forces.

(2) A thin layer of vapour filled wick adjacent to the saturated liquid with in the wick. In this model the liquid is vaporized at the edge of this layer and the resulting vapor finds its way out of the wick along the surface and through large pore size passages in the wick. This model is analogaus to conventional film boiling.

One limit to the radial heat transfer from the evaporator will be set by "wicking", that is when capillary forces are unable to feed sufficient fluid into the evaporator. The limiting flux q_{crit} will be given by the expression,

q_{crit} = mxL/area of evaporator

Ferrell and Davis [19] extended this equation by including a correction for thermal expansion of the wick.

$$q_{crit} = \frac{g\ [h_{co}\ \rho_{lo}\ \sigma_l\ /\ \sigma_{lo} - \sigma_l\ l\ \sin\beta\ (1+\alpha\Delta T)]}{\rho_l\ \mu_l\ [l_e/2 + l_a]\ /\ L\ \rho_l\ K\ d\ (1+\alpha\Delta T)} \tag{3.4.1}$$

Where

h_{co} capillary height measured at a reference temperature

α coefficient of linear expansion of the wick

ΔT the temperature difference between the operating and reference temperature

Their experiments demonstrated that heat flux limits for water uq to 130 kw/m^2.

Liquid vapour interface temperature drop. The magnitude of the temperature drop can be estimated as follows.

The average velocity V_{av} in a vapour at temperature T_v and having molecular mass m is given by kinetic theory as

$$Vave = (8\ k\ T_v/\pi\ m)^{0.5}$$

where k is Boltzmann's constant.

The average flow of molecules in any given direction is nxV_{ave} per unit area times four, and the corresponding flow of heat/unit area

$m\ L\ n\ Vave/4$

Where n is the number of molecules per unit volume and L the latent heat

Also for a perfect gas

Pressure $P_v = n\ k\ T_v$

Hence heat flux $= P_v\ L\ (m/2\pi\ k\ T_v)^{0.5}$ to the liquid surface

heat flux $= P_l\ L\ (m/2\pi\ k\ T_l)^{0.5}$ away from the surface

Net heat flux from the surface is

$$q= \Delta T\ L^2\ P\ /\ (2\pi\ R T_s)^{0.5}\ \ R\ T_s^2 \qquad (3.4.2)$$

Where $T_s \approx T_v \approx T_l$

Wick thermal conductivity. Two models are used in the literature.

(1) Parallel case. Where the wick and working fluid are in parallel. If K_l is the thermal conductivity of the working fluid and K_s is the thermal conductivity of the material and ϵ=is the voidage fraction

$$K_w = (1-\ \epsilon)\ K_s + \epsilon\ K_l \qquad (3.4.3)$$

(2) Series case. If the two materials are assumed to be in series

$$K_w= 1/(1-\epsilon\ /\ K_s\ +\ \ \epsilon\ /\ K_l) \qquad (3.4.4)$$

3.4.2. HEAT TRANSFER IN THE CONDENSER REGION

The condensation mechanism is similar to that of surface evaporation discussed previously. It can occur in two forms, either by the condensing vapour forming a continuous liquid surface or by forming a large number of drops. Nusselt theory given in the standart text books [20] can be used to analyse the former type. The other is more complicated but similar to evaporation. Condensation is seriously affected by the presence of a non-condens gas. Hence the end of the condenser will be effectively shut off because of the concentration of this gas by vapour pumping.

The temperature drop through the saturated wick may be treated in the same manner as at the evaporator. It may be calculated from Equations (3.4.3.) and (3.4.4.)

3.5. LIMITS TO HEAT TRANSPORT

They are illustrated in Fig.3.1.1.

Viscous limit. At low temperatures viscous forces are dominant in the vapour flow. Busse [21] derived the following equation

$$q= r_v^2 \, L \, \rho_v \, P_v \, / \, 16 \, \mu_v \, leff \qquad (3.5.1)$$

where P_v and ρ_v refer to the evaporator end of the pipe.

Sonic Limit. It is given by

$$q= 0.474 \, L \, (\rho_v \, P_v)^{0.5} \qquad (3.5.2)$$

Entrainment Limit. Eg.(3.5.3.) gives the entrainment limiting flux

$$q = (2\pi \rho_v L^2 \sigma_l / Z)^{0.5} \qquad (3.5.3)$$

where Z is a correction factor.

Capillary Limit. In order for the heat pipe to operate,

$$\Delta P_{c_{max}} \geqslant \Delta P_l + \Delta P_v + \Delta P_g \qquad (3.5.4)$$

Burnout. It occurs at the evaporator at high radial fluxes. Eq.3.4.1. gives a limit for a homogeneous wick. This equation, which represents a wicking limit, is shown in section 3.4.1. to apply to water up to 130 kw/m^2. There are no simple correlations about this phemomena.

CH.4. DESIGN CRITERIES

The principle bases to select the three basic components of a heat pipe which are, first, the working fluid, second, the wick or capillary structure, and finally, the container are discussed below.

4.1. WORKING FLUID

A first consideration to select a suitable working fluid is the operating temperature range and some of them are shown in table 4.1. while full properties of most are given in appendix A.

Table 4.1 Heat Pipe Working Fluids.

Medium	Melting point (°C)	Boiling point at atmos.press. (°C)	Useful range (°C)
Helium	-272	-269	-271 - -269
Nitrogen	-210	-196	-203 - -160
Ammonia	-78	-33	-60 - 100
Freon 11	-111	24	-40 - 120
Pentane	-130	28	-20 - 120
Freon 113	-35	48	-10 - 100
Acetone	-95	57	0 - 120
Methanol	-98	64	10 - 130
Flutec PP2	-50	76	10 - 160
Ethanol	-112	78	0 - 130
Heptane	-90	98	0 - 150
Water	0	100	30 - 200
Flutec PP9	-70	160	0 - 225
Thermex	12	257	150 - 395
Mercury	-39	361	250 - 650
Caesium	29	670	450 - 900
Potassium	62	774	500 - 1000
Sodium	98	892	600 - 1200
Lithium	179	1340	1000 - 1800
Silver	960	2212	1800 - 2300

The prime requirements are:

(1) Compatibility with wick and wall materials
(2) Good thermal stability
(3) Wettability of wick and wall materials
(4) Vapour pressures not too high or low over the operating temperature range
(5) High latent heat
(6) High thermal conductivity
(7) Low liquid and vapour viscosities
(8) High surface tension
(9) Acceptable freezing or pour point
(10) Viscous, sonic, capillary, entrainment and nucleate boiling limitations
(11) Cost, availability and the other factors such as toxicity.

4.2. THE WICK OR CAPILLARY STRUCTURE

The prime purpose of the wick is to generate capillary pressure to transport the working fluid from the condenser to the evaporator region. So the principle requirements are

(1) small pore size
(2) high permeability
(3) wick thickness
(4) low thermal resistance
(5) good compatibility and wettability
(6) other factors such as cheap

There is a compromise between (1) and (2). So an aptimum size should be determined for an application.

There are three main types of wick form based on capillarity which are (1) Homogeneous wick (2) groove wick (3) composite wick. Some forms are shown in Fig.4.1.1. And some wick types and their properties are given in appendix B.

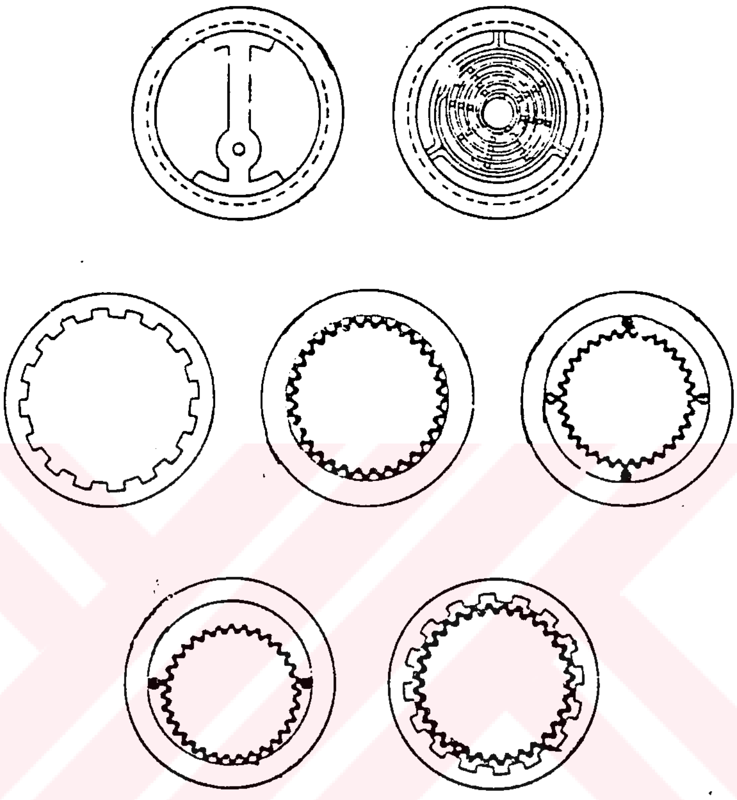


Fig.4.1 Forms of wick used in heat pipe (Courtesy NASA)

Expressions are available for predicting their thermal conductivities in saturated case. For example conductivity of the mesh type of wick can be expressed using Rayleigh's metod such as:

$$k_w = k_l \, (\beta - \varepsilon) \, / \, (\beta + \varepsilon)$$

where

$\beta \;= (1 + k_s \, / \, k_l) \, / \, (1 - k_s \, / \, k_l)$

k_s = thermal conductivity of solid case

k_l = thermal conductivity of liquid case

ε = volume fraction of solid phase

4.3. THE CONTAINER

Its function is to seperate inside of heat pipe from outside since they are in different thermodynamic conditions. Factors to select the container are:

(1) Compatibility
(2) Strength to weight ratio
(3) thermal conductivity (given in Appendix C.)
(4) Ease of fabrication
(5) Porosity
(6) Wettability

The most important factor is the first one, i.e. compatibility. Two major results of incompatibility are corrosion and the generation of the non-condensable gas. Some compatibility data is of course available in the general scientific publications and from trade literature on chemicals and materials.

4.4. FLUID INVENTORY

This is very important especially when considering small heat pipes. To overcome the problem of excess or deficiency of fluid is necessary by providing an excess fluid reservoir. The common practice is to include a slight excess of working fluid over that required to saturate the wick.

CH.5. DOUBLE-WALLED CONCENTRIC HEAT PIPE

5.1. INTRODUCTION

The double wall concentric heat pipe (Fig.5.1.1.) firstly proposed by Faghri consists of two concentric pipes of unequal diameter attached by means of end caps, creating an annular vapor space between the two pipes.

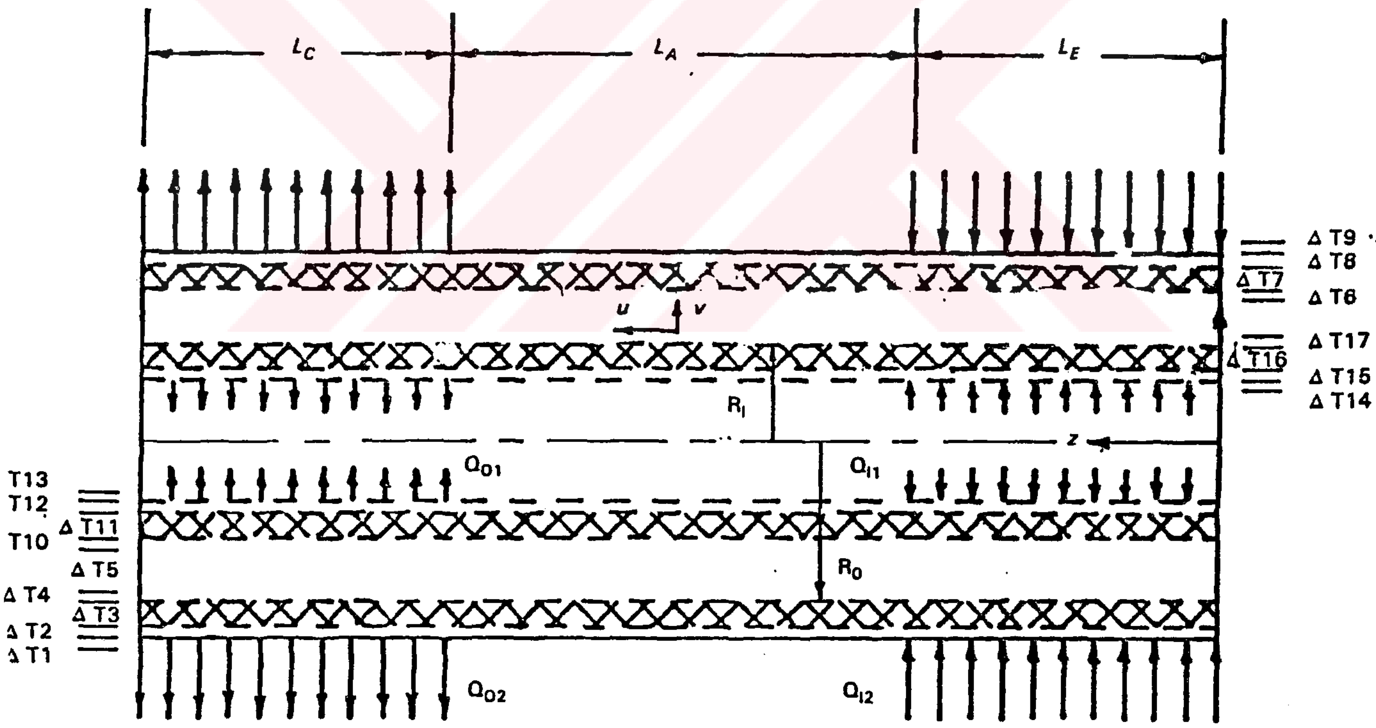


Fig.5.1.1 The double-walled conentric heat pipe

Wick structures may be placed on both the inner surface of the outer pipe and the outer surface of the inner pipe. The main difference from the standart heat pipes may be accepted as the increase in the surface area exposed for heat transfer both into and out of the pipe. The space inside the inner pipe is open to the surroundings. To the author's knowledge, there has been no experimental or theoretical investigation reported in the literature concerning the double-walled concentric heat pipe except investigation on wickless type, i.e. concentric annular two-phase closed thermosyphon by A.Faghri et al. [22].

The advantages of the double-walled concentric heat pipes are as follow:

(1) An increase in performance is expected as a result of the increase in surface area exposed for heat transfer. The heat transport per unit length will be more than the standart heat pipe. The heat pipes have usually small volume since in many applications the space is at a premium, so any increase in heat transport ability for a given pipe size is significant.

(2) Like to conventional heat pipe design, many methods which have been used to increase the capacity of heat pipe can be also used in the design of this type of heat pipe since it is not specific to any working fluid, wick structure, or container material.

(3) Due to the increase in wick or capillary area, performance would be increased because of the increase in capillary pressure.

(4) Needs no expensive tooling or other treatment.

However the main advantage of this type over the conventional one is the increased surface area for heat transfer into and out of the pipe without necessarily increasing the outside pipe diameter.

5.2. ANALYSE

Many researchers have analysed analitically and experimentally for many different types of heat pipe design. Most of them are about the convection heat transfer since heat pipe can be accepted as a device using the mechanism of convection. Since this new type is, of course, in an annuli form there have been many studies in annuli or vertical slots. But most of them have adopted the parallel flow in the same directions. For example, Habchi and Acharya [23]. Approaching to the heat pipe which has a main difference from these as reverse flow, an interesting paper by L.S.Yao and B.B.Rogers [24] is about mixed convection in an annulus of large aspect ratio which is an open thermosyphon. His analitical study proves that the mixed convection is unstable in many regions. The first study about closed annuli wity a reverse flow is about the vapour flow analysis by analitically and about the heat transfer characteristics by both analitically and experimentally.

In conventional heat pipes fluid mechanics and heat transfer problems are of three basic types:

(1) Vapour flow in the core region
(2) Liquid flow in the wick
(3) Interaction between liquid and vapour flows

Most of the analitical research on heat pipes has been done on the vapour flow because liquid flow in wick is complex and difficult to describe withan exact the theoretical model. It requires empirical relations obtained by experimentation.

5.2.1. VAPOR FLOW IN THE CORE REGION

The most recent prediction was made by Busse and Prenger [26] about compressibility effect and choking phenomena as a function of radial Reynolds number in conventional heat pipe. For the concentric case this problem is investigated by A.Faghri analitically and numerically such that.

Using the general Navier-Stokes equations that describe axially symmetric flow in a pipe with suction or injection

$$u \frac{\partial u}{\partial z} + V \frac{\partial u}{\partial r} = - \frac{1}{\rho} \frac{\partial p}{\partial z} + \nu \left(\frac{\partial^2 u}{\partial r^2} + \frac{1}{r} \frac{\partial u}{\partial r} \right) \qquad (5.2.1)$$

$$r \frac{\partial u}{\partial z} + \frac{\partial (Vr)}{\partial r} = 0 \qquad (5.2.2)$$

Where u is the axial component

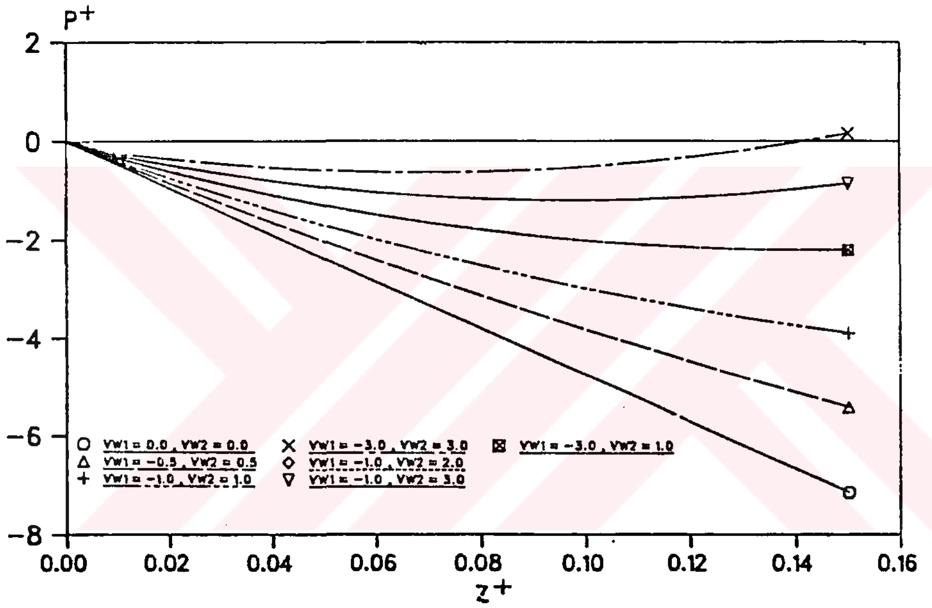
V is the radial component

ρ, ν are density and kinematic viscosity respectively.

He solved these equations numerically by taking the boundary conditions for the evaporator and the condenser region seperately. The boundary conditions are expressed as follows:

$$\begin{aligned} u\,(o,r) &= u_o\,(r) \\ u\,(z,R_1) &= \dot{u}\,(z,R_o)=0 \\ V\,(z,R_1) &= V_{1w}\,(z) \\ V\,(z,R_o) &= V_{ow}\,(z) \end{aligned} \qquad (5.2.3)$$

The analyse was limited to cases in which the suction and injection rates are small compared to the axial velocities. He solved for the cases of K=0.2, K=0.5 and K=0.8 where K= diameter of inner pipe/diameter of outer pipe and obtained the results such as shown in Fig.5.2.1 and 5.2.2 which I used same (K=0.5) value in our experiment. One of the conclusion is that the loss in the vapor pressure increases as K, ratio of pipe sizes, increases together with the radial Reynolds number in the condenser and evaporator region.



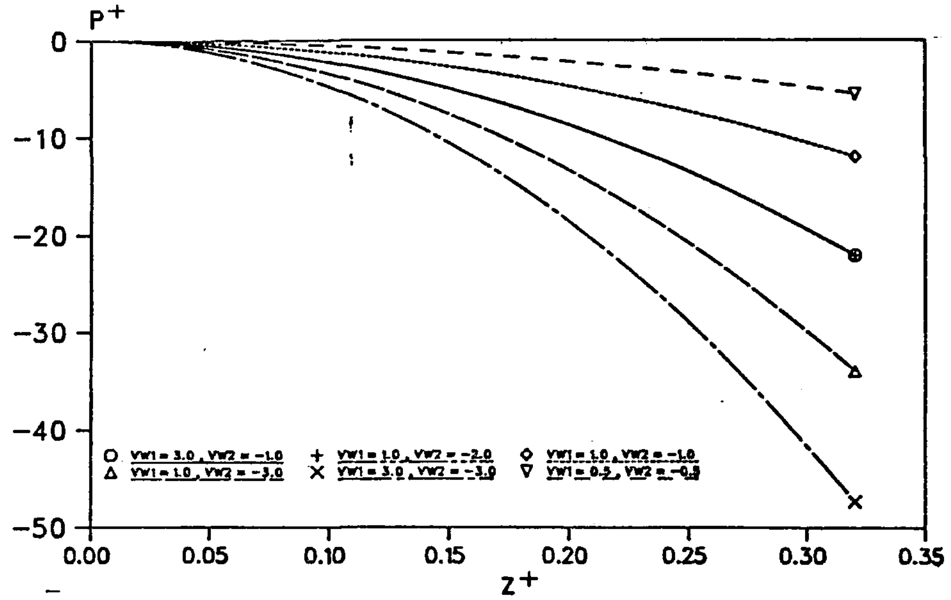
$$P^+ = \frac{P - P_o}{\rho \bar{u}_o^2}$$

$$Z^+ = \frac{\nu z}{R_o^2 \bar{u}_o^2}$$

$$VW1 = \frac{R_o V_{1w}}{\nu}$$

$$VW2 = \frac{R_o V_{ow}}{\nu}$$

Fig.5.2.1 Non dimensional pressure distribution along the condenser section for K=0.5 and different cooling load conditions



where

P_o: Initial pressure

$\bar{u}_o$: Initial mean axial velocity

see Fig.5.1.1 for the others.

Fig.5.2.2 Non dimensional pressure distribution along the evaporator section for K=0.5 and different cooling load conditions

He concludes that, for a radial Reynolds number less than one, viscous effects dominate and axial velocity profile is close to the Poiseuille flow conditions. For a radial Reynolds number greater than one, the evaporation and condensation cases becomes qualitatively different. The reversal flow may occur as a result of increasing of condensation pressure.

Heat transfer characteristics in two-phase closed conventional and concentric annular thermosyphons is the name of most recent paper about this subject . As seen in the name, it is also "wickless" heat pipe, but the first experimental investigation on this new type. This thesis differs from all above in usage of wick.

5.2.2. LIQUID FLOW IN THE WICK

This may also be analysed in three region as evaporator, adiabatic and condenser. But, usually neglecting the axial pressure losses in evaporator and condenser region comparing to that of adiabatic region, many researchers have dealt with this section of heat pipes. Investigations have been focused on wick structures creating different flow conditions. One of them by C.Y.Liu et al. has used the Brinkman-extended Darcy model instead of simple Darcy's law for an analitical analysis of cylindirical heat pipe. His assumptions are

(1) the flow is fully developed and steady, (2) Properties of the fluids and porous medium are constant, (3) The porous medium has a high permeability and is saturated with liquid. His governing equation determining the flow is:

$$\mu_1\left(\frac{d^2w}{dr^2} + \frac{1}{r}\frac{dw}{dr}\right) = \frac{dp_1}{dz} + \frac{\mu_1}{K} w \qquad (5.2.4)$$

Simple Darcy's law is:

$$\frac{dp_l}{dz} = \frac{\mu_l}{K} w \qquad (5.2.5)$$

The another approach takes the variation in porosity and intertial effects into account by A.Özgüç et al. for the planar heat pipes. He used the Darcy-Brinkman-Ergun extended model as shown below.

$$0 = -\frac{dp_l}{dz} - \frac{\mu_l}{K} w - \frac{\rho_l}{K_i} |w| w + \frac{\mu_l}{\varepsilon} \frac{d^2 w}{dy^2} \qquad (5.2.6)$$

where Z is axial coordinate

Y is perpendicular direction

K is the permeability

K_i is the inertial permeability

$$K = \frac{\varepsilon^3}{(1-\varepsilon)^2 g_1 S_w^2} \quad , \qquad K_i = \frac{\varepsilon^3}{(1-\varepsilon) g_2 S_w} \qquad (5.2.7)$$

s_w is the specific surface area of the wick

g_1 , g_2 are two empirical constants.

The analysis of fluid flow in the wick and the interaction between the fluid and vapour in conventional heat pipes is still applicable with small modifications to double-walled concentric heat pipe. The extended model, Darcy-Brinkman-Ergun model, may be used to evaluate the parameters sensitively as, in cylindirical coordinates,

$$0 = -\frac{dp_{l_1}}{dz} - \frac{\mu_l}{K_1} w_1 - \frac{\rho_l}{K_{i1}} |w_1| w_1 + \frac{\mu_l}{\varepsilon_1} \nabla^2 w_1 \qquad (5.2.8)$$

for the inner pipe

$$0 = -\frac{dp_{l_o}}{dz} - \frac{\mu_l}{K_o} w_o - \frac{\rho_l}{K_{io}} |w_o| w_o + \frac{\mu_l}{\varepsilon_o} \nabla^2 w_o \qquad (5.2.9)$$

for the outer pipe

where subscript I shows inner pipe

subscript o shows outer pipe, and $\nabla^2 = (\frac{\partial^2}{\partial r^2} + \frac{\partial}{r\,\partial r})$

Due to the form of wick structure on interaction sections, various boundary conditions may be written.

5.2.3. THERMAL RESISTANCES AND TEMPERATURE DROPS

The double-walled concentric heat pipe has a complexity compared to a conventional heat pipe since a new mechanism, radiation, effects also the performance. The thermal resistances and equivalent circuit are shown in Fig.5.1.1 and 5.2.3. Heat can enter and leave the pipe from the inner wall and outer wall by conduction, convection or radiation. Furthermore, temperature drops may occur by thermal conduction through the heat pipe walls in the evaporator and condenser regions, in the wicks, in the vapor liquid surface, as well as in the vapor core.

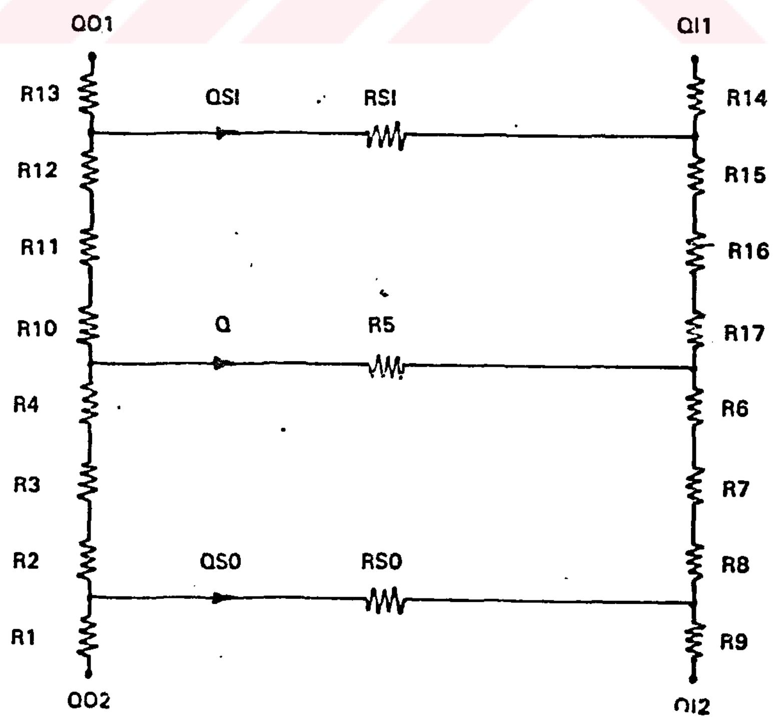


Fig.5.2.3 Equivalent thermal resistance in a double walled concentric heat pipe

5.3. EXPERIMENTAL APPARATUS AND PROCEDURE

To accomplish the objects of the experiment which were to design, and manufacture a new type of heat pipe, and to evaluate its performance, a concentric annular heat pipe was built.

5.3.1. CONSTRUCTION

Two copper pipes were selected as the heat pipe container. The resions of why this material was selected are its high thermal conductivity and compatibility with water and Freon. Its dimensions are given below.

Table 5.1. Heat Pipes Dimensions

Total length	460 mm
Evaporator length	110 mm
Adiabatic length	160 mm
Condenser length	190 mm
Outer pipe (o.d.)	55 mm
Outer pipe (i.d.)	50 mm
Inner pipe (o.d.)	25 mm
Inner pipe (i.d.)	23 mm

The outsides of the tubes on both ends were threaded and threaded brass caps were used to close the annuli.

For returning the fluid from condenser to evaporator 400 mesh stainless steel, 316 type, was used as wick by an amount of 4 layers on both inside of outer pipe and outside of inner pipe.

No extreme precautions other than ordinary cleaning of all components were necessary since operating temperature range are not higher then 400° C approximately.

One of the most difficult problems was the insertion of the wick into the outer pipe. To eliminate this problem a spring steel plate of the width 1/3 of the length of the adiabatic region,instead of spring wire,(but giving same performance) was inserted to retain the wick against the wall. This was done by rolling the plate over a mandrel giving a clearance in heat pipe. After the mandrell was inserted into the pipe, so that the plate is now in adiabatic region, the plate tension was released, and the mandrel then removed. Streching the plate manually caused the wick to hold against the wall. It was easy to wrap the wick over the inner pipe since the process consists of only rolling or wrapping the wick over pipe and a thin steel wire over the wick to prevent unfolding.

There are several methods of fitting of end caps such as by argon arc welding, brazing, soldering, electron beam welding, or by using threated caps. The latter,i.e. threaded cap fitting, used in this experiment was applied on both ends of two pipes in conjuction with TPLE band for providing the sealing. See Fig.5.3.1.

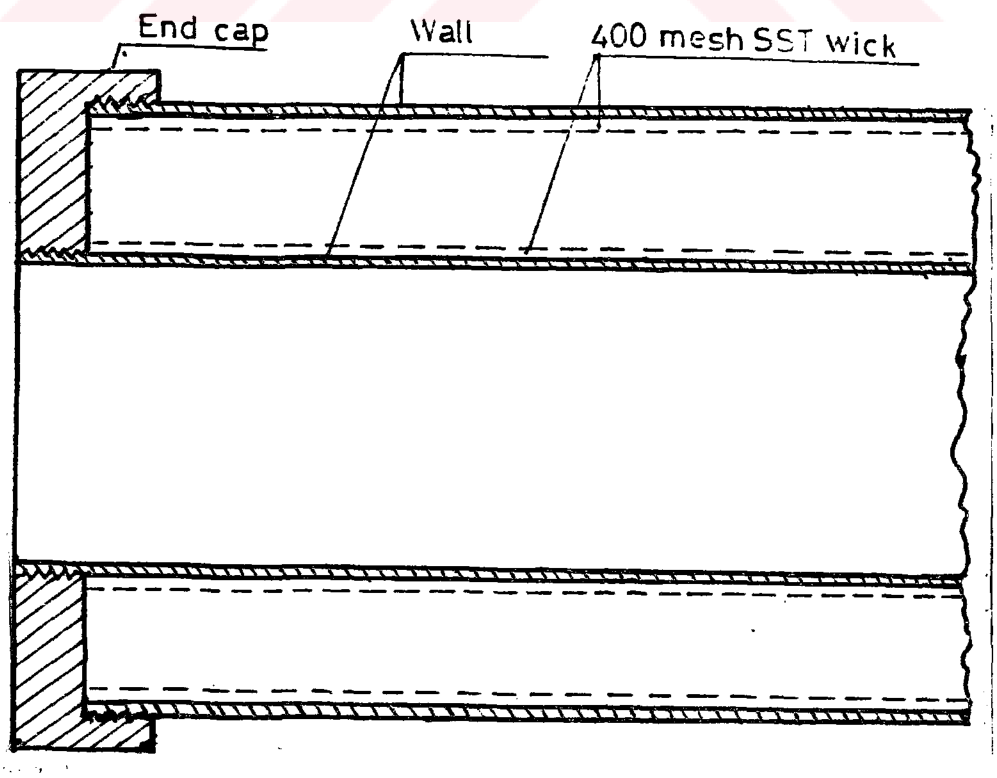


Fig.5.3.1 Fitting of wick and end cap

The heat source used in the inner pipe and the outer pipe of the heat pipe has nicrome electric heater wires. The electric heater of the resistivity 19.1 W/m was wrapped by passing through the holes of electrical isolator of the rod type (but thermally conductive). Then this refractory rod was inserted into the inner pipe. The same electric heater wire was also used for the outer pipe. It was coiled around the outer pipe by bead type electrical insulator. After the aliminium foil was wrapped over, the heater both of the evaporator and adiabatic sections of the heat pipe were isolated thermally from the surrounding. Seperate a-c power supplies were used to supply power to the inner and outer evaporator section heaters. The variac for varying power input, the voltmeter and the ampermeter for measuring power input were used for each of the inner and outer heaters.

The heat sink, on the other end of heat pipe, is composed of two water cooler placed in and out of the pipe. In the inner of the heat pipe, the plastic pipe of considerably smaller diameter than the inner pipe was placed through the condenser length approximately. So, the cooling water will enter to the inner pipe of the heat pipe through the inside of the plastic pipe, then it will return through the annuli between plastic and copper pipes taking the heat rejected from the inner pipe. The outer heat sink is a cooling jacket designed to envelop the condenser section of the outer pipe. The outer condenser made of glass material is a counter flow heat exchanger.

To seperate the evaporator and the condenser section of the inner pipe an amount of glasswool was pressed in the adiabatic section, and a rubber stopper was placed on each end of the adiabatic section to prevent sealing. Then their edges were sticked on the inside of the pipe.

All of the heat sinks were provided with cooling water from a constant head tank which was feeded by a small pump assembled in a hydraulic experimental set up together with two tanks (emptying or drainage and flow measuring tank).

The wall temperatures were measured by chromel-alumel thermo couples located on the surface of the outer pipe. In order to attach the thermocouples to the copper heat pipe, the small cavities were hollowed out of the outer pipe surface at designated locations. After pressing them into these cavities, they were sticked on the pipe. However, this process would create some errors in the measuring of temperatures. The thermocouples were then connected to a device putting them in order manually and to a digital readout device.

5.3.2. PROCEDURE

Since the constructed heat pipe was neither a low nor a high temperature heat pipe it was no needed an extreme cleaning procedure and a material outgassing process.

A simple heat pipe filling rig layout was set up as shown in Fig.5.3.2. After the annuli of the heat pipe was evacuated by the rotary vane pump to 10^{-3} torr which is near the standart value 10^{-4} torr. Then, a measured amount of pure water which is slight excess than the amount necessary to saturate the wick was fed into the annuli of the pipe. Finally, it was disconnected from the layout crimping the connection copper pipe of diameter 5 mm between the jaws of a standart vice as shown in Fig.5.3.3.

The heat pipe was then place on the test set up built up previously. The shematic of the set up is shown in Fig.5.3.4.

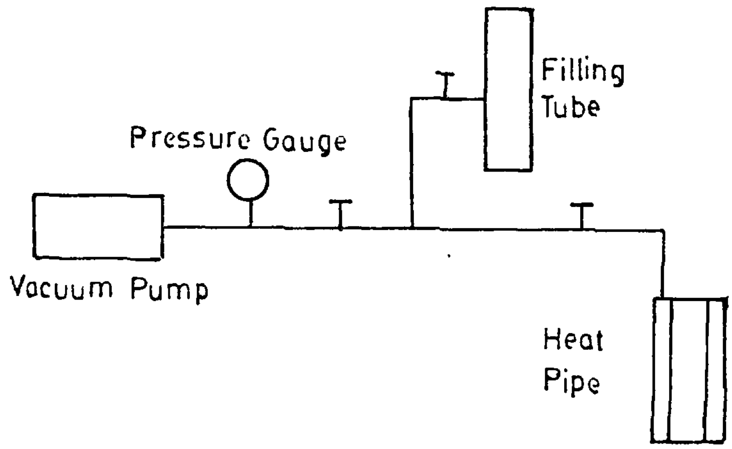


Fig.5.3.2 Heat pipe filling

Before testing it as double-walled concentric heat pipe, it was operated as a concentric thermosyphon to saturate the wicks fully for 3 hours. So, it was ready to test it as a heat pipe.

The first experiment was to find out the temperature distribution over the outer pipe while power input was given by both the inner and outer electrical heaters, and also both, the inner and outer heat sink was in cycle. During the experiment the temperatures at six points over the pipe, inlet and outlet cooling water temperatures, power input and volumetric flowrate was measured and noted at an intervals of 5 minutes for 30 minutes.

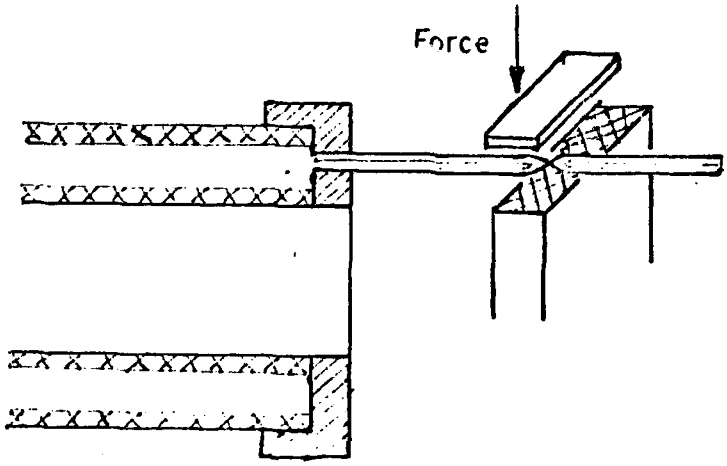


Fig.5.3.3 Sealing of the heat pipe by crimping method

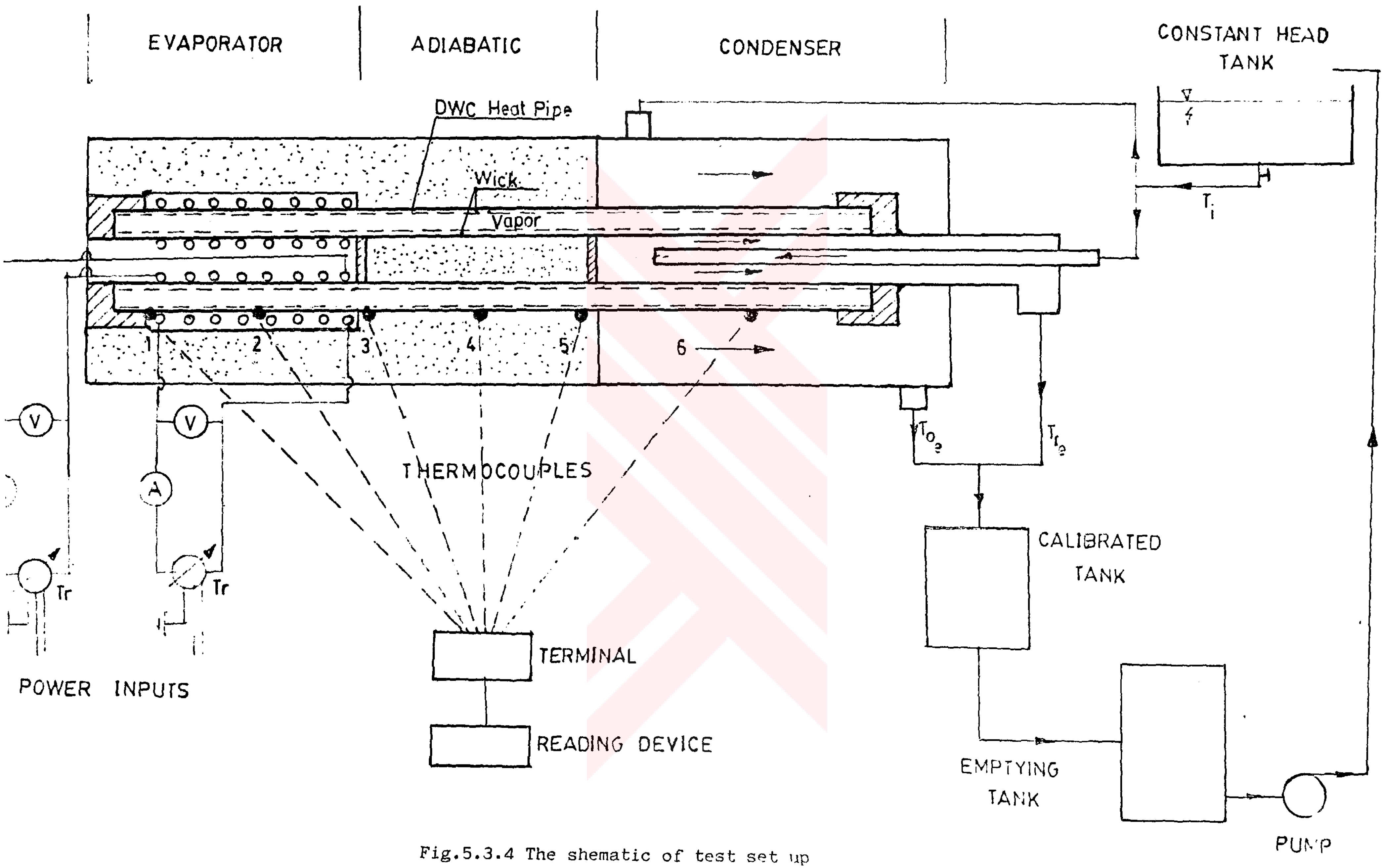


Fig.5.3.4 The shematic of test set up

The second experiment was the same of the first experiment except that no heat input to the evaporator section and no heat output from the condenser section of the inner pipe of the heat pipe were applied. This case may be considered as a standart heat pipe. So, it was thaught that the performance of the double-walled concentric heat pipe would be compared with that of standart pipe, even the conditions were not completely proper. The same parameters were recorded at 5 minutes intervals for 30 minutes. The results are given in section 5.3.3.

5.3.3 RESULTS AND DISCUSSION

Table 5.2 shows the temperature records along the heat pipe including cooling water inlet and outlets in a time interval of 5 min. for 30 min. The temperature distribution of the pipe and variation of the cooling water temperature for two different cases are shown in Fig.5.3.5 and 5.3.6.

It was initially found that the limits other than the wicking limit had no effection on the performance. Due to the wicking limit, an appropriate value of heat power was selected. Firstly, 30 watts to the outer heater and 15 watts to the inner heater, i.e. totally 45 watts, were given, and then while no heat was input to the inner side, the heat power on the outer side was still 30 watts.

As shown in Figures, there is a large temperature gradient between the evaporating and the adiabatic sections in the beginning of the heat transfer process. The temperature rise slowed down considerably at a time of 30 minutes when heat conduction takes place approaching the steady state. In the first experiment, it reached to steady state as the temperature over the pipe is in between 83 and 55. In the second case, the range was 65-35.

Table 5.2. Temperature distribution and cooling water inlet and outlet temperature

Thermocouple locations

TIME (Min)	1	2	3	4	5	6	T_i	Te_O	Te_I
0	25.5	26.3	24.6	23.0	23.0	23.0	20.5	–	–
5	42.0	44.7	34.4	28.0	25.6	23.6	x	20.7	20.6
10	73.1	72.0	57.5	43.1	37.0	28.7	x	21.8	20.9
15	83.0	81.4	75.0	64.6	60.2	42.5	x	22.9	21.5
20	83.2	81.4	75.0	65.0	61.6	53.2	x	23.5	22.0
25	83.5	81.2	75.0	65.8	62.1	54.0	x	23.6	22.1
30	83.5	81.0	75.2	65.8	62.4	54.5	x	23.6	22.2

Q_O= 30 w Q_I= 15 w $\dot{m}_O = \dot{m}_I$ = 6.3 kg/h (Cooling water)

Thermocouple locations

TIME (Min)	1	2	3	4	5	6	T_i	Te_O	Te_I
0	23.6	24.5	22.8	22.7	22.7	22.7	20.5	–	–
5	40.0	41.3	37.8	32.4	29.1	23.0	x	20.6	–
10	57.2	56.4	46.7	39.5	35.0	25.8	x	21.3	–
15	66.1	63.0	55.4	46.0	42.1	30.0	x	22.6	–
20	65.0	63.8	56.0	48.3	44.8	32.6	x	22.9	–
25	65.0	63.5	56.4	49.8	46.0	34.8	x	23.3	–
30	65.0	63.6	57.2	50.0	46.8	35.1	x	23.3	–

Q_O= 30 w Q_I= 0 w $\dot{m}_O$ = 6.3 kg/h $\dot{m}_I$ = 0 kg/h (Cooling water)

Subscripts

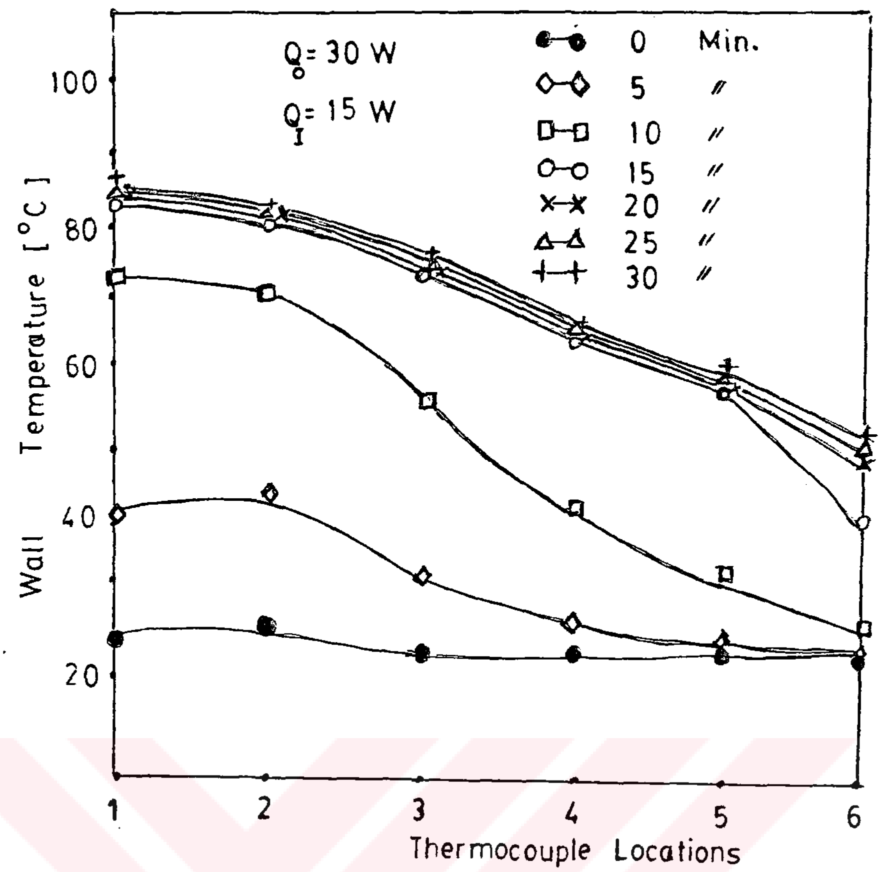
e: exit   o: outer

i: inlet   I: inner

Unfortunately, there is a large temperature difference between the evaporator and condenser section. But the temperature distribution over the adiabatic section may be considered as an indicator of the operation of the heat pipe. However, it did not operate succesfully since the temperature was not nearly constant. The errors may be caused by the filling process, purity of the working fluid, fluid inventory, etc.

Although the heat input in the first case was greater than that given in the second case, the temperature profiles look likes similar. The temperature falling down the evaporator to the condenser was approximately 35-40 % which is the same in both cases. This forces us to think of that double-walled concentric heat pipe has better performance than the second, accepting as if standart heat pipe, case.

The efficiency of the pipe is calculated from the ratio of heat input to heat output. In steady state, it was found as η=0.69 as the heat pipe was operating as a standart heat pipe, and η=0.78 for the double - walled concentric heat pipe. That may also be an indicator of that the latter well operates.



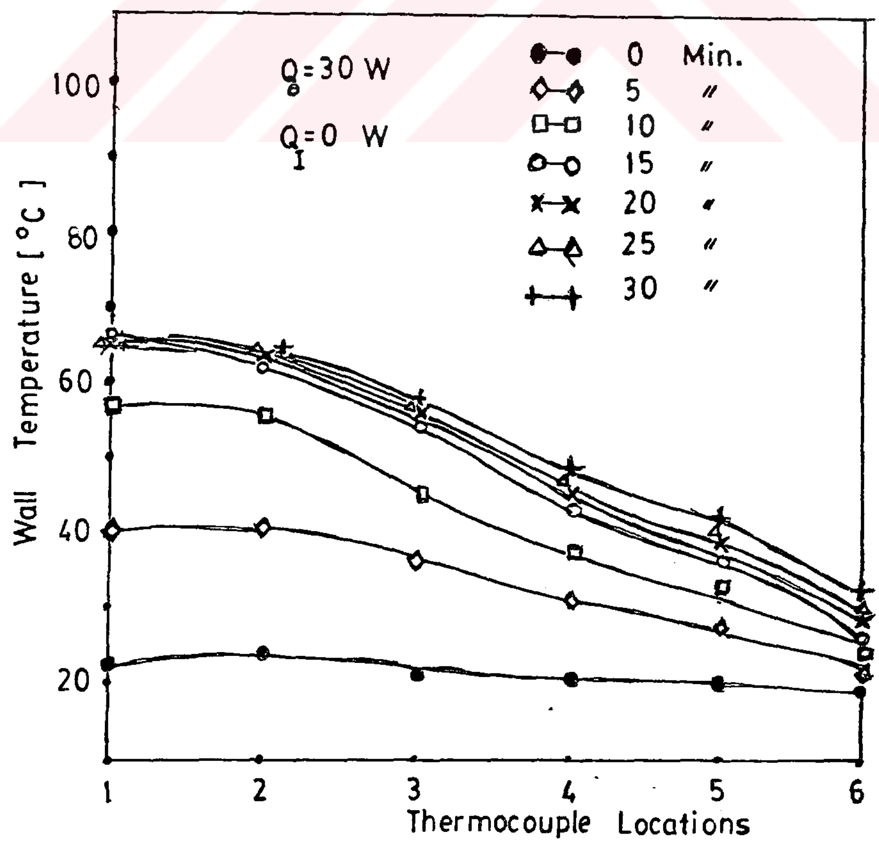


Fig.5.3.5. Temperature distribution over the heat pipe

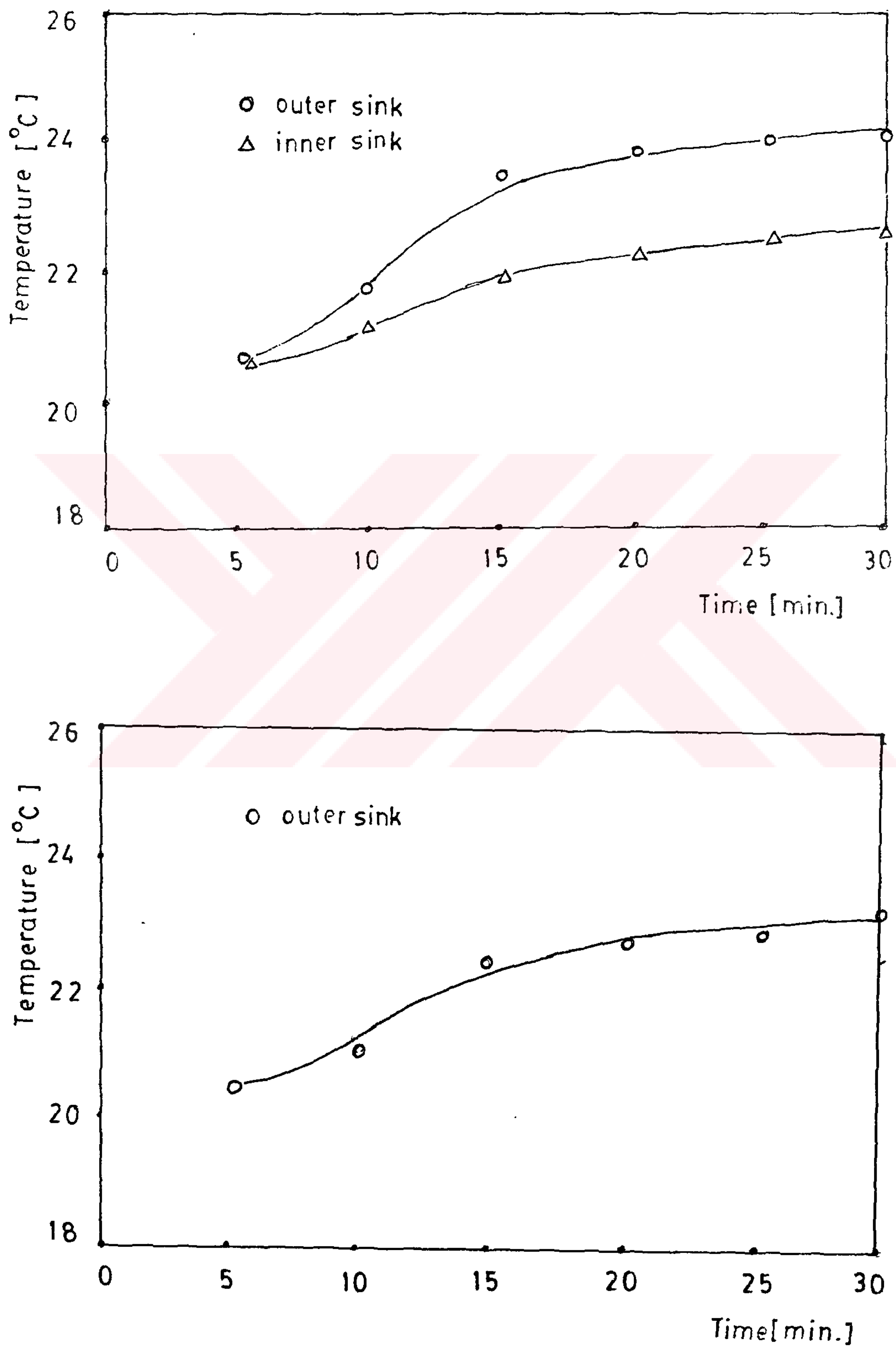


Fig.5.3.6 Temperature variations in cooling waters

CH.6. CONCLUSION

An experimental study of a water heat pipe with a new geometry, called as " double-walled concentric heat pipe", was conducted in a temperature range of 55 to 83°C to observe the superiority over the standart type. As pointed out earlier, its two main advantages are the increase in the surface area exposed to heat transfer and the increase in the pumping power of the liquid due to greater wick volume. This is evident from the theoretical analyse and, even there was no netness, from experimental observation.

However, the errors may caused from many reasons. They were devoted to those; the purity of the water used in the heat pipe as working fluid might be insufficient, a rotary vane pump which gives a vacuum up to 10^{-3} torr was used instead of molecular diffusion vacuum pump which gives standart amount of 10^{-4} torr, and there was a possibility of remaining of the air during the filling even it was small.

The annular heat pipe which has a larger surface area and larger wick volume can be aimed at any application in which there is a fixed space, since its outer dimensions are constant although the inner sizes, such as vapour area, are changable.

REFERENCES

[1] FAGHRİ, A., Vapor Flow Analysis in a Double-walled concentric Heat Pipe Numerical Heat Transfer, vol.10, pp.583-595, 1986.

[2] DUNN , P.D. and D.A. Reay, Heat Pipes, Pergamon press, 1976.

[3] GROOVER,G.M., COTTER, T.P., and ERIKSON, G.F., structures of Very High Thermal Conductance, J.Appl. Phys., vol.35, pp.1190-1191, 1964.

[4] FUKANO, T., KADOGUCHI, K., and TIEN, C.L., Experimental Study on the Critical Heat Flux at the Operating Limit of a Closed Two-Phase Thermosyphon. JSME, 53(B)-487, 1065-1071 (1986).

[5] NGHYEN-CHI, H., et al. A F A A 14 th thermophysics Conf., pp. 79-1106 (1979).

[6] DOBRAN, F., Suppression of the Sonic Heat Transfer Limit in High-Temperature Heat Pipes. ASME paper, vol. 111 pp. 605-610, 1989.

[7] HART, J.E., Observation of Complex Oscillations in a Closed Thermosyphen. Asme paper, vol.107, pp. 833-839, 1985.

[8] LOCK, G.S.H. and LIU, Y., The Effect of Geometry on the Performance of the Closed Tube Thermosyphon at Low Rayleigh Numbers. Int.J.Heat Mass Transfer. Vol.32, no.6, pp.1175-1182, 1989.

[9] GÜRSES, Ç., Su Doldurmalı Isı Boruları Performansında Eğim Açısunun Etkisi. 9 Eylül Üniversitesi, FBE/MAK-86 AR019

[10] SHISHIDO, I., et al., Measurement of the Working Fluid Distribution Within a Heat Pipe Wick.ASME. C, 96, 84 (1984).

[11] KEÇECIOĞLU, I., RUBINSKY, B., Int.J.Heat Mass Transfer Vol.32, No.6, yy.1111-1130, 1989.

[12] LAI, F.C. and KULACKI, F.A., Thermal Dispersion Effects on Won-Darcy Convection over Horizontal Surfaces in Saturated Porous Media. Int.J. Heat Mass Transfer. Vol. 32, No.5, pp.971-976, 1989.

[13] ÖZGÜÇ, A.F., et al., Non-Darcian Effects on the Heat-Pipe Wicking Limit. [Unpublished Paper].

[14] KIM, R.H., Experimental Investigation of a Cornstalk Wick Heat Pipe. Int. Comm. Heat Mass Transfer Vol.16, pp. 367-377, 1989.

[15] KOBAYASHI, Y., Heat Pipe Thermodynamic Cycle and Its Applications. Journal of Solar Energy engineering Vol.107, pp.153-159, 1985.

[16] VASILIEV, L.L., Heat Pipe Research and Development in the U.S.S.R. Heat Recovery Systems CHP Vol.9, No.4, pp. 313-333, 1989.

[17] LIU., et al., Flow in the Adiabatic Section of a Heat Pipe. Int. Comm. Heat Mass Transfer, Vol.16, pp.79-88, 1989

[18] BANKSTON, C.A., Smith, J.H. Incompressible Laminar Vapour Flow in Cylindirical Heat Pipes. ASME Paper NV. 71-WA/H -15,1971.

[19] DAVIS, W.R., FERRELL, J.K.Evaporative Heat Transfer of Liquid Potassium in Porous Media. A.I.A.A./A.S.M.E. 1974. Thermophysics and Heat Transfer Conference, Boston, Mass. July 1974.

[20] ARPACI, V., LARSEN, P.S. Convection Heat Transfer, Prentice-Hall, Inc. 1984.

[21] BUSSE, C.A. Theory of Ulfimate Heat Transfer Limit of Cylindirical Heat Pipes. Inst.J.Heat and Mass Transfer. Vol16, pp. 169-186, 1983.

[22] FAGHRİ, A., et al., Heat Transfer Characteristics in Two-Phase Closed Conventional and Concentric Annular Thermosyphons. Journal of Heat Transfer, Vol.111, pp. 611-618, 1989.

[23] HABCHI,S., and ACHARYA, S., Laminar Mixed Convection in a Symmetrically or Asymmetrically Heated Vertical Channel. Numerical Heat Transfer, Vol.9, pp.605-618, 1986.

[24] YAO, L.S., and ROGERS, B.B., Mixed Convection in an Annulus of Large Aspect Ratio. Journal of Heat Transfer, Vol.111, pp.683-689, 1989.

[25] NARAYANA, K.B., Vapor Flow Characteristics of Slender Cylindirical Heat Pipes a Numerical Approach. Numerical Heat Transfer, Vo.10, pp.79-93, 1986.

[26] BUSSE, C.A., and PRENGER , F.C. Numerical Analysis of the Vapor Flow in Cylindirical Heat Pipes, PRoc.5 th. Heat Pipe Conf., Japon, May 1984.

APPENDIX A

WORKING FLUID PROPERTIES

HELIUM

Temp. ^{o}C	Latent Heat kJ/kg	Liquid Density kg/m^3	Vapour Density kg/m^3	Liquid Thermal Conductivity $W/m^{o}C$	Liquid Viscos. $cP \times 10^2$	Vapour Viscos. $cP \times 10^3$	Vapour Press. Bar	Vapour Specific Heat $kJ/kg^{o}C$	Liquid Surface Tension $N/m \times 10^3$
-271	22.8	148.3	26.0	1.81	3.90	0.20	0.06	2.045	0.26
-270	23.6	140.7	17.0	2.24	3.70	0.30	0.32	2.699	0.19
-269	20.9	128.0	10.0	2.77	2.90	0.60	1.00	4.619	0.09
-268	4.0	113.8	8.5	3.50	1.34	0.90	2.29	6.642	0.01

FREON 113

Temp. °C	Latent Heat kJ/kg	Liquid Density kg/m^3	Vapour Density kg/m^3	Liquid Thermal Conductivity W/m°C	Liquid Viscos. cP	Vapour Viscos. cPx10^2	Vapour Press. Bar	Vapour Specific Heat kJ/kg°C	Liquid Surface Tension N/mx10^2
-50	173.0	1720	0.15	0.120	2.300	0.85	0.01	0.600	2.86
-30	167.8	1683	0.32	0.119	1.604	0.90	0.03	0.613	2.60
-20	165.4	1664	0.46	0.118	1.323	0.92	0.05	0.619	2.47
-10	163.2	1643	0.77	0.118	1.108	0.94	0.09	0.626	2.34
0	160.6	1621	1.26	0.117	0.942	0.97	0.12	0.632	2.21
10	158.0	1599	1.95	0.108	0.812	0.99	0.19	0.644	2.08
20	155.2	1576	3.00	0.098	0.707	1.02	0.37	0.656	1.96
30	152.3	1553	4.34	0.097	0.622	1.04	0.55	0.664	1.84
40	149.2	1529	6.02	0.095	0.553	1.07	0.79	0.669	1.73
50	145.9	1503	8.79	0.094	0.502	1.09	1.11	0.674	1.62
70	139.4	1452	14.34	0.091	0.401	1.13	2.04	0.691	1.40

ACETONE

°C	kJ/kg	kg/m^3	kg/m^3	W/m°C	cP	cPx10^2	Bar	kJ/kg°C	N/mx10^2
-40	660.0	860.0	0.03	0.200	0.800	0.68	0.01	2.00	3.10
-20	615.6	845.0	0.10	0.189	0.500	0.73	0.03	2.06	2.76
0	564.0	812.0	0.26	0.183	0.395	0.78	0.10	2.11	2.62
20	552.0	790.0	0.64	0.181	0.323	0.82	0.27	2.16	2.37
40	536.0	768.0	1.05	0.175	0.269	0.86	0.60	2.22	2.12
60	517.0	744.0	2.37	0.168	0.226	0.90	1.15	2.28	1.86
80	495.0	719.0	4.30	0.160	0.192	0.95	2.15	2.34	1.62
100	472.0	689.6	6.94	0.148	0.170	0.98	4.43	2.39	1.34
120	426.1	660.3	11.02	0.135	0.148	0.99	6.70	2.45	1.07
140	394.4	631.8	18.61	0.126	0.132	1.03	10.49	2.50	0.81

WATER

Temp. °C	Latent Heat kJ/kg	Liquid Density kg/m³	Vapour Density kg/m³	Liquid Thermal Conductivity W/m°C	Liquid Viscos. cP	Vapour Viscos. cPx10²	Vapour Press. Bar	Vapour Specific Heat kJ/kg°C	Liquid Surface Tension N/mx10²
20	2448	998.2	0.01	0.612	1.00	0.96	0.02	1.85	7.40
40	2402	992.3	0.05	0.630	0.65	1.04	0.07	1.86	6.96
60	2359	983.0	0.14	0.649	0.47	1.12	0.20	1.87	6.62
80	2309	972.0	0.29	0.668	0.36	1.19	0.47	1.88	6.26
100	2258	958.0	0.60	0.680	0.28	1.27	1.01	1.88	5.89
120	2200	945.0	1.12	0.682	0.23	1.34	2.02	1.89	5.50
140	2139	928.0	1.99	0.683	0.20	1.41	3.90	1.90	5.06
160	2074	909.0	3.27	0.679	0.17	1.49	6.44	1.91	4.66
180	2003	888.0	5.16	0.669	0.15	1.57	10.04	1.92	4.29
200	1967	865.0	7.87	0.659	0.14	1.65	16.19	1.93	3.89

FLUTEC PP9

°C	kJ/kg	kg/m³	kg/m³	W/m°C	cP	cPx10	Bar	kJ/kg°C	N/mx10²
-30	103.0	2098	0.01	0.060	5.77	0.82	0.00	0.80	2.36
0	98.4	2029	0.01	0.059	3.31	0.90	0.00	0.87	2.08
30	94.5	1960	0.12	0.057	1.48	1.06	0.01	0.94	1.80
60	90.2	1891	0.61	0.056	0.94	1.18	0.03	1.02	1.52
90	86.1	1822	1.93	0.054	0.65	1.21	0.12	1.09	1.24
120	83.0	1753	4.52	0.053	0.49	1.23	0.28	1.15	0.95
150	77.4	1685	11.81	0.052	0.38	1.26	0.61	1.23	0.67
180	70.8	1604	25.13	0.051	0.30	1.33	1.58	1.30	0.40
225	59.4	1455	63.27	0.049	0.21	1.44	4.21	1.41	0.01

SODIUM

Temp. °C	Latent Heat kJ/kg	Liquid Density kg/m^3	Vapour Density kg/m^3	Liquid Thermal Conductivity W/m^oC	Liquid Viscos. cP	Vapour Viscos. cPx10	Vapour Press. Bar	Vapour Specific Heat kJ/kg^oCx10	Liquid Surface Tension N/mx10
500	4370	828.1	0.003	70.08	0.24	0.18	0.01	9.04	1.51
600	4243	805.4	0.013	64.62	0.21	0.19	0.04	9.04	1.42
700	4090	763.5	0.050	60.81	0.19	0.20	0.15	9.04	1.33
800	3977	757.3	0.134	57.81	0.18	0.22	0.47	9.04	1.23
900	3913	745.4	0.306	53.35	0.17	0.23	1.25	9.04	1.13
1000	3827	725.4	0.667	49.08	0.16	0.24	2.81	9.04	1.04
1100	3690	690.8	1.306	45.08	0.16	0.25	5.49	9.04	0.95
1200	3577	669.0	2.303	41.08	0.15	0.25	9.59	9.04	0.86
1300	3477	654.0	3.622	37.08	0.15	0.27	15.91	9.04	0.77

LITHIUM

°C	kJ/kg	kg/m^3	kg/m^3	W/m^oC	cP	$cPx10^2$	Bar	kJ/kg^oC	N/mx10
1030	20500	450	0.005	67	0.24	1.67	0.07	0.532	2.90
1130	20100	440	0.013	69	0.24	1.74	0.17	0.532	2.85
1230	20000	430	0.028	70	0.23	1.83	0.45	0.532	2.75
1330	19700	420	0.057	69	0.23	1.91	0.96	0.532	2.60
1430	19200	410	0.108	68	0.23	2.00	1.85	0.532	2.40
1530	18900	405	0.193	65	0.23	2.10	3.30	0.532	2.25
1630	18500	400	0.340	62	0.23	2.17	5.30	0.532	2.10
1730	18200	398	0.490	59	0.23	2.26	8.90	0.532	2.05

APPENDIX B

EXPERIMENTAL DETERMINED WICK PROPERTIES

Table B-1. Experimentally Determined Wick Properties

No.	Wick Type and Description	Test Liquid	Porosity	$r_p \times 10^{-6}$ m	Method	$K \times 10^{-10}$ m^2	Reference
1	Screen, SST, 120 mesh	H_2O		281	R1	3.02	D
2	Screen, SST, 120 mesh, oxidized at 600°C	H_2O		190.9	R1	1.73	D
3	Screen, SST, 200 mesh	H_2O	.733	58	R4	.52	B
4	Screen, SST, 200 mesh	C_6H_6	.733	57	R4		B
5	Screen, SST, 200 mesh	CH_3OH	.733	56	R4		B
6	Screen, SST, 400 mesh	H_2O		29	R3		C
7	Screen, SST, 400 mesh	C_2H_5OH		30	R3		C
8	Screen, SST, 80 x 700, Double Dutch Twill	$(CH_3)_2CHOH$		< 40	R4	.075	J
9	Screen, SST, 165 x [illegible], Double Dutch Twill	$(CH_3)_2CHOH$		< 18	R4	.044	J
10	Screen, SST, 200 x [illegible], Double Dutch Twill	$(CH_3)_2CHOH$		[illegible]	R4	.024	J
11	Screen, SST, 250 x 1400, Double Dutch Twill	$(CH_3)_2CHOH$		< 12	R4	.011	J
12	Screen, SST, 325 x 2300, Double Dutch Twill	$(CH_3)_2CHOH$		< 10	R4	.006	J
13	Screen, SST, 375 x 2300, Double Dutch Twill	$(CH_3)_2CHOH$		< 8	R4	.006	J
14	Screen, SST, 450 x 2750, Double Dutch Twill	$(CH_3)_2CHOH$		< 7	R4	.006	J
15	Screen, SST, 670 x [illegible], Reverse Dutch Twill	$(CH_3)_2CHOH$		[illegible]	[illegible]		J
16	Screen, SST, 720 x 140, Reverse Dutch Twill	$(CH_3)_2CHOH$		< 21	R4		J
17	Screen, SST, 850 x 155, Reverse Dutch Twill	$(CH_3)_2CHOH$		< 20	R4		J
18	Screen, Nickel, [illegible]	H_2O		1813	R1	5.81	D

Table B-2 Experimentally Determined Wick Properties (Continued)

No.	Wick Type and Description	Test Liquid	Porosity	$r_p \times 10^6$ m	Method	$K \times 10^{10}$ m^2	Reference
19	Screen, Nickel, 50 mesh, sintered @ 600°C	H_2O		580.2	R1	6.47	D
20	Screen, Nickel, 50 mesh, sintered	H_2O	.625	305	R1	6.63	A
21	Screen, Nickel, 100 mesh, sintered	H_2O	.679	<131 [illegible]	R1	1.52	A
22	Screen, Nickel, 100 mesh, sintered	H_2O	.678	84	R1		A
23	Screen, Nickel, 200 mesh, sintered	H_2O	.676	64	R1	.77	A
24	Screen, Nickel, 500 mesh	H_2O	.60	12.5	R4		J
25	Screen, Nickel, 1000 mesh	H_2O	.48	13.0	R4		J
26	Screen, Nickel, 1000 mesh	CH_3OH	.48	16.6	R4		J
27	Screen, Nickel, 1500 mesh	H_2O	.36	6.7	R4		J
28	Screen, Nickel, 2000 mesh	H_2O	.22	6.1	R4		H
29	Screen, Nickel, 2000 mesh	CH_3OH	.22	5.3	R4		H
30	Screen, Copper, 60 mesh	H_2O		481	R1	4.20	D
31	Screen, Copper, 60 mesh, oxidized	H_2O		[illegible]	R1	2.38	D
32	Screen, Phosphor Bronze, 120 mesh	C_2H_5OH		102	R3		C
33	Screen, Phosphor Bronze, 120 mesh	H_2O		105	R3		C
34	Screen, Phosphor Bronze, 200 mesh	C_2H_5OH		54	R3		C
35	Screen, Phosphor Bronze, 200 mesh	H_2O		60	R3		C
36	Screen, Phosphor Bronze, 250 mesh	C_2H_5OH		51	R3		C

Table B-3 Experimentally Determined Wick Properties (Continued)

No.	Wick [illegible] Description	Test Liquid	Porosity	$r_p \times 10^6$ m	Method	$K \times 10^{10}\ m^2$	Reference
37	Screen, Phosph[illegible], 250 mesh	H_2O	—	48	R3		C
38	Screen, Phospho[illegible], 270 mesh	C_2H_5OH		41	R3		C
39	Screen, Phospho[illegible], 270 mesh	H_2O		43	R3		C
40	Screen, Phosph[illegible], 325 mesh	C_2H_5OH		32	R3		C
41	Screen, Phosphor [illegible], 325 mesh	H_2O		33	R3		C
42	Felt, Sintered, [illegible]	H_2O	.822	110	R1	11.61	A
43	Felt, Sintered, [illegible]	H_2O	.916	94	R1	5.46	A
44	Felt, Sintered, [illegible]	H_2O	.808	65	R1	1.96	A
45	Felt, 347 SST, C38 (FM 134)	H_2O	.35	49	R1	1.04	L
46	Felt, 347 SST, [illegible] 123)	H_2O	.35	42.5	R1	.64	L
47	Felt, SST, A-8 (FM [illegible], 1106, 1111)	H_2O	[illegible]	26.5	R1	.56	L
48	Felt, SST, A-8 (FM 1102, 1107, 1112)	H_2O	.80	13.5	R1	.087	L
49	Felt, SST, A-8 (FM 1103, 1108)	H_2O	.70	7.5	R1	.028	L
50	Felt, SST, A-8 (FM 1104, 1109)	H_2O	.60	5	R1	.013	L
51	Felt, SST, A-8 (FM 1105, 1110)	H_2O	.40	2.3	R1	.0009	L
52	Felt, 430 SST, B-[illegible] [illegible] 1305, 1309)	H_2O	[illegible]	260	R1	30.6	L
53	Felt, 430 SST, B-[illegible] FM 1302, 1306, 1310)	H_2O	[illegible]	191	R1	13.6	L
54	Felt, 430 SST, [illegible] 1303, 1307, 1311)	H_2O	[illegible]	[illegible]	R1	6.44	L

APPENDIX C

THERMAL CONDUCTIVITY OF HEAT PIPE CONTAINER AND WICK MATERIALS

Material	Thermal Conductivity ($W/m^{o}C$)
Aluminium	205
Brass	113
Copper (0 - 100^{o}C)	394
Glass	0.75
Nickel (0 - 100^{o}C)	88
Mild Steel	45
Stainless Steel (Type 304)	17.3
Teflon	0.17

BIOGRAPHY

The author, Erol Abit, was borned in Eğrikuyu/Konya on date 1964. After he has completed the educations of Elementary, Medium and Industrial School (Motor dept.) in Akşehir, he has enrolled to Mechanical Engineering Depatment of Middle East Technical University (M.E.T.U.) Gaziantep Extension Campus, in 1981. He has worked in the industry for four months in 1988 after his graduation at the end of 1987. Then, in 1988, he has enrolled to İstanbul Technical University (İ.T.U.) Institute of Natural Science and Technology Energy Programme of Mechanical Engineering Department. By now, he has been working as a research assistant in Yıldız University (Y.U.) Kocaeli Engineering Faculty.