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Experimental Investigation Of The Performance Of Gravity Assisted Heat Pipes Having Wicks Filled With Porous Medium

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ABSTRACT

The aim of the present work is to investigate experimentally the heat transfer performance of gravity assisted heat pipes having wicks filled with porous medium, using fluorocarbon R-11 refrigerant as the working fluid. The particles of the porous media used in the experiments were made of nickel, mild steel, granite, marble, ceramic, glass, and gravel covering a wide range of solid thermal conductivities and particle diameters. The effects of the evaporator heat flux, and evaporating vapor pressure on both the evaporator heat transfer coefficient, h, and the overall thermal conductance, C, of the heat pipe (tested heat transfer parameters) were also studied. The present results for smooth heat pipe (without porous medium) show good agreement with the available previous published results. For porous wicked heat pipe, it was found that the tested heat transfer parameters were strongly improved with increasing particle to fluid thermal conductivity ratios (for same packing material diameter), and with decreasing particle to heat pipe diameter ratios (for same material). Moreover, it was concluded that, for porous wicked heat pipe using nickel packing material, the average maximum value of (h/h_{sm}) is about 2.40, and of (C/C_{sm}) is about 2.10. These values are obtained for smaller particle to heat pipe diameter ratio ($d/D = 0.0912$). Finally, correlations for the heat transfer parameters of the heat pipe (h & C) as functions of the different tested parameters were deduced.

Keywords

Heat transfer performance, heat pipe, porous medium, porous wick, thermal conductivity.

1. INTRODUCTION

The heat pipe is an efficient heat conductor which is used to transfer heat from one location to another with small temperature gradient. It is divided into three sections, evaporator, adiabatic, and condenser. Heat is added through the evaporator section, where the working fluid is vaporized. The vapor then passes through the adiabatic section. The vapor releases its latent heat and gets liquefied in the condenser. The circulation is completed by the return condensed liquid to the evaporator through the wick or packed bed under the driving action of capillary forces.

Due to their high efficiency and reliability, heat pipes have been used in different applications including energy conversion systems, cooling of nuclear reactors, cooling of electronic equipment and high performance space applications [1&2]. Over the past 40 years, extensive studies have been conducted in order to provide a deep understanding of the heat pipe operation and hence to improve its performance. Faghri et al. [3] investigated experimentally and theoretically the condensate heat transfer coefficient and the flooding limit of conventional and annular two-phase closed thermosyphon using R-113 and acetone as working fluids. It was found that, a significant increase in the heat transfer coefficient was obtained for annular thermosyphons than conventional ones. Faghri and Thomas [4] studied experimentally and theoretically the capillary limit of the concentric annular heat pipe (two concentric pipes of unequal diameters). The capillary wicks can be placed on both the inside of the outer pipe and the outside of the inner pipe. The results show a significant increase in the heat capacity per unit length compared to conventional heat pipe with the same outer diameter. Sakr and Abd El-Aziz [5] , and Abd El-Aziz et al. [6] and carried out an experimental investigation to study the heat transfer performance for gravity assisted heat pipes with different working fluids. They found that, the heat pipe performance is affected by the working fluids and is improved with increasing the operating pressure. Abo El-Nasr and El-Haggar [7] performed an experimental investigation to study the effect of the wick structures on the heat transfer characteristics of the heat pipes. They found that, increasing the number of screen wick layers (up to 16 layers) enhance the ability of the heat pipe to transfer heat. Zuo and Faghri [8] presented a thermodynamic analysis of the heat pipe. It was concluded that, in order to transport heat through a longer distance, a working fluid with a larger latent heat of vaporization, a smaller vapor viscosity and a larger vapor density is preferred. Taher et al. [9] carried out an experimental investigation to study the heat transfer parameters (h & C) of gravity assisted heat pipes with inserted concentric finned tubes. It was found that, the tested heat transfer parameters of a smooth heat pipe were significantly improved by inserting a concentric finned tube. Osman [10] conducted experimentally the effect of inserting screen layers from brass material, on the heat transfer performance of gravity assisted heat pipe, using R-11 as the working fluid. The results indicated that, the heat transfer parameters of the heat pipe were improved by increasing the wire mesh number and the number of mesh layers. Berbish [11] studied experimentally the effect of surface roughness on the heat transfer performance of gravity assisted heat pipes with wire mesh screen wicks, using R-12 as the working fluid. It was concluded that, the tested heat transfer parameters of the heat pipe were increased with the increase of the surface roughness, screen thermal conductivity, and screen wick layers number. El-Kady et al. [12] carried out an experimental investigation to study the performance of three geometries of heat pipe, using pure water as the working fluid. The effects of applied heat flux and the geometry of the outer surface of the condenser section was examined. The results show that the overall heat transfer coefficient, and boiling and condensation heat transfer coefficients have higher values for knurled sample and longitudinal grooved sample compared with smooth sample. Heat and mass transfers in porous media are of growing interest in a wide range of engineering domains. A wide variety of applications involving convective transport in porous media include utilization of geothermal energy, and design of packed bed reactors. Udell [13] conducted a one-dimensional, steady state analysis of the heat and mass transfer in porous media saturated with the liquid and vapor phases of a single component fluid. The effects of capillarity, gravity forces, and phase change were included. It was found that the heat transfer was increased several orders of magnitude beyond pure conduction due to an evaporation, convection and condensation phenomenon similar to conventional heat pipe operation. De Vries [14] established a phenomenological theory of combined heat and moisture transfer in porous media. The study revealed that, water movement through porous media is often caused by a temperature gradient. Water evaporates from hot regions and moves across the gas-filled pores by diffusion and condenses on the cold region thus releasing its latent heat of vaporization. Crowe et al. [15] performed an experimental and theoretical investigation to study the performance limits of a copper heat pipe with copper porous wick using water and R-11 as working fluids. The results showed that, shorter adiabatic section length gives better performance. It was also found that, the maximum heat flow rate was obtained for heat pipes having low porosity wick. Bouddour et al. [16] developed a theoretical model to study the heat and mass transfer in wet-porous wick in presence of evaporation-condensation. It was found that the heat transfer occurs by conduction and convection in presence of phase change by evaporation-condensation. From the previous survey, the heat transfer performance of heat pipes having wicks filled with porous medium has not been studied extensively. Accordingly, the purpose of the present work is to explore some aspects of the heat transfer characteristics for heat pipe with wicks filled with porous medium. The effects of the particle diameter and the porous medium thermal conductivity on the heat pipe performance are examined experimentally.

2. EXPERIMENTAL APPARATUS

The experimental apparatus which is used to investigate the heat transfer characteristics of gravity assisted heat pipe with wick filled with porous medium is shown schematically in Fig. (1). It consists mainly of the vertical heat pipe, insulation layers, the heating wire, the cooling water circuit and different measuring instruments. A heat source is applied to the lower section of the heat pipe (evaporator section) to evaporate the liquid. The vapor flows upward to the upper section of the pipe (condenser section) where the latent heat of condensation is removed by circulating cooling water. The condensed liquid returns back to the evaporator by gravity. A mild-steel wire screen mesh of thickness 0.8 mm is fixed rigidly and concentrically inside the heat pipe in a hollow cylindrical form of about 8.0 mm outer diameter. The heat pipe wick (space between the screen outer diameter and the heat pipe inner diameter) is filled with porous medium, as shown in Fig.(2). The tested heat pipe is made of hard copper material, of 28.5 mm inside diameter, 1.5 mm thickness, and a total length of 400 mm. The length of the evaporator section is 140 mm, adiabatic section is 120 mm, and condenser section is 140 mm. The cooling water flow rate is measured using a calibrated orifice-meter. A manual ball valve is installed before the orifice-meter to control the water flow rate. The electrical heater is covered with an electric insulation. A cylindrical rock wool insulation of thickness of 25.4 mm sheltered with aluminum foil is used for covering the whole heat pipe to minimize the heat loss. The outer and inner insulation surface temperatures are measured at different axial positions along the pipe to calculate the radial heat loss. Heat input is determined from the measured electrical power input as corrected for heat loss through outer wall, based on the average inner and outer insulation temperatures. The heat loss is found to be less than 2 %. Figure (2) show the details of cross-sectional view in the evaporator zone of the heat pipe. Surface temperatures are measured at eleven axial stations along the heat pipe as shown in Fig. (3). Calibrated thermocouples are fixed at each test position with the hot junction of each embedded in 0.5 mm deep drilled holes to be in good contact with the pipe surface. The calibrated thermocouples are made from copper-constantan wires of 0.25 mm diameter with $\pm 0.1\ ^{o}C$ accuracy. The fluid temperatures are measured using six thermocouples inserted inside the heat pipe, where two thermocouples are press fitted into the evaporator, adiabatic, and condenser sections as shown in Fig. (3). The fluid temperature in the adiabatic section is used to measure the average saturation temperature corresponding to the operating pressure. This location is selected to avoid the temperature of the superheated boiling liquid in the evaporator or the sub-cooled condensate in the condenser [3&4]. The heat pipe is designed to have the facility of changing the porous medium with different particles diameters and materials. At first the heat pipe without packing material (smooth) is used for calibrating the experimental apparatus and also for comparison with cases using packing material. The tested parameters are shown in Table (1).

Table (1): The tested parameters

Operating pressure, (bar)	2.0, 2.85, 3.95 and 5.10
Porous medium particles diameters to inner heat pipe diameter ratios, (d/D)	0.1684, 0.1404, 0.1123 and 0.0912
Porous medium particles material	Gravel, glass, ceramic, marble, granite, mild-steel, and nickel
k_s/k_f for gravel, glass, ceramic, marble, granite, mild-steel and nickel respectively.	5.2, 12.5, 14.9, 32.3, 36.1, 533.6 and 1156.7

Seven different packing materials are tested. The ratios of packing materials conductivity to liquid refrigerant R-11 conductivity at pressure equals to 2.0 bar (k_s/k_f) are given in Table (1).

3. TEST PROCEDURE AND METHOD OF CALCULATION

The power is supplied via a variac transformer to adjust the heat flux. At each power setting, after reaching the steady state condition, data are recorded concerning the water flow rate, inlet and outlet water temperatures, current, voltage, heat pipe wall temperatures, the outer wall insulation temperatures, and the working pressure.
The main heat transfer parameters for the tested heat pipe are the evaporator heat transfer coefficient and the overall thermal conductance.

The evaporator heat transfer coefficient, h, is calculated as:

$$h = q/(T_e - T_a) \qquad (1)$$

where: q : the heat rate flux based on the evaporator outer surface area, (W/m^2).

T_e : average surface temperature along evaporator section, (K).

T_a : fluid saturation temperature (along adiabatic section), (K).

The overall thermal conductance of the heat pipe is calculated as:

$$C = Q / (T_e - T_c) \qquad (2)$$

where: Q : electric heat rate input, (W).
T_c : average surface temperature along condenser section, (K)].

4. RESULTS and DISCUSSION

The tested heat transfer parameters of a gravity-assisted smooth heat pipe and heat pipe having a wick filled with porous medium at different tested parameters are presented.

4.1. Results of Smooth Heat Pipe

Figures (4) and (5) show the variations of the evaporator heat transfer coefficient, h_{sm}, and the overall thermal conductance, C_{sm}, (tested heat transfer parameters) for a smooth heat pipe without porous medium wick, against the heat flux at different operating pressures, respectively. It is found that the increase in both heat flux and operating pressure enhance significantly the tested heat transfer parameters. For a constant operating pressure, increasing the heat flux, the liquid in contact with the pipe surface will become progressively heated and bubbles will form nucleation sites. These bubbles will transport some energy to the heat pipe surface by latent heat of vaporization and will also increase the heat transfer parameters [6&17]. At constant heat flux, an increase in heat transfer parameters is found with the increase of working pressure according to the dependence of the temperature drop of the working fluid on the pressure drop along the heat pipe. Consequently the pressure drop and the temperature drop along the heat pipe decrease by increasing the working pressure as a result of decrease in vapor specific volume for a constant vapor mass flow rate by transferring a given heat input [5, 6&10].

To check the validity of the present apparatus, comparisons are made between the present results of the tested heat transfer parameters and the existing previous published data (Sakr and Abd El-Aziz [5], and Osman [10]). The comparisons are indicated in Figs (6) and (7), and good agreement is obtained.

The evaporator heat transfer coefficient, h_{sm}, and the overall thermal conductance, C_{sm}, are correlated as functions of operating pressure and heat flux using the measured data plotted in Figs. (4) and (5) for smooth heat pipe, as:

$$h_{sm} = 0.43\, P^{0.18}\, q^{2/3} \tag{3}$$

and

$$C_{sm} = 0.014\, P^{0.25}\, q^{0.51} \tag{4}$$

The present correlations for the evaporator heat transfer coefficient, Eq.(3), and the over all thermal conductance, Eq. (4) are valid with the present experimental results within ±5.8% and ±6.7 % deviations, as shown in Figs. (8) and (9), respectively for all tested parameters ranges. Correlation (3) agrees with the previous published correlations achieved by Casarosa [18] and Abd El-Aziz et al [6], and Taher et al. [9], as h_{sm} is proportional to the product ($P^{0.18}$ $q^{2/3}$) and the constant of propotionality depends on the thermophysical properties of the working fluid, and the heat pipe dimensions and materials.

4.2. Results of Porous Wicked Heat Pipe

The particles of the porous media used in the experiments were made of gravel, glass, ceramic, marble, granite, mild steel and nickel covering a wide range of solid thermal conductivities and particle diameters.

4.2.1. Effect of porous medium thermal conductivity

For heat pipes having wicks filled with solid particles, of special interest for studies of heat transfer augmentation is the question of how much the heat transfer parameters are improved relative to an equivalent smooth pipe (without porous medium) at similar conditions. The variations of the evaporator heat transfer coefficient and the overall thermal conductance (tested heat transfer parameters) versus the heat flux for different packing material thermal conductivities at constant particle to the heat pipe diameter ratio of about d/D=0.1684, are shown in Figs. (10) and (11), respectively. The results show that the use of porous wick inside heat pipe increases considerably both the evaporator heat transfer coefficient and the overall thermal conductance. The porous media solid particles serve as effective enhancers for heat pipes. The heat transfer augmentation produced by the porous matrix is attributed to a combination effects, including thinning the thermal boundary layer of the pipe, and direct conduction through the porous matrix. Moreover, it is found that the tested heat transfer parameters increase with the increase in particle thermal conductivity and the highest values are associated with the use of nickel packing material. This result is due to the high conduction to nickel solid particles (higher conductivity) touching the heat pipe surface [19, 20& 21].

For the investigated packing bed materials, the enhancement ratios of both the evaporator heat transfer coefficient (h/h_{sm}) and the overall thermal conductance (C/C_{sm}) are indicated in Figs. (12) and (13), respectively. It is seen that the enhancement ratios are independent of the heat flux and the maximum values of both (h/h_{sm}) and (C/C_{sm}) are averaged at nearly about 1.90 and 1.70 for nickel packing material at d/D = 0.1684.

4.2.2. Effect of porous medium particle diameter

The variations of the tested heat transfer parameters with the heat flux for different particle to heat pipe inner diameter ratios (d/D) are illustrated in Figures (14) and (15). The particles of the porous medium were made of nickel material with $0.0912 \leq d/D \leq 0.1684$. For the same packing material, it is observed that the tested heat transfer parameters are increased with the decrease of the particle to the heat pipe diameter ratio. This may be attributed to increase the contact conduction area of solid particles touching the heat pipe surface with the smaller particle diameter [19&22].

Figures (16) and (17) indicate the enhancement ratios of the evaporator heat transfer coefficient, h/h_{sm}, and the overall thermal conductance, C/C_{sm}, for porous wicked heat pipe using nickel packing material, respectively. It is found that, the average maximum value of (h/h_{sm}) is about 2.40, and of (C/C_{sm}) is about 2.10. These values are obtained for smaller particle to heat pipe diameter ratio (d/D = 0.0912). Also, for the same operating pressure, it is observed from these figures that the enhancement ratios (h/h_{sm}) and (C/C_{sm}) are nearly constant over the present heat flux range.

Finally, correlations for the enhancement ratios of both the evaporator heat transfer coefficient (h/h_{sm}) and the overall thermal conductance (C/C_{sm}) are obtained as functions of the particle to heat pipe diameter ratio, (d/D) and the particle to liquid thermal conductivity ratio, (k_s/k_f) using the present experimental data, as:

$$h/h_{sm} = [0.644 + 0.048\,(D/d)](k_s/k_f)^{0.102} \qquad (5)$$

and

$$C/C_{sm} = [0.768 + 0.017\,(D/d)^{1.32}](k_s/k_f)^{0.085} \qquad (6)$$

These correlations, Eqs. (5) and (6), for both (h/h_{sm}) & (C/C_{sm}) are in good agreement with the present experimental data within ±9.1% & ±9.5% maximum deviations as shown in Figs. (18) and (19), respectively within the tested ranges of (d/D) and (k_s/k_f).

5. CONCLUSIONS

In the present study, an experimental investigation is conducted to study the tested heat transfer parameters of gravity assisted heat pipes having wicks filled with porous medium. From the previous results and discussion, the following conclusions can be obtained:

1. The evaporator heat transfer coefficient and the overall thermal conductance (tested heat transfer parameters) of the heat pipe increase with increasing both the operating pressure, and the heat flux.
2. The tested heat transfer parameters for smooth heat pipe are correlated as a function of the heat flux, q, and the operating pressure, P, as:

   $$h_{sm} = 0.43\,P^{0.18}\,q^{2/3}$$

   and

   $$C_{sm} = 0.014\,P^{0.25}\,q^{0.51}$$

   The obtained correlations for h_{sm} & C_{sm} fit the present experimental data within ± 5.8% and ±6.7% maximum deviations, respectively.
3. The tested heat transfer parameters for porous wicked heat pipe increase significantly with the increase in the packing material conductivity, and with the decrease in the particle to the heat pipe diameter ratio.
4. For porous wicked heat pipe using nickel packing material, it is found that the average maximum value of (h/h_{sm}) is about 2.40, and of (C/C_{sm}) is about 2.10. These values are obtained for smaller particle to heat pipe diameter ratio (d/D= 0.0912).
5. Correlations for the augmentation of the tested heat transfer parameters are obtained as functions of the particle to heat pipe diameter ratio, (d/D) and the particle to liquid thermal conductivity ratio, (k_s/k_f), as:

   $$h/h_{sm}=[0.644+0.048(D/d)](k_s/k_f)^{0.102}$$

   and

   $$C/C_{sm}= [0.768+0.017(D/d)^{1.32}](k_s/k_f)^{0.085}$$

These correlations for (h/h_{sm}), and (C/C_{sm}) are in good agreement with the present measured data within ±9.1% and ±9.5 % maximum deviations, respectively.

NOMENCLATURE

C	Overall thermal conductance of the heat pipe, W/K.
D	Inner heat pipe diameter, m.
d	Solid particle diameter of the porous medium, m.
h	Evaporator heat transfer coefficient, W/m^2 K.
k	Coefficient of thermal conductivity, W/m K.
P	Operating pressure, bar.
Q	Rate of heat flow (input power), W.
q	Rate of heat flux, W/m^2.
T	Temperature, K.

Subscripts

c	Condenser.
e	Evaporator.
f	Liquid.
s	Solid.
sm	Smooth heat pipe.

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- *General Manager of General Administration for International Cooperation, GOTEVOT, KSA. (Nov. 2003- till now).*

- *General Manger of many international projects.*
- *Also, he is a member of many international societies and national committees, such as:*
- *American Society of Mechanical Engineers (ASME), 2002-2004.*
- *Member of Scientific Council of GOTEVOT.*
- *AIAA member.*
- *President of Financial Committee for the 3rd and 4th Saudi International Conference and Exhibition, 2004 and 2006.*

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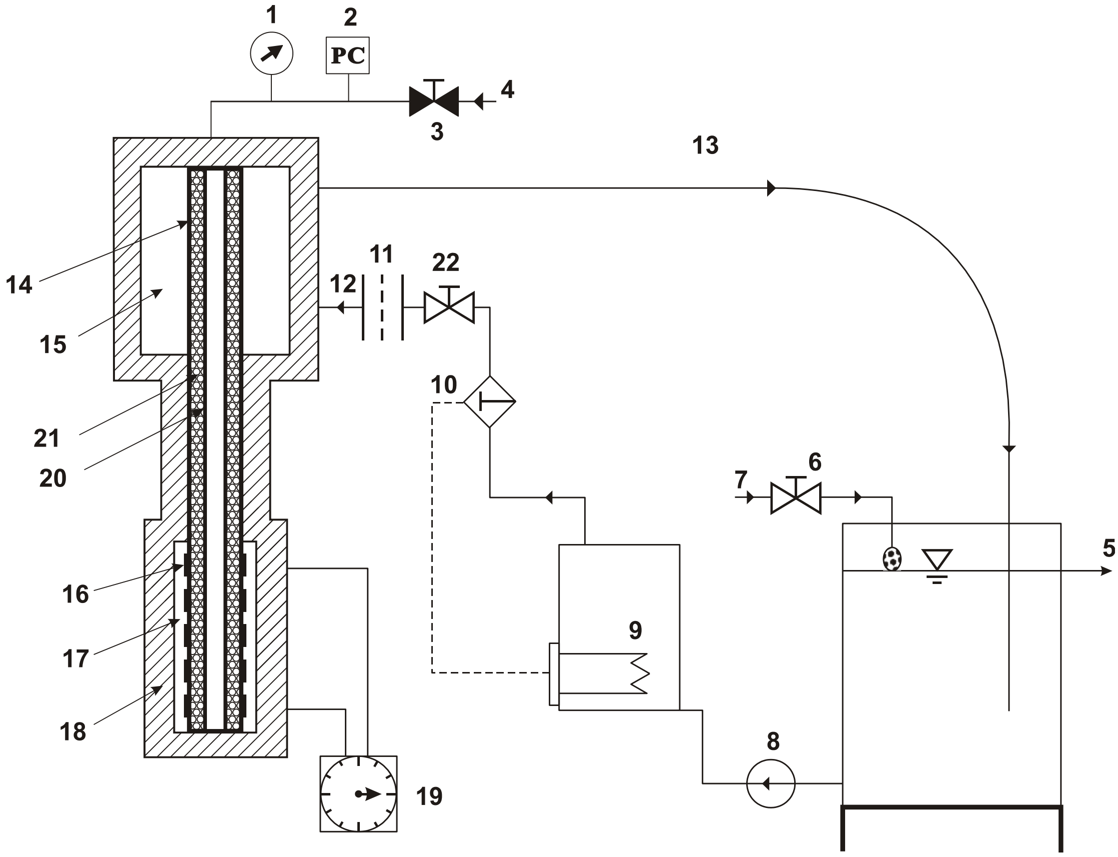
ملخص البحث:

"دراسة معملية لآداء الأنابيب الحرارية المساعدة بالجاذبية والمزودة بطبقة على سطحها الداخلي من الوسط المسامي"

في هذا البحث أجريت دراسة معملية لدراسة آداء الأنابيب الحرارية المساعدة بالجاذبية والمزودة بطبقة من الوسط المسامي على سطحها الداخلي. الوسط المسامي المستخدم عبارة عن حبيبات نسبة قطرها المتوسط إلى القطر الداخلي للأنبوب الحراري يتراوح بين $(0.0912 \le d/D \le 0.1684)$ ومصنعة من مواد النيكل، الحديد، الجرانيت، الرخام، السيراميك، الزجاج، والزلط. في هذه الدراسة تم تصميم وتنفيذ دائرة إختبار معملية تحتوي على أنبوب حراري مصنوع من النحاس الأحمر بقطر داخلي 28.5 مم وسمك 1.5 مم وطول 400 مم. وتم إستخدام فريون 11 كمائع للتشغيل.

للتأكد من دقة الجهاز المعملي وأجهزة القياس المستخدمة فقد تمت مقارنة النتائج المعملية لكل من معامل انتقال الحرارة لمقطع المبخر ومعامل التوصيل الحراري (معاملات انتقال الحرارة للأنابيب الحرارية) للأنبوب الحراري الأملس بدون وسط مسامي بالنتائج المعملية الأخرى وكانت المقارنة جيدة. وقد أظهرت النتائج المعملية أن معاملات انتقال الحرارة للأنابيب الحرارية (h & C)، تزداد بزيادة كل من ضغط مائع التشغيل، معدل الفيض الحراري لمقطع المبخر، ومعامل التوصيل الحراري لمادة الوسط المسامي، وبصغر قطر حبيبات مادة الوسط المسامي.

في هذه الدراسة وعند تغيير قطر حبيبات مادة الوسط المسامي المستخدم في الأنبوب الحرارى، فقد بينت النتائج أن متوسط الزيادة في كل من معامل انتقال الحرارة لمقطع المبخر ومعامل التوصيل الحراري للأنبوب الحراري يصل إلى 2.40 و 2.10 بالمقارنة للأنبوب الحرارى الأملس على الترتيب، وقد تم الحصول على هذه النسب عند استخدام وسط مسامي من حبيبات مادة النيكل (ذات أعلى معامل توصيل حراري) وذلك عند أصغر نسبة لقطر الحبيبات إلى قطر الأنبوب الحراري وهي $(d/D=0.0912)$. وأخيراً تم إستنتاج معادلات تجريبية لكل من معامل انتقال الحرارة لمقطع المبخر ومعامل التوصيل الحراري للأنبوب الحراري كدالة في كل المتغيرات التي تم دراستها.



1. **Pressure gauge**
2. **Pressure cut-out**
3. **Refrigerant valve**
4. **Charged line**
5. **Water overflow**
6. **Water supply valve**
7. **Water supply**
8. **Pump**
9. **Water heater**
10. **Thermostate**
11. **Orifice-meter**
12. **Cooling water inlet**
13. **Cooling water outlet**
14. **Heat pipe**
15. **Cooling water jacket**
16. **Heating wire**
17. **Asbestos**
18. **Rock wool insulation**
19. **Power supply**
20. **Wire screen**
21. **Porous medium**
22. **Ball valve**

Fig. (1) Experimental test rig layout

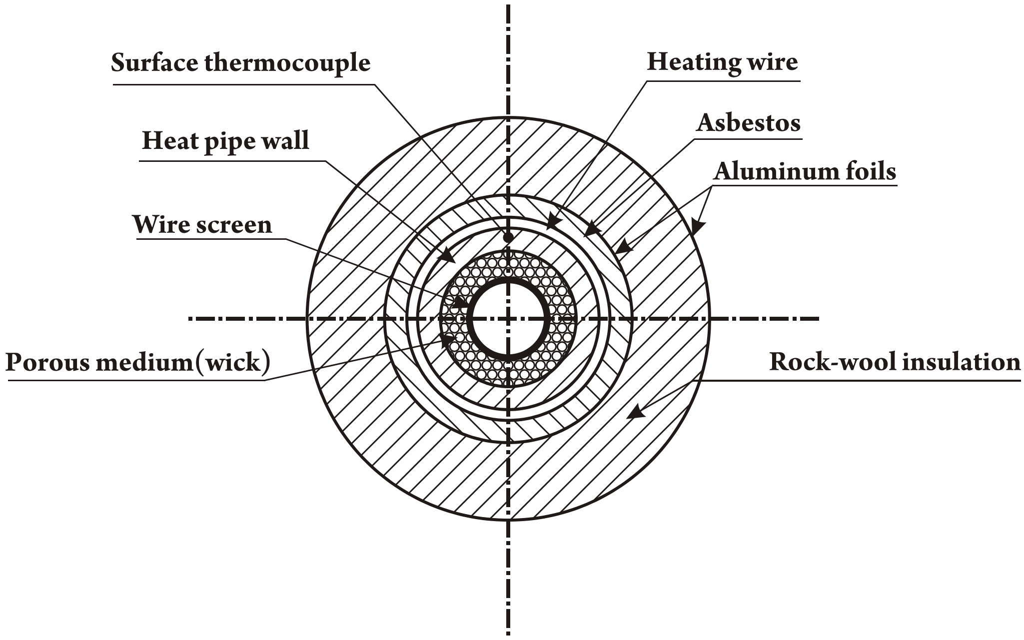


Fig. (2) Cross sectional view in the evaporator zone of the heat pipe

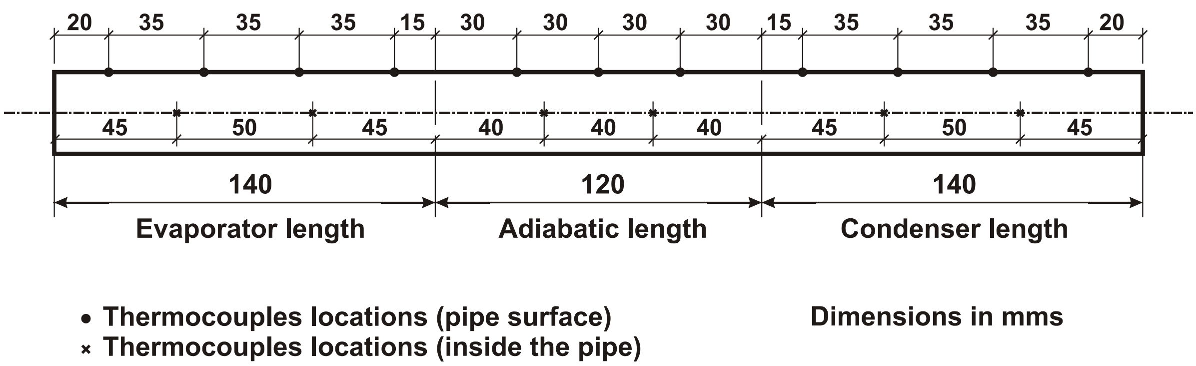


Fig. (3) Details of the thermocouples locations

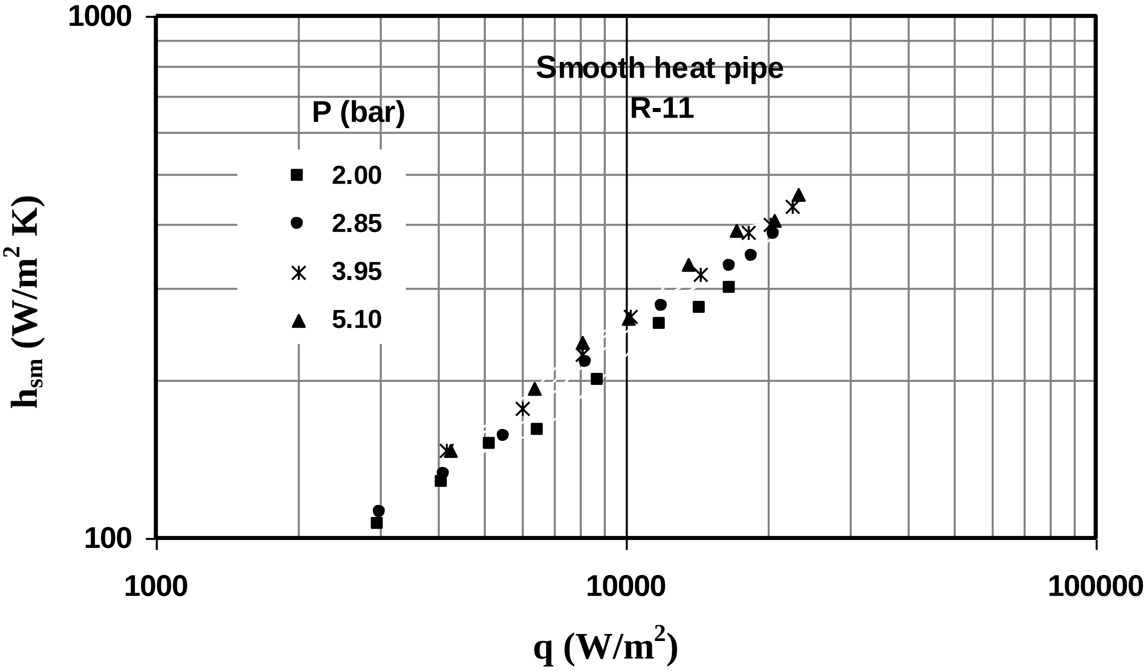


Fig. (4) Variation of the evaporator heat transfer coefficient, h_{sm}, with heat flux, q, for smooth heat pipe

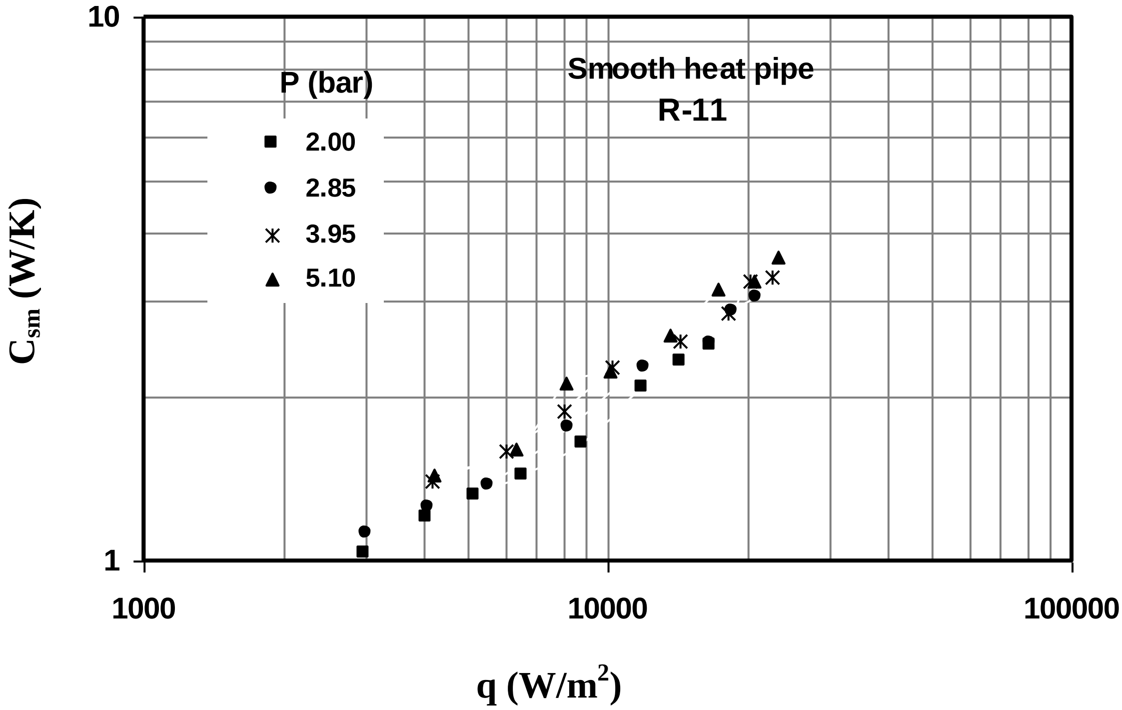


Fig. (5) Variation of the overall thermal conductance, C_{sm}, with heat flux, q, for smooth heat pipe

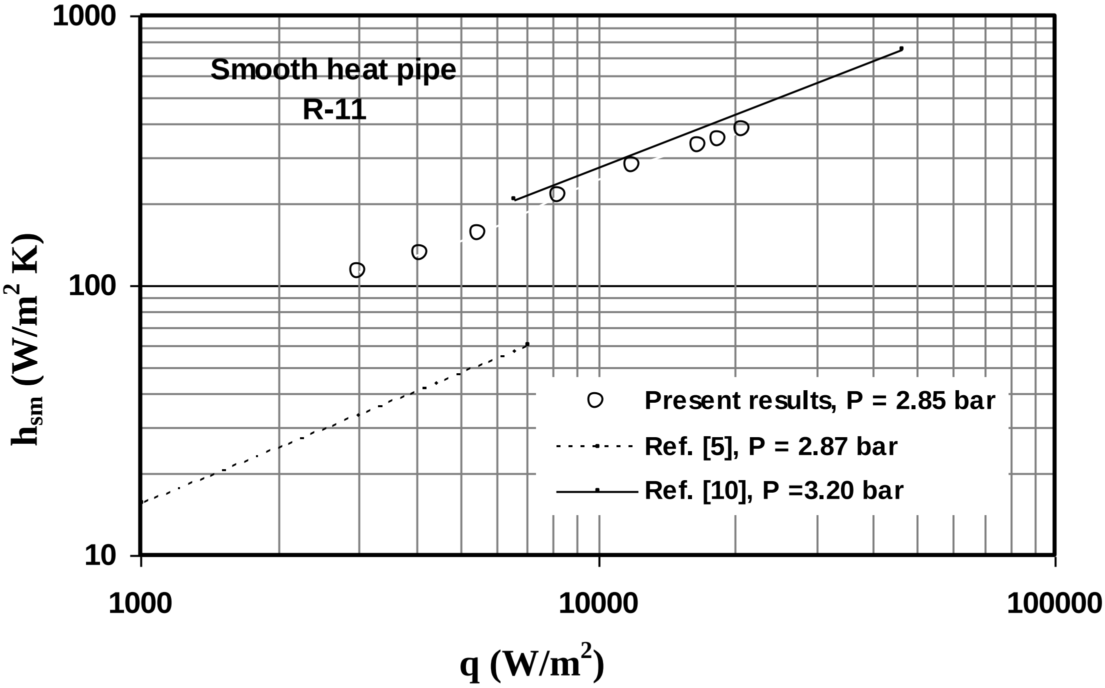


Fig. (6) Comparison of measured evaporator heat transfer coefficient with other published results for smooth heat pipe

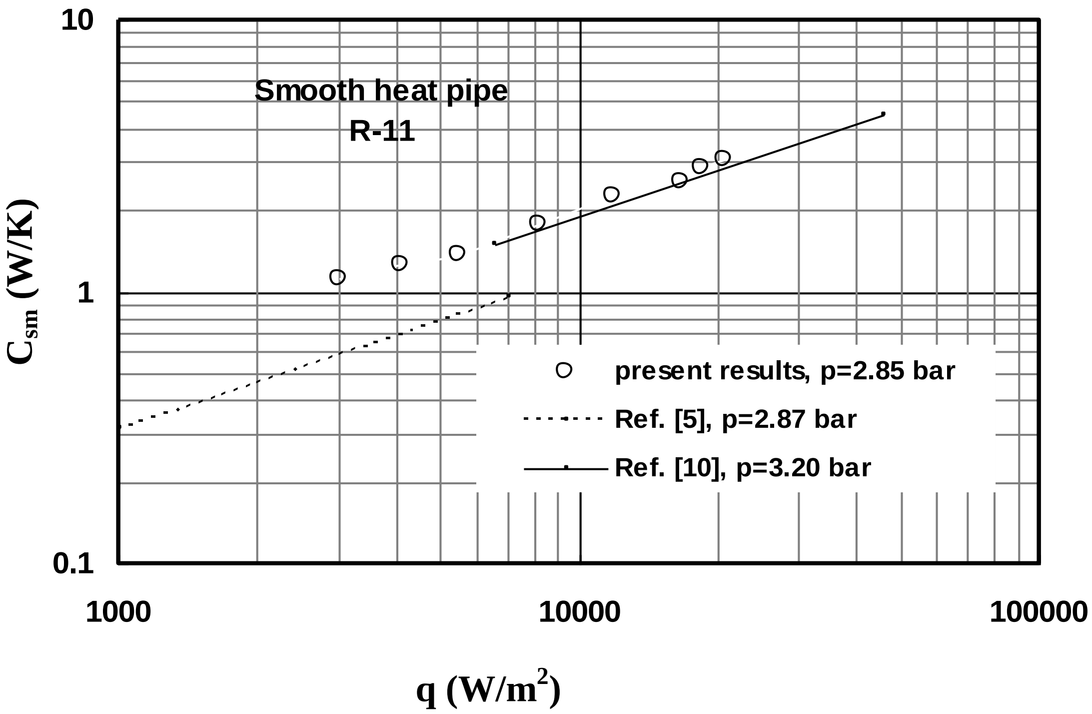


Fig. (7) Comparison of measured overall thermal conductance with other published results for smooth heat pipe

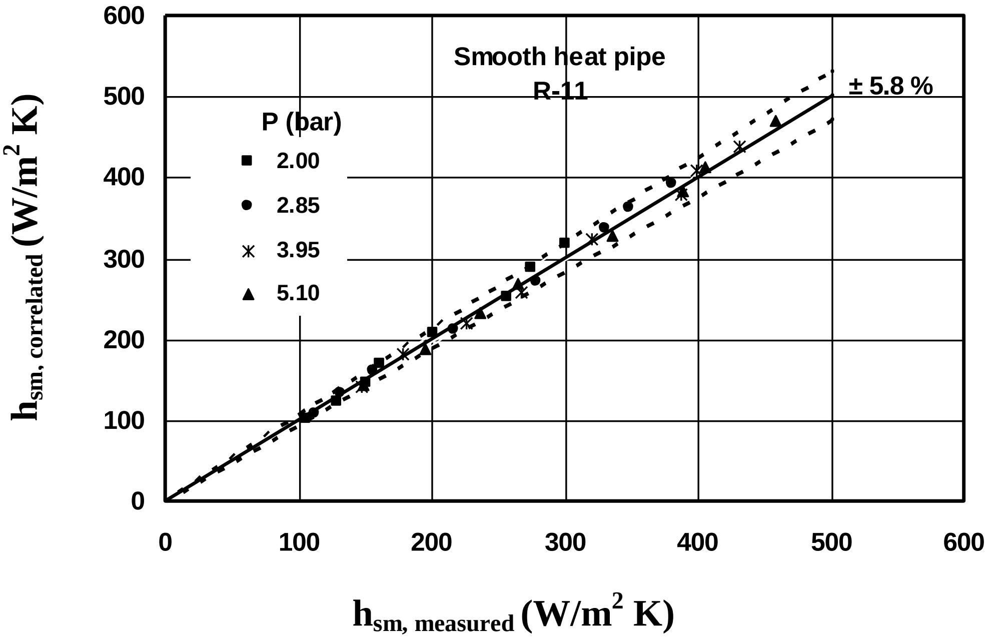


Fig. (8) Comparison of measured evaporator heat transfer coefficient of smooth heat pipe with those obtained from present correlation, Eq. (3)

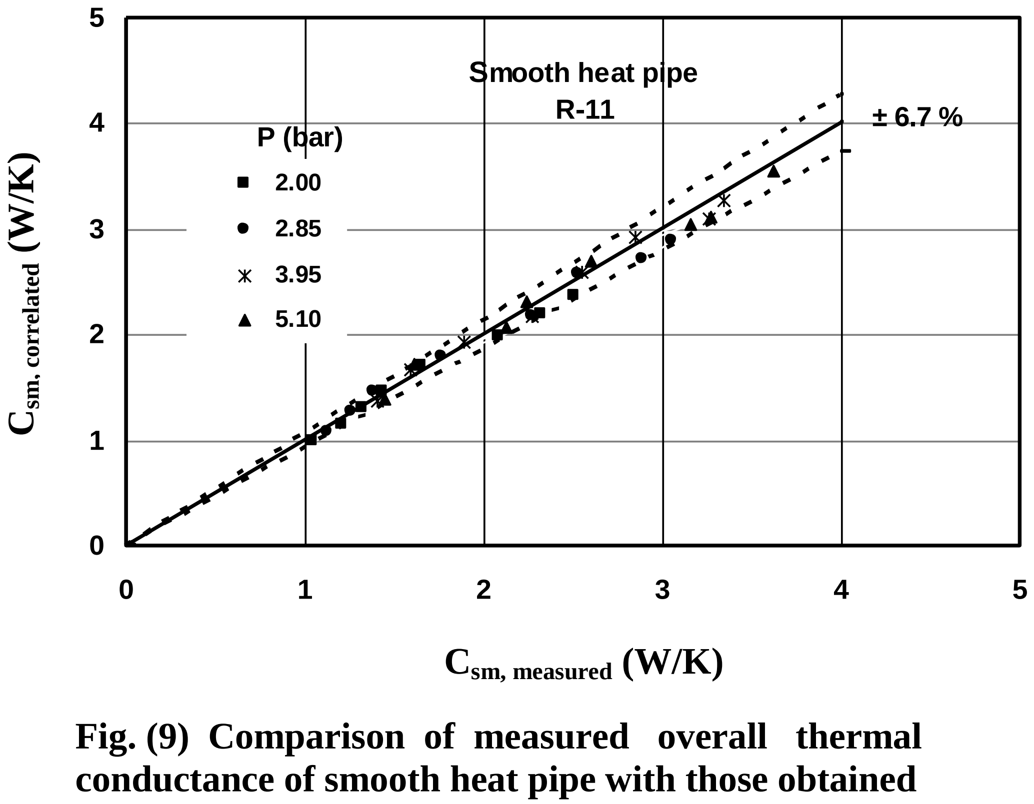


Fig. (9) Comparison of measured overall thermal conductance of smooth heat pipe with those obtained from present correlation, Eq. (4)

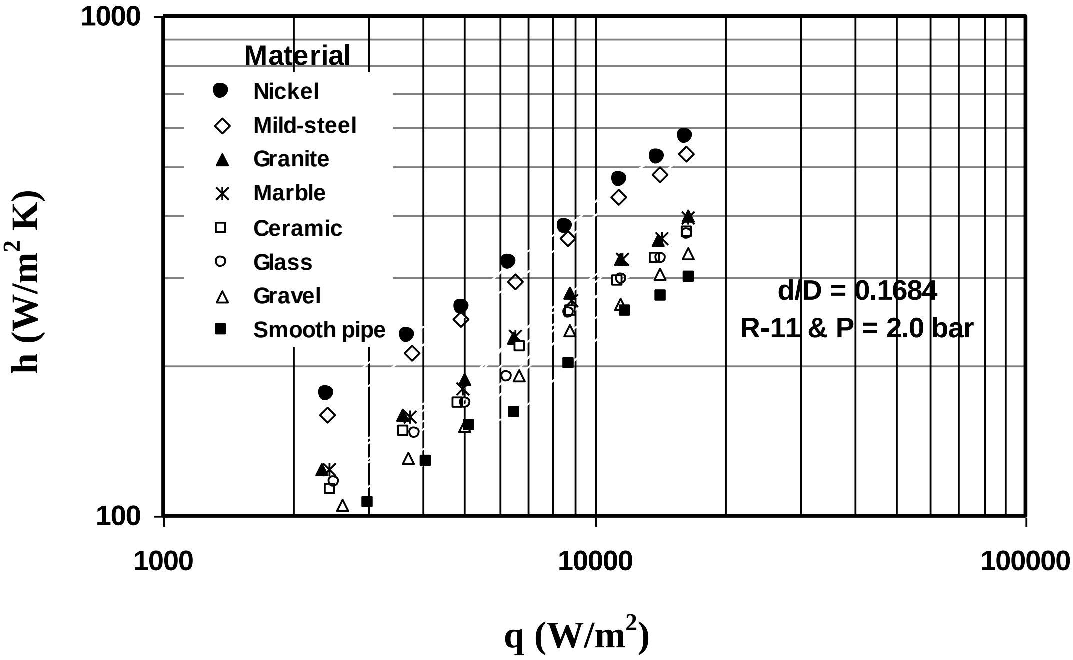


Fig. (10) Variation of the evaporator heat transfer coefficient, h, with heat flux, q, for porous wicked heat pipe at different packing materials and d/D=0.1684

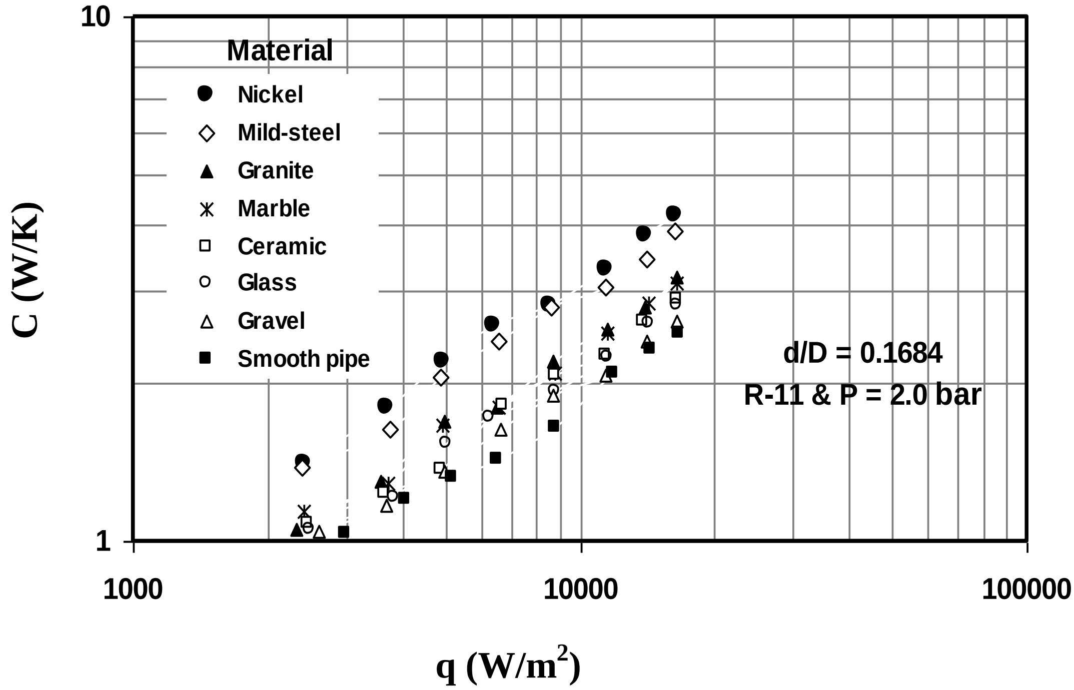


Fig. (11) Variation of the overall thermal conductance, C, with heat flux, q, for porous wicked heat pipe at different packing materials and d/D=0.1684

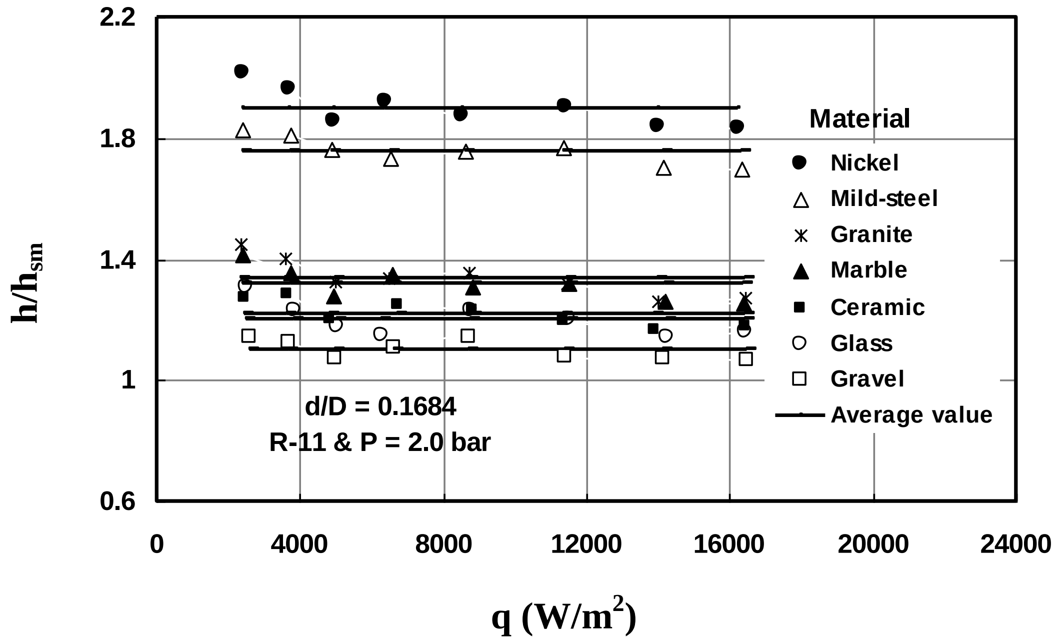


Fig. (12) Enhancement ratio of the evaporator heat transfer coefficient, h/h_{sm}, with heat flux, q, for porous wicked heat pipe at different packing materials and d/D=0.1684

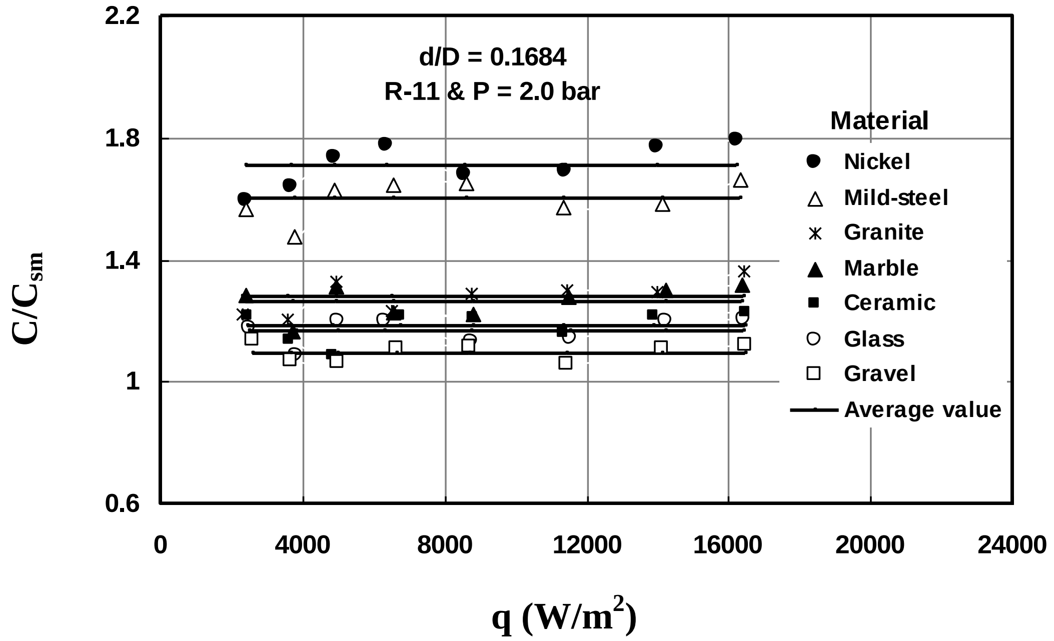


Fig. (13) Enhancement ratio of the overall thermal conductance, C/C_{sm}, with heat flux, q, for porous wicked heat pipe at different packing materials and d/D=0.1684

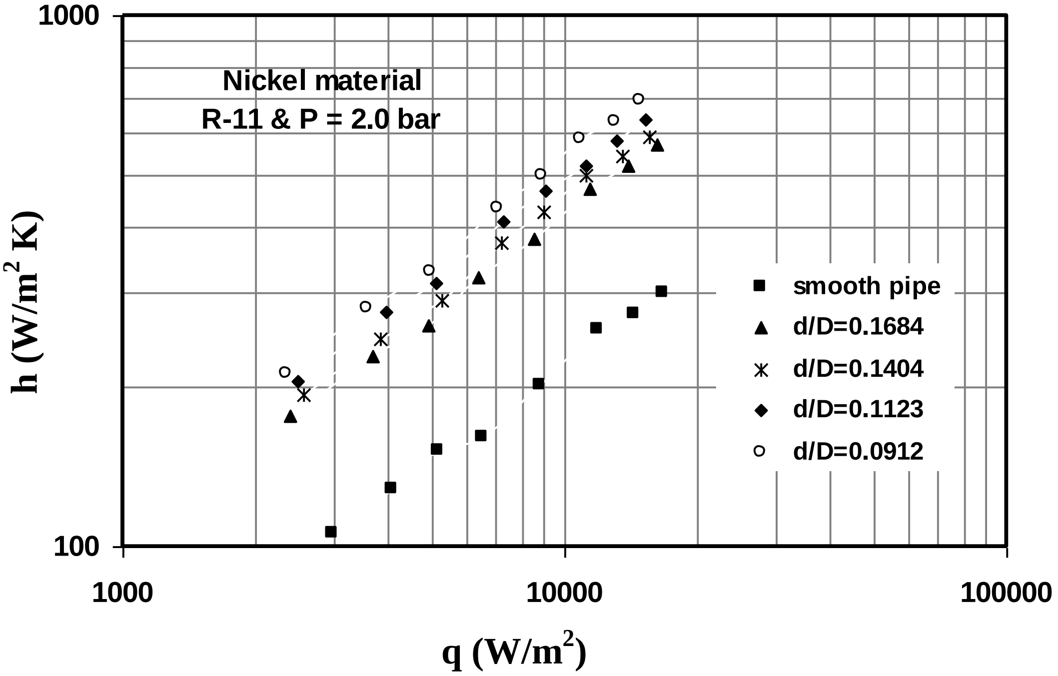


Fig. (14) Variation of the evaporator heat transfer coefficient, h, with heat flux, q, for porous wicked heat pipe at different nickel particle to heat pipe diameter ratios

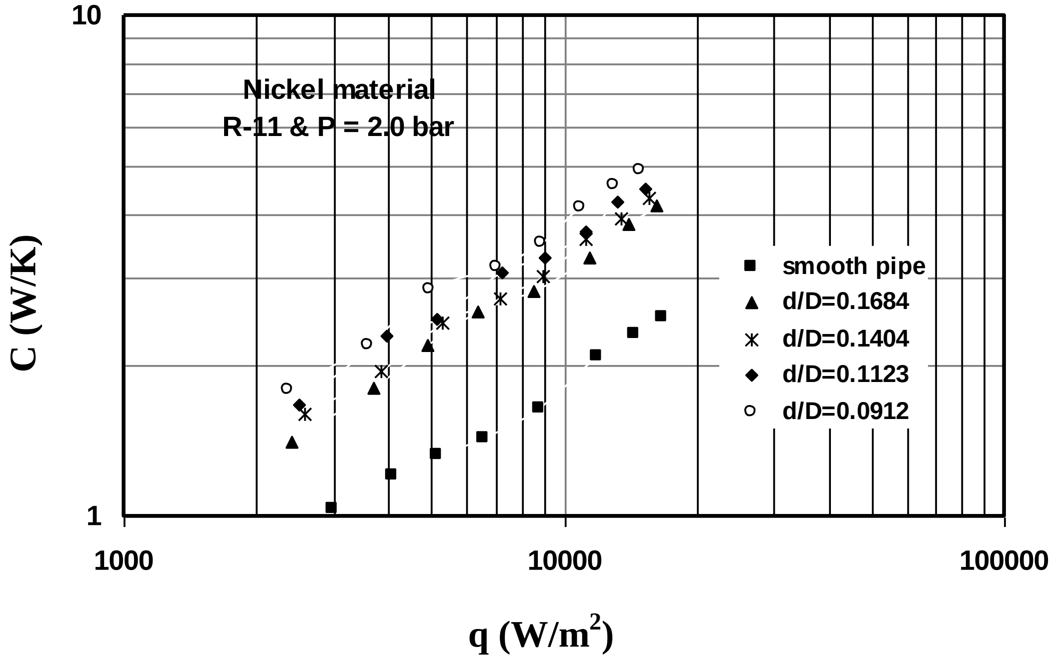


Fig. (15) Variation of the overall thermal conductance, C, with heat flux, q , for porous wicked heat pipe at different nickel particle to heat pipe diameter ratios

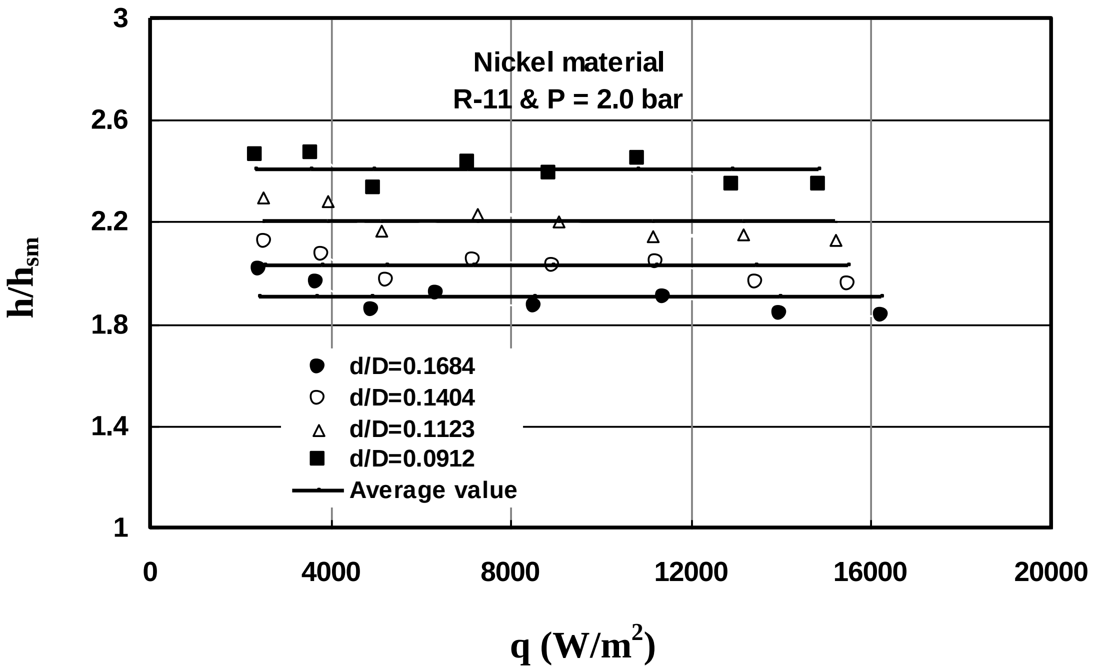


Fig. (16) Enhancement ratio of the evaporator heat transfer coefficient, h/h_{sm}, with heat flux, q, for porous wicked heat pipe at different nickel particle to heat pipe diameter ratios

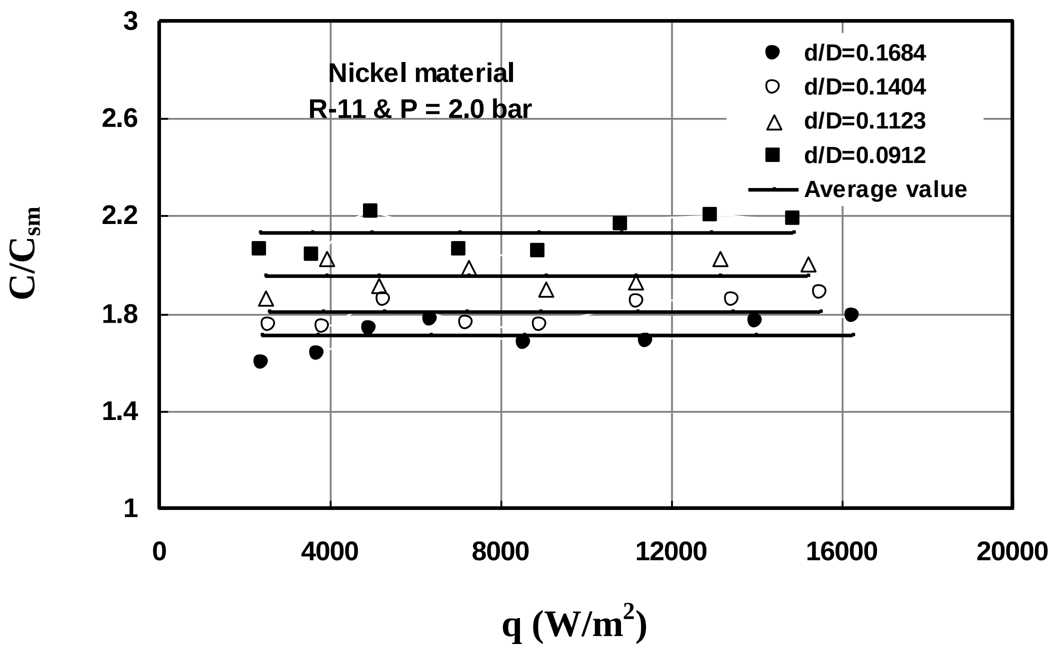


Fig. (17) Enhancement ratio of the overall thermal conductance, C/C_{sm}, with heat flux, q, for porous wicked heat pipe at different nickel particle to heat pipe diameter ratios

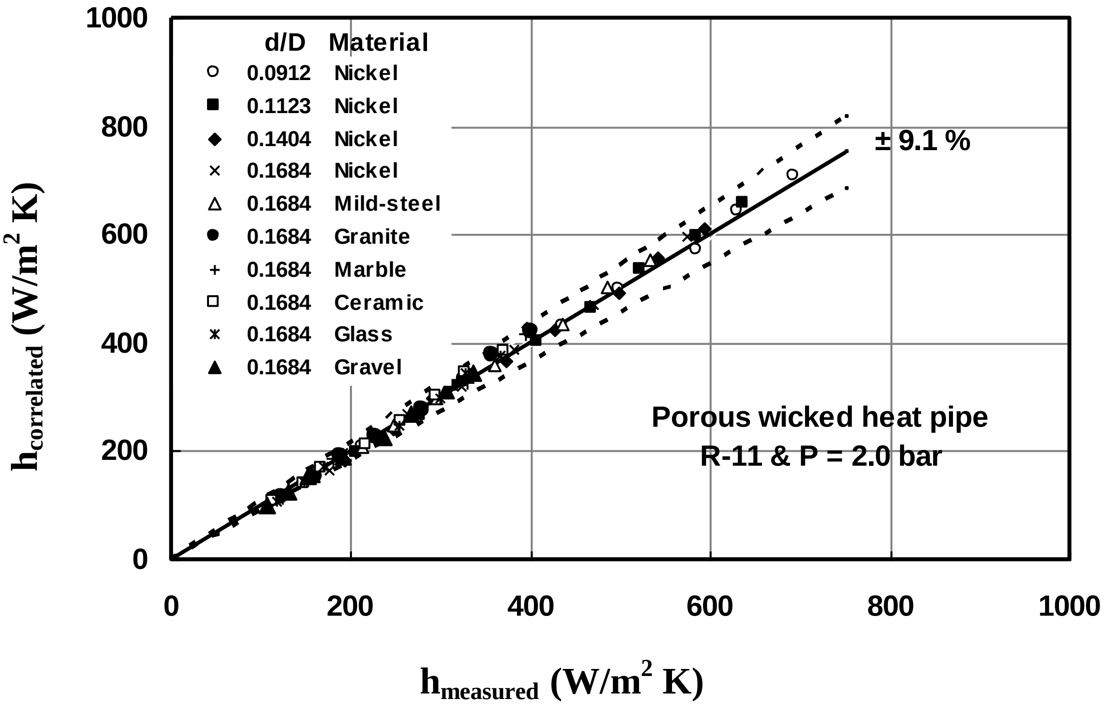


Fig. (18) Comparison of measured evaporator heat transfer coefficient of porous wicked heat pipe with those obtained from present correlation, Eq. (5)

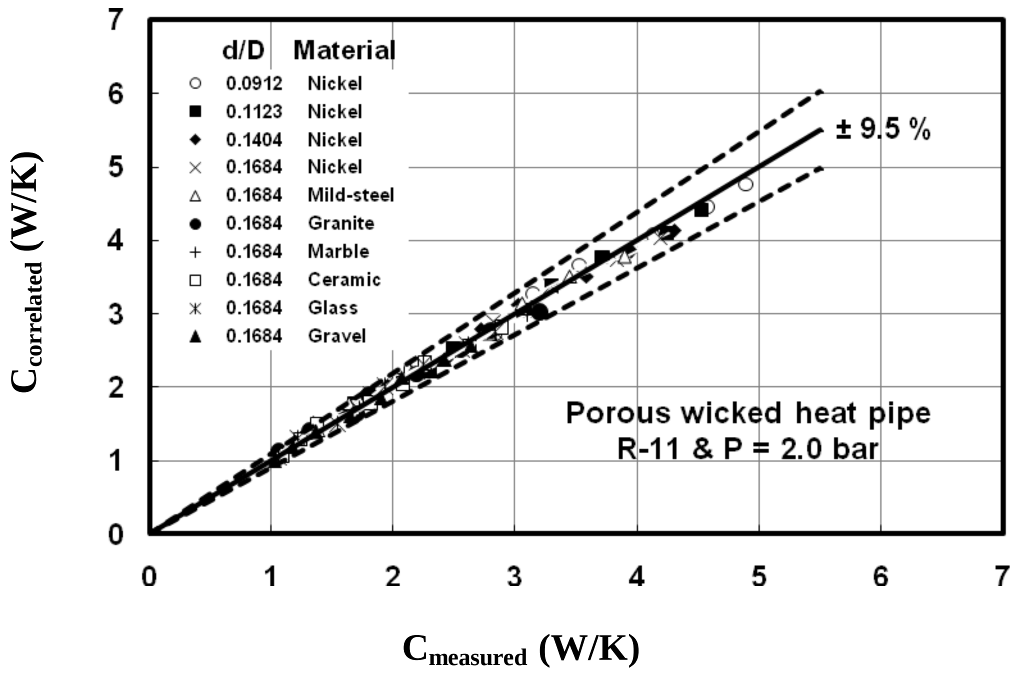


Fig. (19) Comparison of measured overall thermal Conductance of porous wicked heat pipe with those obtained from present correlation, Eq. (6)