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Thermal Applications of Cellular Lattice Structures

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Abstract. Numerous methods have recently emerged for fabricating cellular lattice structures with unit cells that can be repeated to create 3D space filling systems with very high interconnected pore fractions. These lattice structures possess exceptional mechanical strength resulting in highly efficient load supporting systems when configured as the cores of sandwich panels. These same structures also provide interesting possibilities for cross flow heat exchange. In this scenario, heat is transported from a locally heated facesheet through the lattice structure by conduction and is dissipated by a cross flow that propagates through the low flow resistant pore passages. The combination of efficient thermal conduction along the lattice trusses and low flow resistance through the pore channels results in highly efficient cross flow heat exchange. Recent research is investigating the use of hollow truss structures that enable their simultaneous use as heat pipes which significantly increases the efficiency of heat transport through the lattice and their mechanical strength. The relationships between heat transfer, frictional flow losses and topology of the lattice structure are discussed and opportunities for future developments identified.

Introduction

Sandwich panel structures based upon highly porous, periodic cellular metal structures have attracted significant interest for load supporting structural applications [1,2]. Traditionally corrugated (prismatic) cellular structures are widely used to dissipate heat because they provide ample opportunity to conduct the heat from the hot facesheet into the web structure [3]. These structures have a single easy flow direction. However, such structures have low through thickness strength. Cellular metals with open cell topologies have interconnected pore channels in two or three in-plane directions and are attractive as heat exchange media where dissipation of high intensity heat is required, while multifunctionaly providing structural support.

Consider a typical application where a high heat flux is deposited on one of the facesheets of a sandwich panel structure. This results in a temperature gradient through the thickness of the core. If a coolant flows through the core, heat transfer at the metal – coolant interface occurs and this heat is transported away from the webs raising the average temperature of the coolant and reducing that of the cellular core.

The performance of a forced coolant, sandwich core heat exchanger can be characterized by the heat removed under constant fluid flow velocity and by the pressure difference required to propagate the coolant at that velocity through the flow channels [4]. The thermal efficiency of a sandwich panel can be fully described by three dimensionless parameters, the Nusselt number (Nu_H), Reynolds number (Re_H) and friction factor (f_H), where H is a characteristic dimension (e.g. the cell or core height). The friction factor is given by $2H(\Delta p/L)/\rho_f U_m^2$ where U_m is the mean coolant velocity at the inlet of the heat exchanger, ρ_f is the coolant density and $(\Delta p/L)$ is the pressure drop per unit length [4].

Any open cell metal structure that allows a coolant flow to pass through the pore structure can be used as a heat exchange medium. Both stochastic (metal foams and sintered powders) and periodic topology structures have been made from a wide variety of metals and their heat exchange performance explored [4-7]. Metal foams with open cells have inferior load supporting capability compared to periodic structures of the same weight. However, they provide a high thermal conductivity path for thermal transport, possess a very high surface area for heat transfer to a cooling fluid and provide a contiguous (though tortuous) path for coolant flow through the structure [8,9]. Periodic cellular structures have anisotropic pore structures. For instance, prismatic structures have one low friction flow direction, pyramidal lattices have two and the 3D Kagome and tetrahedral topologies have three easy flow directions. Textile and co-linear structures have one very easy flow direction while flow in others lies between that of the lattices and prismatic structures. The thermal characteristics of periodic cellular structures are therefore orientation dependent.

Textile Based Lattice Structures

The details of manufacturing brazed textile based copper lattice sandwich structures has been described in detail by Tian et al. [10] and the reader is referred to it for details. The overall heat transfer for this forced convective sandwich configuration consists of the contribution of heat transfer from both the solid lattice and cooling fluid. The three main mechanisms of forced convective heat transfer through a cellular textile structure (and other periodic lattice structures also) are: (1) heat transfer through the solid struts by conduction, (2) heat transfer through the textile core by convection and (3) heat transfer from the facesheets by convection, Fig. 1.

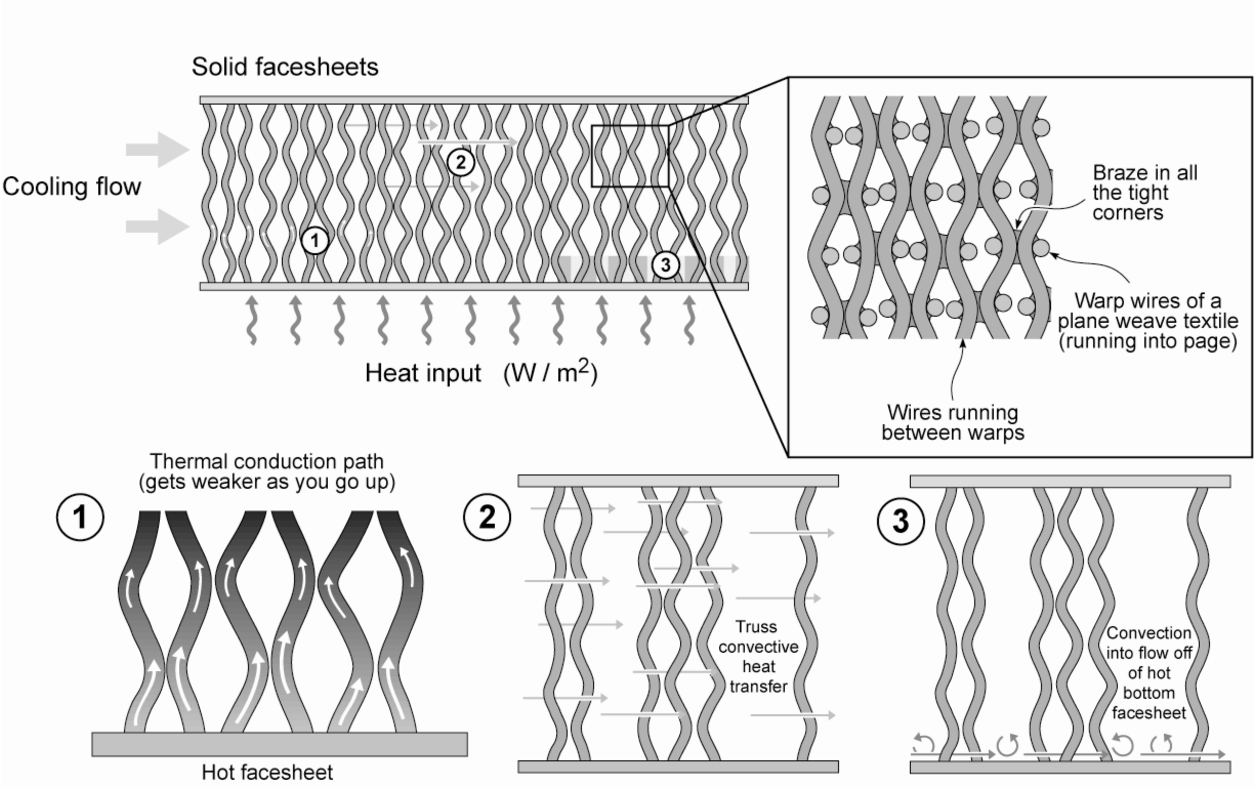

Figure 1 – Schematic illustration of a textile lattice cross flow heat exchanger and the three main mechanisms of forced convective heat transfer.

In general, increasing the thermal conductivity of the textile core increases the heat transfer by mechanism 1, increasing the surface area density increases the heat transfer by mechanism 2, and lowering the core relative density increases the significance of mechanism 3. An increase in the turbulence of the cooling flow causes an increase in heat transfer by mechanism 2 and 3. Typically, increasing the porosity of a textile core increases the surface area density, and hence may increase heat transferred by mechanisms 2 or 3; on the other hand, because of the high porosity (low relative density), there is limited solid conduction passages for heat to be transferred by mechanism 1. It is therefore anticipated that an optimal porosity exists (balance between surface area density and relative density) that maximizes the heat transfer characteristics for textile based sandwich structures.

There have been numerous studies that have sought to characterize the cross flow heat exchange performance of the various cellular structures [4-10]. Several recent attempts have been made to compare them in terms of the dimensionless metrics described previously as a function of the flow velocity expressed in the form of the dimensionless Reynolds number based upon H defined as the core thickness used for measurements [7,10-12]. Figure 2 shows a recent comparison of the performance of aluminum, copper and iron base alloy foams, copper textile structures and various truss structures measured in the easiest flow direction for air and water coolant flows. The metrics for various reference configurations are also shown including Moody's result for an empty channel (a panel with the core removed), a corrugated duct and a louvered fin structure. The most promising structures have a high Nusselt number and low friction factor at the coolant velocity of interest (set by the input thermal flux, the coolant, the required operating temperature and the available fluid pumping capacity).

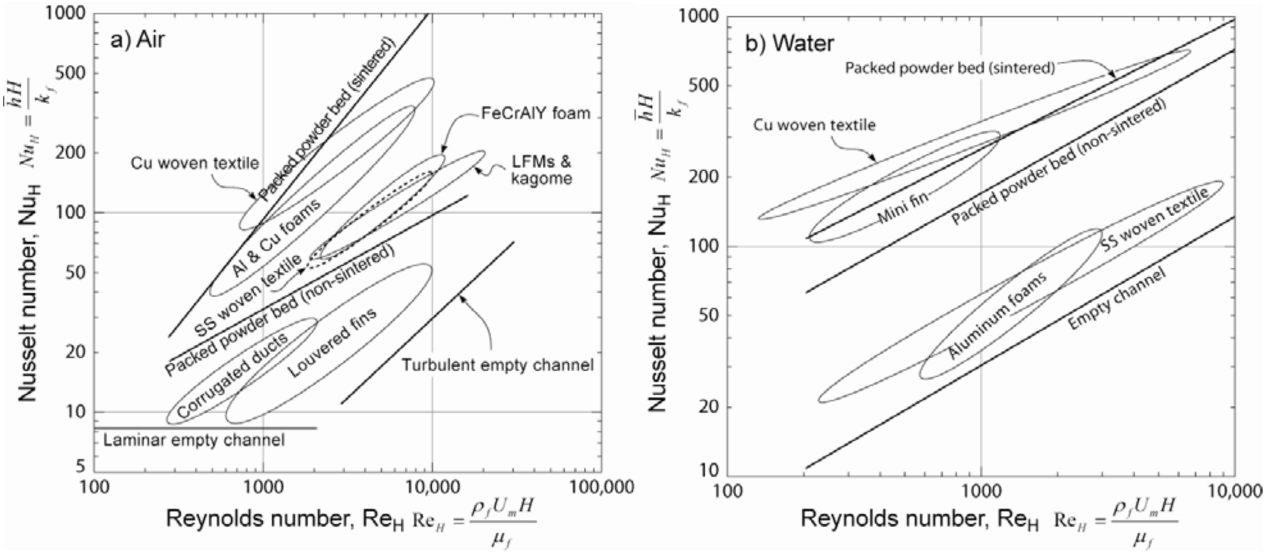

Figure 2 – Comparison of the dimensionless parameters, a) Nusselt number in air and b) Nusselt number in water of textile lattice structures with other heat dissipation media, where $\bar{h}$ is the heat transfer coefficient between the core elements and the coolant, k_f the thermal conductivity of the coolant and μ_f the coolant viscosity.

Several observations can be deduced from these efforts to compare structures. If heat removal and friction factor are equally important, Louvered fins and corrugated structures have the best performance because of their low frictional losses. As friction becomes less of a constraint, or the need to remove heat dominates, the textile structures become interesting because they have a large area of contact with the hot facesheet and present a high surface area to the flow. There are important caveats yet to be resolved for multifunctional structures which must support loads and enable heat transfer to a cross flow. The optimum heat exchanger structures balance the conduction of heat through webs and trusses (maximized by increasing the relative density) against the need to create easy flow paths (maximized by minimizing the relative density). The optimum for heat removal is material selection and application specific but lies in the 10 - 20% relative density range. Strength based sandwich panel optimization (for bending) typically results in structures with much lower core relative densities (~ 2%).

It is interesting to note that the porosity of wire screens tested in this study ranged from 0.68 to 0.8 are significantly smaller than that of typical metal foams, which are > 0.9, but much larger than that of typical packed powder beds, which are < 0.35. Therefore, conduction through the solid ligaments is expected to be more important in bonded woven textile structure than that in metal foams, especially with high thermal conductivity materials.

At high Reynolds numbers, the fluid flow in the textile-based lattice structures is form dominated: the friction factor in all cases is independent of the coolant velocity. The transfer of heat through these textile-based lattice structures depends on two competing mechanisms: conduction through the solid ligaments and forced convection to the applied coolant flow. At a given Reynolds number, the porosity and surface area density are the two key parameters controlling heat transfer. At a given porosity, the heat dissipation rate increases as the surface area density is increased. However, with increasing porosity, conduction decreases and convection increases. Consequently, for a fixed surface area density, there exists an optimal porosity for maximum heat dissipation (a balance between the two mechanisms). For copper textile lattice structures, this optimal porosity is ~ 0.75.

Heat Pipe Lattice Structures

One approach to resolving this dilemma is to increase the thermal conductivity of the lattice structure. This then enables an increase in the conductive transport of the thermal flux using smaller cross sectional area webs or trusses decreasing the flow resistance. The integration of heat pipes into a lattice truss sandwich core offer a novel potential route. An additional advantage of the approach arises when hollow truss structures are utilized since these possess exceptional strength characteristics when compared an equivalent relative density solid lattice trusses [13].

Metal lattice sandwich structures with core relative densities of 10 – 25% are very efficient load supporting structures and the transport of heat from one face to the other occurs partly by conduction via the core elements. In the case of a localized heat source, heat must first be transported via conduction within the hot facesheet and then through the truss core elements to the cool facesheet. Much more effective transport can be achieved by phase-change processes and the integration of heat pipes seek to exploit this phenomenon.

These structures contain hollow interior regions connecting areas close to the source of heat with others that are cooler. When the interior surfaces of this hollow space are coated with a microporous wick structure that promotes capillary driven fluid flow and an appropriate fluid is added to saturate the wick; the system once evacuated and sealed, acts like a closed-loop, two-phase convective system. Heat applied locally to the structure evaporates the fluid and this vapor is rapidly transported to cooler regions where condensation occurs. Evaporation of the working fluid occurs in the hot region and the latent heat of vaporization is absorbed during the vaporization process. The evaporation results in a slight internal temperature increase and, hence, a pressure differential that causes the vapor to flow from the hot evaporator region to the cooler condenser region. The vapor travels rapidly to the condenser region, where it condenses, releasing the latent of heat condensation. This closed cycle has two important consequences: it can result in structures that posses very high specific thermal capacities (because of the very high heat of vaporization for some fluids) and the systems acts as though it has a very high “effective” thermal conductivity because significant thermal energy is transported in the vapor by the latent heat of vaporization/condensation.

This concept has been used to effectively integrate heat pipe technology into the lattice truss elements of a conventional load supporting sandwich panel with a pyramidal lattice core resulting in a truly multifunctional structure capable of supporting load and thermal management. An aluminum sandwich panel with a pyramidal lattice core has been fabricated, integrated with a stochastic open-cell nickel foam wick and the system evacuated and filled with deionized water as the working fluid to form a large-scale multifunctional heat pipe sandwich panel consisting of an integrated heat plate and lattice truss heat pipes. As a result of the container and working fluid materials choices, an electroless nickel barrier layer was plated onto all internal surfaces which were contacted by the working fluid in order to prevent the generation of hydrogen gas within the heat pipe.

The operation of the heat pipe pyramidal lattice has been experimentally validated by applying a constant non-uniform heat flux distribution on bottom facesheet reservoir and monitoring the temperature distribution along the lengths of the four lattice unit cell truss elements. Heat was applied via a small propane torch for ~300 s and removed for ~200 s, which constituted one-cycle which was repeated four times to observe the transient and steady state behavior of the system. Figure 3 (a) – (c) show infrared thermal images of the heat pipe pyramidal lattice at times 200, 600 and 1500 s, respectively. The experimental set-up was arranged such that heating was from the bottom and isolated from the images shown in Fig. 3.

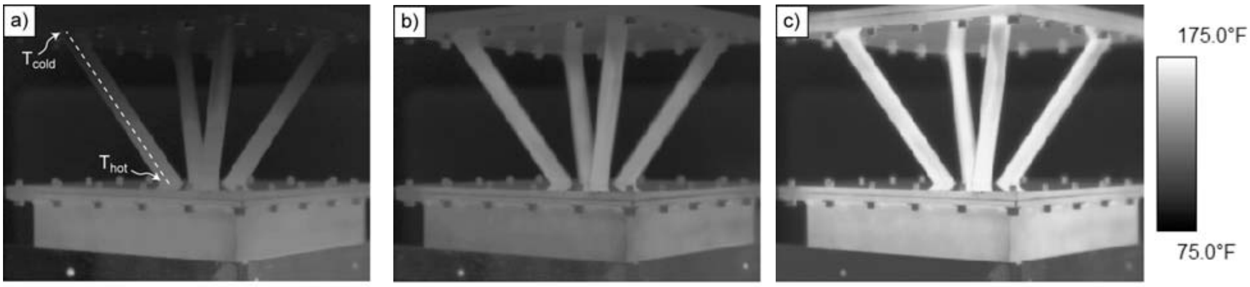

Figure 3 – Infrared thermal images of the heat pipe pyramidal lattice at times a) 200 s, b) 600 s and c) 1500 s.

Two temperature profiles are devised as a measure of the heat pipe performance; the average temperature, T_{avg}, and the temperature difference, $\Delta T = T_{hot} - T_{cold}$, along the length of the pyramidal lattice truss heat pipe, Fig. 4 (a) and (b) respectively. During the initial heating transient, 0 – 200 s, the T_{avg} gradually increases and ΔT rapidly increases. The rapid increase in ΔT is attributed to slow rate of heat transferred into the lattice structure via conduction along the core elements. From 200 – 250 s, T_{avg} rapidly increases and ΔT rapidly decreases corresponding to transient vapor generation within the heat pipes, whereas between 250 – 300 s steady-state operation was observed. During cooling, 300 – 500 s, T_{avg} decreases and ΔT rapidly increases indicating the loss of vapor flowing from T_{hot} to T_{cold}. Subsequent heating and cooling cycling exhibited similar behavior (without the initial transient period). As time progressed, the T_{avg} increased through the heating/cooling cycles because heat was only being removed by radiation and natural convection. However, the ΔT decreased over time (during steady-state operation) indicative of the uniform temperature of the heat pipes. It can also be seen from Fig. 4 (b) that during the fourth heating cycle, the ΔT is ~2°F at T_{avg} of ~150°F along the entire length of the heat pipe. This results in a high "effective" thermal conductivity of the lattice trusses and also increases the forced convection heat removal mechanism by increasing the T_{avg} along the entire length of the heat pipe lattice trusses.

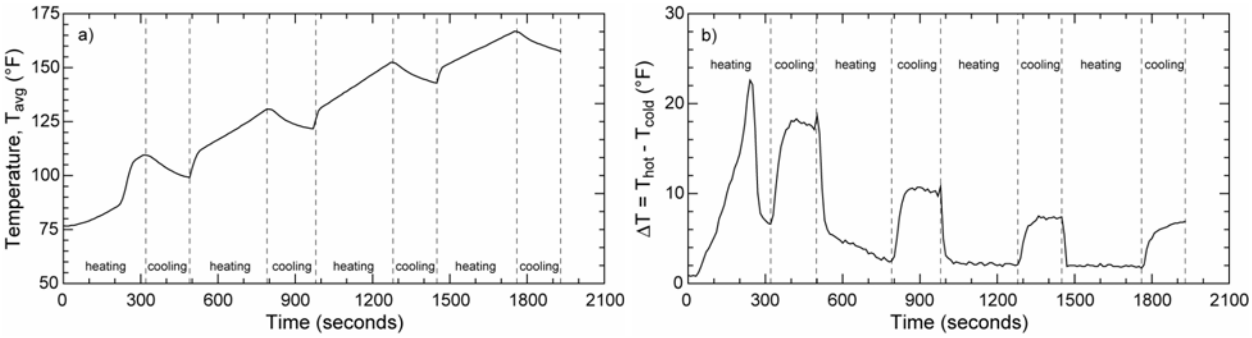

Figure 4 – Plots showing a) the average temperature along the length of a pyramidal heat pipe truss and b) the temperature difference between the hot and cold ends of a pyramidal heat pipe truss.

Summary

Performance charts are presented to compare copper textile-based lattice structures with selected heat dissipation media including stochastic open-cell metal foams, periodic Kagome and lattice frame materials, corrugated ducts and louvered fins. Thermally, the copper textile-based lattice structures perform as well as the stochastic open-cell metal foams (aluminum and copper based), where both have large surface area densities. However, the pumping power required is significantly lower for the textile-based structures, because of the periodic topology. Therefore, their overall thermal efficiency is ~3 times greater than that of comparable weight open-cell copper foams.

An aluminum sandwich panel with a pyramidal lattice core has been fabricated to form a large-scale multifunctional heat pipe sandwich panel consisting of an integrated heat plate and lattice truss heat pipes. This concept has been used to effectively integrate heat pipe technology into the lattice truss elements of a conventional load supporting sandwich panel with a pyramidal lattice core resulting in a truly multifunctional structure capable of supporting load and thermal management. The integration serves as a means for transporting thermal energy from a hot face into the lattice structure both increasing and isothermalizing the temperature of the lattice core thereby increasing the effective thermal conductivity of the lattice truss elements.

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References

[1] A.G. Evans, J.W. Hutchinson, N.A. Fleck, M.F. Ashby, H.N.G. Wadley: Prog. Mater Sci. Vol. 46 (2001), p. 309.

[2] H.N.G. Wadley, N.A. Fleck, A.G. Evans: Comp. Sci. Technol. Vol. 63 (2003) p. 2331.

[3] W.M. Kays, A.L. London: Compact Heat Exchangers (Krieger Publishing Company, Florida 1984).

[4] S. Gu, T.J. Lu, A.G. Evans: Inter. J. Heat Mass Transfer Vol. 44 (2001) p. 2163.

[5] T. Boger, A.K. Heibel: Chem. Eng. Sci. Vol. 60 (2005) p. 1823.

[6] B.M. Dempsey, S. Eisele, D.L. McDowell: Inter. J. Heat Mass Transfer Vol. 48 (2005) p. 527.

[7] T.J. Lu, L. Valdevit, A.G. Evans: Prog. Mater Sci. Vol. 50 (2005) p. 789.

[8] K. Boomsma, D. Poulikakos: Inter. J. Heat Mass Transfer Vol. 44 (2001) p. 827.

[9] A. Bhattacharya, V.V. Calmidi, R.L. Mahajan: Inter. J. Heat Mass Transfer Vol. 45 (2002) p. 1017.

[10] J. Tian, T. Kim, T.J. Lu, H.P. Hodson, D.T. Queheillalt, D.J. Sypeck, H.N.G. Wadley: Inter. J. Heat Mass Transfer Vol. 47 (2004) p. 3171.

[11] T. Wen, J. Tian, T.J. Lu, D.T. Queheillalt, H.N.G. Wadley: Inter. J. Heat Mass Transfer, 2005, submitted.

[12] J. Tian, T.J. Lu, H.P. Hodson, D.T. Queheillalt, H.N.G. Wadley: Inter. J. Heat Mass Transfer, 2005, submitted.

[13] D.T. Queheillalt, H.N.G.Wadley: Mater. Sci. Eng A, Vol. 397 (2005) p. 132.