HEFAT2010
7th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics
19-21 July 2010
Antalya, Turkey

OPEN CELL ALUMINUM FOAM APPLIED IN AN AUTOMOTIVE HEAT EXCHANGER

De Jaeger P.[(1)]*, T'Joen C.[(2)], Huisseune H.[(2)] and De Paepe M.[(2)]
*Author for correspondence
[(1)] NV Bekaert SA,
Bekaertstraat 2, 8550 Zwevegem, Belgium
[(2)] Department of Flow, Heat and Combustion Mechanics,
Ghent University – Ugent,
Sint-Pietersnieuwstraat 41, 9000 Gent, Belgium
E-mail: peter.dejaeger@bekaert.com

ABSTRACT

Heat transfer with open cell aluminium foam has been studied mainly for electronics cooling. In this contribution, we want to show the possibilities of using this foam in an automotive heat exchanger application. More specific, a water cooler for a high performance race car was build and tested. Bonding of the foam was done with an alumina particles enriched epoxy, which introduces an extra thermal resistance compared with brazing technology. Furthermore, it is known that pressure drop of open cell foam versus louvered fins, is rather at the high end. To overcome these issues, use was made of the relatively more isotropic nature of the foam compared to finned structures. This resulted in a completely different shape of the cooler when comparing with conventional rectangular coolers. As a result, heat exchanging surface area could be increased dramatically. The drawback is an increased complexity to design such coolers. Preliminary results have shown a 50% increase of heat transfer compared with a conventional cooler mounted at the same position. This was measured during a wind tunnel test campaign and confirmed by real-time measurements when driving the car. Recalling that the cooler was manufactured with a single epoxy, it is believed that performance can be augmented significantly by a metallic bonding.

INTRODUCTION

Thermal management is critical in automotive systems, with strong influence on aerodynamics, fuel economy, weight, emission and safety. At the heart of such a system lie the heat exchangers which are subject to strict manifold design criteria: high thermal effectiveness per unit pressure drop, low weight, compactness, reduced fluid inventory, mechanical strength, reliability, durability, low cost and low environmental impact during production and disposal. For engine cooling, cross flow heat exchangers with louvered fins are most frequently used and thus can be considered as the reference technology.

A schematic representation is depicted in **Figure 1**. Such coolers yield high heat rejection with low flow resistance, the latter due to a good alignment of the fins with the ambient air flow direction.

NOMENCLATURE

A_{FAD}	[m²]	Air duct frontal surface area
A_{FC}	[m²]	Cooler frontal surface area
A_{FT}	[m²]	Test cooler frontal surface area
c_P	[J/kgK]	Specific heat
d_C	[mm]	Cell diameter
d_S	[μm]	Equivalent strutdiameter
F	[-]	Correction factor
k	[W/mK]	Thermal conductivity
$\dot{m}$	[kg/s]	Mass flow rate
$\dot{q}$	[W]	Rejected heat
R_I	[m^{-1}]	Inertial flow resistance
RP	[%]	Relative performance
R_V	[m^{-2}]	Viscous flow resistance
UA	[W/K]	Thermal conductance
ΔT	[K]	Temperature difference
x	[m]	Cartesian axis direction
y	[m]	Cartesian axis direction

Special characters

ρ	[kg/m³]	Density

Subscripts

Spec	Specification
LM	Log Mean

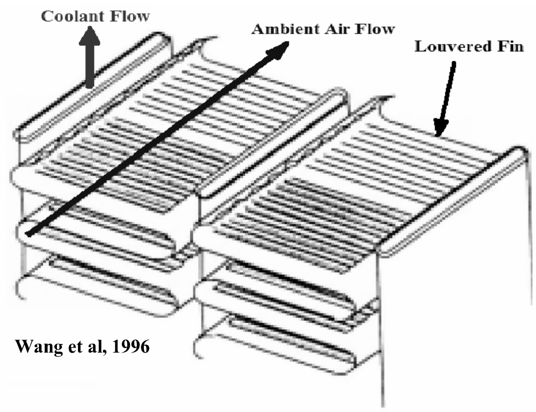

Figure 1 Conventional cross flow heat exchanger with louvered fins

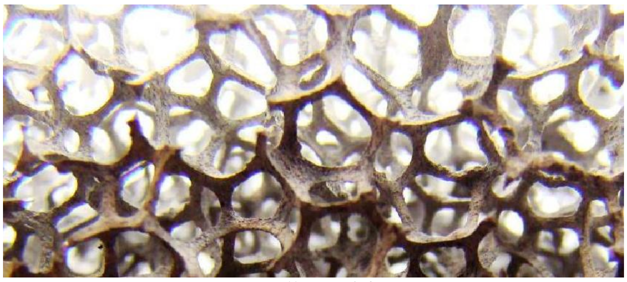

Figure 2 Open cell metal foam structure

To meet the thermo-hydraulic aspects, designers can control three parameters: *heat transfer coefficient*, *heat transfer area* and *flow arrangement* [1].

To improve heat transfer coefficients, research focuses on fin geometry adaptation as well as using relatively new materials like open cell aluminum foam (depicted in **Figure 2**). Because such foams have a high surface to volume ratio, (very) strong mixing capability due to the tortuous flow paths, low weight and relatively high toughness, they are believed to be a potential material for compact heat exchangers.

Tadrist *et al.* [2] described glycol/water-air cooling with aluminum foam brazed between flat tubes. The Colburn j factor for three foam types (10, 20 and 40 PPI) and two tube spacings (2.5 mm and 5.3 mm) was reported with no significant difference. Unfortunately, no comparison with conventional fins was presented. Kim *et al.* [3] experimentally compared thermo-hydraulic properties of conventional louvered fins with ERG aluminum foams. Foam was clamped between isothermal walls to ensure minimal thermal contact resistance. Air was duct-guided through the foam, as occurs in conventional automotive heat exchangers, but at relatively low air velocity (<5 m/s). It was noted that metal foam has comparable thermal performance, albeit at a greater pressure drop; especially at higher air velocities. Thus replacing louvered fins with metal foam in a conventional cooler most likely will result in a less effective heat exchanger.

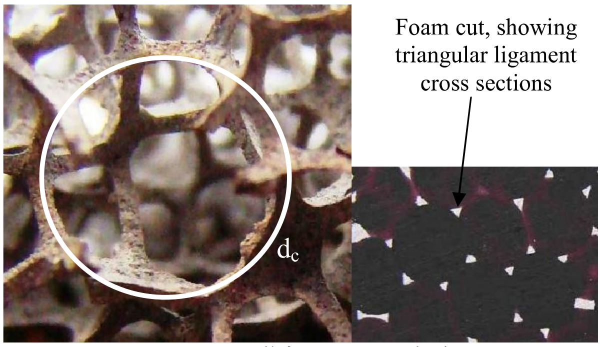

Figure 3 Open cell foam geometrical parameters

The aluminum foam used in this study was manufactured by Bekaert (a Belgian multinational) via a lost wax casting process. The resulting structure is stochastic and statistically described by two parameters: cell diameter (d_c) and a characteristic length for the ligaments between two vertices, called equivalent strutdiameter (d_s). In this study, the latter is defined as the diameter of a circle which results in the same surface area as a cross section of a ligament. Both parameters are shown in **Figure 3** and can be measured by e.g. optical microscopy or X-ray tomography.

Optimization of the foam geometrical parameters makes it possible to maximize heat transfer with minimal flow resistance. Current models demonstrate this possibility [4, 5], but still lack accuracy for automotive heat exchanger design. The reason for this is twofold. First of all, there is still a considerable measurement uncertainty on geometrical parameters (more specifically for the strutdiameter). Secondly, the way the foam is handled during the manufacturing of heat exchangers can change its properties significantly, resulting in totally unexpected results. Typically, the reason is local compression, locally reducing porosity and thus resulting in an increased pressure drop penalty.

An important foam property is that its structure can be considered isotropic. Unlike louvered fins, foam structure has a much less preferential direction concerning air flow. As a consequence, a foam based heat exchanger can be shaped with a higher degree of design freedom, possibly resulting in more heat exchanging surface area for a given volume. Thus the second parameter which designers control is altered (heat transfer area)

The objective of this paper is to demonstrate this design freedom with foam based heat exchangers and compare its thermo-hydraulic performance with conventional louvered fin cross flow heat exchangers. This will be done via a user case in a high performance sports car, namely the Gumpert Apollo.

USER CASE DESCRIPTION

The maximum engine power is 650 HP, producing a potential heat load of ±144 kW. To remove this heat, the car is equipped with 3 conventional coolers, as shown in **Figure 4**. Both side coolers (left and right) are placed vertically, but the middle cooler (shown in detail) is tilted at 32 degrees with respect to the bottom of the air duct.

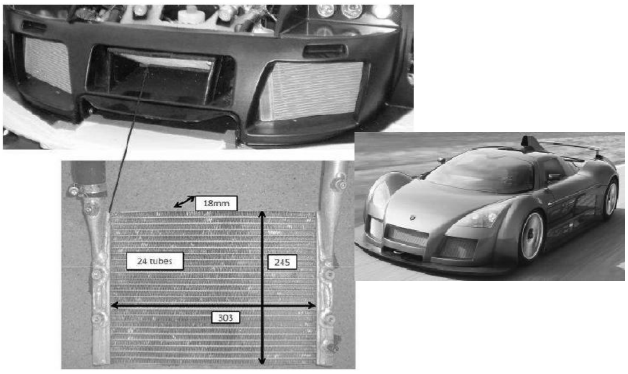

Figure 4 The high performance sports car user case

Further analysis of the manufacturer's data revealed that 30% of the heat load is removed by this middle cooler, representing 43 kW at full throttle. The car manufacturer is seeking to improve its cooling capacity as this opens opportunities to increase engine power. An additional design consideration for racing conditions has to be added. Increasing frontal surface area isn't an option, as it results in poor aerodynamics. A more efficient design will minimise drag and weight while meeting the required cooling specification.

The design specification for the foam based prototype is that cooling capacity should be at least $\dot{q}_{SPEC}$ = 20 kW, under following conditions:

1. It has to fit in current air duct, with frontal inlet dimensions of 0.14 m heigth and 0.41 m width. Thus A_{FAD} = 0.047 m².
2. Coolant is a water/glycol (40%) mixture, with c_p = 3800 J/kgK and ρ = 980 kg/m³
3. Incoming water temperature is 90 °C
4. Water flow rate is 40 l/min
5. Ambient air temperature upfront the cooler is 20 °C
6. Air velocity in front of the cooler is 10 m/s

The last specification is not in contradiction with the earlier mentioned 43 kW, as this occurs at full throttle. Air velocity then exceeds 25 m/s, according the manufacturer's data.

METAL FOAM SELECTION

In a previous study [5], thermo-hydraulic performance of different foams was characterised in a conventional cooler (depicted in **Figure 5)**. Tube spacing was chosen equal to a high performance louvered fin cooler, at 4.85 mm. Frontal surface area is A_{FT} = 210x255 mm².

A qualitative measurement was done in a windtunnel, by mounting the testcooler in the free stream air flow such that air could bypass the cooler. Thus pressure drop penalty is present in the thermal result. Heat rejection ($\dot{q}$) was determined on the water side. To have acceptable accuracy, water flow rate ($\dot{m}$) was measured with a mass flow meter and three PT100 sensors were placed up- and downstream the cooler, from which an average temperature difference (ΔT) could be obtained. Thus rejected heat is given by:

$$\dot{q} = \dot{m} c_p \Delta t \qquad (1)$$

Figure 5 Cooler configuration, used to characterise thermo-hydraulic perfomance of different aluminum foams

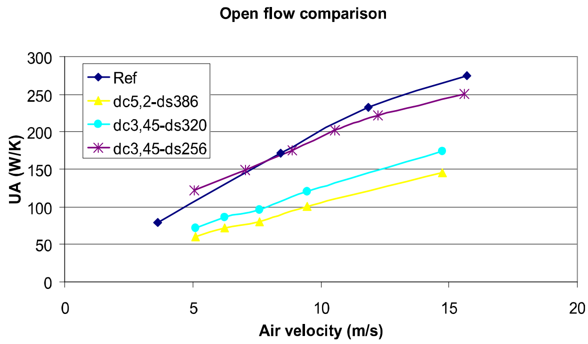

Figure 6 Thermal conductance of aluminum foam versus louvered fins (Ref).

At the air side, flow rate was measured with an orifice, calibrated according ISO5167. Based on this measurement, the average air velocity inside the windtunnel was calculated and used as a reference to compare the coolers. Air side temperature was measured with 2 miniature PT100 sensors before the heat exchanger and 8 sensors 2 cm behind the heat exchanger. As the air flow through the heat exchanger is unknown, heat balance couldn't be calculated.

As all coolers were manufactured with the same type of tubes and in the same manner, convective resistance on water side and conductive resistance towards the foam are assumed to be equal. Thus, differences are completely accounted for by the different foam structures. Thermal conductance was calculated with the LMTD method:

$$\dot{q} = UAF\Delta T_{LM} \qquad (2)$$

The correction factor F was calculated and found to be nearly equal for all coolers (>0.99).

The reference cooler was manufactured by an industrial partner via a conventional Nocolok® brazing technology, with the same type of tubes and inherently will have a lower conductive resistance then the epoxy bonded foam coolers.

The resulting thermal conductance (UA) for the foamed and the louvered fin coolers (labelled Ref) are shown in **Figure 6**. Based on this data, foam type "dc3.45-ds256" was chosen for the prototype. This means that the open cell foam has an average cell diameter of 3.45 mm and the equivalent strutdiameter was 256 µm on average.

COOLER DESIGN AND MANUFACTURING

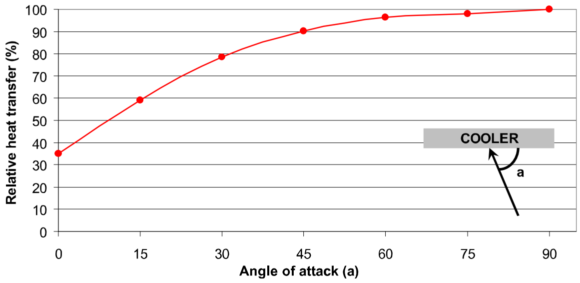

Figure 7 Relative performance by varying the angle of attack.

As the cooler shape will be altered, air flow direction can not be guaranteed to be normal towards the cooler's frontal surface. Extra experiments were conducted to estimate heat transfer variation by varying the air flow direction towards the cooler's frontal surface (defined as angle of attack). This was done at a constant air velocity of 10m/s, with a testcooler manufactured with the "dc3.45-ds256" foam type. The result is depicted in **Figure 7**. A clear drop in thermal performance can be seen by decreasing the angle of attack.

Using these data, the required frontal cooler surface area (A_{FC}) can be rated. Assuming an average angle of attack of 30°, then it can be read from Figure 7 that relative heat transfer performance is 80% compared with a normal incoming air flow (or RP = 0.8). From Figure 6, the thermal conductance at normal airflow at the specified air velocity (10 m/s) is UA = 190 W/K. Based on the earlier discussed problem specifications, the log mean temperature difference can be computed to be ΔT_{LM} = 52.8°C. This assumes that the heat rejected from the water, can be retrieved entirely in the air (20 kW according design specification). If flow depth of the final cooler is the same as of the test cooler, than frontal cooler surface area can be estimated by:

$$A_{FC} = \frac{\dot{q}_{SPEC}}{UA.RP.\Delta T_{LM}} A_{FT}\,, \qquad (3)$$

resulting in A_{FC} = 0.133m². This was rounded to A_{FC} = 0.14 m², to compensate inaccuracy of the data used to determine this estimation.

The final design is shown in **Figure 8**, forming a W-shape. It comprises of 3 header tanks, where coolant is supplied via the middle tank, flowing through the bended tubes towards the left or right tank. Foam is mounted between the tubes spaced at 4.85mm. This final design was obtained iteratively on a computational basis. Foam was modelled as a porous medium with viscous resistance R_V = 2959146 m^{-2} and inertial resistance R_I = 284m^{-1}. The tip position of the cooler was changed along its y-coordinate (see **Figure 9**), to provide a more evenly distributed air flow through the foam. Calculated pressure drop is 1626 Pa for a local air velocity of 20 m/s in front of the cooler, in the symmetry plane.

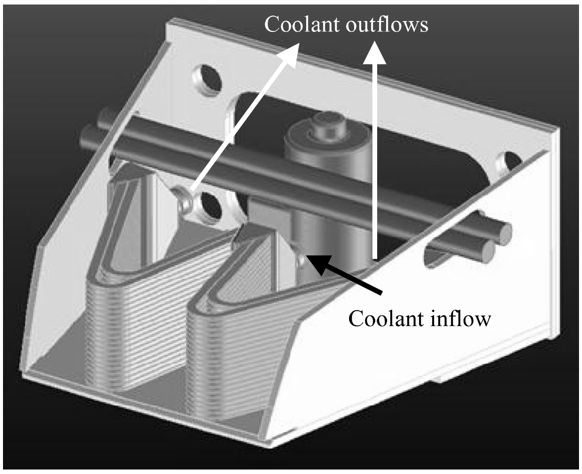

Figure 8 CAD design of the foam based cooler, mounted in the air duct.

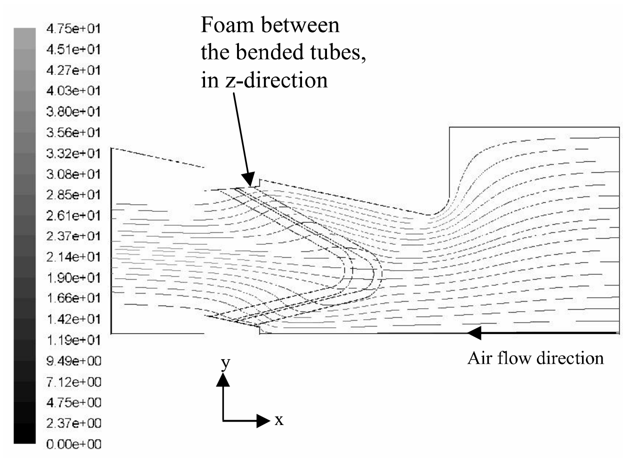

Figure 9 Pathlines of air flow through the air duct and foamed W-cooler

Note that the pathlines indicate a smooth redirection of air flow through the foam, instead of an abrupt direction change perpendicular to the cooler's frontal surface.

Based on this design, a prototype was built, shown in **Figure 10**). Three major manufacturing steps are involved:

1. ***Tube bending*** is done along their most difficult side. This can only be done with tubes of maximum 12 mm width. Burst pressure tests up to 50 bar ensured tube strength after bending. Two stacks are placed after each other for thermal effectiveness.
2. ***Header tanks*** are fabricated in aluminum. Tubes are mounted in the tanks with an aluminum-oxide particles enriched single epoxy.
3. ***Foam bonding*** is also done with the earlier mentioned single epoxy. A thin layer of 200 µm was applied with thermal conductivity k = 0.66 W/mK.

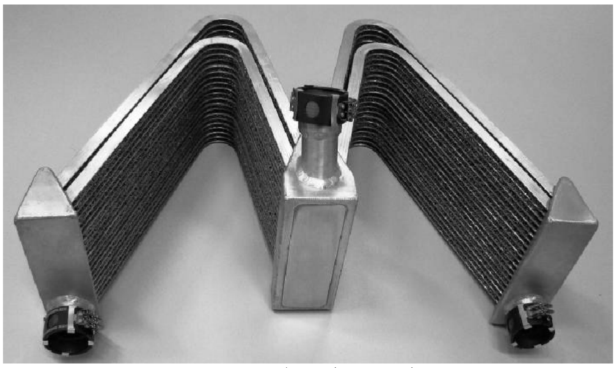

Figure 10 Foam based W-cooler prototype

It should be noted that epoxy bonding is qualitatively inferior to brazing [6]; indicating thermal performance of the prototype will not be optimal. This is also the case from pressure drop standpoint because epoxy layer thickness partially blocks the tube spacing.

EXPERIMENTAL RESULTS

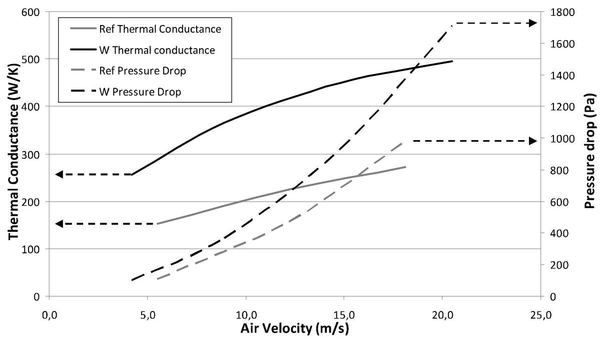

Figure 11 Comparison of foam based W-cooler with conventionally shaped louvered fin reference cooler.

First evaluation is done in a wind tunnel, under steady state conditions. The middle air duct of the sports car was replicated to mount the coolers. Inlet and outlet temperatures of both fluids, flow rates and pressure drop are measured. The thermal conductance is calculated using the earlier mentioned LMTD-method. Air velocity is measured locally in the middle of the air duct. Results are presented in **Figure 11**.

Pressure drop of the foam based cooler exceeds the reference. The reason is an air flow depth of 24 mm through foam (two stack of 12 mm in succession), versus 18mm in the reference cooler. Measured pressure drop at 20 m/s is ±1700 Pa, well fitting the predicted value (1626 Pa). Regarding heat transfer, the foamed cooler is clearly outperforming the reference. Referring to the design spec, a thermal conductance of 380 W/°C can be found at 10m/s air velocity. With a measured LMTD of 52.8°C, this results in 20 kW heat transfer, just meeting design requirements.

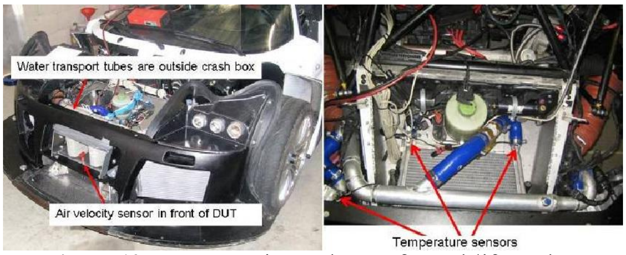

Figure 12 In-car experimental setup for real-life cooler comparison

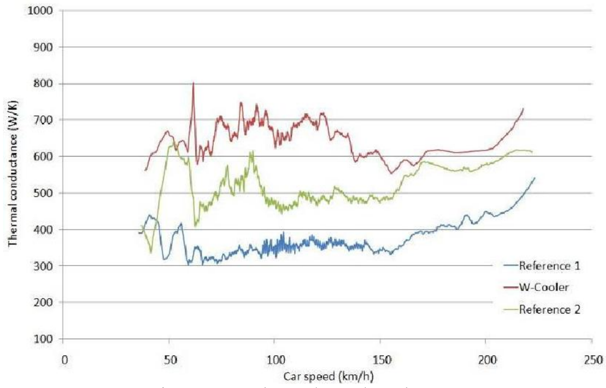

Figure 13 Cooler comparison, based on in-car measurements

Next, an evaluation is done by driving a car with the foamed cooler installed. Test setup is shown in **Figure 12**. A real time data logger records car speed, engine RPM (determines water flow), water temperatures and air velocity in the middle of the air duct. A second reference cooler was also tested. It was mounted like the first reference (see Figure 1), but with core dimensions 303x255x40 mm³. For each cooler, averaged data over 5 laps was taken. Thermal results are shown in **Figure 13**.

Also under highly transient conditions, the foamed cooler is outperforming both reference coolers. Most gain can be observed at relatively lower car speeds (<150 km/h).

CONCLUSIONS

A cooler based on metal foam was designed and built for a high performance sports car. Thermal evaluation on a test rig and in-car, have shown clear benefit in comparison with conventional louvered fin coolers. The freedom to shape foamed coolers for a given volume, in contrast with the conventional method where a volume is designed around a given cooler, is demonstrated.

Besides thermo-hydraulic results, the in-car test drive on a race track also indicated that mechanical stability of the prototype foamed looks promising.

Next step is to further optimize this design and compare with an optimally shaped louvered fin cooler to fully understand the cooling potential of foam.

ACKNOWLEDGEMENTS

The authors are grateful to the Sportwagenmanufaktur Gumpert GmbH for the ability to perform in-car cooler evaluations. This study is supported by the Institute for the Promotion of Innovation by Science and Technology in Flanders, Belgium, through a PhD grant for Peter De Jaeger.

REFERENCES

[1] Shah, R.K., and Sekulic, D.P., Fundamentals of Heat Exchanger Design, *John Wiley & Sons, Inc.*, Hoboken, New Jersey, 2003.

[2] Tadrist, L., Miscevic, M., Rahli, O., Topin, F., About the use of fibrous materials in compact heat exchangers", *Experimental Thermal and Fluid Science*, 28, pp. 193–199, 2004.

[3] Kim, S.Y., Paek, J.W., Kang, B.H., Flow and heat transfer correlations for porous fin in a plate-fin heat exchanger, *Transactions of the ASME: Journal of Heat transfer*, 122, pp. 573-578, 2000.

[4] Schmierer, E.N., Razani, A., Self-Consistent Open-Celled Metal Model for Thermal Applications, *Transactions of the ASME: Journal of Heat transfer*, 128, pp. 1194-1203, 2006.

[5] De Jaeger, P., De Paepe, M., A Simple Model for Foam Based Heat Transfer, *Proceedings of the CellMet 2008 Conference*, Dresden, 2009.

[6] Sekulic, D. P., Dakhoul, Y.M., Zhao, H., Liu, W., Aluminum Foam Compact Heat Exchanger: Brazing Technology Development vs. Thermal Performance, *Proceedings of the CellMet 2008 Conference*, Dresden, 2009.