[Mansoura Engineering Journal](https://mej.researchcommons.org/home)

Volume 25 | Issue 1 Article 6

2-4-2021

Enhancement of Mixed Convection in a Channel with Discrete Heat Sources by Using a Highly Conducting Porous Medium.

M. El-Kady
Mechanical Power Engineering Department., Faculty of Engineering., El-Mansoura University., Mansoura.,Egypt., mselkady@mum.mans.eun.eg

Mansoura Engineering Journal, (MEJ), Vol. 25, No. 1, March 2000. M. 1

Enhancement of Mixed Convection in a Channel with Discrete Heat Sources by Using a Highly Conducting Porous Medium

تحسين انتقال الحرارة بالحمل المختلط فى مجرى به منابع حرارية باستخدام وسط مسامى عالى الموصلية

M. S. El-Kady
Mechanical Power Engineering Department
Mansoura University, Egypt
Email: mselkady@mum.mans.eun,eg

خلاصـــــــــــة

تحتوى الاجهزة الالكترونية الحديثة على العديد من المكونات الباعثة للحرارة، ويعتـــبر نظـــام التحكـــم الحرارى الامثل أحد المتطلبات الرئيسية فى تصميم وتشغيل هذه الاجهزة، حيث يؤدى عدم وجود نظـــام فعـــال لازالة الحرارة المنبعثة من المكونات سببا لارتفاع درجة الحرارة داخل الاجهزة عن المعدل المسموح به والـــذى يؤدى بدوره الى انهيار أداء هذه الاجهزة الحساسة.

يتناول هذا البحث دراسة عملية على انتقال الحرارة بالحمل المختلط فى مجرى به منبعين حراريين افقيا فى أسفله، تهدف الدراسة الى ايجاد مدى تأثير استخدام الاوساط المسامية داخل المجرى على خـــواص انتقـــال الحرارة المتمثلة فى درجات حرارة المنبعين الحراريين ومعدل انتقال الحرارة منهما. ولذا تمت الدراسة العمليـــة على ثلاثة حالات، تناولت الحالة الاولى مجرى اختبار خالى من الوسط المسامى، والثانية مجرى اختبار مـــلىء بالكامل بالوسط المسامى والثالثة مجرى ملىء جزئيا بالوسط المسامى (ملء الفراغ أعلى المنابع الحرارية بالوسط المسامى وترك الفراغ الباقى خاليا). وقد استخدم مجرى اختبار بمقطع رأسى ١٢٠×١٢٠مم وبطول ٢٠٠مم به منبعين حراريين افقيين كل بطول ٤٠مم وعرض ١٢٠مم. وقد اقحم مجرى الاختبار فى مجرى هوائى يمكـــن رفع سرعة الهواء به حتى ٥ م/ث والتى تعطى امكانية تغيير رقم رينولدز الى ١٠^٤، كما تم تغيير معدل انبعـــاث الحرارة من المنبعيين الحراريين حتى ٧٠١٧ وات/م^٢ والذى مكن من تغيير رقم جراشوف حتـــى ٥٨×١٠^٦ ، وكوسط مسامى استخدمت شرائح مسامية من معدن الالومنيوم بسمك حوالى ٣مم وكثافة قدرها ١٢٢,٨٥ كجم/م^٣ برقم دارسى ٨×١٠^-٦ . بينت النتائج أنه على الرغم من أن استخدام الشرائح المسامية من معدن الالومنيـــوم داخل المجرى قد سبب مقاومة عالية لسريان وسط التبريد (الهواء) وادى الى زيادة الانخفاض فى ضغط الـــهواء فى المجرى الا انه قد قلل بشدة من درجات الحرارة الموضعية والقصوى للمنابع الحرارية الى حوالـــى الثلـــث عنها فى حالة عدم استخدام الاوساط المسامية، كما أدى الى الزيادة فى معدل انتقال الحرارة من المنبعين بنســـبة ٢٥٣ و٢٩٦% عند كل من رقم رينولدز ١٧٢٣ و٥٥٣١ على التوالى. كما بينت النتـــائج ايضـــا أن اســـتخدام الوسط المسامى جزئيا (فى الفراغ أعلى المنابع الحرارية) قد أعطى تقريبا نفس الزيادة فى معدل انتقال الحـــرارة ونفس النقصان فى درجات حرارة المنابع الحرارية، مع الميزة الكبيرة والمتمثلة فى نقص يعادل ٦٠% فى معدل الانخفاض فى الضغط وذلك بالمقارنة مع استخدام الوسط المسامى كليا فى المجـــرى، ممـــا يجعـــل تكنولوجيـــا استخدام الاوساط المسامية جزئيا فى الفراغ أعلى المنابع الحرارية تكنولوجيا واعـــدة الاســـتخدام فـــى تـــبريد الالكترونيات.

ABSTRACT

Laminar mixed convection in a porous channel with two discrete heat sources on the bottom wall was performed experimentally. The main object is to show the effect of using the porous medium in the channel on the heat transfer characteristics. Therefore, three cases were presented; non-porous channel, fully filled porous channel, and partially porous channel, in which the passages above the heat sources are filled with porous layers. A channel of 120x120mm cross section and length of 200mm is fitted in a low turbulence wind channel. The air flow velocity was varied up to 5 m/s which gives a variation of Reynolds number up to 10^4, heat flux was varied up to $7017 W/m^2$.

Accepted January 24, 2000

Aluminum porous filtration screen sheets of 3mm thickness, with porosity of 0.95, permeability of $1.28x10^{-8} m^2$, density of 122.85 kg/m^3 and Darcy number = $8x10^{-6}$ were used as porous layers to form the porous medium. The results show great effect of using the porous layers on both the temperature and heat transfer. The presence of the porous matrix causes a high resistance to the flow and slows it down and increases the pressure drop along the test section. It also, decreases the local and maximum temperatures along the heat sources to about one third of that for the non-porous channel. It increases the average heat transfer coefficient and average Nusselt number of both heaters by about 253 and 296 percent at Re = 1723 and 5531 respectively. The results for the partially porous channel indicate that almost the same increase in heat transfer, the same decrease in the temperatures and a significant reduction in pressure drop can be obtained compared with the fully porous channel. Therefore, the technique of using partially porous channel appears to be promising for application to electronics cooling.

INTRODUCTION

Due to the continuous miniaturization of electronic devices, the thermal design of these devices is becoming more and more important. Since the purpose of the thermal design is to predict or to control equipment reliability, the temperatures of heat generating components where device failure occurs must be known.

Recently the porous substrates are used to improve convection heat or mass transfer in channels or ducts in practical applications and in many technological processes in thermal and chemical engineering. As examples, heat exchangers, heat pipes, electronic cooling, filtration, and chemical reactors are mentioned. Indeed, filling the entire channel with a high conductivity solid matrix can significantly enhance the heat or mass transfer rate but at the expense of a considerable increase of the pressure drop.

The vast literature devoted to the various modes of convective heat transfer and relevant configurations along with the associated heat transfer and other correlations has been reviewed on several occasions by Incropera [1], Papanicolaou and Jaluria [2], Bejan and Ledezma [3] and El Kady [4]. Large number of practical situations involves mixed convective heat transfer in which both modes of forced and natural convection effects are dominant. Such circumstances arise when a fluid flows over a heated surface with relatively low velocity. There are some works in the literature dealing with mixed convection in the cooling of protruding heat sources of electronic components. In the work of Habchi and Acharya [5], a numerical investigation was made of mixed convection of airflow in a vertical channel containing partial rectangular blockage (single electronic module) on one channel wall. The wall containing the blockages was assumed to be heated while the other wall was assumed to be adiabatic or heated. Maughan and Incropera [6] made experimental measurements in the thermal entry for various channel inclinations. Kang et. al. [7] obtained experimental results for the cooling of a protruding heat generating module on a horizontal plate in an externally induced forced flow. Mahaney et al. [8] studied the mixed convective heat transfer from an array of discrete heat sources in a horizontal rectangular channel. A comprehensive analysis is made by Kim et. al. [9] of the flow and heat transfer characteristics of a mixed convection in a channel with rectangular blocks on channel wall. Papanicolaou and Jaluria [2] studied numerically the cooling of heat dissipating electronic components, located in a rectangular enclosure, and cooling is effected externally through flow of air. Experimental and numerical study has been carried out by El Kady and Sultan [10] of the mixed convection heat transfer from a constant heat flux, protruding heat source module located on an adiabatic floor of a horizontal channel, and exposed to an externally induced forced air flow.

Few investigators studied forced convection in ducts fully packed with a porous material numerically and experimentally. For example, the numerical work of Koh and Colony [11], based on the Darcy flow model, showed that the insertion of solid matrices in a cooling passages produces a significant decrease of the wall temperature. The same problem was experimentally investigated by Koh and Stevens [12]. Kaviany [13] performed a numerical study based on the Darcy Brinkman model to account for the wall effect in the case of a porous channel bounded by isothermal parallel plates. Poulikakos and Renken [14] numerically examined the effects of solid boundaries, flow inertia, and variable porosity on the fluid flow and heat transfer through porous media bounded by parallel plates and circular pipes. An analytical solution, based on the general flow model, was derived by Vafia and Kim [15] for fully developed forced convection in a porous parallel-plate channel for the case of constant wall heat flux. El Kady et al [16] and El Kady [17] studied numerically and experimentally the fluid flow and heat transfer through cylindrical porous media exposed to both constant heat flux and constant wall temperature.

As recently noted by Huang and Vafia [18], very little work has been done on internal forced convection on porous-fluid composite systems. Forced developed convection in a channel partially filled with a porous medium was studied by Poulikakos and Kazmierczak [19] for a parallel-plate channel and circular pipe, and by Chikh et al. [20] for annuli. For both studies, the Dracy Brinkman model was considered, and analytical solutions were derived for the condition of constant heat flux at the wall. It was found that there exists a critical thickness of the porous layer at which the Nusselt number reaches a maximum. Hadim [21] studied numerically the forced convection in a porous channel with localized heat sources.

In spite of the large number of mixed convective heat transfer applications, the previous studies illustrated the lack of knowledge in this field. In the above literature review, the use of porous medium in the cooling of channels, which have concentrated discrete heat sources, is very rare. Therefore, in this study an experimental work is performed to estimate the effect of using porous layers in the passage above the heat sources on the temperatures and heat transfer characteristics. The present study shows also the difference in the effect of using fully porous channel and partially porous channel, on the enhancement of heat transfer and the power penalty which must be paid to overcome the accompanied pressure drop.

EXPERIMENTAL APPARATUS

Figure 1 shows the test rig, which is constructed for the experimental work. The test section (4) in Fig. 1-a is made of wood of 50mm thickness with 120x120mm square cross section and 200 mm long. Two discrete heat sources (5) are embedded in the bottom wall of the test section, to make the heat source face in the same horizontal inside plane of the bottom wall. Each heat source (5) has dimensions of 40 mm long, 2.2mm thickness, and 120 mm spanwise with a fixed source spacing ratio $S/L = 1.0$.

The test section (4) is inserted, as shown in Fig. 1-a, in a low-turbulence wind channel with 120x120mm square cross section and 1300 mm long (6). A centrifugal blower (10), driven by an electric motor (12) is used to draw the air through the system. The blower discharges to a graduated throttle valve (11) by means of which the air velocity through the apparatus may be regulated. To ensure a fairly uniform flow with negligible turbulence through the test section (4), air is drawn through the apparatus by way of a bell mouth (1) and a fine mesh screen (2) as a setting chamber.

In order to show the effect of partially filling the test section with porous medium, three cases as shown in Fig. 1-b were studied. In the first case, (non-porous case) the air flows through the test section without any obstacles. In the second ease the test section is fully filled with porous layers. In the third ease, the spaces above the heat sources are filled with porous layers, and the test section is non-porous elsewhere.

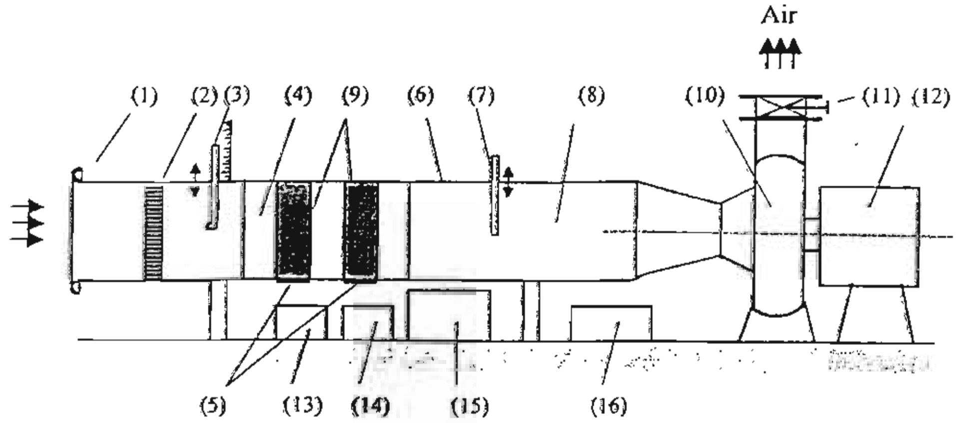
Aluminum filtration porous screen sheets, which are used for the purpose of air filtration in the air conditioning devices, are used to form the porous medium. Each aluminum porous sheet is of 120mm wide, thickness of 3 mm and consists of 3 layers of fine aluminum screens. The porous medium is formed from several layers of the aluminum porous screen sheets, which are fitted horizontally above each other to fill the space above the heaters.

The permeability of the porous medium K is evaluated experimentally as mentioned in Appendix (1) and is found to be $K = 1.28 \times 10^{-8}\ m^2$. The density of the porous sheets was determined experimentally by measuring the mass and volume of the used porous sheets. It took the value of 122.85 kg/m^3. The porosity is calculated as [1.0 – (density of the porous sheets/density of the aluminum metal)]. The porosity was also checked by putting the used aluminum porous sheets in a bottle, which has the same dimensions of the porous block, and the bottle was then filled with water. The porosity was calculated as the ratio of the water volume to the total volume of the bottle. The porosity took the value of 0.95.

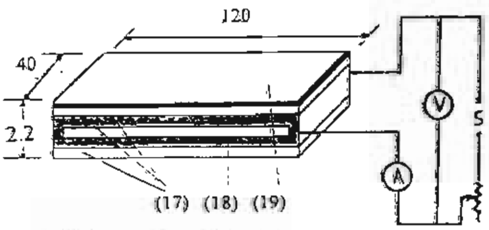
The heat source face is made, as shown in Fig. 1-c, of polished stainless steel sheet (19) with 0.2 mm thickness to minimize the conduction through the heat source surface in the x and z directions. The stainless steel sheet was heated electrically by an electric heater. Two equal lengths of nickel-chromium wire (18) were used as electric heaters for the heat sources. The wire was wounded around a mica sheet (17) of 0.5mm thickness and then was sandwiched between other two-mica sheets (17). An auto-transformer (15) is used to control the heat input to the heat sources as well as one voltmeter (13) and an ammeter (14).

The test section is instrumented by 42 copper-constantan thermocouples. The surface temperatures of each heater were measured by 13 copper-constantan thermocouples; nine of them are arranged in the section z = 0 and the other four thermocouples are distributed at $z = \pm 3$ cm to ensure the spanwise uniformity. The surface temperatures of the unheated sections were measured by 11 thermocouples of the same kind. The thermocouple distribution along the heater surfaces and the base plate is shown in Fig. 1-b. The upper channel surface temperature was measured by means of 4 thermocouples. The entire junction beads (about 0.25 mm in diameter) of the thermocouples are carefully embedded into the wall, and then grounded flat to ensure that they are flush with the surfaces. The temperature signals are then transferred to a data acquisition unit (Yokogawa) of a sensitivity of 0.1°C (16).

The air velocity is measured with the help of a Pitot tube (3) and an inclined alcohol manometer at the centers of eleven imaginary equal areas into which the Pitot

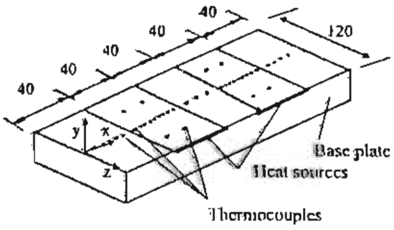


a . Test rig

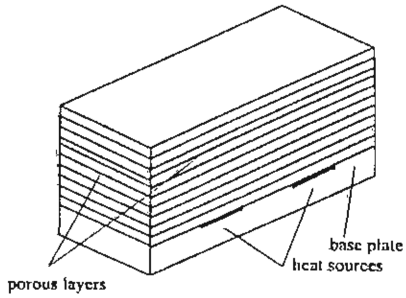


c. Construction of the heat source

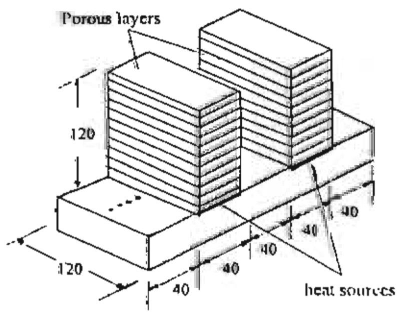
Dimensions in mm.



case 1: non-porous



Case 2: Fully porous



Case 3: Partially porous

b. Channel base plate with heat sources and porous media

Fig. 1 Schematic diagram of test rig

(1) bell mouth,
(2) fine mesh screen,
(3) pitot tube,
(4) test section,
(5) heat sources,
(6) wind tunnel,
(7) hot wire anemometer,
(8) transition piece,
(9) porous media,
(10) blower,
(11) blower control valve,
(12) electric motor,
(13) voltmeter,
(14) ammeter,
(15) auto transformer,
(16) temperature recorder,
(17) mica sheets,
(18) nickel-chromium wire,
(19) Stainless steel sheet

tube is situated and then integrated numerically to obtain the mean air flow velocity. The air velocity is also measured by a hot wire probe (7). The difference in velocity values measured by the two methods is less than ± 1.5%.

The pressure drop across the test section is measured in the cases of porous channels by means of an inclined water manometer. In the case of the non-porous channel the pressure drop has a very small value compared with those of the other two cases of porous layers. It is calculated in this case by the equations of the pressure drop of the flow through the channel.

During the course of the experimental work nearly 1.5 hours were needed to reach the steady state condition. This condition was satisfied when there were no change in the temperature reading within a time period of about 15 minutes.

DATA REDUCTION

The rate of convective heat transfer from the heat source Q was determined from the electrical power input to the heat source Q_e using $Q = Q_e - \Delta Q$, where ΔQ is a small correction for conduction and radiation heat losses from the heat source. The radiative losses are kept low in the non-porous channel case by employing the polished stainless steel sheet as the heater surface. The emmisivity of polished stainless steel is of order 0.17 as mentioned by Incropera and DeWitt [22]. The radiation heat loss was calculated to be less than 2.5 percent of the total electrical power dissipation, using a simple analytical model. In the case of the porous channel, the radiative losses from the porous surfaces (sides) to the channel surfaces tend to be zero because of the very small difference between their temperatures. The conduction heat losses have been minimized by fixing the heat sources to a wood plate of 5cm thickness. The conduction losses through such surface never exceeds 3.5% by the non-porous case and 2.5% by the porous case of the heat electrical heat input to the heat source.

The average convective heat transfer coefficient $\bar{h}$ for a specific heated section can be defined by:

$$\bar{h} = q / (\bar{T} - T_r) \tag{1}$$

Where q is the convective heat flux and is calculated via $q = Q/A$ where A is the exposed surface area of each heater, $\bar{T}$ is the average temperature of each heated surface and T_r is a reference temperature.

The temperature of the flowing air adjacent to the top surface of the channel was measured by 4 thermocouples as mentioned before. The values of the measured temperatures were always equal to the uniform inlet temperature T_o, which means that the thermal boundary layer over both heat sources occurs inside the channel. Therefore, the reference temperature T_r can be taken as the uniform inlet temperature T_o. This value is taken also by Habchi and Acharya [5], Kang et al. [7], Hadim [21], Kim and Annand [23], Nakayama and Park [24], Young and Wafia [25] and Tso et al. [26] among others. The convective heat transfer coefficient in this case is function of the conduction heat transfer coefficient of the porous medium which is an aggregate property of the fluid-saturated porous medium. I.e. it is a function of the conductivity of the solid material, the conductivity of the fluid and the porosity.