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Thermohydraulics of Porous Heat Exchangers: Full or Partial Blockage?

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THERMOHYDRAULICS OF POROUS HEAT EXCHANGERS: FULL OR PARTIAL BLOCKAGE?

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ABSTRACT

This paper examines the thermo-hydraulic performance of different porous liquid-gas heat exchangers. Two categories of such heat exchangers are considered being fully and partly blocked ones. The former completely fills the space between heated pipes or plates containing liquids while the latter only partly fills those spaces. Two different types of heat exchangers in each category are investigated being a shell and plate and a tube bundle. Heat transfer versus pressure drop is plotted as these are the determining factors in most engineering applications. It has been shown that full blockage of the available gas flow area is not necessarily the best design as it can lead to unnecessary higher pressure drop and even lower heat transfer rates compared with partially blocking porous inserts. Hence, proper performance indicators are presented and discussed in details providing enough information for a design engineer to select the best option in each of the three above-mentioned cases.

INTRODUCTION

Porous heat exchangers are receiving considerable attention as their application can lead to high heat transfer rates usually within a limited footprint which could be of significant importance in some engineering applications including air-cooled condensers wherein heat exchanger size determines the fan or the cooling tower size. Like other surface extension approaches, however, this heat transfer augmentation technique causes extra pressure drop. As such, it makes perfect engineering sense to try to minimize the total pressure drop and keep the augmented heat transfer.

Without lose of generality, we focus on gas-liquid heat exchangers where heat has to be transferred between a gas and a liquid. Obviously, the two phases should be separated using a wall. In most engineering applications of this type, the overwhelming resistance is that of the gas side. Therefore, the gas side area has to be increased while the liquid side area is almost always untouched. The increase in the gas side area can come through different techniques among which fins are currently the most popular ones in industry. Fins can be of different shapes, types, and material but the ultimate goal is for them to lead to least possible flow resistance with additional heat transfer, compared to no fin case, of course at a reasonably low price. They have reached a stage that fins can be referred to as a very mature technology with plenty of information about, and even software packages to design, them for specific applications.

Recently, porous heat exchangers, like metal foams, are also suggested as alternatives to fins [1-5]. Even the applications of fin-foamed structures have been reported in the literature [6]. It can be argued that such porous heat exchangers are not understood well and thereby not optimized yet for engineering applications as heat exchangers despite the enormous effort that the heat transfer community has already put in them. One reason that comes to mind is that porous heat exchangers are designed using the same knowledge that we gathered about fins; of course over the years. This, however, is not the best analogy. Recent experimental results, for instance, showed that the wake behind a porous-covered pipe is completely different from those of bare and finned tubes in cross flow [7]. So are the flow structures detaching from the wake [8]. This is to be expected as fins act like narrow channels to guide the gas flow in the preferred direction(s). While similar to fins in leading to boundary layer interruption, porous covers lead to a random flow distribution within the pores with different local heat transfer patterns and wall heat flux split [9]. Furthermore, taking a finned tube bundle in cross-flow as an example, like an air-cooled condenser in a power plant, finned-tubes are spaced very close to one another mainly because the created jet, as a result of the dense tube bundle, significantly enhances the heat transfer and improves the turbulence as the gas flows across the bundle. This, of course, leads to higher pressure drops compared to a single finned-tube in cross-flow as one would anticipate. The immediate question, however, is if we have to design a bundle of tubes with porous covers in a similar way, i.e. dense and thick (porous layers like fins). The effective fin height has been known to us for a while but, to the author's knowledge, there is not effective porous layer height concept in the literature. That is, we still do not know how thick the porous layer has to be in a porous-covered tube bundle. Using the method of Intersection of Asymptotes, this has been partly addressed by Odabaee et al. [10] for a single porous-wrapped tube in cross flow but the work has not been extended to tube bundles which are of significant engineering interest. Some authors, tried to cover the whole available flow area using foams [11,12]. This significantly simplifies the manufacturing process (despite the obvious concerns about thermal contact resistances [13-15]) but, at the same time, leads to significantly higher pressure drops.

Here, one can ask if partial blockage of the available flow area using porous materials and spacing the liquid-gas interface walls away is an answer. The aim of this paper is to answer this question. One, however, notes that with any partial blockage of the flow area, one adds another unknown to the problem being the interface modeling of a porous and non-porous region. As recently underlined by Nield and Kuznetsov [16], this interface modelling remains an open question in the literature. While physically one expects much lower fluid velocity in the pores compared to that of free flow, capturing this sharp gradient at the interface can add to the difficulties of numerical simulation. Experiments addressing this issue are, surprisingly, rare. Beavers and Joseph [17] were amongst the first to show that sharp gradients at the interface between the porous and fluid regions exist. Their work highlighted the existence of a slip velocity at the interface. From there, authors have established different interface conditions that can be classified into two main types according to Alazmi and Vafai [18]: slip and no-slip boundary conditions. Those authors then establish five main categories for the hydrodynamic interface conditions and four categories for the thermal interface conditions that they critically examined. The different models mostly lead to comparable results except for few specific cases. To show the complexity of the problem, it is interesting to note that all these works were conducted for duct flows where there is no recirculation or wakes which cannot be modeled as internal flows. This paper does not aim at solving the interface problem but it presents a critical analysis of the available experimental data in the literature to comment on the overall comparison between the thermohydraulic performances of heat exchangers composed of passages which are fully or partially blocked by porous inserts. Further to information in the literature, some of the experimental data obtained from our experiments at The University of Queensland are presented where data were not available in the literature. Details are, however, not reported to allow for the focus on the main question posed here being about the overall performance comparison of a fully or partly blocked passage of a heat exchanger using porous medium.

NOMENCLATURE

H	=	Height
L	=	Length
p	=	Pressure
Q	=	Heat transfer
W	=	Width

Subscripts

f	=	foam

1 External Flow

Let's start the analysis of the problem by investigating external flow over tubes bundled in a heat exchanger. Sertkaya et al. [11] have tested a radiator-type heat exchanger which can be thought of as a parallelepiped filled with foam. Holes are then drilled in the foam to house the pipes in which heated water is flowing. Air is pushed to flow normal to the water pipes. Those authors tried different air flow rates and plotted the Nusselt number versus the Reynolds number. This is an example of a case when the whole flow area is covered with a porous material. Interestingly, the authors reported higher pressure drops and lower heat transfer rates from the porous structure compared to their tested finned-tube alternatives.

Khasehchi et al. [7,8] and Chumpia and Hooman [19] tested a foam-wrapped pipe as well as a finned pipe in cross flow using simple measurements as well as more involved PIV and hot wire anemometry as Figure 1 schematically shows. Their porous samples were identical to those of Sertkaya et al. [11]. Subsequently, Chumpia [20] tested a single row dense bundle of foamed and finned pipe. The bundle uses the same pipes that were tested as single pipes in cross-flow in [19]. In what follows a brief description of the experiments is provided before results are discussed.

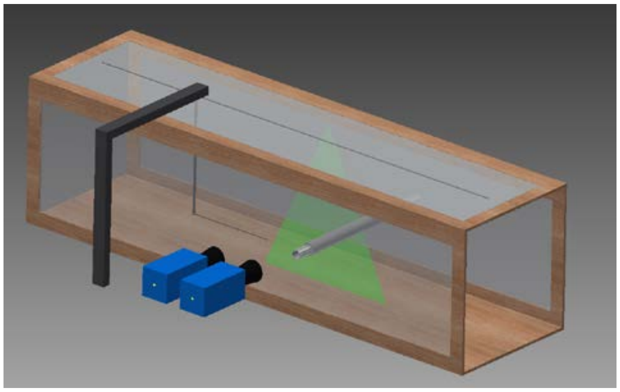

Figure 1: SCHEMATIC DESCRIPTION OF THE CHANNEL

1.1 PIV Measurement Details

Particle Image Velocimetry (PIV) technique has been applied to measure the air velocity outside the porous region and over the interface. Aiming at increasing the spatial resolution of the collected data, two adjacent

CCD cameras (1356×1048 pixel resolution) were used where fine oil droplets (2μm mean diameter) were used as trace particles to move with the flow. Further details of the seeding particles, illumination, optics and the cameras are given in [7] and are not repeated here. The two cameras are synchronized together with the laser pulse at frequency 5Hz. The cameras are fitted with a Micro-Nikkor 60mm lens. For both of them the #f was set at 4 providing 2.5mm depth of view. The time between the laser pulses was set based on the different flow speed to fulfill the one quarter rule (Kean and Adrian [21]). The calibration target including a matrix of 0.5mm diameter dots spaced 5mm apart in the laser sheet position. The displacement vectors are mapped from the image plane to the object plane via a third-order polynomial function; see Soloff et al. [22], to account for any aberrations due to the lenses, Perspex or glass medium and air.

In order to analyse the PIV images, the Dantec PIV software was used. Therefore, single-exposed image pairs were analysed using adaptive cross-correlation algorithm designed for a two-pass multi-grid cross-correlation digital PIV (MCCDPIV) analysis. The first pass used an interrogation window of 64 pixels, while the second pass used an interrogation window of 32 pixels with a discrete interrogation window offset to minimize the measurement uncertainty. The sample spacing between the centers of the interrogation windows was 16 pixels (50% overlap). Flow features were investigated for a range of Reynolds number values were for each of them a total of 3000 images were acquired over different streamwise and transverse locations in each experiment.

1.2 Hotwire Anemometry Details

A Dantec 55P15 single sensor hot-wire probe, 1.25 mm long platinum-plated tungsten wire sensing elements of 5μm diameter, is operated in constant temperature mode. Streamwise velocity fluctuations were acquired at linearly spaced stations along the flow with sufficient sampling frequency to resolve the smallest scales and sufficiently long sample lengths for statistical convergence. Details of these measurements are given in [8] and are not reported here for the sake of brevity. PT-100 RTD probes, accurate within ±0.03˚C, are also used for temperature measurements at the inlet. Downstream of the heated tubes, a traversing system with four PT-100 probes is mounted to scan the exit area using a 100 mm x 100 mm grid area. Liquid inlet and exit temperatures are measured using K-Type thermocouples calibrated against a FLUKE-9142 Field Metrology Well to an accuracy of ±0.001˚C. Data logging and control of different parts of the system such as air velocity and exit air temperature scanning are coordinated by a host computer as described in [19].

1.3 Results

Figure 2 shows the total heat transfer versus pressure drop for different experiments including those of single tubes and single row experiments (for both finned and foamed tubes). These are results for partial blockage which are contrasted to those of full blockage, i.e. those of Sertkaya et al. [11] which are pertinent to a three-row bundle. Bundle results are reported as per-tube heat transfer and per-row pressure drop. One can argue that the single tube is a limiting case for a very sparsely arranged bundle. As seen, both fins and foams are showing higher heat transfer per pressure drop when they are not bundled. This is partly because of extra local (contraction and expansion) losses which are present in bundles. More interestingly, the results of a single tube finned and single row bundle are closer compared to those of foamed tube and bundle. What is even more interesting is the comparison between data from a fully blocked design and that of an isolated tube with partial blockage. As seen, for a given pressure drop the heat transfer to or from a single tube exceeds that of a tube in a fully blocked bundle by an order of magnitude. For instance, with a fan that can overcome 25 Pa of total flow resistance as the pressure drop, a single tube can transfer as much heat as an eight tube single-row bundle if the bundle is fully blocked by the same foam.

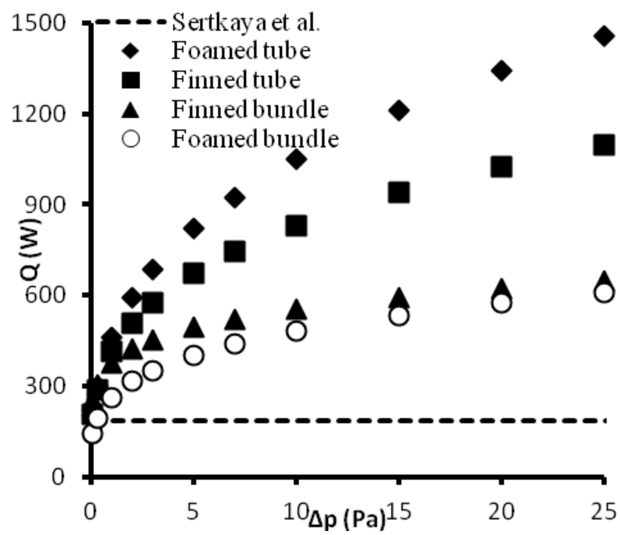

Figure 2: Q (PER TUBE) VS Δp FOR FULLY AND PARTIALLY BLOCKED BUNDLES

2 Internal Flow

There are a large number of papers in the literature looking into thermohydraulics of a porous-saturated duct of rectangular cross-section. Some of them use electric resistance heaters for generating the heat [23,24]. This can be a good model for a shell and plate heat exchanger where the liquid flows in the shell side and the gas is pushed through the plates. As a sample of the available data, we present data from Calmidi and Mahajan [3] where one plate is heated and the other is insulated. A theoretical model was also developed and validated against those experimental data as well as those reported in [23]. Hence, results from theoretical model are also presented to extrapolate the reported data. These results, for a fully blocked channel, are then compared against recent experimental data reported in [26] where only half of the duct cross-section area was covered with the same porous material. In [26] heat is transferred from a hot gas through a thin plate to water flowing in the shell. Similar to the previous case, the total heat transfer is plotted against the pressure drop across the channel.

2.1 Results

Figure 3 is presented to illustrate a comparison between the fully and partially blocked cases. As seen, the results for a fully blocked duct are not as impressive as those of partial blockage. In [26], only half of the cross-sectional area is covered with the same porous material as those in [3]. However, higher heat transfer rate (almost twice) is observed with a fixed pressure drop. One also notes that the results presented in [26] are not optimized ones. That is, one might even get higher heat transfer rates and lower pressure drops if one blocks less or more of the cross-sectional area of the duct.

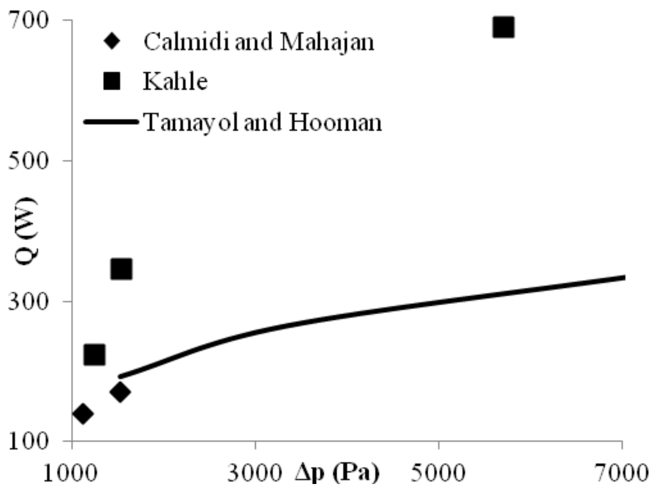

Figure 3: Q VS Δp FOR FULLY AND PARTIALLY BLOCKED DUCTS

The experiments cited in this study were all aiming at heat transfer augmentation to or from a plate separating a gas from a liquid. The gas flow is then pushed through a porous medium layer which covers that separating wall, right under the region marked as foam, as indicated by Figure 4. One can argue that only a part of the gas flows through the porous domain as it offers higher resistance to fluid flow compared to non-porous region. Hence, a thin layer of the porous cover will be conducting heat away from the wall (in case of a gas-cooled heat exchanger). This heat is then convected away mostly at the interface where the resistance to flow is minimal, at least compared to what the flow experiences inside the porous medium.

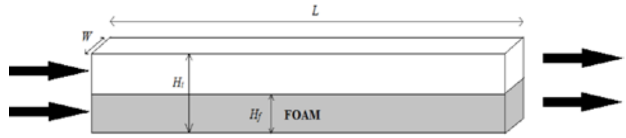

Figure 4: SCHEMATIC VIEW OF THE PARTIALLY BLOCKED DUCT

On the other hand, one expects the flow to only penetrate 3-4 pores deep into a foam layer. As such, one anticipates that the rest of the pores will not participate in convection heat transfer process while the pores closer to the interface are only receiving a portion of the incoming flow rate. An important question here is about determining how much of the approaching air will actually flow through the pores. One argument is to say, on the limit when the porosity of the layer goes to zero, i.e. a solid obstacle is faced, there is no flow split and all the flow has to avoid the solid block. With that, a maximum for pressure drop through a partly porous passage can be obtained, it can be formulated as the sum of a contraction, shear (though a narrower channel), and expansion. The dominant resistant, however, will depend on flow rate and blockage portion. This will be an extremely useful formulation only if one can assume that the streamlines through a porous medium, like the one shown in Figure 4, do not deflect upward. This, however, seems to be the case according to our latest hot wire data collected in our wind tunnel experiment where porous layers, similar to those used in [26], are examined with the main goal of finding the local velocity distribution at the porous-air interface. More experiments were then conducted using PIV to observe a similar trend to what we expected. That is, a part of the flow that enters the porous layer eventually leaves it before reaching the end of the channel (porous layer). This could partly be due to the formation of a recirculation region right downstream of the porous plate. We are currently post-processing the collected data to be reported soon.

CONCLUSIONS

We have exclusively relied on experimental data to show that, with the same pressure drop, higher heat transfer can be obtained by only partially blocking the available gas flow area using a porous medium as opposed to full blockage. This proved to be the case for both internal and external flow as shown by examples. The available data in the literature were used and when not available, experimental data collected at Heat Exchangers laboratory at Queensland Geothermal Energy Centre of Excellence are presented. Modeling the porous-gas interface, especially with external flow, can be a challenging task for which more accurate experimental and numerical modeling is called for. Our preliminary investigation using hot wire and PIV visualization shows very interesting flow features which are not expected based on current theories developed for parallel flows, i.e. those encountered in a partly porous duct. Sample of obtained results will then be presented.

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