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 have 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 particulate fouling of heat exchangers, particularly in the exhaust gas recirculation system. Some past approaches to investigate fouling, particle transport and deposition in the metal foam heat exchangers for many different applications are also included. In addition, this study also includes the challenges that lie ahead in implementing the metal foam heat exchangers in the industries.

INTRODUCTION

Metal Foam Microstructural & Thermo-physical Properties

Metal foams are classified into two categories; (1) open-cell or closed-cell structure and (2) cell arrangement either stochastic or periodic [1], [2]. The metal foams have been used in many industries such as 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 [3]. 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 as shown in Figure 1. The dodecahedron shape of open-cell foam is usually modeled as a cubic unit cell model or tetrakaidecahedron in numerical or mathematical studies [4]–[6]. The metallic material of foam prospered the heat transfer performance through conduction processes. An aluminum metal foam has been widely considered for heat exchangers due to its low density, high thermal conductivity and relatively low in price [7].

The open-cell metal foams have considerable advantages in thermal management [8] and heat recovery [9], [10]. The thermo-physical properties of high porous metal foams have been investigated aggressively for the last 15 years [4], [11]–[13]. Manufacturers usually have classified the microstructure of open-cell metal foams based on porosity and pore density. The porosity can be estimated by using the weight of a given volume of the sample and the density of the metal, meanwhile the pore density is the number of pores per unit length of the material specified as PPI (pores per inch), which usually being supplied by the manufacturer [14]. The relative density of metal foam which is defined as the volume of solid foam material relative to the total volume of metal foam significantly influences the heat transfer performance. The open-cell metal foam with low relative densities and large surface area per unit volume promotes eddies for better thermal management, and compactness. While the porous structure (low relative density) foam offers high convective heat transfer by thermal dispersion and permeability, the high relative density one offers high thermal conductivity [2]. By increasing the porosity and pore diameter, the permeability can be increased significantly. However, only the metal foam porosity has influenced the effective thermal conductivity [4], [15]. In addition, the increased porosity reduces the ligament (also known as strut or fiber) diameter, which consequently affects the conduction and the overall heat transfer coefficient significantly [16].

In terms of heat transfer performance, the complex structure of open-cell metal foam induces two different modes of heat transfer; convection and conduction which vary significantly based on the foam microstructure and operating condition. The effective thermal conductivity, the permeability, and the inertial impaction could be influenced by the foam microstructural properties such as the pore density (PPI), mean pore diameter, and surface-area to volume ratio [15]–[17]. Consequently, the microstructural properties of metal foams may also affect the overall heat transfer performance [15], [18]. The convective heat transfer within a porous structure metal foam is more dominant than conduction since its thermal conductivity is one order of magnitude lower than their parent material [1]. 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 [2]. Chumpia and Hooman [18] proved that the thermal performance and pressure drop of 20 PPI metal foam wrapped on an aluminum cylinder could be increased with foam thickness of 5 – 20 mm. But, when comparing the foam to a finned tube at the same pressure drop of 25 Pa, the foam with 15 mm thickness has a higher heat transfer up to 37%. The study also claimed that the same foam thickness could have different thermal resistances due to other factors such as bonding method, air velocity, porosity and surface material. Generally, the overall thermal resistances of a heat exchanger include the convective and fouling resistances for both free stream fluids as well as the wall (surface) conductive resistance. For a metal foam heat exchanger, some additional thermal resistances are expected if the fouling layer is insignificant or did not clog the porous structure completely due to its bonding (contact) method [2], [5] and the ligaments layout (convective and conduction resistances). Figure 2 shows a descriptive comparison of thermal resistances between a flat surface and a brazed metal foam based on the heat transfer mediums.

The foam 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*) [9], [19]. Furthermore, the ligament diameter and the interfacial velocity, which is calculated as Darcy velocity have been proven to influence the Reynolds number [19]. The open-cell metal foam heat exchangers show advantages in term of physical and mechanical properties as follows [1], [2], [19]:

- 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.

Due to the promising foam properties, many studies have been conducted in different heat exchanger application, e.g. EGR cooler [15], [20], [21], air-cooled condenser [18], [22], and waste heat recovery [10]. A good review on the heat exchanger and heat sink using metal foam and metal matrix composites can be found from Han *et al.* [1]. As overall, the review supported the benefits of metal foam with better thermal conductivity, lighter in weight, higher surface-area to volume ratio, and convective heat transfer compared to the other conventional heat exchanger. However, the study also stated that no mass and cheap production are available for the metal foam, and there are lacks of open literatures on the thermo-hydraulic characterization based on a full scale testing for diverse application, which required more information for system integration and design optimization. Especially, to implement metal foam as a heat exchanger in diverse industries. Therefore, this study reviewed the metal foams as a potential heat exchanger to reduce particulate fouling and enhance the heat transfer performance. In addition, the particle transport and deposition process within different types of heat exchangers, configurations and operating condition are also included, mainly for an exhaust gas recirculation (EGR) cooler application. The possible challenges related to the metal foam heat exchangers are also identified which should be solved in the near future.

Exhaust Gas Recirculation (EGR) System

Modern diesel engines have been installed with the EGR system to reduce nitrogen oxides (NO_x) emission by recirculating a fraction of the exhaust gas into an engine combustion chamber. The exhaust gas contains a lower oxygen concentration, but it has a higher heat capacity compared to the fresh air. The recirculation process reduces the combustion temperature through a dilution of fresh air (oxygen concentration) which allows higher mass concentration of the exhaust gas in the combustion chamber [23], [24]. Zheng *et al*. [24] determined the EGR rate by subtracting the measured fresh intake air (using a mass air flow sensor) from the estimated mass flow of a cylinder charge and attempted to reduce the combustion temperature in their study. Their result showed that an enhanced cooled EGR reduced about 200 ppm of NO_x at 20% EGR rate as compared to a hot raw EGR at 2100 rpm and brake mean effective pressure of 900 kPa bar as shown in Figure 3. Therefore, the cooled EGR is preferred in attaining much lower NO_x emission. Principally, the operation between the hot EGR and cooled EGR systems are different due to several additional components like EGR cooler (a heat exchanger), EGR valve, etc. The hot EGR system recycles the exhaust gas directly to the combustion chamber, meanwhile the cooled EGR system will reduce the exhaust gas temperature at the first hand before entering the chamber [24], [25]. The details literature on the EGR system could be found from Zheng *et al*. [24]. Even though, the cooled EGR showed greater reduction of NO_x by lowering the combustion temperature using a heat exchanger, a very low combustion temperature may cause incomplete combustion process which consequently, may produce more soot particles [26]. Therefore, a trade-off design between the soot particle deposition and NO_x reduction is critical in designing the EGR system. The first EGR cooler design was a shell and smooth tube type, and it is commonly used in many heavy-vehicles. The EGR cooler development is continued with corrugated tubes, and rectangular corrugated tubes with housing to attain EURO 5 standard, and later, the internal fins are considered to be attached to different geometries e.g. tubes or plates to attain EURO 6 standard [27].

PARTICULATE FOULING

Particulate Fouling in EGR Coolers

Fouling in EGR coolers is a combination of chemical reaction, corrosion fouling, or particulate fouling. The condensation of unburned hydrocarbon (HC), water and acid as suggested by Kahle [21] and Abarham *et al*. [28] may initiate corrosion on the heat transfer surfaces. Meanwhile, biofuel as an alternative fuel could produce different chemical properties of a fouling layer [29]. However, the particulate fouling is a dominant fouling type in the EGR system. It is always related to one of the main exhaust gas composition, Particulate Matter (PM) which is a combination of soot particles, soluble organic fraction (SOF), sulfates and unburned hydrocarbon [30]–[32]. Additionally, the condensed HC was also considered as the main constituent of the fouling layer due to its significant effects on fouling growth [33]. Storey *et al*. [29] investigated fouling using two different fuels; ultra-low sulfur diesel (ULSD) and biodiesel blends, B20. The study proved the insignificant difference in the fouling rate based on the different fuels, HC level, or surface treatment and concluded that the fouling layer is porous with the thermal conductivity was approximate to 0.04 W/m K. Thus, the very low thermal conductivity of the fouling layer is considered as a barrier to heat transfer process.

Different types of heat exchanger exhibit dissimilar heat transfer characteristics due to the different fluid flow pattern, total heat transfer surface areas and surface material. For a smooth surface such as a shell and tube heat exchanger, the deposited particles forms a layer and subsequently, reduces the cross-sectional area of the tubes. However, a process of particle removal is expected after some time, since the reduced cross-sectional area could induce high shear forces to drag or roll the particles before an asymptotic fouling condition is achieved. Furthermore, the temperature-dependent fouling in a heat exchanger becomes insignificant at that condition. The temperature of the outer layer 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 4 [30], [34]. 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 [30], [35], [36].

The deposit layer increased the pressure drop thus, affecting the thermo-hydraulic performance of the heat exchanger [21], [34], [37]. 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 position of the actuator [27]. The increased pressure drop across the EGR cooler may increase the pumping power [32]. In addition, the exit gas temperature is higher than initial EGR cooler design, thus reducing the EGR efficiency [27]. Moreover, the condensation of unburned HC of the diesel exhaust gas [25], [38], excessive fuel consumption in the engine chamber [27], fouled EGR cooler design, operating mode based on the engine load and corrosion by SOF and PM [31] as well as carbon monoxide [25] would intensify the fouling problem. Table 1 shows the typical soot particles and operating conditions in investigating the fouling problem in the EGR cooler.

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 [23], [32] 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 [34]. However, the thermophoresis is the dominant fouling mechanism for the EGR cooler due to fine particle size and high temperature gradient [21], [32], [34]. In the meantime, van der Waals forces initiated opposite charges and forced the particles to move toward each other [29]. Thus, particle agglomeration occurs and the bigger particles deposit on the EGR cooler surface due to gravitational forces (sedimentation). The diffusion occurs as the EGR cooler surface temperature is lower than the material dew point temperature at local pressure, which then, initiates the SOF and HC condensation [31], [38]. According Abd-Elhady & Malayeri [34], 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 5 shows the acting forces on a particle on a flat plate surface.

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*. [2] experimental work for their geometry consideration [5], [39] and validation [39]. Earlier, T'Joen *et al*. [2] 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 [5], [39], [40]. 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 [41], [42]. Odabaee *et al*. [5] 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*. [40] 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 smaller, 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 [39].

Both studies from Sauret *et al*. [40], and Sauret & Hooman [39] set the top, bottom and side surfaces of the domain to symmetry and atmospheric pressure at the outlet, but the former study set the wall temperature at 353 K and the inlet axial velocity of 3 m/s meanwhile, the latter has 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 the complexity 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 [39]. 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 [39], [40]. Figure 6 shows the recirculation and dead zones of metal foam at the rear of the tube, marked as ‘x’ and Figure 7 shows the preferential particulate deposition area which relates the particle size and particle volume fraction. The stagnant and recirculation zones exist behind the ligaments [1] possibly increased the particle deposition within the porous structure, which contradicts the ligaments function to increase the shear force to remove the deposited particles.

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 [17]. 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 [5], [21]. 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 [16]. 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 [39].

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

Sauret *et al.* [39] investigated the interface boundary condition between the 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.* [9], [43] worked on 10 PPI metal foam, and claimed that a double wake size appeared behind the foam cylinder with 10% higher shedding frequency compared to a bare tube due to its rougher surface. In addition, the experimental work has shown no wake region within Field of View (FOV) at *Re* = 8000, compared to *Re* = 2000 which proved the relationship between swirl strength and Reynolds number. Meanwhile, the geometrical effects in their study could be observed only at high gas velocity. Furthermore, Hooman, [44] 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. Meanwhile, Sauret *et al*. [39] also agreed that gas velocity was already fully developed at a quarter of a channel length of porous structure. The porous structures of a metal foam heat exchanger may create tortuous flows and boundary layer disruption all the time. Therefore, the particle deposition seems likely to be influenced by the foam geometrical and microstructural aspects. Moreover, the flow regimes are also classified based on the pore-based Reynolds number, Re_p, which stated as follows [45]:

- 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*. [35] stated that the irregular size of particles reduced the critical flow velocity as some of the larger particles could roll 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*. [29] proved that different fuel types, HC level, and surface treatment have insignificant effects on the fouling process as their results showed similar particle deposition rates. Besides, the study also questioned the fact of increasing shear force as building up the fouling layer as the effect seems insignificant due to the porous structure of fouling layer. In addition, a part of the fouling layer microstructure and thermal conductivity are inconsistent throughout the fouling process due to an ageing process (sintering process)[46]. The fouling layer starts to harden gradually and under a hot gas stream, the layer become denser and stronger [35], [47]. 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) [46]. Meanwhile, Salvi [48] who investigated the nano-particulate layers in the engine exhaust gas heat exchangers using a visualization rig and in-situ measurement proved that the particulate layer thickness has insignificant effects on thermal conductivity. Since, the layer 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 [30]. Particularly, Abd-Elhady & Malayeri [34] stated that if the gas velocity is higher than the critical flow velocity (at maximum based on particle size), the fouling layer formation could be avoided. However, the study also claimed the gas velocity in the current EGR cooler is very small, about 10 - 30 m/s as compared to the required critical flow velocity of 40- 280 m/s for typical size of soot particles in between 10 - 300 nm. Despite, the low gas velocity had increased the radial and circumferential growth of fouling layer [35], but a very high gas velocity may cause pressure drop and required an alteration of the existing EGR system [49].

Thus, improving the geometrical aspect of a heat exchanger is preferred rather than considering the particle (foulant) chemical properties and diverse operating condition. A stack-type heat exchanger exhibited 25 – 50% higher effectiveness compared to a shell & tube heat exchanger [50], [51] and an oval spiral type had less particle deposition than a spiral EGR cooler [52] due to two main reasons (1) better mixing flows and (2) larger surface area. Meanwhile, the effectiveness of finned EGR coolers reduced gradually under different cycles [53] 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 [54]. 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 [21]. Meanwhile, as comparing the same foam size of 30 mm (width), 195 mm (length) and different thickness of 3 mm and 4 mm, Ackermann [20] has agreed with Kahle [21] that lower PPI (e.g. 20 PPI) had better performance and the pressure drop could be decreased by reducing the foam thickness. Like the metal foam which acts as a turbulence inducer, other studies on fouling have attempted to improve the EGR cooler surface by creating inducers [55] or using a corrugated tube [27]. 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 [31].

Mathematical Equation for Fouling Measurement

This study includes some common mathematical equations in determining the heat exchanger effectiveness, heat transfer rate, and thermal resistances. The effectiveness of a heat exchanger could be calculated by using (1) effectiveness-NTU method and (2) EGR cooler (heat exchanger) temperature ratio. The former has been used widely to compare various types of heat exchangers in order to determine the best-suited type of heat exchanger in a certain application [52]–[54]. It is defined as a ratio of the actual heat transfer rate, (Q) and maximum possible heat transfer rate (Q_{max}) in a system, with the equation is written as [52]:

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

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}. Meanwhile, equation (2) shows the effectiveness by describing the actual changes in the EGR gas temperature to the maximum change in the EGR gas temperature being cooled by the coolant temperature [29], [31], [56].

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

The overall thermal resistance, R_{th} of an EGR cooler could be calculated by using equation (3) [32]. As shown in equation (3) and (4), LMTD is an abbreviation for logarithmic mean temperature difference, defined as an average temperature difference of the hot gas temperature and can be written as [21], [57]:

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

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

Equation (5) is used to determine the heat transfer rate, which involved the gas mass flow rate, $\dot{m}$ and heat capacity, c_p [21]:

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

The fouling thermal resistances, R_f can be determined using equation (6), which is the difference of the thermal resistance between a clean condition and a fouling condition at a certain time [32], [46]:

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

Commonly, the 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 [46].

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 geometrical parameters, prophecy and validation of thermo-hydraulic performance, finding alternative measures and design improvement for excessive pressure drop and fouling [15]. Besides, a very limited published literatures on fouling of metal foam heat exchangers, current studies on the porous structure metal foam could not explain the insight of fouling phenomenon due to its complex structure. Previously, the numerical studies were conducted to explain the propensity of fouling within the foam structure and fluid-solid interface with some assumptions as follows [5]:

- 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 [40]. Kahle [21] 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 of heat exchanger. Besides, the numerical studies 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 microstructure are unclear and fall behind compared to other types of heat exchanger. Furthermore, as the deposited particles are subjected to the ageing process [46], 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 [20], [21] and no foulant was involved physically in the past experimental study, even though the effect on velocity profile and flow separation were identified to increase the understanding on porous and non-porous interface [10]. Moreover, there are some conflicting results on the fouling effects with nano-sized particles on metal foam regardless of the particle chemical properties, such as 40 nm Al_2O_3 nanoparticles showed no deposition or the effects was insignificant for flowing fluid to block the porous cylinder with 0.249 porosity [58]. The nanofluids (based-fluid suspended with nano-size particles) have discrepancies in results probably due to the foam porosity or particle size effects. Since, 100 nm Al_2O_3 particles had deposited on the higher porosity (0.98) copper and nickel foams [59].

The optimum pore diameter is required so that more particles can pass through the foam microstructure to reduce fouling, without deteriorating the heat transfer performance. However, the interconnected ligaments can act like the barriers and to be considered as the nearby walls for the flowing fluid. Thus, the particle deposition may happen either partially or fully blocked the pores due to the ligament construction. Practically, it is difficult to obtain the exact fouling rates and resistances at every point of the EGR cooler surface due inconsistent thermo-physical properties [46], moreover within a complex porous foam. Besides, 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 both non-isothermal and isothermal condition. The partial and fully blocked metal foam heat exchanger [44], may also possibly have different effects on the particle deposition because of the fluid flow disturbance. Furthermore, the foam thickness showed significant effects on the pressure drop [60] even though the fouling effects are not being considered at all.

Free Flow Region, Interface and Porous Structure

Sauret *et al*. [10] have suggested further fouling studies 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). Particularly, as their results could not identify any sharp gradient at the porous and non-porous interface as assuming a continuous shear stress has appeared in that specific region. Moreover, the idea of particles to move from the fluid-foam interface to the tube wall in building-up fouling layer could not be true if considering the rebound and impaction velocities of the particles [39]. Besides, the numerical simulation of an interface boundary condition between the fluid and porous region in their study hardly simulates the experimental studies or represent real condition. More studies on slip velocity preferred through mathematical modeling [61]–[63], instead of the experimental work. For example, Xu, *et al*. [63] investigated on velocity slip, thermal slip, local thermal non-equilibrium (LTNE) effect, and asymmetric heat fluxes in two-parallel plate sandwiched with microfoams. The study derived an explicit expression for velocity and temperature of solid and fluid where their analytical solutions for wide ranges of Knudsen number and heat flux showed in agreement with existing references. In terms of fouling, the process of particle transport near the interface (boundary layer) could not be explained thoroughly. It shows the needs of experimental studies with a slip condition in investigating the fouling and the thermo-hydraulic properties effects at the free fluid-porous interface. The thermo-hydraulic properties of fluid flow affect 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 [17] .

Bonding Method and Overall Thermal Resistance

An appropriate bonding method of the metal foam heat exchanger could reduce the thermal resistances [19]. 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 a uniform fouling layer formation, but truthfully, they are changing with time [49]. Specifically for metal foam heat exchangers, the net radiation between the foam surface and surroundings was neglected from the overall thermal resistance as it was insignificant [18], [22]. The prediction of radiative heat transfer of metal foams is difficult due to the foam architecture, inherent complexity and transport mechanism within the ligaments layout [6]. However, the radiation effects can be significant for high temperature heat exchanger applications. One of the early attempts to consider the radiative heat transfer of metal foam was done by Contento *et al*. [6] who used a tetrakaidecahedron model as the basic unit cell to represent the foam microstructure. By combining the numerical simulations and analytical models based on ray-tracing Monte Carlo method and an iterative procedure, their model showed in agreement with experimental results from the past literatures. They have suggested to prepare an accurate foam morphological characteristics for a reliable model in considering the radiative conductivity.

Fouled Foam Cleaning Methods

Due to limited past literatures on fouling of the metal foam heat exchangers, there is no established fouling cleaning measure for that kind of heat exchanger. However, Kahle [21] has compared three different off-line cleaning method for a fouled metal foam by using a brush, an ultrasonic device and an oxidization method; see also Hooman & Malayeri, [64]. The study concluded that the best cleaning procedure for soot particle deposition on the porous structure metal foam was when combining two methods using the brush and the ultrasonic device together. To the best authors' knowledge, no studies or published literatures have considered the effects of corrosion fouling or other types of fouling on the metal foam heat exchanger which specifically may influence the cleaning measures. Current mitigation and cleaning strategies for the other types of heat exchanger reviewed by Müller-Steinhagen *et al*. [65] could also be considered for the metal foam heat exchanger, as long as, the cleaning process does not deteriorate the foam microstructure – ligament layout.

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 heat exchanger applications. In addition to that, more researches on metal foam are required to be compared with other types of heat exchangers in term of compactness and cost, before details of fouling studies are included. For further studies in the fouling aspect, the future work will be aimed to gain the insight of fouling within a porous foam plate and identifying the preferential particle deposition areas through a visualization technique. The study will investigate the propensity of particulate fouling by considering the effects of foam geometry and operating condition. Diverse foam thickness and initial air velocities up to 7 m/s with different particle size will be considered. Ultimately, all regions along the plate as well as the upstream and downstream flow will be considered as the main points of visualization to investigate the particles transport and deposition process, the fluid flow pattern as well as the pressure drop along the foam plate.

ACKNOWLEDGEMENT

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NOMENCLATURE

A	heat transfer area, m^2
C	heat capacity rate, $W.K^{-1}$
CFD	Computational fluid dynamics
C_p	specific heat at constant pressure, $J.kg^{-1} K^{-1}$
d	diameter, m
EGR	exhaust gas recirculation
FOV	field of view
HC	hydrocarbon
LMTD	logarithmic mean temperature difference
$\dot{m}$	mass flow rate, $kg.s^{-1}$
NTU	number of transfer units
PPI	pore per inch
PM	particulate matter
R	thermal resistances, $m^2K.W^{-1}$
Re	Reynolds number, dimensionless
SOF	soluble organic fraction
T	temperature, K
U	overall heat transfer coefficient, $W.m^{-2} K^{-1}$

Q	heat transfer rate, W
v	fluid velocity, m.s^{-1}

Greek Symbols

ε	heat exchanger effectiveness
ρ	density, kg.m^{-3}
μ	dynamic viscosity, kg.m^{-1} s^{-1}

Subscripts

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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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 [34]
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*. [66]

List of Figure Captions

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

Figure 2 Heat transfer mediums for a flat surface (left) and a metal foam (right) - No deposition within the porous structure

Figure 3 Exhaust nitrogen oxides, (NOx) versus EGR rate [24]

Figure 4 Stage of particulate fouling

Figure 5 Forces acting on a resting particle

Figure 6 Axial velocity distribution with 3 m/s inlet air velocity (top) and axial velocity profile in the metal foam (below) [39]

Figure 7 Average volume fraction of particle with particle size distribution [39]

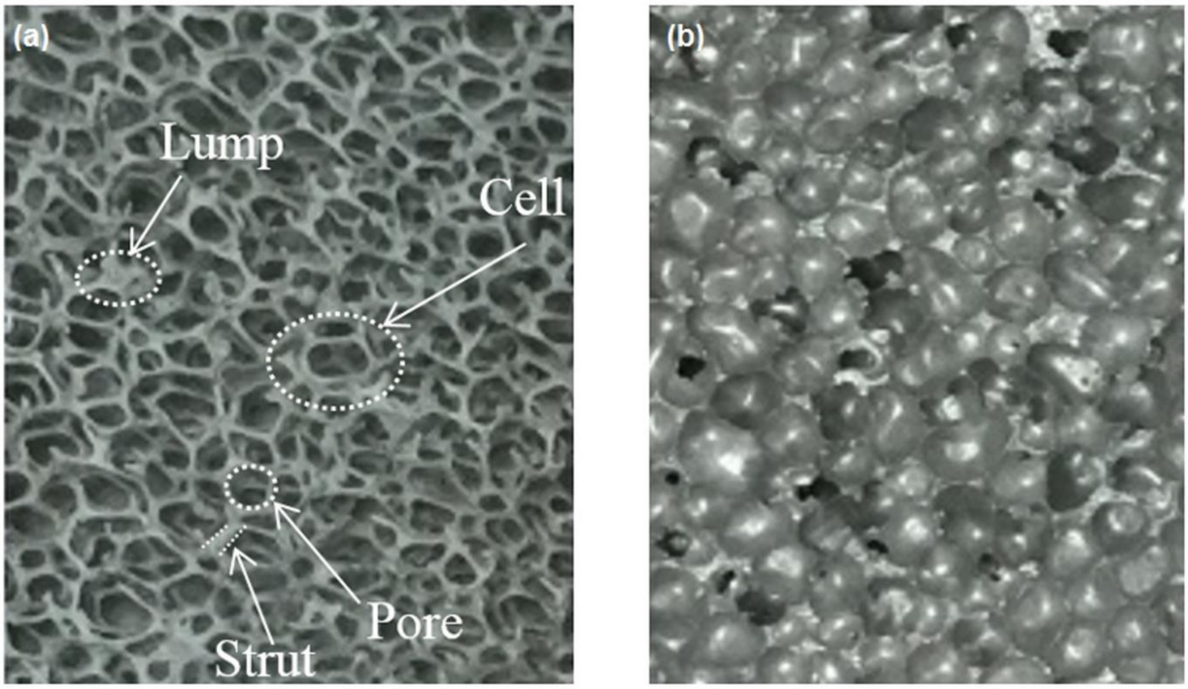


Figure 1 (a) 10 PPI open-cell metal foam (b) Closed-cell metal foam

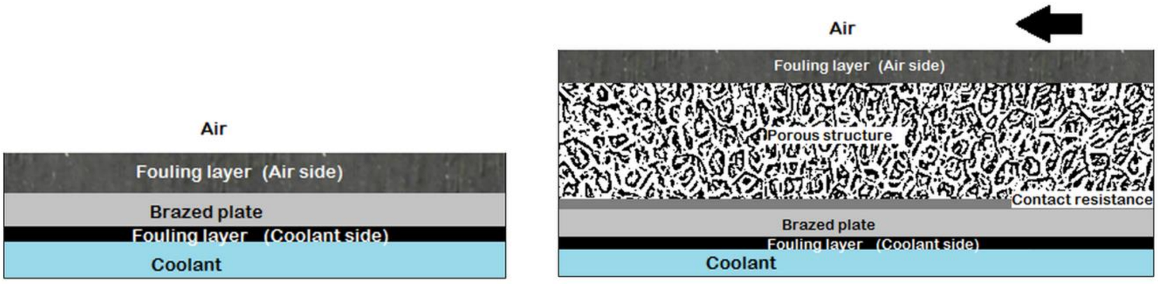


Figure 2 Heat transfer mediums for a flat surface (left) and a metal foam (right) -No deposition within the porous structure

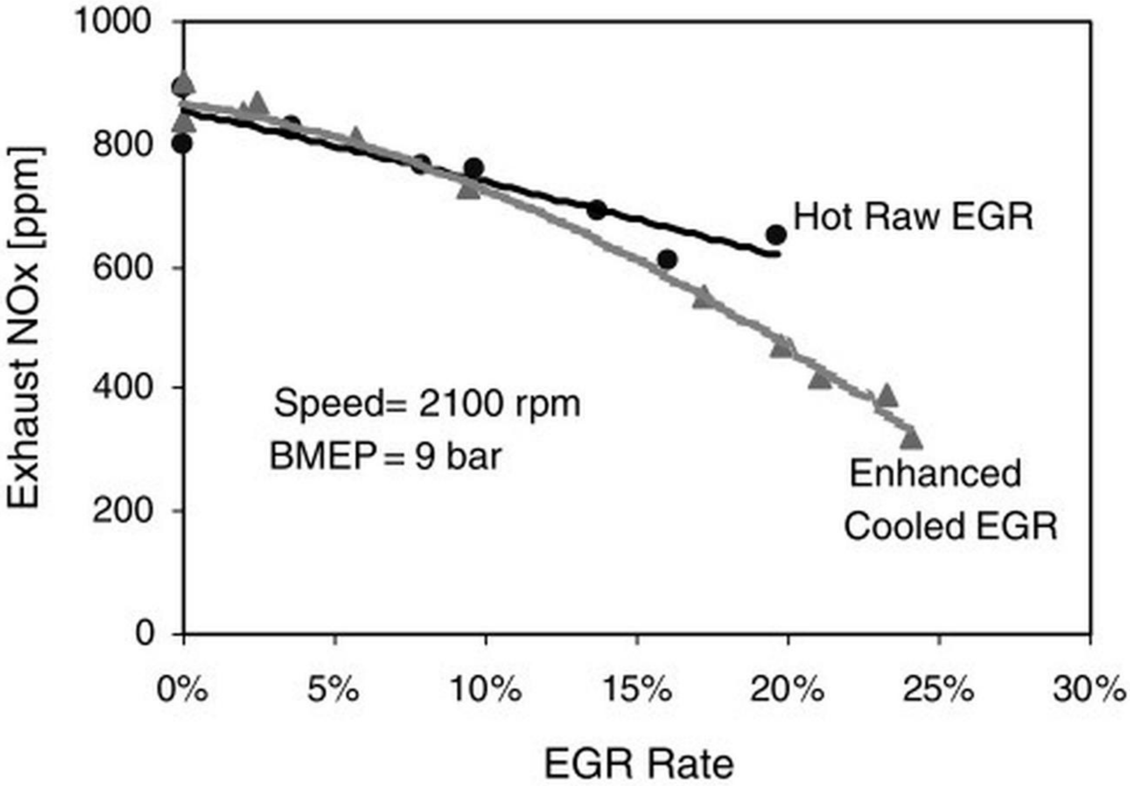
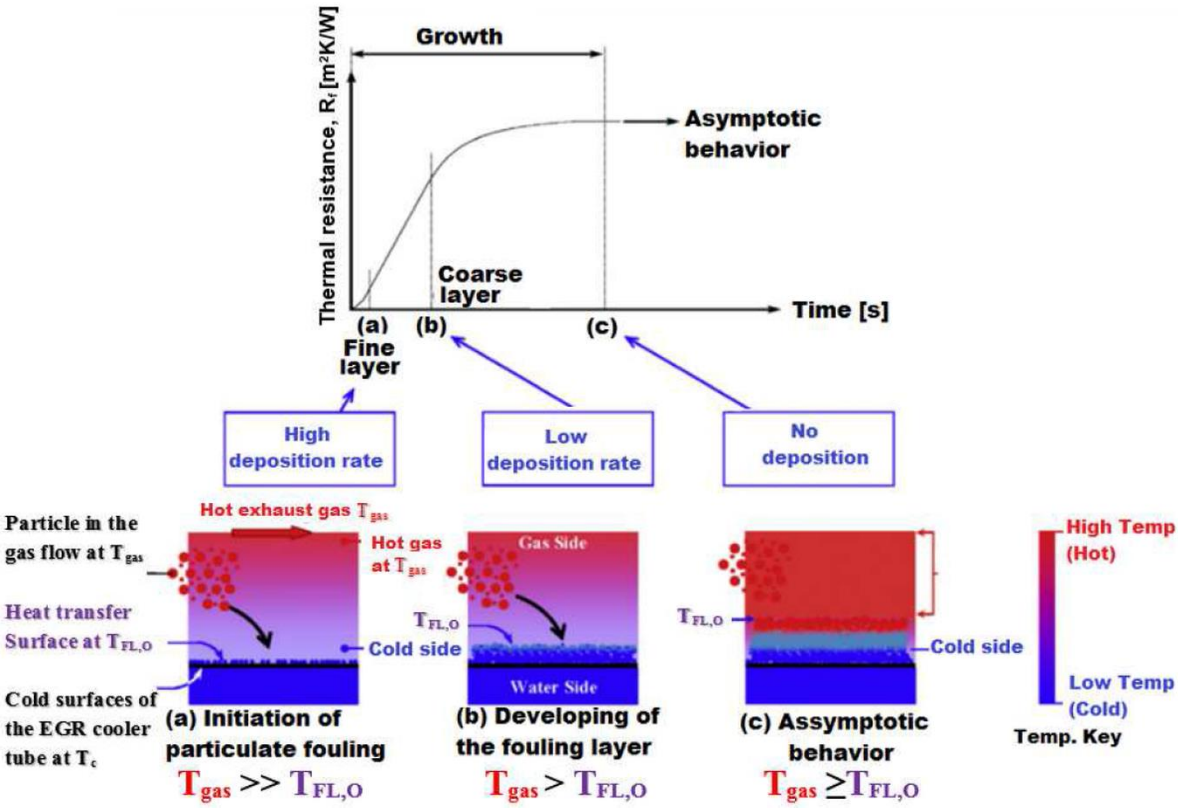


Figure 3 Exhaust nitrogen oxides, (NO_x) versus EGR rate [24]



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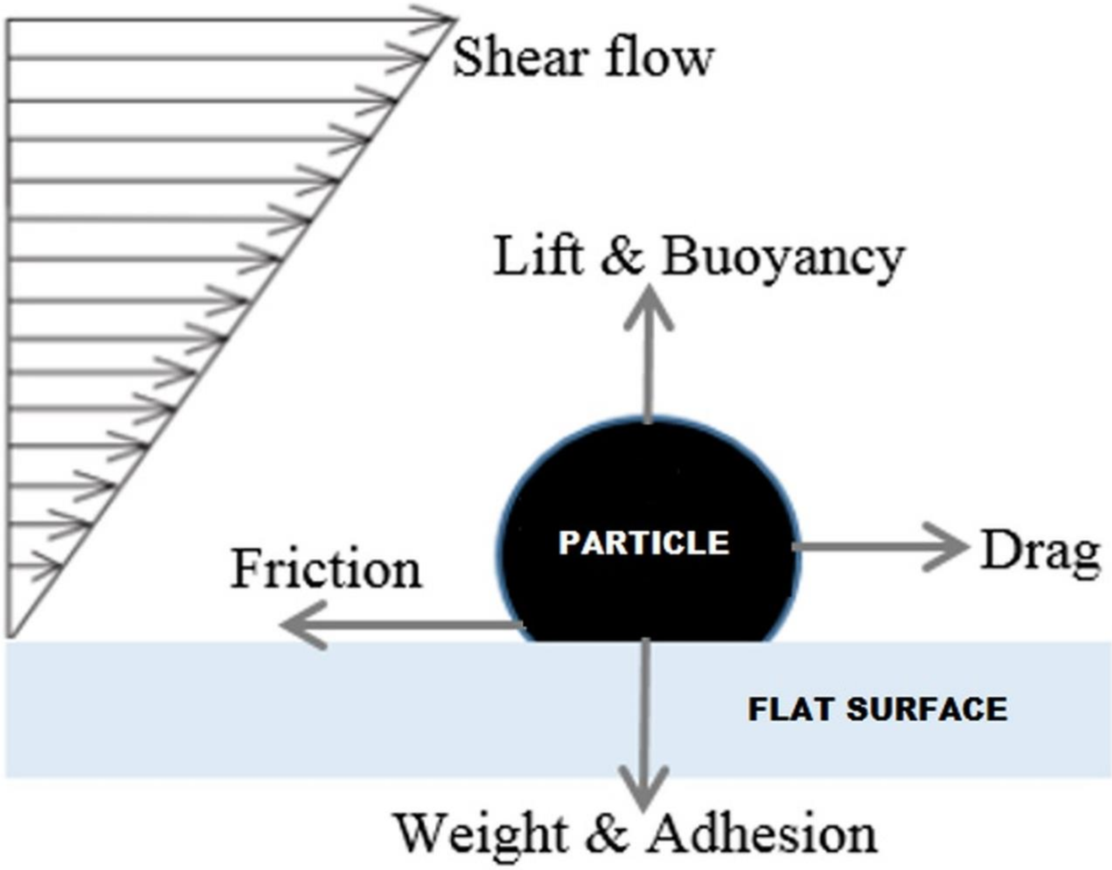

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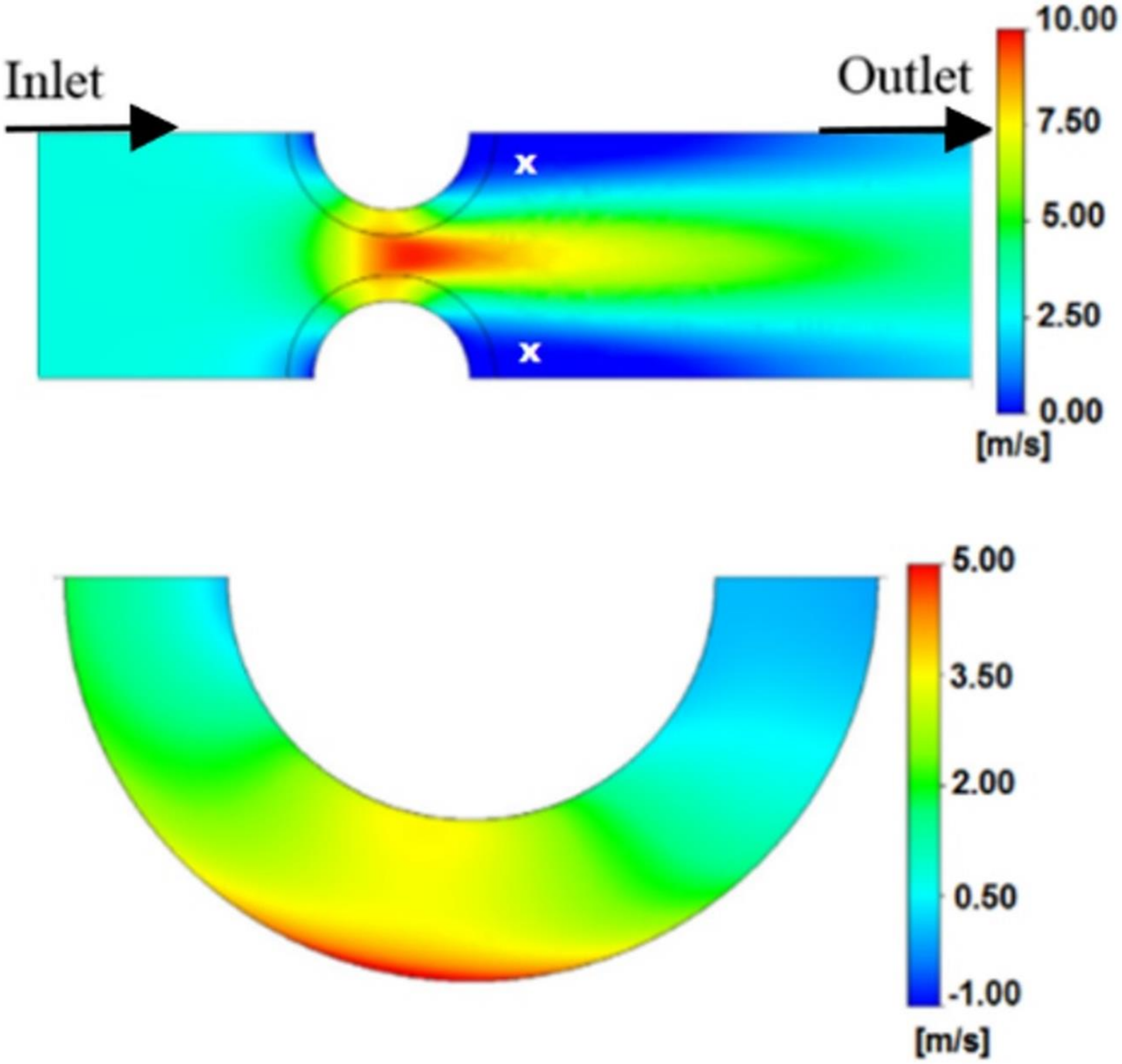

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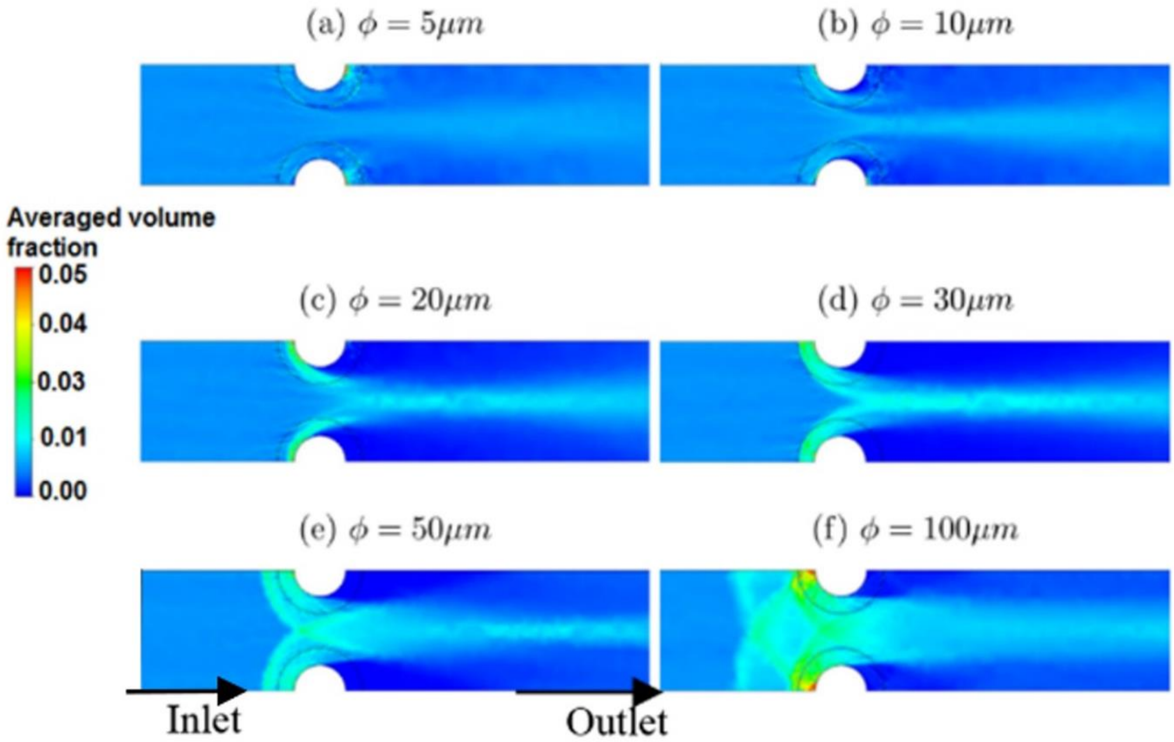

Figure 7 Average volume fraction of particle with particle size distribution [39]

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