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A REVIEW OF PARTICULATE FOULING AND CHALLENGES OF METAL FOAM HEAT EXCHANGERS

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ABSTRACT

In recent years, open-cell metal foam has gained attention for utilization as Exhaust Gas Recirculation (EGR) coolers due to its large surface area and porous structure. Theoretically, the porous foam structure would has better transfer heat through conduction and convection processes. However, the exhaust gases that enter the cooler would carry particulate matter (PM) which may deposit within the foam structure. The existing fouling studies cannot explain the underlying mechanisms of particulate deposition thoroughly within the foam structure. This study reviews the present approaches to investigate fouling in the metal foam structured EGR coolers, as well as the fouled metal foam heat exchanger from other applications. In addition, this study also includes the challenges that lie ahead for implementing the metal foam heat exchangers in the industries.

INTRODUCTION

Metal Foam Microstructural & Thermo-physical Properties

The metal foams have been used in the biomedical industry because of their bio-compatibility, in the automotive, aerospace, ship, and railway industries for their light-weight, crash energy absorption, and noise control properties, besides being proposed for heat exchanger industries due to its highly conductive and porous structure (Banhart, 2001). The metal foams are classified into two main categories: (1) open-cell or closed-cell structure and (2) cell arrangement - stochastic or periodic (Han et al., 2012; T'Joen et al., 2010). The open-cell metal foam consists of interconnected cells like dodecahedron shape which allow fluids to flow and the closed-cell metal foam has individual enclosure within the material. The dodecahedron shape is usually modeled as a cubit unit cell in numerical or mathematical studies (Bhattacharya et al., 2002; Odabaee et al., 2013). Figure 1 shows the open-cell and closed-cell metal foams. The open-cell metal foam has been classified based on porosity and pore density– pore per inch (PPI) and it has superior surface area density and thermal performance (Bhattacharya et al., 2002; Ghosh, 2009; Odabaee et al., 2013).

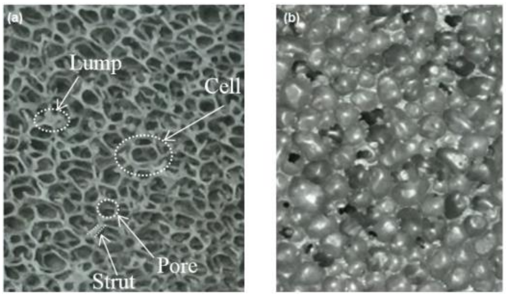

Fig 1. (a) 10 PPI open-cell metal foam (b) Closed-cell metal foam

The high relative density offers high thermal conductivity, while the porous structure offers high convective heat transfer by thermal dispersion and permeability (T'Joen et al., 2010). The permeability increases with porosity and pore diameter, but solely porosity influences the effective thermal conductivity significantly (Bhattacharya et al., 2002; Muley, Kiser, Sundén, & Shah, 2012). However, the convective heat transfer within porous structure is more dominant than conduction as its thermal conductivity is one order of magnitude lower than their parent material (Han et al., 2012). By increasing the porosity, the ligament diameter could be decreased (Bhattacharya et al., 2002), which consequently affects the conduction and the overall heat transfer coefficient significantly (Ghosh, 2009).

Besides, the ligaments show similar concepts of corrugated fins or vortex generators which increase the heat transfer rate by imposing higher mixing flow, especially, at high Reynolds numbers (*Re*) (Ashtiani Abdi et al., 2014; De Schampheleire et al., 2013). The ligament diameter and interfacial velocity influenced the values of Reynolds number (De Schampheleire et al., 2013). The open-cell metal foam heat exchangers show advantages as follows (De Schampheleire et al., 2013; Han et al., 2012; T'Joen et al., 2010):

- Light weight as composed about 90% of air.
- Large specific surface area i.e. 500 to 10,000 m^2/m^3
- High gas permeability and thermal conductivity.

- Resistive to high temperatures, humidity and thermal cycling.
- Excellent fluid mixing as it offers a tortuous flow path.

The thermal performance and the pressure drop of 20 PPI metal foam wrapped on an aluminum cylinder increased with thickness (5, 12, 15 and 20 mm) but comparing the foam to a finned tube at same pressure drop of 25 Pa, the foam with 15 mm thickness has higher the heat transfer up to 37% (Chumpia & Hooman, 2014). By increasing the foam thickness, the air side convective resistance could be reduced as the air penetrated within the foam structure up to 3–5 mm into the foam (T'Joen et al., 2010). However, the same foam thickness could have different thermal resistances due to other factors such as bonding method, air velocity, porosity and surface material (Chumpia & Hooman, 2014). In general, overall thermal resistances of a heat exchanger includes the convective and fouling resistances for both free stream fluid as well as the wall (surface) conductive resistance. For metal foam, the overall thermal resistances also includes the bonding (contact) resistances (Odabaee et al., 2013; T'Joen et al., 2010), and conductive and convective resistances due to ligaments structure.

Exhaust Gas Recirculation (EGR) System

Modern diesel engines have been installed with the EGR system to reduce NO_x emission by recirculating part of the exhaust gas into engine combustion chamber. The exhaust gas contained lower oxygen, but higher heat capacity than fresh air. The recirculation process lowers the combustion temperature through a dilution of fresh air (oxygen concentration) which allows higher mass concentration of the exhaust gas in the combustion chamber (Mirsadraee & Malayeri, 2013; Zheng et al., 2004). Figure 2 shows the NO_x reduction as a function of increased EGR rates for both hot and cooled EGR system (Zheng et al., 2004). Zheng et al. (2004) determined the EGR rate by subtracting the measured fresh intake air using a mass air flow with the estimated mass flow of the cylinder charge. At the same operating condition and 20% EGR rate, the cooled EGR has reduced about 200 ppm of NO_x as compared to the hot raw EGR through a reduction of combustion temperature. Hence, the cooled EGR is favorable in attaining lower NO_x emission.

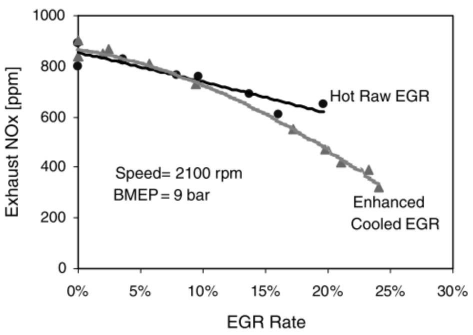

Fig. 2. Exhaust NO_x emission vs. EGR rate (Zheng et al., 2004)

Principally, the hot EGR recycles the exhaust gas directly to the combustion chamber, while the cooled EGR system reduces the exhaust gas temperature using a heat exchanger called as EGR cooler before entering the chamber (Wei et al., 2012; Zheng et al., 2004). Unfortunately, very low combustion temperature may cause incomplete combustion process consequently, more soot particles should be expected (Agarwal et al., 2011). Therefore, a trade-off design between the particle deposition and NO_x reduction is critical in designing the EGR system (Kahle, 2012; Kim e al., 2008; Park et al., 2010). The first EGR cooler design was a shell and smooth tube then, the EGR cooler development continued with corrugated tubes, and rectangular corrugated tubes with housing to optimize the EGR cooler volume and thermal efficiency to match EURO 5 standard and later, the designs considered internal fins to be attached to different geometries e.g. tubes or plates to attain EURO 6 standard (Bravo et al., 2013).

PARTICULATE FOULING

Particulate Fouling in EGR Coolers

Particulate fouling is always related to the EGR diesel engine due to the exhaust gas composition - Particulate Matter (PM) which is a combination of soot, soluble organic fraction (SOF), sulfates and unburned hydrocarbon (M. S. Abd-Elhady, Malayeri, & Müller-Steinhagen, 2011a; Lee & Min, 2014; Warey, Balestrino, Szymkowicz, & Malayeri, 2012). Furthermore, the condensed HC was also considered as the main constituent of the deposit layer due to its significant effects on fouling growth (Warey et al., 2013). The fouling layer has a porous structure with very low thermal conductivity approximate to 0.04 W/m K which acts as an insulating layer (Storey et al., 2013). The temperature of outer layer of the deposit is close to the hot gas temperature. This in turn causes the temperature gradient as well as the particle deposition rate to decrease as shown in Figure 3 (Abd-Elhady et al., 2011; Abd-Elhady & Malayeri, 2013). The fine particles deposited on the heat transfer surface at high deposition rate at the beginning of the deposition process prior to coarse particles, as they have higher sticking velocity compared to the coarse particles (Abd-Elhady et al., 2011a; Abd-Elhady et al., 2004; Santhyanarayanarao et al., 2011).

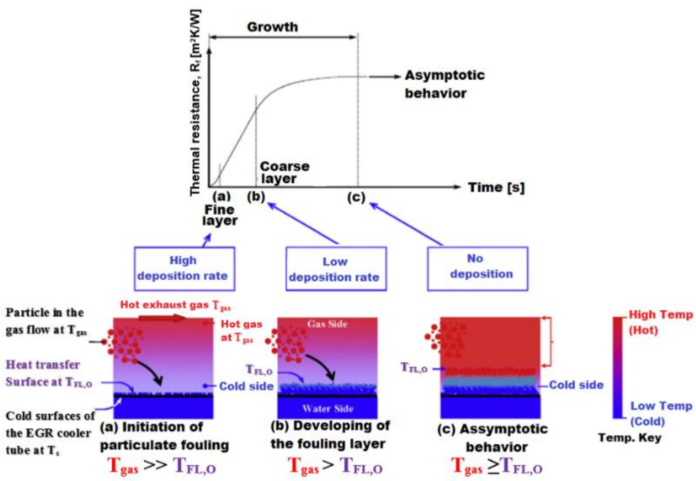

Fig. 3. Stage of particulate fouling

The deposit layer increased the pressure drop thus, affecting the thermo-hydraulic performance of the heat exchanger (Abd-Elhady & Malayeri, 2013; Bott, 1990; Kahle, 2012). The high pressure drop may increase the NO_x as affecting the engine operation but low pressure drop could be handled by the EGR valve by modifying the positon of actuator (Bravo et al., 2013). The increased pressure drop across the EGR cooler may increase the pumping power (Warey et al., 2012). In addition, the exit gas temperature is higher than initial EGR cooler design, thus reducing the EGR efficiency (Bravo et al., 2013). Moreover, the condensation of unburned HC from the diesel exhaust gas (Hong et al., 2011a; Wei et al., 2012), excessive fuel consumption in the engine chamber (Bravo et al., 2013), fouled EGR cooler design, operating mode based on the engine load and corrosion by SOF and PM (Lee & Min, 2014) as well as carbon monoxide (Wei et al., 2012) would intensify the fouling problem. Table 1 shows the typical soot particles and operating conditions in investigating the fouling problem in the EGR cooler.

Table 1. Typical particle properties and operating conditions for EGR cooler fouling test.

Properties	Values	References
Soot particle size	10 – 300 nm	Abd-Elhady & Malayeri, (2013)
Soot particle mass concentration	100 mg/m^3	
Exhaust gas temperature	300 - 400°C (Depend on the load and speed)	
Exhaust gas velocity	30 m/s	
Coolant temperature	80°C	
Coolant flow rate	>1 LPM	Lance et al., (2015)

Fouling Mechanisms and Particle Deposition on Metal Foam Heat Exchanger

Fouling mechanisms in EGR coolers include thermophoresis, electrostatic, eddy diffusion, turbulent impaction and gravitational force (Mirsadraee & Malayeri, 2013; Warey et al., 2012) which depend on the operating mode and the particle properties. Any particle size smaller than 1 µm could be transported through thermophoresis, while the larger particles may experience inertial impaction for flowing exhaust gas in the EGR cooler at 400°C and 30 m/s (Abd-Elhady & Malayeri, 2013). However, the thermophoresis is the dominant fouling mechanism for the EGR cooler due to fine particle size and high temperature gradient (Abd-Elhady & Malayeri, 2013; Kahle, 2012; Warey et al., 2012).

In the meantime, van der Waals forces initiated opposite charges and forced the particles to move toward each other (Storey et al., 2013). Thus, particle agglomeration occurs and the bigger particles deposit on the EGR cooler surface due to gravitational forces (sedimentation). The diffusion occurs as the cooler surface temperature is lower than the material dew point temperature at local pressure, which then, initiates the SOF and HC condensation (Hong et al., 2011a; Lee & Min, 2014). According to Abd-Elhady & Malayeri (2013), the shear forces exerted by the gas flow can dislodge the deposited particles either by rolling or sliding the particles over the surface. The adhesion force and weight could be overwhelmed by higher rolling moment, meanwhile, the drag force may slide (drag) the particles if it is higher than the friction force. Figure 4 shows the acting forces on a particle on a flat plate surface.

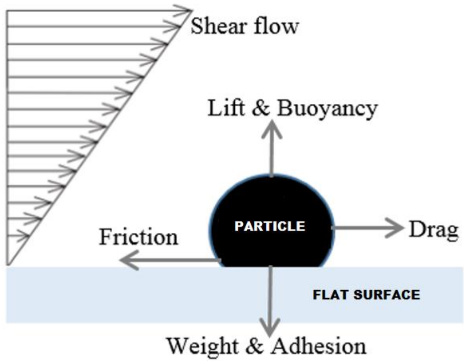

Fig. 4. Forces acting on a resting particle

In addition to the operating conditions, the fouling in structurally various types of heat exchangers could be different, especially for the unique structure likes metal foam. Most numerical studies on fouling of metal foam have considered T'Joen et al. (2010) experimental work for their geometry consideration (Odabaee et al., 2013; Sauret et al., 2013) and validation (Sauret & Hooman, 2014). Earlier, T'Joen et al., (2010) investigated the thermo-hydraulic performance of metal foams with different thickness, 4, 6 and 8 mm glued on aluminum tubes (0.01 m internal diameter and 0.012 outer diameter) and different dimensionless transversal tube pitch of 2.38, 2.68, 3.06 and 3.57 under diverse inlet air velocity of 0.75 – 7.7 m/s. The past numerical studies have considered no slip condition as assuming the particles have the same velocity as the continuous fluid flow (Odabaee et al., 2013; Sauret & Hooman, 2014; Sauret et al., 2013). However, the slip velocity depends significantly on the geometrical parameters of the foam which influence the sharp gradient at the interface of fluid and porous structure (Beavers & Joseph, 1967; Sauret, Hooman, & Saha, 2014a). Odabaee et al. (2013) looked into the effect of dust deposition thickness in the range of 0.01 - 0.2 mm on metal foam thermo-hydraulic properties using CFD simulation. Based on the contours of normalized velocity for both clean and fouled foam with the inlet velocity of 3 m/s, the study has concluded that the air velocity reduced to the lowest values at the forward and backward stagnation point of the foam tube inside the porous region, meanwhile a large recirculation zone appeared at the downstream, right after the tube.

Sauret et al. (2013) investigated the preferential areas of particulate deposition in one row tube bundle wrapped with metal foam by injecting 5000 particles (mean diameter of 50 µm, standard deviation of 50 µm) into the air stream which developed 4 mm deposition thickness. The injected particles were about 3 orders of magnitude larger than those encountered in the EGR cooler work, as indicated in Table 1. The results showed that the backward velocity is noticeably small, about -0.6 m/s (no particle movement) compared to the main jet velocity of 10 m/s. Their results on the backward velocity is similar to another numerical study by Sauret & Hooman (2014). Both studies, (Sauret & Hooman, 2014; Sauret et al., 2013) set the top, bottom and side surfaces of the domain to symmetry and atmospheric pressure at the outlet, but Sauret et al. (2013) set the wall temperature at 353 K and the inlet axial velocity of 3 m/s meanwhile, Sauret & Hooman (2014) used 353 K and 450 K as the wall temperature and 1 – 7 m/s as the inlet axial velocity. However, the ligaments inside the porous medium in the between of foam and tube radius and tube radius were not modeled due to complex of real foam microstructure. By adding more particles up to 7500, the results showed no significant difference as compared to 5000 particles, but reducing the particles amount to 2500 showed significant effects on the deposition process (Sauret & Hooman, 2014). The high deposition rate occurred mostly at the front tube which involved large particles with higher momentum whereas for the tube rear region was subjected to large recirculation zone thus moving the particles back to the tube wall especially for particles < 20 µm (Sauret & Hooman, 2014; Sauret et al., 2013). Figure 5 shows the recirculation and dead zones of metal foam at the rear of the tube, marked as ‘x’ and Figure 6 shows the preferential particulate deposition area which relates the particle size and particle volume fraction (Sauret & Hooman, 2014). The stagnant and recirculation zones exist behind the ligaments (Han et al., 2012) possibly increased the particle deposition within the porous structure, which contradicts the ligaments function to increase the shear force to remove the deposited particles.

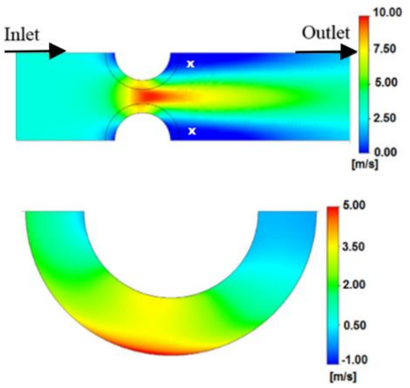

Fig. 5. Axial velocity distribution with 3 m/s inlet air velocity (top) and in the metal foam (below)

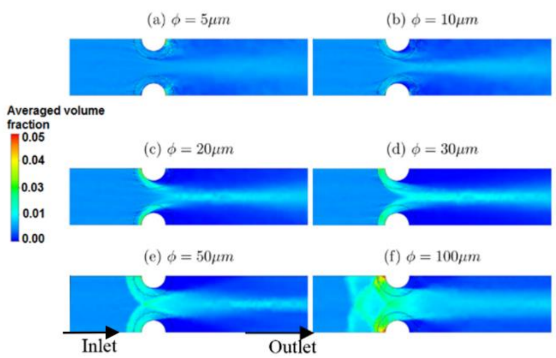

Fig. 6. Average volume fraction of particle with particle size distribution

Besides, the ligaments may block the particle motion, accumulate the particles and develop a fouling layer. The fouling severity depends on the particle size, the pore size and the heat transfer characteristics (Hooman et al., 2012). By considering 60 - 130 nm soot particle, stated that the top layer of a foam plate was fully blocked but partial blockage for underneath area within the ligaments structure and higher PPI clogged faster in a duration of 100 minutes. The clogged porous structure consequently increased the pressure drop (Kahle, 2012; Odabaee et al., 2013). At high velocities, the metal foam area goodness factor reduced significantly due to the increased drag component of the pressure while the skin friction is dominant on the pressure drop at low velocity (Ghosh, 2009). In general, three factors in estimating the propensity of particulate fouling in the gas - side of metal foam are (1) particle volume fraction, (2) particle velocity and (3) particle travelling time (Sauret & Hooman, 2014).

Metal Foam Fluid Flow, Solid-Fluid Interface and Boundary Layer

Sauret, et al. (2014a) investigated the interface boundary condition between gas and porous layer of metal foam with different pore density, foam height and gas inlet velocity. The study stated that at a low gas velocity, the disagreement between numerical and experimental studies occurred due to high uncertainties in the experimental data for low pressure drop condition and the model inaccuracy in predicting the fluid behavior at foam-fluid interface. Ashtiani Abdi et al. (2013; 2014) for 10 PPI metal foam, double wake size appeared behind the foam cylinder with 10% higher shedding frequency compared to a bare tube due to its rougher surface. The study showed no wake region within Field of View (FOV) at Re = 8000, compared to Re = 2000 which showed the relationship between swirl strength and Re, moreover, the geometrical effects could be observed only at high gas velocity. Meanwhile, Hooman, (2014) stated that a fluid flow within a porous layer was leaving earlier before reaching the end because of a recirculation region exits at the downstream of the porous medium. Sauret et al. (2014a) also agreed that gas velocity was already fully developed at a quarter of a channel length. The porous structures of a metal foam heat exchanger may create tortuous flows and boundary layer disruption all the time. Therefore, the particle deposition could be influenced by the foam geometry as the flow regimes are also classified based on the pore-based Reynolds number, Re_p, which stated as follows (Dukhan, 2013):

- Darcy dominated laminar flow region, $Re_p < 1$
- Forchheimer dominated laminar flow region, $1–10 < Re_p < 150$
- Post-Forchheimer, unsteady laminar flow regime, $150 < Re_p < 300$
- Fully turbulent flow, $Re_p > 300$

Effect of Particles, Operating Conditions, and Heat Exchanger Design

Abd-Elhady et al. (2004) stated that the irregular size of particles reduced the critical flow velocity since the larger particles rolled the fine particles and the particle deposition likely to occur at the rear end of the tube due to lower shear force presented at that region. Storey et al. (2013) proved that different fuel types, HC level, and surface treatment were insignificant in the fouling process as the results showed similar particle deposition rates. In addition to that, the study stated that the increasing shear force due to the fouling layer formation seems inconsequential since the layer is porous. The microstructure and thermal conductivity of the layer are inconsistent throughout the fouling process due to an ageing process (sintering process) (Lister et al. 2013). The fouling layer started to harden gradually and the hot gas stream made the layer become denser and stronger (Abd-Elhady et al., 2007; Abd-Elhady et al., 2004). Therefore, the fouling layer could be divided into two different layers called as coke (an aged deposit with higher thermal conductivity) and gel (a fresh deposited particle with low thermal conductivity) (Lister et al., 2013). However, Salvi (2013) investigated the nano-particulate layers in the engine exhaust gas heat exchangers using a visualization rig and in-situ measurement. The study showed that the particulate layer thickness has insignificant effects on thermal conductivity, as densification which influenced the layer porosity did not occur with deposition. Nevertheless, the thermal resistance, fouling layer thickness and fouling rate could be reduced by increasing the gas velocity (Abd-Elhady et al., 2011b). If the gas velocity higher than a critical flow velocity, the fouling layer formation could be avoided (Abd-Elhady & Malayeri, 2013). Nevertheless, the gas velocity in the EGR cooler is about 10 - 30 m/s which is very small compared to the required critical velocity of 40- 280 m/s for typical size of soot particle, 10 - 300 nm (Abd-Elhady & Malayeri, 2013). The low gas velocity had increased the radial and circumferential growth of fouling layer (Abd-Elhady et al., 2004) but very high velocity may cause pressure drop and required an alteration of the existing EGR system (Abd-Elhady et al., 2011b).

A stack-type had 25–50% higher effectiveness compared to a shell & tube type (Jang et al., 2010; Kim et al., 2008) and an oval spiral had less particle deposition than a spiral (Park et al., 2014) due to better mixing flows and larger surface area. Meanwhile, the effectiveness of finned EGR coolers reduced gradually under different cycles (Park et al., 2010) and the EGR cooler with a higher fin pitch of 4 mm had similar effectiveness with 2.5 mm fin pitch, but showing a lower pressure drop (Jang et al., 2011). The effectiveness of a metal foam EGR cooler was higher than a flat plate after 4.5 hours under a fouling condition at 10 m/s gas velocity, but 40 PPI foam has more deposited particles than 20 PPI (Kahle, 2012). Meanwhile, Ackermann (2012) agreed with Kahle (2012) that lower PPI (e.g. 20 PPI) had better performance and the pressure drop could be decreased by reducing the foam thickness as comparing the same foam size of 30 mm (width), 195 mm (length) and different thickness, of 3 mm and 4 mm, respectively. Like the metal foam which acts as a flow inducer, other studies on fouling attempted to improve the EGR cooler surface by creating inducers (Mohammadi & Malayeri, 2013) or using a corrugated tube (Bravo et al., 2013). Meanwhile, an oval type EGR cooler with 4 mm fin pitch had better efficiency than 6 mm fin pitch since its wavy finned had created turbulence flow for self-purity (Lee & Min, 2014).

Mathematical Equation for Fouling Measurement

This study includes common mathematical equations in determining the heat exchanger effectiveness, heat transfer rate, and thermal resistances. The effectiveness could be calculated through (1) effectiveness-NTU method and (2) EGR cooler temperature ratio. The former has been used widely to compare various types of heat exchangers to determine best-suited type of heat exchanger for a certain application (Jang et al., 2011; Park et al., 2010; Park et al., 2014). It is defined as a ratio of the actual heat transfer rate, (Q) and maximum possible heat transfer rate (Q_{max}) in a system (Kim et al., 2008; Park et al., 2010) as shown in Eq. (1) (Park et al., 2014),

$$\varepsilon = 1 - \exp\left[\frac{\exp(-NTU \times C^{*}n) - 1}{C^{*}n}\right] \tag{1}$$

where $n = NTU^{-0.22}$ and number of transfer units (NTU) to represent the size of EGR cooler is equal to $UA/C_{\min}$. Meanwhile, C_{min} and C_{max} are the minimum and maximum capacity rates, and C^* is the ratio of C_{min} and C_{max}. Equation (2) shows the effectiveness which describes actual changes in the EGR gas temperature to the maximum change in the EGR gas temperature being cooled by the coolant temperature (Hong et al., 2011b; Lee & Min, 2014; Storey et al., 2013):

$$\varepsilon = \left(\frac{T_{gas,in} - T_{gas,out}}{T_{gas,in} - T_{coolant,in}}\right) \times 100 \tag{2}$$

The overall thermal resistance of an EGR cooler (R_{th}) could be calculated by using Eq. (3) (Warey et al., 2012). The $LMTD$ is a logarithmic mean temperature difference which is defined as an average temperature difference of the hot gas temperature and the cooled surface temperature as stated in Eq. (4) (Hatami et al., 2014; Kahle, 2012):

$$R_{th} = \frac{A_o \times LMTD}{Q} \tag{3}$$

$$LMTD = \frac{(T_{gas,in} - T_{coolant,out}) - (T_{gas,out} - T_{coolant,in})}{\ln\left(\frac{T_{gas,in} - T_{coolant,out}}{T_{gas,out} - T_{coolant,in}}\right)} \tag{4}$$

To determine the heat transfer rate, Eq. (5) could be used which involved the gas mass flow rate, $\dot{m}$ and heat capacity, c_p (Kahle, 2012).

$$Q = \dot{m}c_p(T_{gas,in} - T_{gas,out}) \tag{5}$$

The fouling thermal resistances, R_f could be determined using Eq. (6) which is the difference of thermal resistance under a clean condition and a fouling condition at a certain time (Lister et al., 2013; Warey et al., 2012):-

$$R_f = R_{th,(fouled)} - R_{th,(clean)} \quad (6)$$

Commonly, fouling resistances are obtained by considering the process is a thermal steady state which assumed that the temperature profiles through fouling layer are lines, the heat flux through fouling layer is constant and the duration of heat transfer through the fouling layer is so much faster than the fouling layer growth (Lister et al., 2013).

CHALLENGES OF METAL FOAM HEAT EXCHANGERS

Complex Structure of Open-Cell Metal Foam

In general, the metal foam heat exchanger challenges included the accurate characterization of foam S.Ds, prophecy and validation of thermo-hydraulic performance, alternative measures and design improvement for excessive pressure drop and fouling (Muley et al., 2012). Current fouling studies on the open-cell metal foam heat exchanger could not explain the insight of fouling phenomenon due to its complex structure. The past numerical studies were conducted to explain the propensity of fouling within the foam structure and fluid-solid interface with some assumptions as follows (Odabaee et al., 2013):

- Uniform deposition of particle within foam structure
- Uniform free stream velocity
- Constant temperature across the foam structure
- Constant local thermal equilibrium through porous medium by using thermal conductivity effectiveness from another experimental study on the heat transfer performance of a metal foam

However, the fluid velocity varies within the foam structure and it causes uneven particle deposition on the ligaments surface (Sauret et al., 2013). Kahle (2012) also showed different severity of fouling in bulk within the ligaments and interfaces. However, no reference to local scales is made which seems to be controlling the overall performance. Besides, the numerical and mathematical modeling used a cubic cell unit to represent the porous structure of metal foam. In real practice, the foam ligament diameter/shape, porosity and thickness affect the thermo-hydraulic properties significantly, which require experimental studies to explain the fouling process within the metal foam heat exchangers.

Range of Operating Condition, Metal Foam Geometry and Foulant Properties

Due to diverse heat exchanger operating conditions in many industries, the optimization of metal foam geometrical parameters are not well established to match a particular application especially the trade-off design between heat transfer and pressure drop performances. In addition to that, the particle removal and mitigation techniques from the foam structure are unclear and fall behind compared to other types of heat exchanger. Furthermore, as the deposited particles are subjected to the ageing process (Lister et al., 2013) more experimental and numerical studies are required as it may affect the thermal performance as well as the particle removal process. In addition to that, the effects of foam geometrical on the fouling propensity are not well established due to small parameter ranges (Ackermann, 2012; Kahle, 2012) and no foulant was involved physically (Sauret, Abdi, & Hooman, 2014b) in the past experimental studies.

Moreover, conflicting results on the fouling effects with nano-sized particles on metal foam as Al_2O_3 nanoparticles showed no deposition or the effects was insignificant for flowing fluid to block the porous cylinder (Miguel, 2015). Practically, it is difficult to obtain the exact fouling rates and resistances at every point of the EGR cooler surface and also within a foam structure. In addition to that, the reliable dominant fouling mechanism in the metal foam heat exchanger is unknown, which requires the manipulation of foam geometry, foulant properties, as well as the fluid velocities under non-isothermal and isothermal condition. The partial and fully blocked metal foam heat exchanger (Hooman, 2014) also possibly has different effects on the particle deposition because of the fluid flow disturbance. Furthermore, the foam thickness showed significant effects on the pressure drop (Dukhan & Patel, 2011) even though the fouling effects are not being considered at all.

Free Flow Region, Interface and Porous Structure

Sauret et al. (2014b) suggested further fouling studies are required to explain the particle deposition across the interface of the porous and non-porous regions (transverse direction) as well as the particle motion along the foam length (streamwise direction). Since, their results could not identified any sharp gradient at the porous and non-porous interface as assuming a continuous shear stress appeared in that specific region. Moreover, the idea of particles to move from the fluid-foam interface to the tube wall in the developing fouling layer could not be true if considering the rebound and impaction velocities of the particles (Sauret & Hooman, 2014). The numerical simulation of an interface boundary condition between the fluid and porous region hardly simulates the experimental studies (Sauret et al., 2014a). Therefore, the process of particle transport near the interface (boundary layer) could not be explained thoroughly. It shows the needs of experimental studies and slip condition consideration in investigating the thermo-hydraulic properties for free fluid-porous interface. The thermo-hydraulic properties of fluid flow influenced the fouling process which required more explanation as the deposition process varied based on the surface roughness, surface area to volume ratio, pore size, and pore distribution over the time (Hooman et al., 2012).

Bonding Method and Overall Thermal Resistance

An appropriate bonding method of the metal foam heat exchanger could reduce the thermal resistance (De Schampheleire et al., 2013). The manufacturing of metal foam should be improved in producing uniform ligaments throughout the foam structure. Hence, the lumps at the ligaments intersection and the irregularity of the ligament diameter could be avoided. Past fouling studies assumed that the thermal resistances in the EGR cooler were constant, as assuming the uniform fouling layer, but they were changing with time (Abd-Elhady et al., 2011b). The net radiation between the foam surface and surroundings was excluded from the overall thermal resistance as it was insignificant

(Chumpia & Hooman, 2012; 2014). However, the radiation effects could be significant in the heat exchanger for high temperature applications. A radiation heat transfer model was developed by Contento et al. (2014) who concluded that their model was in agreement with the experimental results, and suggesting an accurate foam morphological characteristics for a reliable model.

Fouled Foam Cleaning Methods

Due to limited fouling literatures on the metal foam heat exchangers, there are no established fouling cleaning measures. However, Kahle (2012) compared three different off-line cleaning method for fouled metal foam, using a brush, ultrasonic device and oxidization method. The study concluded that the best cleaning procedure was when using both brush and the ultrasonic device together. No studies considered the effects of corrosion fouling or other types of fouling on the metal foam heat exchanger which may influence the cleaning measures. Probably, current mitigation and cleaning strategies for the other types of heat exchanger (Müller-Steinhagen et al., 2011) could be considered for the metal foam heat exchanger, as long as, the cleaning process does not deteriorate the foam structure.

CONCLUSION AND FUTURE WORK

The underlying knowledge of metal foam heat exchangers is important especially the thermo-hydraulic properties and the fouling effects. At least to be accepted as a potential heat exchanger, the metal foam heat transfer performance should be superior compared to other types of heat exchangers for an identical pressure drop. The optimization of foam geometry and operating conditions should be determined for diverse applications. In addition to that, more researches are required to be compared with the current heat exchangers in term of compactness and cost, before details of fouling studies are included. For further studies in fouling aspect, the future work should be aimed to gain the insight into the open-cell metal foam fouling process and the particle preferential deposition area through visualization techniques. The study should investigate the propensity of particulate fouling by considering the effects of foam geometry and operating condition. Diverse foam thickness and air velocities up to 7 m/s with different particle size will be considered. The study should observe the particle deposition over a 270 mm x 320 mm foam plate by using a Particle Image Velocimetry (PIV). Ultimately, all regions along the plate as well as the upstream and downstream flow will be considered as the main points of visualization.

NOMENCLATURE

A	heat transfer area, m^2
C	heat capacity rate, W/K
C_p	specific heat at constant pressure, J/(kg K)
CFD	Computational fluid dynamics
d	diameter, m
EGR	exhaust gas recirculation
FOV	field of view
HC	hydrocarbon
LMTD	logarithmic mean temperature difference
$\dot{m}$	mass flow rate, kg/s
NTU	number of transfer units
PIV	particle image velocimetry
PPI	pore per inch
PM	particulate matter
R	thermal resistances, m^{2}K/W
Re	Reynolds number, $\rho v d/\mu$, dimensionless
SOF	soluble organic fraction
T	temperature, K
U	overall heat transfer coefficient, W/(m^2 K)
Q	heat transfer rate, W
v	fluid velocity, m/s
ε	heat exchanger effectiveness
ρ	density, kg/m^3
μ	dynamic viscosity, kg/(m s)

Subscript

c	cold surface
f	fouling
FL,O	outer fouling layer
in	inlet
max	maximum
min	minimum
o	overall
out	outlet
p	pore
th	thermal

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