Metal foam heat exchanger for dry cooling

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1 EXECUTIVE SUMMARY

1.1 CONCLUSIONS AND RECOMMENDATIONS

Although foams provide an improvement in heat transfer over finned tubes conventionally used for air cooling, this comes at a cost of very much increased pressure drop. Under conditions of high air flow, heat transfer is increased by up to 75%, but the pressure drop for air flow is increased almost ten-fold compared with conventional finned tube systems.

Even at quite low air flows, the pressure drop is several times higher. Since parasitic power consumption (to provide the cooling air flow) is a significant consideration for air large cooled heat exchangers, this trade-off is so unfavourable to metal foam systems that they become unsuitable for these applications.

As a consequence of this finding the project has been terminated after about 1 year into the 2 year original proposal. Since the early outcome demonstrated that the process was unlikely to be successful for the target applications in practice, further aspects of the project (mainly modelling of how a system might fit into a power station as a case study, and a roadmap for further development) have been abandoned.

Foam heat exchangers may have application where heat exchanger selection is dominated by volume, weight or footprint considerations and (gas side) pressure drop is not an important selection criterion.

1.2 BACKGROUND

This project was directed at understanding the opportunities for air-cooled condensers or heat exchangers to replace evaporative cooling in the power industry. For forced air systems, the parasitic power requirement for air cooling is very high. Typically, depending on a number of parameters including the ambient conditions, between 1-10% of a power plant net power goes to air cooling fans. Consequently reducing the work to these directly affects overall total exportable energy. For natural draft systems the available pressure drop due to the chimney effect is intrinsically low. Further, since dry cooling is less efficient than wet systems, the heat exchangers are physically much bigger, which translates into more materials, higher capital costs and large land areas for their location.

Since the overwhelming resistance in air coolers is on the air side, enhancing the air side heat transfer area offers the best opportunity for improvement. This is normally done by using finned tubes.

The technology to be examined in this project is metal foam heat exchangers. These offer the opportunity for a step change in dry cooling systems for large scale applications, initially with forced air systems, but also potentially opening the way for dry natural draft cooling towers.

The concept arises from the from much higher area/volume ratio (around 1000 m2/m3), which is almost 5 times higher than finned-tube bundles. This means that the same weight of metal foam should be 5 times more compact than the fins and consequently also require much less footprint. On this basis, both capital cost and fan power requirements are hypothesised to significantly favour the use of thin layer of metal foams on the outer surface of the condenser tubes, in place of conventional fins.

If very low pressure drops can be accommodated, this opens the possibility for dry natural draft cooling towers - a very attractive prospect with no parasitic power losses, which is currently disregarded as being impossible for power plants in Australia. With a saving in air-side pressure drop, a horizontal arrangement of bundles is possible (instead of currently preferred vertical or inclined arrangements). This gives the heat exchanger a better protection against dust and severe ambient temperature variation.

Further improvement in the performance of metal foams may be possible by compressing the foams, albeit at the expense of higher pressure drops. Given the fact that the manufacturers currently provide uncompressed samples only, an important question is to determine the best compression for different foams, gaining the advantage of improved heat transfer without trading too much additional pressure drop.

1.3 LABORATORY PROGRAM

The laboratory experiments aimed to assess and compare the performance of a finned tube heat exchanger with that of a foam-wrapped exchanger. The tests encompassed two different categories of experiments looking at both closed (double pipe or shell and tube style) and open channel, cross flow arrangements. The former provide better control of the experiments and conditions, and consequently are most useful for developing a physical and modelling understanding of the thermohydraulics, while the latter more closely represent the crossflow, multi-tube arrangements seen in practical air cooling layouts. These tests represent the bulk of the work that was conducted and provide the basis for the studies' conclusions.

2 Scope and Objectives

2.1 Hypothesis

Metal foam heat exchangers offer the opportunity for improvement in dry cooling systems for large scale applications using either forced air or natural draft systems. The hypothesis arises from literature reports that heat exchanger tubes encased in metal foams as a means of extending air side surface area may provide more efficient heat transfer than alternative configurations like finned tubes. Higher efficiency implies better cooling (lower exit temperatures) or smaller heat exchange footprints.

While metal foam encased tubes are currently more expensive than finned tubes, this cost gap may be expected to close as the scale of market for metal foams increases and the foams become 'productized'. While metal foams are currently almost twice as expensive as finned-tube bundles they offer a good potential opportunity for significant cost reduction through improved manufacturing. On an area/volume ratio they are up to 5 times better than the finned-tube bundles, which should translate into a very compact design.

2.2 Technical Scope

The technical scope as outlined in the proposal document encompassed:

1. Provide a process design case study of metal-foam air-cooled dry coolers in large scale cooling duty (e.g. a 450MWe power station based on the Tarong N station, a modern supercritical coal fired
   power station). The goal is to obtain a 50% increase the overall heat transfer and at the same pressure drop, compared to conventional finned-tube bundles. This will be a cooling tower substitution study, not a full process re-optimisation (for different cooling water temperature). The objective was to be completed in 2 years.
2. Develop optimised design case studies for forced and natural draft systems to illuminate the benefits that may reasonably be achievable with further technical development of foam heat exchangers. The objective was to be completed in 2 years.
3. Cost out the operation benefits and capital cost window for large scale metal foam heat exchanger deployment, and develop a road map for this. The road map was to be completed in 1 year. The opportunity window, based on a typical power station case study, was to be progressively evolved over 2 years.
4. Conduct trials on small scale bundles (e.g. 5-10 tube systems) to establish performance experimentally, conduct detailed technical evaluations, understand the dusting/fouling issues and investigate through wind tunnel experiments, mitigation options. The experimental phase was to span 2 years.

In implementation, the experimental trials were performed first in order to provide the technical information necessary for the modelling and economic study. In the event, the experiments have demonstrated that there is little prospect that metal foams will sufficiently outperform conventional finned tubes. On this basis the cooling substitution study, and associated modelling, optimisation and 'roadmapping' are not warranted and it was consequently recommended and accepted that the project be terminated.

3 INTRODUCTION

The major use of water in power generation is for cooling to condense the low-pressure steam and recycle it. As condensate is formed, the heat removed from it needs to be dissipated to the environment. Cooling may be accomplished by:

1. **Direct or "once-through"** cooling, which can only be done if the plant is close to a very large water body.
2. **Recirculating or indirect cooling** where water is recooled through an evaporative cooling tower. Typically the cooling tower evaporates ~ 5% of the flow, so water must be continually replaced.
3. **Dry cooling** where the cooling is achieved by dumping heat directly to air using a forced or natural draft of air passing across (typically) finned heat exchange tube bundles.
4. **Hybrid systems** which combine wet and dry cooling, for example by using dry cooling for most heat dumping, but augmented by evaporative cooling during particularly hot periods.

In many locations, in particular in the Australian context, water conservation or availability preclude the use of wet cooling, since there may simply be no access to sufficient water being available to the plant site. This obliges major processes to use dry cooing and drives the search for opportunities for water demand reduction for power generation by improving the effectiveness of dry-cooling and hybrid dry-cooling.

Dry cooling suffers two major disadvantages relative to evaporative cooling: the approach temperature is constrained by the dry bulb rather than the wet bulb temperature; and the (sensible) cooling capacity of air is much lower than the (sensible and latent) cooling capacity of evaporative systems.

Despite significant advances in dry cooling systems, there remains significant opportunity to improve these devices further. The goals are to achieve close approach temperature and maximize the heat transfer efficiency.

Improvements may be accomplished by increasing the heat transfer area; increasing the heat transfer coefficient; or by increasing the temperature driving heat exchange ("delta T"). Since the dominant resistance in air coolers is on the air side, it is here where the best opportunity for improvement occurs.

A constraint is that improvements must be achieved with minimum pressure drop. For forced air systems, the parasitic power requirement for dry cooling is very high. Typically, depending on a number of parameters including the ambient conditions, between 1-10% of a power plant net power goes to air cooling fans, so reducing the work to these directly affects overall total exportable energy; and for natural draft systems the available pressure drop due to the chimney effect is intrinsically low.

A common method of improving the heat transfer performance of a heat exchanger is to extend the exchange area. Currently heat transfer augmentation in dry coolers is mainly achieved by fins - but at the expense of significantly increasing the pressure drop. Thus, the target is to find a more efficient heat transfer augmentation method with low pressure drop.

An alternative to finned-tubes is a class of porous materials called metal foams. These provide low densities and novel thermal, mechanical, and acoustic properties mainly because the foams are lightweight with high strength and rigidity and high surface area.

These characteristics are helpful for heat exchange providing enhanced heat conduction pathways in the material struts, very large accessible surface area per unit volume for air contact, opportunities for turbulence enhancement and low weight.

Metal foam heat exchangers offer the possibility for a step change in dry cooling systems for large scale applications, initially with forced air systems, but also potentially opening the way for dry natural draft cooling towers.

In conventional current systems, the heat exchange tubes are finned to increase the air-tube contact area. Metal foam heat exchangers offer an alternative method for extending the surface area, and the possibility for new and highly efficient designs for air-cooled condensers.

Metal foams are highly porous materials consisting of mostly inter-connected and randomly distributed voids. Typically, a foam cell is a near-spherical polyhedron having 14 faces. Each cell face forms an open passage to adjacent surrounding cells in all directions. This porous structure provides high open space but also very high interfacial surface area, reported [T'Joen et al 2010] to be in the range of 500-10000 m2/m3. This makes the material a good candidate for high efficiency, compact heat exchangers.

Foams may use non-metal or metal materials, including carbon, nickel, aluminium, and copper [Mancin et al 2012; Jamin & Mohamad 2007; Zhao et al 2006; Noh et al 2006]. These foams can be manufactured to have a range of pore sizes, usually designated in pores per inch (ppi), and porosities. These two factors may be expected to influence greatly the hydrothermal performance of the foam. Smaller cells lead to a more compact heat exchanger but the pressure drop to push the cooling air through the smaller foam passages will increase.

The optimal design of such heat exchangers and the extent of the tradeoff between high efficiency (small footprint, compact design, possibly lower capital costs) and pressure loss (operating cost), in cross-flow, multitube bundles which might be applied for an air-cooled condenser in a power plant remains essentially unexplored. Further, little is known about the interaction between the two adjacent foam-covered surfaces (two boundary layers). ***This represents the main objective of this project.***

The particular aims, as provided in the project proposal sought to address this in two main aspects:

<u>Commercial:</u> A roadmap including possible deployment costs for metal foam heat exchangers on a large scale is required. Currently, only small scale trials are possible because of supply and (small scale manufacturing) cost constraints. Design requirements, optimization studies to understand the best combination of porosity, permeability, and plant performance in terms of pressure drop and heat transfer are necessary.

<u>Operational:</u> A key issue for metal foam systems is the air side pressure drop, which may also suffer performance deterioration due to dusting and fouling within the foam structure.

4 LITERATURE

4.1 OVERVIEW:

Metal foams represents a new, as yet imperfectly characterized, class of material with low densities and novel physical, mechanical, thermal, electrical and acoustic properties [Dukhan 2012]. These properties have made them popular for industrial applications as filters, batteries, biomedical implants, catalytic reactors and heat exchangers [Dukhan 2012]. They offer potential for lightweight structures for energy exchange and thermal management, exploiting their highly tortuous flow paths which disrupt the thermal and hydrodynamic boundary layers.

Open foams are highly porous and permeable, providing opportunity for good fluid mixing, while the struts in the foam provide a very high interfacial surface area between the void and its solid backbone. Due to their other unique properties of high strength, high absorption to impact, low weight, excellent noise attenuation and other properties [Ashby 2000], metal foams offer new possibilities in emerging industries where these combined properties are sought.

Studies in the literature have used of non-metal and metal foams including carbon, nickel, aluminium, and copper [Mancin et al 2012; Mancin et al 2011; Jamin and Mohamad 2007; Zhao et al 2006; Noh et al 2006]. These foams can be manufactured to have a range of pores per inch (ppi) and porosities. These two properties greatly influence the hydrothermal performance of the foam. Smaller foam cells lead to a more compact heat exchanger but at the cost of higher pressure drop.

Mancin et al (2012) tried aluminium and copper foams and observed higher heat transfer with copper foam, demonstrating that the heat transfer properties of the foam material is important. Jamin and Mohamad (2007) studied a vertical, forced convection tubular heat exchanger in which the tubes were covered with various thicknesses of low porosity carbon foam. The study measured heat transfer rate and pressure drop for different foam thickness and also for a comparable aluminium finned tube. The largest increase in heat exchange was achieved by aluminium fins, which was about three times greater than the best carbon foam case. The largest pressure drop occurred when the heat exchanger shell was completely filled with foam through which all of the air was forced to pass.

Zhao et al (2006) examined heat exchange in a double pipe arrangement where both inner and outer tubes were filled with different metal foams and various fluids. Heat transfer performance was compared with those of two other concentric tube-in-tube designs; one with radial fins and the other with spiral fins, fixed on the outer surface of the inner tube. They reported a significant improvement of heat transfer in the foam-filled tube compared with the internally finned designs. Unfortunately, they presented no information regarding pressure drop for the different configurations.

Noh et al (2006) tested a foam-filled annulus and reported higher heat transfer and pressure drop compared to no-foam double pipe heat exchangers. Mahjoob and Vafai (2008) investigated a counter flow double pipe heat exchanger theoretically for cold water flowing through the inner pipe and exhaust gas flowing in the annular section. According to these authors, heat transfer is increased 8-13 times, compared to no-foam case. Pressure drop also increased, but the authors did not explicitly give the comparison between the foam and bare conditions.

Another consideration is the design and arrangement of the heat exchanger, importantly the extent to which foam fills the flow channels. In general, the thicker he foam layer the better the heat transfer, but also, the higher the pressure drop for a given flow rate.

One distinct application which can take a maximum advantage of all metal foam features and properties mentioned above is that involving high efficiency heat exchange. Three niche technological areas that fit within this broad application are; thermal processes demanding high rate of simultaneous chemical reactions [Boyd & Hooman 2012; Weclas & Cypris 2012], fast heat removal from high power electronic components, and highly efficient heat rejection in power cycles [Ejlali et al 2009] operating at low temperature differentials.

As a consequence the major proportion of open literature on metal foam studies in the past decade has been centred on high performance heat exchangers [Boomsma et al 2003; Hsieh et al 2004; Kim et al 2000; Mahjoob and Vafai 2008; Tadrist et al 2004]. A large number of these studies either concern fundamental investigations of the materials themselves or practical applications dealing with relatively small metal foam volumes. Compact heat sinks for high density and high power electronic components are obvious examples [Moffat et al 2009; Hsieh et al 2004; Battacharia & Mahajan 2004; Boomsma et al 2003].

4.2 PERFORMANCE AND MODELLING

Air cooled tube banks, which are of interest here, are a class of open flow heat exchanger, where one of the flows (the heat sink, air) is not confined or specifically piped within the equipment. Nevertheless, there is a significant and relevant literature associated with more conventional heat exchangers of tubular design which incorporate metal foams, inside or outside the tubes, for enhancement of thermal exchange. This more constrained arrangement informs modelling of how foams operate to change the energy exchange and affect flow dynamics. This section consequently reviews both closed and open flow heat exchangers. Mahjoob and Vafai (2008) have provided a survey on existing heat transfer coefficient and pressure drop correlations of foam heat exchangers from the literature. They considered three main categories of work: (i) correlations based on micro-structural properties of the metal foams; (ii) correlations specific to metal foam shell and tube heat exchangers; and (iii) correlations specific to metal foam open channel heat exchangers.

Closed shell and tube style heat exchangers

Correlations are generally based on the work of Calmidi and Mahajan (2000). This was applied, extended and tested by Zhao et al (2006); Lu et al (2006) and Xu et al (2011) which, together, can predict thermo-hydraulic performance of concentric tube heat exchangers for three main cases, namely where the inner tube of the heat exchanger is filled by a metal foam; where the inner tube is surrounded by a metal foam; and where the inner tube is filled and surrounded by metal foams.

Lu et al (2006) studied forced convective heating of a round tube fully filled with a high porosity metal foam. Uniform heat input was applied at the tube external surface along its total length and a coolant such as air or water flowed inside the tube, through the foam, to remove heat. They applied the Brinkman-extended Darcy momentum model and two-equation heat transfer model based on the work of Calmidi and Mahajan (2000) for porous media to model temperature and velocity distributions of the flow field. These were compared to the measured pressure drop and heat transfer. The results showed the pressure drop varied exponentially with the foam pores density (ppi). Heat transfer depended on four dimensionless parameters, viz., the ratio of the tube radius to the foam pore size (geometry parameter), volume porosity (f), Reynolds number using the tube diameter as the characteristic length, and fluid-solid thermal conductivity ratio. Most importantly, they found that metal-foam filled tubes could significantly enhance the heat transfer, *up to forty times* that of the plain tubes of comparable dimensions.

Further research from the same group [Zhao, et al 2006] used a double pipe exchanger with the annulus filled with foam. The outside wall was well insulated so that there was no heat loss from the exchanger to the surroundings. The exchanger was operated in counter-current flow, with the hotter fluid flowing inside the inner tube while the cooler fluid flowed in the annular section in the opposite direction.

The heat transfer model was again based on the work of Calmidi and Mahajan (2000). Zhao et al (2006) explored different test conditions by varying the diameter of the inner tube and filling the two fluid passages with different foams.

Thermal performance was compared with other extended surface designs; one with radial fins, and the other with spiral fins, fixed inside the annulus. A significant improvement of heat transfer in the foam-filled tube compared with other designs tested. It was also shown that, on both sides of the inner wall separating the two fluid streams, heat transfer performance of the metal-foam filled heat exchanger is a function of the ratio of the flow cross-sectional area and relative pore density of the metal foam filling the respective area. Pressure drop information was not disclosed.

Mahjoob and Vafai (2008) compare experiment to model prediction for a counter flow system exchanging heat between cold water in the inner tube and hot exhaust gas in the annulus. For the conditions tested, the results showed a considerable increase in the heat transfer rate of 8-13 times compared with no metal foam in the inner tube. Regarding pressure drop, the authors did not explicitly give the comparison between the with-foam and without-foam conditions, but observed that for cases when the inner tube was filled with foam, pressure drop was insensitive to the tube diameter, being dominated by the foam micro-structural properties.

Open channel heat exchangers

Jamin and Mohamad (2007) studied external heat transfer by cross flow from a single tube, comparing a bare tube, a conventionally finned tube, a tube covered in (carbon) foam, and a tube partly covered in the foam (tubes with foam covered sections and bare sections alternating so that the foam sections were like wide fins). They concluded that the best results in terms of heat transfer and pressure drop were obtained, by the conventionally finned tubes and this arrangement was the most suitable medium for cross-flow heat exchangers.

A metal foam exchanger comprising a single row of aluminium foam covered tubes was studied by T'Joen, et al (2010), with the aim to achieve a low pressure drop on the air side. The tubes were covered with various thin layers of foam (4-8mm) and with various foam properties. The cores were aluminium tubes with outside and inside diameters measuring 12mm and 10mm, respectively. The effect of different parameters on the thermohydraulic performance was considered, including the Reynolds number, the tube spacing, the foam thickness, bonding material, and the type of foam. The results showed that, providing a good bonding between the foam and the tubes can be achieved, metal foam covered tubes with a small tube spacing, thin foam layer, and made of foam with a high specific surface area potentially offer strong benefits at higher air velocities (> 4 m/s), compared to helically finned tubes. It was also reported that the air only penetrates the foam to a certain depth, resulting in a declining performance improvement as the foam thickness increases. The authors also found that good thermal contact between the foam and the tube was critical to heat transfer and noted that more research is required to develop a cost-effective and efficient brazing process to attach metal foams to the tube cores.

In this regard, de Jaeger et al (2012) have identified four possible methods of connecting the foam to the tube: brazing, co-casting, thermal glue bonding, and mechanical press-fitting. For heat transfer, they found brazing to be the best technique while press-fitting was the worst.

5 LABORATORY PROGRAM

Tests were conducted to compare the performance of a finned tube heat exchanger with that of a foam-wrapped. Two different categories of experiments were conducted being internal and external flow tests. The former refers to a case where air flow is bounded by tube walls while the latter takes into account cross flow over tubes. Internal flow experiments preceded those of cross flow mainly because these tests allow for permeability and inertial coefficient measurement, i.e. foam characterization. In what follows details of the experiments and obtained results are discussed.

5.1 DOUBLE PIPE

Figure 1 shows the schematic view of the test rig while Fig. 2 depicts a photo taken from the rig (with the sensors removed). It consists of two loops; one for the heating water and the other for the air. Hot water at a constant from a Julabo circulation heater (Fig. 3) was passed through the inner pipe in counter flow. The circulation heater can keep the bath temperature constant (here 80oC) and circulate it through a loop. The water flow rate is measured using a VorTek Instruments M22 vortex volumetric flow meter (Fig. 4). The temperature was measured at inlet and outlet for both of the fluids. To avoid heat loss, the heat exchanger is covered with 25mm thick insulation (see Fig. 2)

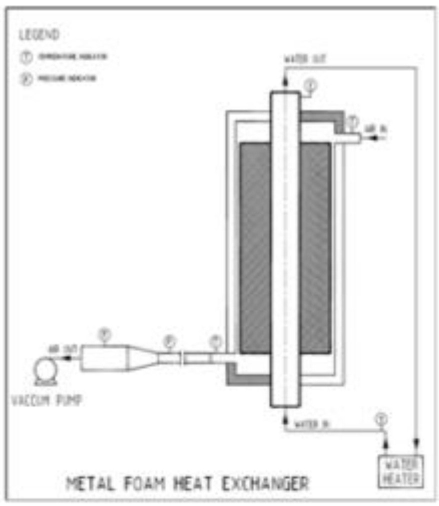


Figure 1 Schematic of experimental setup

Figure 2 Photo of double pipe heat exchanger and foam sample

Figure 3 Photo of temperature controlled circulation heater

Figure 4 Water flow meter

Air was forced through the annulus using a vacuum pump at the outlet side (far left corner of Fig. 2). The air flow was controlled by a valve located before the vacuum pump and the flow rate is obtained from a calibrated Venturi flow meter. The pressure drop across heat exchanger, for various air flow rates, was measured using a Digitron digital manometer, which ranges between 25 mbar up to 10 bar gauge with six pressure and temperature taps measuring at various locations inside the test section were used to measure air pressure and temperature.

As a benchmark case, a conventional double pipe counter-flow heat exchanger (with no foam) is examined where hot water flows in the inner pipe and ambient air is forced through the annulus. For all the experiments the system dimensions are the same: shell 56mm ID, tube 32mm OD, and tube length 44mm. Four cases are reported: (1) with the annulus empty (conventional double tube); (2) with the annulus completely filled with foam; (3) tube wrapped in 5mm layer of foam , PPI=20 and void fraction = 0.94; tube wrapped in 5mm layer of foam, PPI=20 and void fraction = 0.9. The conditions are summarized in Table 1. Each tube arrangement was tested with different air flow rates. To ensure the repeatability of the experiments, each test was repeated three times. Experiments were done for an air superficial velocity range of 1.5m/s-7.5m/s for the different physical systems shown in Table 1.

Table 1 Sample description

Sample	Description	Foam layer thickness	Porosity (manufacturer data)
1	Without foam	-	-
2	With foam	12 mm	0.9
3	With foam	5 mm	0.9
4	With foam	5 mm	0.94

5.2 ANALYSIS AND RESULTS

An Ergun-type pressure drop model is usually used for metal foams, i.e.

$$\frac{\Delta p}{L} = \frac{\mu V}{K} + \frac{C_F \rho V^2}{\sqrt{K}} \tag{1}$$

Dividing both side of the above equation by the velocity, one has

$$\frac{\Delta p}{LV} = \frac{\mu}{K} + \frac{C_F \rho V}{\sqrt{K}} \tag{2}$$

Plotting the left hand side of Eq. (2), termed as the reduced pressure drop, versus fluid velocity can give the permeability and inertial factor of the foam as Fig. 5 does.

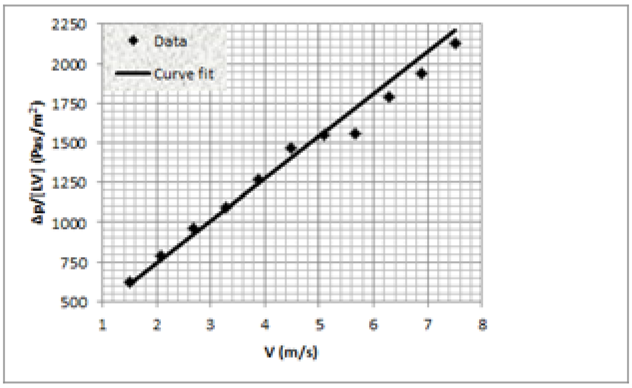


Figure 5 reduced pressure drop vs velocity

In the above plot, the intercept shows the viscosity to permeability ratio from which, with air viscosity measured at room temperature, we have K = 1.04 x10^{-7} m^2. Furthermore, the slope gives the inertial coefficient which is obtained as C_F=0.08.

Figure 6 is presented to show a complete picture of the problem. One observes that the induced extra pressure drop for the thin foams is negligible. This hints that the thin foams result in only marginal extra pressure drop penalties. The fully-foamed case, however, results in significantly higher pressure drop compared with the no foam and thin foam cases. In fact, the total pressure drop is over two times that of no-foam case. The thin foam with the highest porosity can cause up to 9% higher resistance compared to the no-foam case whereas the thin foam with lower porosity (0.9) can increase the pressure drop by a maximum of 20%.

Results, as depicted by Fig. 7, show that heat transfer for metal foam with 24mm thickness are the highest amongst the rest of the other samples and it is more than 3 times that of the no-foam case. According to this figure, heat transfer augmentation rate with velocity varies from one sample to another. The heat transfer rate is lower for lower velocities and it is improved by additional metal foam thickness.

One notes, however, that this increase in heat transfer comes at the expense of a higher pressure drop especially with the thickest foam layer. It is interesting to note that while the thin foams showed almost the same pressure resistances; their heat transfer performance differs by about 30% on average.

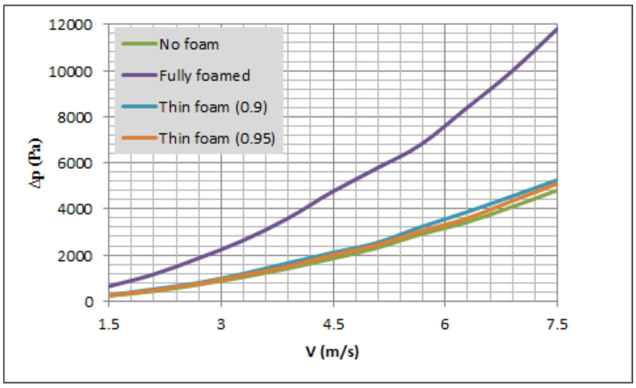


Figure 6 Measured pressure drop for various sample, see Table 1

Overall, performance of these two thin foams seems to be more promising, compared to the fully-filled case. While the pressure drop is almost the same as the no-foam case, the heat transfer augmentation starts from 42% to a maximum of 100% compared to the no-foam case. This further demonstrates the fact that there is an optimal design for the thickness of the foams in heat exchangers and that filling all of the available area in a conventional heat exchanger with foams to maximize heat transfer compromises other performance criteria (like pressure drop) and is not necessarily the best arrangement. However, moving the foam covered tubes further apart to a wider pitch to reduce the pressure drop increases the bundle size, which obviates a key advantage of foam cover. While an optimal compromise can be achieved, the heat transfer improvement never compensates adequately for the pressure drop, compared with finned tubes.

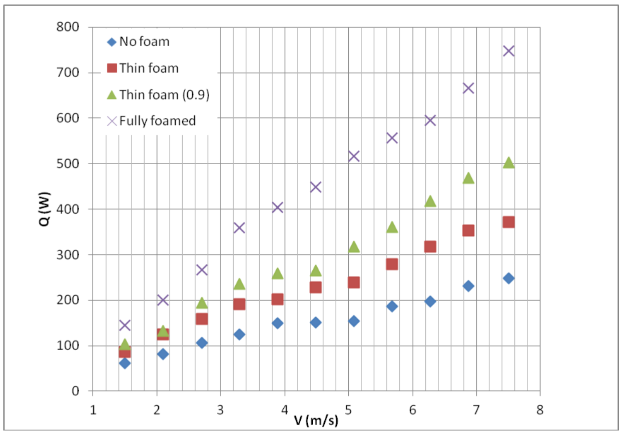
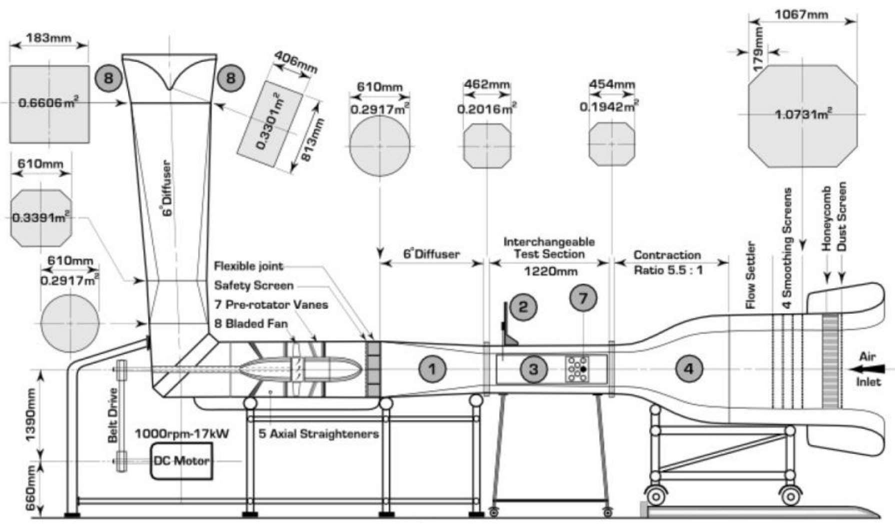


Figure 7 Measured heat transfer for various samples, see Table 1

5.3 Tube bundles

External flow experiments were conducted in the low speed wind tunnel at UQ. Fig. 8 shows a schematic description (a) and a photo (b) of the wind tunnel and its accessories. A uniform air flow through the test section can be obtained and, with no obstruction, a maximum air speed of 40 m/s can be achieved. However, as our heat exchangers induced extra load on the suction fan only a maximum approach velocity of 5 m/s (for the highest cross-sectional area blockage) could be realized which is well within the range of air speed for practical dry cooling applications.

The test facility is an open circuit wind tunnel where the air is drawn into the tunnel from the right-hand side through a dust filter, a honeycomb separator, and 4 sets of smoothing screen. It then passes through the settling section, the constriction plenum of 5.5:1 ratio (4) into the test section (3). In the test section, the air mass flows over the hot surface of the test specimen, takes up heat, and flows into the downstream stabilizing chamber (1). The hot air exits the tunnel through an elbow bend which diverts the air stream out of the system via the ceiling (8). Just before the elbow, the suction blower is installed in-line and the driving shaft extends out to the prime driver which is a large 17kW DC motor. The constriction section (4) has one pressure ring at its inlet, immediately after the flow settler and another at the exit where it joins the test section.



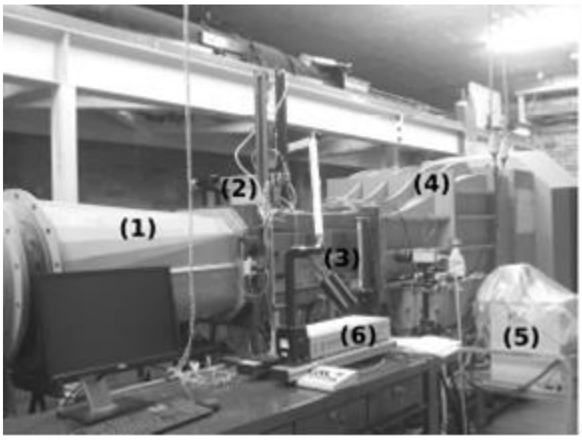


Figure 8 Schematic diagram (top) and photo (bottom) of wind tunnel and accessories

The pressure differentials of the two rings are input to a transducer which generates a signal to drive the control unit of the blower motor. The air velocity is controlled by a PID closed-loop control strategy implemented using LabVIEW software suite. Before the test, the chosen range of air velocity from 0.5 m/s to 5.0 m/s is verified by a Particle Image Velocimetry (PIV) (6) under the empty chamber condition. The test section has its cross-sectional areas measured 454mm x 454mm at the inlet and 462mm x 462mm at the exit. It is 1220mm long and divided into three compartments horizontally by two sets of flexi-glass baffle (see Figure 8). The middle compartment has the cross-sectional area of 454mm (width) x 210mm (height) at its inlet. During the experiments, the heat exchanger specimen is installed in the centre line of the middle section. A pair of pitot tubes are installed either side of, and at the same level to, the specimen whose pressure drop is to be measured. The upstream pitot tube, the specimen, and the downstream pitot tube, are located 193mm, 455mm, and 810mm, respectively from the test section inlet. The pressure drop is measured by a high accuracy pressure differential transducer with a resolution of 0.01 Pa at 200 Pa full scale. A pair of PT-100 RTD probes is installed in the bottom compartment near the test section inlet to measure the air inlet temperature. Exit temperature was measured by an XY traversing system (2) where four PT-100 probes are mounted and can scan the designated exit area of the three compartments at the grid size of 10mm x 10mm, using similar movement to dot matrix printers.

On the liquid side, a hot liquid mixture -made of 1 part glycol + 2 parts water by volume- is heated and maintained at 75^{o}C by the Julabo circulation heater (5). The hot liquid is circulated around a closed circuit through the core tube of the heat exchanger by an in-line pump installed in series with an accurate volumetric flow meter. Liquid inlet and exit temperatures are measured by installing two RTDs on the inlet and exit metal fittings with the tip of each RTD sitting at the centre of the liquid stream.

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. The data file logs air inlet and exit temperatures, hot liquid inlet and exit temperatures, liquid flow rate, and pressure drop across the test specimen. For bundle experiment, two headers are used to distribute the heated water into the heat exchanger tubes and also to return the water to the circulation heater. The liquid pressure and temperature is then monitored and recorded at the headers, see Fig. 9. For each specimen under test, the air flow is set to 0.5 m/s and the liquid temperature at heat exchanger inlet is monitored until it is settled, all relevant data are logged every second for 10 mins. The air speed is then increased to the next step of 0.5 m/s increment and when the liquid temperature re-settles, the process is repeated until air velocity reaches 5 m/s.

Figure 9 Inlet header showing three foam wrapped pipes in wind tunnel

Single tube experiments:

Before moving on to discuss the single row bundle results, a single heated foam-wrapped tube was tested against a finned tube. Four samples of aluminum, and one stainless-steel, foam-wrapped tubular heat exchanger (see Fig. 10) are being tested for heat transfer performance and pressure drop characteristics. The foam layer has thickness (or height) varied from 5mm to 20mm.

Figure 10 Samples tested

The tests are carried out on each heat exchanger, installed horizontally in a cross-flow arrangement inside our wind tunnel, one at a time. Heat transfer rate from 75°C hot liquid, circulating through the core tube, to external air is evaluated. These results, together with temperature differential between the ambient air and the foam surface, allow evaluation of the overall thermal resistance, as illustrated by Fig. 11. Here, thermal resistance is defined as the bath temperature-inlet air temperature difference divided by the heat transferred from the tube. As seen, the heat transfer advantage does not increase linearly with foam thickness - signifying the existence of an optimum thickness when an increase in pressure drop at increased air velocity is taken into account. According to Fig. 11, the heat exchangers with smaller foam layer thickness is less efficient in rejecting heat, i.e. has higher thermal resistance, at the same range of air velocity. This is as expected in all cases of the specimens. Another notable result is that the two samples with the same foam thickness (5mm) have a significant difference in their thermal resistances. The 5mm-SS (s316 stainless steel core tube) performs poorer. This can be attributed to three possible reasons. Firstly, the 5mm-SS sample uses thermal glue bonding method between its foam layer and the core tube. We have previously examined thermal contact resistance (TCR) in a separate study. Depending on air velocity, it was found that thermal glue bonding had TCR between 10% and 19%, (corresponding to air velocity 1.5 m/s to 5.0 m/s) of the total thermal resistance if brazing method TCR is treated to be small and can be disregarded. Secondly, the 5mm-SS has a higher porosity compared to the 5mm-Al (0.937 and 0.901). Higher porosity is associated with less interfacial surface area and it is usually the case that the heat transfer is decreased. Thirdly, sample 5mm-SS has stainless steel core tube with higher wall thickness and lower thermal conductivity value than those of the core tube of 5mm-Al sample. The effect of different surface types on heat transfer can be assessed by comparing the results of Fin-15mm and Foam-15mm. In this comparison, the finned tube has no TCR between all its 89 annular fins and the core tube because the heat exchanger was made as a single piece from a solid aluminum round bar. In contrast the Foam-15mm sample has the maximum TCR because the foam layer and the core tube were press-fit together. Despite the double disadvantage, the Foam-15mm still performs considerably better than the same thickness Fin-15mm. To make the final decision, however, as to go with fin or foam, one has to measure the pressure drop which, inevitably, comes with heat transfer augmentation.

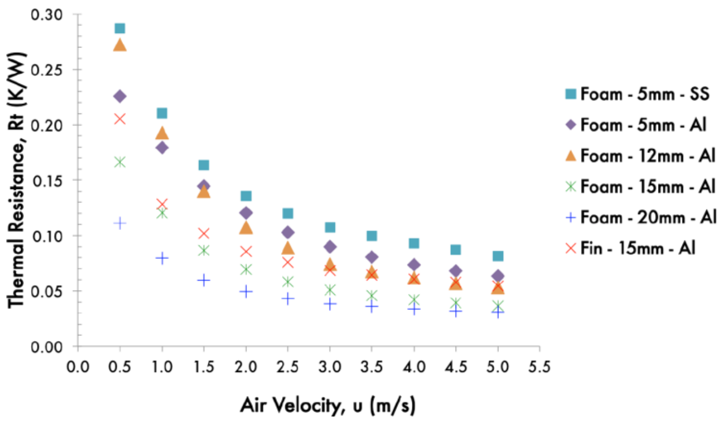


Figure 11 thermal resistance of different foam and finned tubes vs air velocity

Pressure drops across each sample are measured at two imaginary planes perpendicular to the direction of the air flow. The plane upstream locates at 200mm away from the centre line of the test sample, and the one downstream locates 420mm away from the same reference. At each plane, a pitot tube was installed from the top panel to the depth horizontally aligned with the centerlines of the test section and the heat exchanger tube being tested. The difference in total pressure between the two pitot tubes is taken as the pressure drop across the sample. The foam materials covering all foam samples are of similar alloy and having the same PPI density. If two heat exchangers are wrapped with the same thickness, the expectation on pressure drop characteristics should likewise be similar. The pressure drop for all specimens under test, plotted against the air velocity, is shown by Fig. 12. The pressure drop is known to be affected by tangible, macroscopic properties of the specimens. Because the two 5mm foam heat exchangers have very similar physical dimensions and foam specifications, their pressure drop results are therefore similar. The thin layer of thermal conductive glue being added on the external surface of 5mm-SS tube core to bond its foam matrix does not manifest a different effect to the results. Over the whole range of designated air velocity, the general trend of foam thickness toward pressure drop it generates is as expected for all test samples. The sample with the highest add-on thickness generates the maximum pressure drop while those with the lowest thickness generate the minimum pressure drop. The curves diverge toward the maximum air velocity. The effect of different modified surface structure of the same thickness or height, i.e. foam 15mm vs. finned 15mm, to pressure drop is not significant with the finned tends to cause less pressure drop especially toward the higher air elocities. For all samples, their pressure drop up to the air velocity of 2.5 m/s are not varied greatly apart.

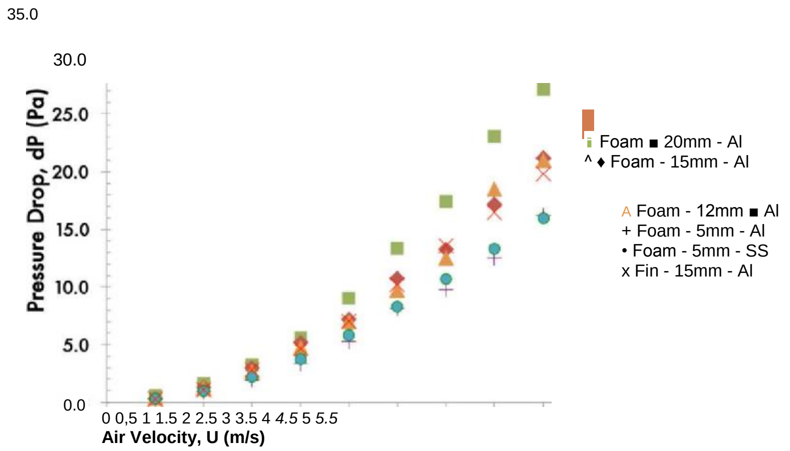


Figure 12 Total pressure drop for different tubes vs air velocity

Finally, Fig. 13 is presented to compare the thermo-hydraulic performance of 15mm foam versus that of the 15mm finned-tube. This plot shows the heat transfer versus pressure drop for the two samples. As seen, the trends of the two curves diverge as the pressure drop increases. This shows that while the increase in the rate of heat transfer from the finned tube slows down, total heat transfer from the foamed tube continues to rise with the increase of pressure drop (i.e. corresponding to the increase of air velocity of the air flow). Put it differently, over the range of the air flow rate under this test conditions, the foam tube exhibits a better heat transfer performance at the same pressure drop. For example, at about 25Pa (corresponding to air speed of 5m/s) Q(foam) = 1450W and Q(fin) = 1060W, a 37% enhancement.

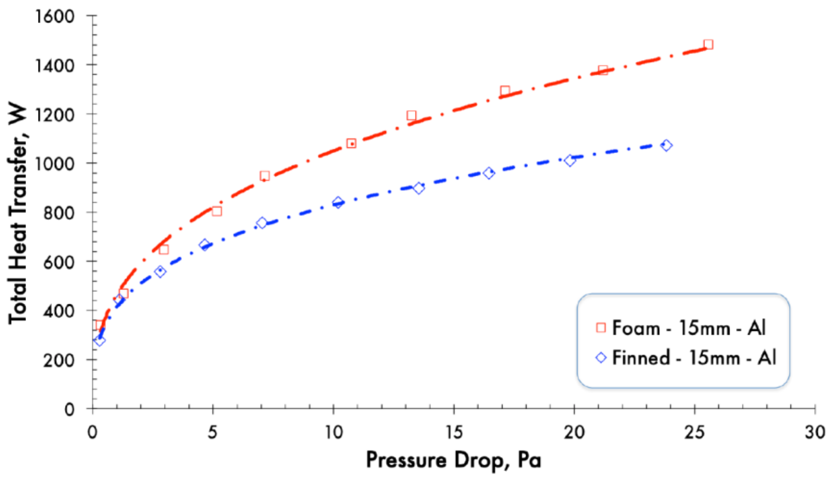


Figure 13 Performance comparison of aluminium foam and finned tube, same ID and OD

Figure 13 performance comparison of aluminium foam and finned tube, same ID and OD

In view of the above, foams show better thermohydraulic performance compared to a finned tube as single cross-flow obstruction. This is in line with our previous numerical simulations, eg Odabaee and Hooman (2011).

Finned and foamed tube bundles

In a final experiment, a single row bundle of finned tubes (see Fig. 14) are tested and compared with an identical foam-wrapped tube bundle. The test procedure is similar to the one described above with the only difference that the heated liquid is now flowing to a header and then is distributed to the tubes instead of going through the tubes directly from the circulation heater. The vertical pitch for the finned and foamed tube bundle is the same and so is the ID and OD (blockage) of the tubes in the two bundles. Fig. 14 shows a photo of the foam-wrapped tube bundle taken inside the wind tunnel.

Figure 14 Front view of foam wrapped bundle

Focusing on fan-cooled condensers, results pertinent to single row bundles are presented in Figures 15 and 16. Fig. 15 presents the total heat transfer from the bundles. A maximum of 75% increase in the total heat transfer is obtained when foams are used instead of fins. While this is significant, Fig. 16 shows the increased pressure drop associated with this heat transfer enhancement is far too high to make foams the favorable option against finned-tube bundles. The foam-wrapped bundle pressure drop is higher by an order of magnitude when the air speed is about 5 m/s. With lower air flow rates, the foam/fin pressure drop ratio is about 5 but the heat transfer gain, according to Fig. 15, is only about 20%.

From these results, foam heat exchangers offer good opportunity for significantly reducing overall heat exchanger size or footprint (by almost half), but this comes at a very substantial cost in terms of pressure drop. Since pressure drop is a major consideration for utility heat exchangers, the cost penalty for incurring this extra pressure drop will never sufficiently compensate for the size reduction in these applications. Indeed, it is probably only where size or space is an over-riding priority, and cost a secondary consideration, that foam heat exchangers will find justifiable application.

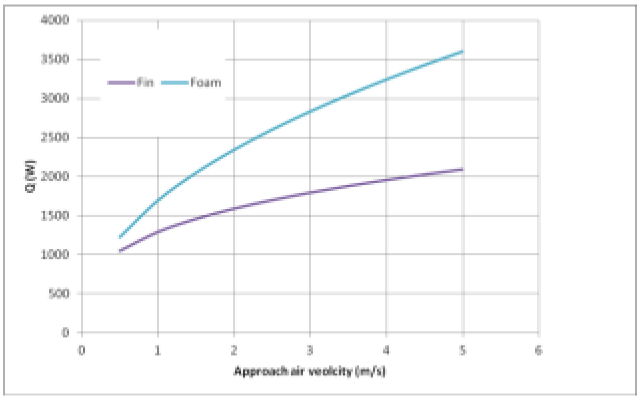


Figure 15 Total heat transfer vs approach air speed for finned and foamed bundles

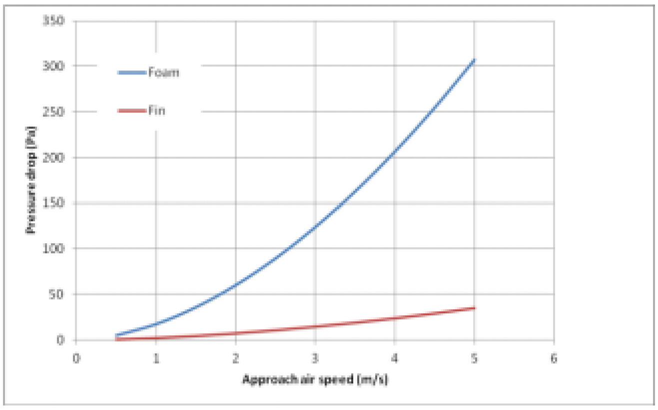


Figure 16 Pressure drop vs approach air speed for finned and foamed bundles

6 KEY OUTCOMES

For open channel cross-flow heat exchangers, foam covered tubes improve heat transfer over finned tubes by up to 75%. The improvement is greater for higher gas velocities. There is diminishing benefit from increasing the thickness of the foam layer.

Pressure drop is substantially higher in multi-tube crossflow exchangers using metal foam for surface extension compared with finned exchangers, by up to an order of magnitude at the higher gas side velocities and by several fold at lower velocities.

The much higher pressure drop makes foam covered heat exchangers unsuitable for large utility type coolers eg for power utilities.

Metal foam heat exchangers may have application where heat exchanger pressure drop is a less important consideration than the size, that is to say where the heat exchanger volume, mass, footprint area or aesthetic appearance is the dominating selection concern.

7 References

Ashby, M.F., Metal Foams : A Design Guide, Butterworth-Heinemann, Boston, 2000.

Bhattacharya, A. and Mahajan, R.L., Finned Metal Foam Heat Sinks for Electronics Cooling in Forced Convection, J. Electronic Packaging, 124, 2002, 155-163.

Boomsma, K., Poulikakos, D. and Zwick, F., Metal Foams As Compact High Performance Heat Exchangers, J. Mech. of Materials, 35, 2003, 116-1176.

Boyd, B. and Hooman, K., Air-cooled Micro-porous Heat Exchangers for Thermal Management of Fuel Cells, Int. Comm. in Heat and Mass Transfer, 39, 2012, 363-367.

Calmidi, V.V. and Mahajan, R.L., Forced Convection in High Porosity Metal Foams, J. Heat Transfer - Trans. Of the ASME, 122, 2000, 557-565.

Cavallini, A., Mancin, S., Rossetto, L. and Zilio, C., Air Flow in Aluminum Foam: Heat Transfer and Pressure Drops Measurements, J. Experimental Heat Transfer, 23, 2010, 94-105.

Chumpia, A. and Hooman, K., Quantification of Contact Resistance of Metal Foam Heat Exchangers for Improved, Air-cooled Condensers in Geothermal Power Application, 18th Australasia Fluid Mechanics Conference, Australian Maritime College - UTAS, Launceston, Tasmania, 3-7 December 2012.

De Jaeger, P., T'Joen, C., Huisseune, H., Ameel, B., De Schampheleirea, S. and De Paepe, M., Assessing the influence of four bonding methods on the thermal contact resistance of open-cell aluminum foam, Int. J. Heat and Mass Transfer, 55, 2012, 6200-6210.

Ejlali, A., Ejlali, A., Hooman, K. and Gurgenci, H., Application of High Porosity Metal Foams as Air Cooled Heat Exchangers to High Heat Load Removal Systems, Int. Comm Heat and Mass Transfer, 36, 2009, 674-679.

Hsieh, W.H., Wu, J.Y., Shih, W.H. and Chiu, W.C., Experimental Investigation of Heat-transfer Characteristics of Aluminum-foam Heat Sinks, Int. J. Heat and Mass Transfer, 47, 2004, 5149-5157.

Jamin, Y.L. and Mohamad, A.A., Enhanced Heat Transfer Using Porous Carbon Foam in Cross Flow - Part I: Forced Convection, J. Heat Transfer - Trans. ASME, 129, 2007, 735-742.

Kim, S.Y., Paek, J.W. and Kang, B.H., Flow and Heat Transfer Correlations for Porous Fin in a Plate-Fin Heat Exchanger, J. Heat Transfer - Trans. ASME, 122, 2000, 572-578.

Lage, J.L., P. S. Krueger, and A. Narasimhan, Protocol for measuring permeability and form coefficient of porous media, Phys. Fluids, vol. 17, 2005, 088101-5.

Lu, W., Zhao, C.Y. and Tassou, S.A., Thermal Analysis on Metal-foam Filled Heat Exchangers. Part I: Metal-foam Filled Pipes, Int. J. Heat and Mass Transfer, 49, 2006, 2751-2761.

Mahjoob, S., and K. Vafai, A synthesis of fluid and thermal transport models for metal foam heat exchangers, *Int. J. Heat Mass Transf.* vol. 51, 2008, 3701-3711.

Mancin, S., Zilio, C., Rossetto, L. and Cavallini, A., Heat Transfer Performance of Aluminium Foams, J. Heat Transfer - Trans. of the ASME, 133, 2011, 060904-1-060904-9.

Mancin, S, C. Zilio, A. Diani, and L. Rossetto, Experimental air heat transfer and pressure drop through copper foams, *Exp. Therm. FluidSci.* vol. 36, 2012, 224-232.

Mancin, S., C. Zilio, A. Diani, and L. Rossetto, Pressure drop during air flow in aluminum foams, *Int. J. Heat Mass Transf.* vol. 53, 2010, 3121-3130.

Moffat, R.J., Eaton, J.K. and Onstad, A., A Method for Determining the Heat Transfer Properties of Foam-fins, J. Heat Transfer - Trans. of the ASME, 131, 2009, 011603-1-011603-7.

Noh, S., K. Lee, and C. Lee, Pressure loss and forced convective heat transfer in an annulus filled with aluminum foam, *Int. Comm. Heat Mass Transf.* vol. 33, 2006, 434-444.

Odabaee, M., K. Hooman, Metal Foam Heat Exchangers for Heat Transfer Augmentation from a Tube bank, *Appl. Therm. Engng.* vol. 36, 2012, 456-463.

Straatman, A.G., Gallego, N.C., Thompson, B.E. and Hangan, H. Thermal Characterization of Porous Carbon Foam - Convection in Parallel Flow, Int. J. Heat and Mass Transfer, 49, 2006, 1991-1998.

Tadrist, L., Miscevic, M., Rahli, O. and Topin, F., About the Use of Fibrous Materials in Compact Heat Exchangers, J. Exper. Therm. Fluid Sci., 28, 2004, 193-199.

T'Joen, C., De Jaeger, P., Huisseune, H., Van Herzeele, S., Vorst, N. and De Paepe, M., Thermo-hydraulic Study of a Single Row Heat Exchanger Consisting of Metal Foam Covered Round Tubes, Int. J. Heat and Mass Transfer, 53, 2010, 3262-3274.

Weclas, M. and Cypris, J., Characterization of the Distribution-nozzle Operation for Mixture Homogenization by A Late-diesel-injection Strategy, in Procs. of the Inst. of Mechanical Engineers, Part D: J. Automobile Engineering, 226, 2012, 529-546.

Xu, H.J., Z. G. Qu, and W.Q. Tao, Analytical solution of forced convective heat transfer in tubes partially filled with metallic foam using the two-equation model, *Int. J. Heat Mass Transf.* vol. 54, 2011, 3846-3855.

Zhao, C.Y., W. Lu, and S.A. Tassou, Thermal Analysis on Metal-foam Filled Heat Exchangers. Part II: Tube Heat Exchangers, *Int. J. Heat Mass Transf.* vol. 49, 2006, 2762-2770.