ABSTRACT
The present study presents CFD simulations of a liquid-piston compressor with metal foam inserts. The term “liquid-piston” implies that the compression of the gas is done with a rising liquid-gas interface created by pumping liquid into the lower section of the compression chamber. The liquid-piston compressor is an essential part of a Compressed Air Energy Storage (CAES) system. The reason for inserting metal foam in the compressor is to reduce the temperature rise of the gas during compression, since a higher temperature rise leads to more input work being converted into internal energy, which is wasted during the storage period as the compressed gas cools.

Liquid, gas, and solid coexist in the compression chamber. The two-energy equation model is used; the energy equations of the fluid mixture and the solid are coupled through an interfacial heat transfer term. The fluid mixture, which includes both the gas phase and the liquid phase, is modeled using the Volume of Fluid (VOF) method. Commercial CFD software, ANSYS FLUENT, is used, by applying its default VOF code, with user-defined functions to incorporate the two-energy equation formulation for porous media.

The CFD simulation requires modeling of a negative momentum source term (drag), and an interfacial heat transfer term. The first one is the pressure drop due to the metal foam, which is obtained from experimental measurements. To obtain the interfacial heat transfer term, a compression experiment is done, which provides instantaneous pressure and volume data. These data are compared to solutions of a zero-dimensional compression model based on different heat transfer correlations from published references. By this comparison, a heat transfer correlation which is most suitable for the present study is chosen for use in the CFD simulation.

The CFD simulations investigate two types of metal foam inserts, two different layouts of the insert (partial vs. full), and two different liquid piston speeds. The results show the influence of the metal foam inserts on secondary flows and temperature distributions.

1. INTRODUCTION
Metal foam provides large surface area per unit volume. It can be used to absorb thermal energy in gas compressors for applications in Compressed Air Energy Storage (CAES). Integrating the CAES approach in power producing systems can overcome the mismatch between power demand and power generation by compressing air during low-power-demand periods and expanding the compressed air during high-power-demand periods. An important aspect for efficient and effective CAES operation is to achieve near-isothermal compression. A technology of using inter-stage cooling between compressors proposed in [1] has demonstrated the importance of cooling in CAES. A study on liquid-piston compression has shown that liquid piston has an advantage over a traditional solid piston in terms of power consumption [2]. Another advantage of the liquid piston is that it does not compromise the compression ratio or the use of heat-absorbing media in the
compressor, as liquid can flow through porous medium. A CAES system design proposed in [3] uses hydraulic circuits and liquid pistons to compress air, making it possible for exploring advanced cooling methods, such as inserting solid material in the liquid-piston compressor and using droplet-sprays. Analyses based on CFD simulations in [4] have shown that compression efficiency is higher when the compression process is more isothermal for the same pressure compression ratio. Physically, the reason for the need of near-isothermal compression is that typical compression results in a temperature rise of the compressed air. This temperature rise occurs because part of the input work is being converted into an increase in internal energy of air. This thermal energy, however, is wasted during the storage period as the compressed air cools. This can be clearly shown by a P-V diagram (Fig. 1). The total work input is represented by the integral under the curves. The amount of recoverable energy is represented by the integral under the isothermal curve. The total work input of a non-isothermal process (including the compression and cooling processes) is always greater than that of isothermal compression, but the amount of recoverable energy is the same. Therefore, cooling the air during compression is important for lowering input work and maintaining high compression efficiency. The present study investigates compression processes when metal foams are used as heat-absorbing media.

Porous media have been shown to be successful heat-absorbing materials in engineering applications. A numerical study on a pulse tube cryocooler has demonstrated an effective regenerator made of porous media [5]. In another study, porous media are used in catalytic converters and the performance is shown [6]. A new technique is presented in [7] which can be used to fabricate one chunk of metal foam of variable pore size for heat exchanger applications. Although the optimum porous media distribution for the liquid-piston compression chamber is currently unknown, the present study explores the effects of different metal foam layouts in the chamber.

CFD simulations on fluid flow and heat transfer in a liquid-piston compressor with inserted metal foam are done in the present study. The volume-averaging technique based on [8], a powerful method for simulating flow through porous media without the need for resolving the pore-scale activities directly, is applied. The continuity, momentum, and energy equations are volume-averaged on the scale of a Representative Elementary Volume (REV) of the porous medium. Instead of resolving the flow through the exact shape of the porous medium, these volume-averaged equations are solved. Due to volume averaging, a negative momentum source term arises in the momentum equation. This term has been proposed to be modeled by a Darcian and a Forchheimer extension term [9].

For formulating the energy transport in the porous media, two methods are available. One method is the one-energy equation method, which solves an artificial temperature that represents an “average” of the solid temperature and fluid temperature [10]. This requires thermal equilibrium between fluid and solid phases. The other one is the following two-energy equation method. The volume-averaged energy equation for each of the solid and fluid phases is solved. They are coupled through an interfacial heat transfer term. Different interfacial heat transfer correlations for flow in porous media have been proposed [11]–[15]. Each of these correlations will be assessed in this paper.

Application of the two-energy equation method can be found in a number of single-phase flow studies, including: fully developed flow through a metal-foam-filled pipe [16], laminar flow in a channel passing through porous medium made of small particles [17], a double-pipe heat exchanger with metal foam inserts [18], and flow through a ceramic structure used in a solar energy receiver [19]. The present study will apply this method for a two-phase flow.

A solid piston compressor with a partial insert of a porous medium has been solved computationally in [20]. It shows that the porous insert inhibits the formation of flow vortices. The study has great value for understanding the flow phenomena in porous media during compression, yet it solves the one-energy equation model. Because it solves a solid piston compression problem, the numerical model cannot be used for the present study where the piston is a liquid column and both the fluids and solid temperatures are needed a solution.

The Volume of Fluid (VOF) method proposed by [21] can be used to numerically simulate immiscible multiphase flows. The volume fraction scalars are solved to track bulk locations of different fluid phases; one set of momentum and energy equations are shared by all fluid phases. In a more recent study, the VOF method has been applied in a study solving, at the pore scale, two-phase flow in a porous medium [22]. In the present study, the VOF model and the two-energy equation model are applied to solve the global-scale fluid flow and heat transfer problem in a porous medium.

2. PROBLEM FORMULATION

A cylindrical, liquid-piston compression chamber is occupied by a porous metal foam. The present study investigates two situations: foam in the entire volume and foam in a portion of the chamber. A schematic of a partially occupied chamber is shown in Fig. 2. The fully occupied case uses the same coordinate system and coordinate layout. The compression chamber is studied in cylindrical coordinates. The x axis is along the centerline of the compression chamber. The gravitational field points opposite to the axis direction x. At time t = 0, water pumping into the chamber at location x = 0 begins. Boundaries x = L and r = R represent walls of the chamber. For the partial insert case, the metal foam is inserted over the region of length L_{ins}; for the full insert case, the metal foam is inserted over the entire length L. The material of the metal foam is aluminum.

During compression, water, air, and solid coexist in the chamber. The two-energy equation model for porous media and the VOF method are used to formulate the transport equations.
The continuity equation for each of the two fluid phases is solved. The fluid density of a particular phase that is transported by the continuity equation is the phase density times its phase volume fraction, \( \alpha \). Let subscripts 1 and 2 represent air and water, respectively. Then, for air:

\[
\frac{\partial \rho_1 u_1}{\partial t} + \nabla \cdot (\rho_1 \mathbf{u}_1) = 0
\]  

(1)

For water, since the density is a constant, the continuity equation becomes a transport equation of volume fraction only,

\[
\frac{\partial \rho_2 \alpha_2}{\partial t} + \nabla \cdot (\rho_2 \mathbf{u}_2) = 0
\]  

(2)

The velocity field and temperature field are shared by air and water. Thus, one set of momentum and fluid energy equations are solved, based on the properties of the fluid mixture.

\[
\frac{\partial \rho \mathbf{u}}{\partial t} + \nabla \cdot (\rho \mathbf{u} \mathbf{u}) = -\nabla p + \nabla \cdot \mathbf{\tau} + \mathbf{\tilde{S}}_m
\]  

(3)

where \( \rho = \rho_1 \alpha_1 + \rho_2 \alpha_2 \)

(4)

The stress tensor, \( \mathbf{\tilde{S}}_m \), is based on viscosity of the fluid that is made up of air and water,

\[
\mathbf{\mu} = \alpha_1 \mu_1 + \alpha_2 \mu_2
\]  

(5)

The negative momentum source term, \( \mathbf{\tilde{S}}_m \), represents the resistance of the metal foam to the flow. It can be modeled based on a Darcian and a Forchheimer term using the following form [9]:

\[
\mathbf{\tilde{S}}_m = -\frac{\mu \mathbf{\tilde{u}}}{K} - \frac{1}{2} b \rho |\mathbf{\tilde{u}}| |\mathbf{\tilde{u}}| \mathbf{\tilde{u}}
\]  

(6)

The energy equation for the fluid is,

\[
\frac{\partial (\rho c_p T)}{\partial t} + \nabla \cdot (\rho c_p \mathbf{u} T) = e\nabla \cdot \mathbf{\tilde{\tau}} + h_v (T_s - T) + e \frac{\partial p}{\partial t}
\]  

(7)

where \( \rho c_p = \alpha_1 \rho_1 c_{p,1} + \alpha_2 \rho_2 c_{p,2} \)

(8)

\( k_f = \alpha_1 k_1 + \alpha_2 k_2 \)

(9)

The volumetric heat transfer coefficient can be written in terms of surface heat transfer coefficient,

\[
h_v = a_v h_{sf}
\]  

(10)

The air density follows the ideal gas law,

\[
\rho_1 = \frac{p}{RT}
\]  

(11)

The energy equation for the solid is

\[
(1 - e) \frac{\partial}{\partial t} (\rho_s c_s T_s) = (1 - e) \nabla \cdot k_s \nabla T_s - h_v (T_s - T)
\]  

(12)

The boundary conditions for velocity and temperature variables are given in Table 1.

![Fig. 2 Schematic of compression chamber with partial insert](image)

Table 1. Boundary conditions for velocity and temperature

<table>
<thead>
<tr>
<th>Condition</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \bar{u}(x = L) = \bar{u}(r = R) = 0 )</td>
<td>( \bar{u}(x = 0) = (\bar{U}_0, 0) )</td>
</tr>
<tr>
<td>( T(x = 0) = T(x = L) = T(r = R) = 297K )</td>
<td></td>
</tr>
<tr>
<td>( T_s(x = 0) = T_s(x = L) = T_s(r = R) = 297K )</td>
<td></td>
</tr>
<tr>
<td>( \alpha_1(x = 0) = 0 )</td>
<td>( \alpha_2(x = 0) = 1 )</td>
</tr>
<tr>
<td>( \frac{\partial T}{\partial r}</td>
<td>_{r=0} = \frac{\partial \bar{u}}{\partial r}</td>
</tr>
<tr>
<td>( T_s(r = R) = T_0 )</td>
<td></td>
</tr>
<tr>
<td>For full insert: ( T_s(x = 0) = T(x = L) = T_0 )</td>
<td></td>
</tr>
<tr>
<td>For partial insert: ( T_s(x = L - L_1 - L_{ins}) = T(x = L - L_1) ), ( T_s(x = L - L_1 - L_{ins}) = T(x = L - L_1 - L_{ins}) )</td>
<td></td>
</tr>
</tbody>
</table>

CFD simulations based on the above formulation require models for the negative momentum source term, \( \tilde{S}_m \), and the interfacial heat transfer coefficient, \( h_v \). In the next section, the first will be obtained by measuring the pressure drop of the metal foam, and the second will be selected by comparing simple, zero-dimensional (Zero-D) model solutions to measured volume and pressure data from compression experiments. Then, using the experimentally validated models, eight CFD runs with different compression speeds, different metal foams (10PPI and 40PPI), and different layouts of the metal foams (full-insert vs. partial-insert) will be studied.

3. INVESTIGATIONS ON PRESSURE DROP AND HEAT TRANSFER MODELS

3.1. Pressure Drop

The pressure drop values of the metal foams are measured. A schematic of the experimental setup is shown in Fig. 3 (a). A fan moves air as shown. Two kinds of metal foams are inserted and measured for pressure drop: one with 10 pores per inch (10PPI), and one with 40PPI. They both have 93% porosity. A picture of the metal foam pieces is shown in Fig. 3 (b). The flow rate of air can be adjusted by a manual valve. A Sierra TopTrack 822S flow meter is used to measure the volumetric flow rate. The gauge pressure, also considered as the pressure drop of the porous foam, is measured from the pressure tap by a micro-manometer.

Results are shown in Fig. 4. The micro-manometer has an uncertainty of 0.125Pa. The uncertainty of the measured Darcian velocity is 1.5%. The data are fit into the form of Eq. (6), from which the permeability, \( K \), and the coefficient in the Forchheimer term, \( b \), are obtained:
10PPI metal foam: $K = 2.397 \times 10^{-7} m^2$, $b = 570.1/m$

40PPI metal foam: $K = 9.913 \times 10^{-8} m^2$, $b = 1152.0/m$

MATLAB/Simulink using the built-in Proportional Integrator (PI) Controller. Due to the nature of the control valve, the flow rate is only approximately constant. Downstream of the control valve is the compression chamber. The chamber is apolycarbonate pipe with a length of 353 mm and an internal diameter of 50.8 mm. It is marked with graduations so that the water level in the chamber can be recorded. A PCI-DAS1602/16 16-bit analog input board is used to send and receive signals. Using xPC target, the I/O board is connected to a MATLAB/Simulink model for control and data acquisition. The sampling frequency is chosen to be 4000 Hz to correctly acquire the signal from the turbine meter. The three measured signals are: upstream pressure, turbine meter flow rate, and chamber pressure.

Two experimental runs are made, one with the chamber filled with 10PPI metal foam, and the other with 40PPI metal foam. The conditions for these runs are given in Table 2. Due to a difference in the lengths of the compression chamber and the metal foam inserts, there remains at the bottom of the chamber a region that does not have insert material. In these experiments, the liquid piston is brought to the bottom of the metal foam at the beginning of compression so that the entire active length of chamber is filled with the metal foam insert. The instantaneous volume of air in the compression chamber is found by subtracting the cumulative amount of water added to the chamber from the initial volume of air. The instantaneous pressure of the compression chamber air is found directly from the pressure transducer at the top of the chamber. The instantaneous temperature is not measured directly; it is calculated from pressure and volume measurements using the ideal gas law. Although previous efforts had been made to measure the air temperature using a thermocouple, the ideal gas law method is preferred for several reasons. A thermocouple measurement is only one point in a thermal field that contains large thermal gradients. Additionally, the relatively fast temperature change during the transient process is difficult to capture accurately with the relatively slow responding thermocouple. Additionally,

The Zero-D numerical model has been used for calculating the transient bulk temperature and pressure in a compressor with heat exchanger inserts [23]. The model is used to compare with experimental results. The model is given by the following energy equations for air and solid, respectively:
The two equations are coupled through an interfacial heat transfer coefficient, \( h_v \).

\[
h_v(T_{air} - T_{solid}) = (\frac{\varepsilon}{\varepsilon}) + 1)\varepsilon A U P + \frac{c_p}{\varepsilon} \varepsilon A U m V \frac{dp}{dv} \quad (13)
\]

\[
c_s \rho_s (1 - \varepsilon) A U m \frac{d T_{solid}}{dv} = -h_v(T_{air} - T_{solid}) \quad (14)
\]

The two equations are coupled through an interfacial heat transfer coefficient, \( h_v \).

### Table 2. Conditions of experimental runs

<table>
<thead>
<tr>
<th>Experimental Runs</th>
<th>Initial length of air region (m)</th>
<th>Chamber radius (m)</th>
<th>Initial and ambient temperature (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>10PPI Run</td>
<td>0.294</td>
<td>0.0254</td>
<td>297</td>
</tr>
<tr>
<td>40PPI Run</td>
<td>0.272</td>
<td>0.0254</td>
<td>297</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Initial Pressure (Pa)</th>
<th>Compression time (s)</th>
<th>Average Flow Rate (m/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>10PPI Run</td>
<td>101644</td>
<td>2.54</td>
</tr>
<tr>
<td>40PPI Run</td>
<td>102632</td>
<td>3.78</td>
</tr>
</tbody>
</table>

A number of references have given interfacial heat transfer correlations often used for porous media. Wakao and Kaguei proposed a correlation for packed beds of 40% porosity based on the length scale of particle diameter [11]. The correlation is converted for use in a form which is based on mean pore diameter by assuming a ratio of pore volume to particle volume of 4/6. The converted Wakao and Kaguei correlation reads,

\[
\frac{h_{sf}}{k} = 1.748 + 1.19\left(\frac{\rho u d m}{\mu}\right)^{0.6} P r^{0.3} \quad (15)
\]

Kuwahara, et al. proposed a correlation for packed square rods in cross flow based on the rod edge size [12]. The exact layout of the porous medium was given. After conversion, the correlation based on mean pore diameter for a 93% porosity medium reads,

\[
\frac{h_{sf}}{k} = 9.48 + 0.29\left(\frac{\rho u d m}{\mu}\right)^{0.6} P r^{0.3} \quad (16)
\]

Zukauskas developed heat transfer correlations for cylinders in cross flow [13]. The correlations have been further modified to be used for porous media [16]. They are given by:

\[
N u = \frac{h_{sf}}{k_f} = 0.76 R e_d^{0.4} P r^{0.37}, \quad (1 \leq R e_d \leq 40) \quad (17)
\]

\[
N u = \frac{h_{sf}}{k_f} = 0.52 R e_d^{0.5} P r^{0.37}, \quad (40 \leq R e_d \leq 10^3) \quad (18)
\]

\[
N u = \frac{h_{sf}}{k_f} = 0.26 R e_d^{0.6} P r^{0.37}, \quad (10^3 \leq R e_d \leq 2 \times 10^5) \quad (19)
\]

where \( d_f \) is the filament diameter of the metal foam. Nakayama, et al. developed a heat transfer correlation for open cell porous media based upon mean pore diameter [14],

\[
\frac{h_{sf}}{k} = 0.07\left(\frac{\rho u d m}{\mu}\right)^{0.2} P r \quad (21)
\]

In addition, they converted two other correlations for open-cell porous media to be based upon mean pore diameter:

Kamiuto and Yee

\[
\frac{h_{sf}}{k} = 0.996\left(\frac{\rho u d m}{\mu}\right) P r^{0.791} \quad (\varepsilon = 0.93) \quad (22)
\]

Fu, et al.

\[
\frac{h_{sf}}{k} = 0.275\left(\frac{\rho u d m}{\mu}\right)^{1.01} \quad (P r = 0.72) \quad (23)
\]

The Zero-D model is solved to simulate the experimental run by substituting the different heat transfer correlations shown above. An iterative implicit method used in [23] is used here to solve Eqns. (13) and (14) numerically. When evaluating the Reynolds number in the heat transfer correlations, a velocity that is half the piston velocity is used to approximate the overall mean velocity of the air in the chamber. The physical properties of air and metal foam (solid) used in the computation are given in Table 3. The air Prandtl number is 0.713. A fit of conductivity data from [24] with temperature over 273K to 573K is used. The metal foam is aluminum. The average cell size is used as the characteristic pore diameter in Eqns. (15), (16), and (21) – (23). The averaged cell size values of the 10PPI and 40PPI metal foams are, respectively, 3.61mm and 2.38mm. An average filament size of 0.6mm is used in Eq. (17) – (19). The values of surface area per volume for the 10PPI and 40PPI metal foams are, respectively, 697/m and 1677/m.

### Table 3. Physical Properties

<table>
<thead>
<tr>
<th>( \varepsilon )</th>
<th>( \rho )</th>
<th>( k )</th>
<th>( c_p )</th>
<th>( c_v )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.65</td>
<td>1.002 \times 10^{-3} Pa · s</td>
<td>2719 kg/m^3</td>
<td>205 W/(mK)</td>
<td>1005 W/(mK)</td>
</tr>
<tr>
<td>0.56</td>
<td>1.000 kg/m^3</td>
<td>41813 J/(kgK)</td>
<td>0.56 W/(mK)</td>
<td>871 J/(kgK)</td>
</tr>
</tbody>
</table>

The computed and the experimentally measured air temperature values are normalized based on the initial temperature. The results are shown in Fig. 6. In the experiment, the temperature is calculated from the ideal gas law, based on instantaneous pressure and volume measurements. The uncertainties of the initial volume and initial temperature measurements are respectively 1.013 cm^3 and 0.8 K. The uncertainties in the transient pressure and volume measurements are, respectively, 1724 Pa and 3 cm^3. Using an uncertainty propagation equation from [25] yields a temperature of 15 – 17K. Despite local mismatches and uncertainties in the experiments, one sees from Fig. 6 that the solutions obtained based on the Kamiuto and Yee correlation agree with experimental measurements best throughout the whole compression process. Therefore, the Kamiuto and Yee correlation (Eq. (22)) is chosen for CFD modeling to calculate interfacial heat transfer coefficients in Eqns. (7) and (12).

### 4. NUMERICAL METHOD, VERIFICATION, AND VALIDATION

#### 4.1 Numerical Method

The transport equations, Eqns. (1) – (3), (7), and (12) are
solved by the finite volume method using the commercial CFD Software ANSYS FLUENT including its VOF solver. User-Defined-Function (UDF) codes are written to solve the negative momentum source term, and the interfacial heat transfer term based on the Kamiuto and Yee correlation is solved for energy transport in the two-energy equation model. It is assumed that the permeability and coefficient for the Forchheimer term apply equally to water and air. The following numerical methods are applied using ANSYS FLUENT. The 1st order implicit method is used on the transient discretization. Spatial derivatives of velocity, density, and temperature are differenced using a 2nd order upwind scheme. The algorithm of Pressure Implicit with Splitting Operators [26] is used for pressure-velocity coupling. The Pressure Staggering Option scheme [27] is used for calculating discretized pressure values on a staggered grid. The gradient terms are handled using the Green-Gauss Cell-Based method, which calculates the face value of a variable based on the arithmetic average of the values at its adjacent cell centers. In each time step, convergence is satisfied when the residual is smaller than $10^{-5}$. Eight CFD runs are calculated by varying the inlet water velocity, the layout of the insert (full-insert vs. partial-insert), and the pore size. The cases are shown in Table 4. The schematic of the chamber is shown in Fig. 2. In all cases, the dimensions of the chamber are: $L = 0.294m$, $R = 0.0254m$. For the four full-insert cases, the entire chamber length $L$ is occupied by the insert. For the four partial-insert cases, the insert length is: $L_{ins} = \frac{L}{2} = 0.147m$, and the distance between the top boundary of the insert and the top cap of the chamber is: $L_1 = 0.013m$. The initial temperature (for both fluid and solid) and pressure are: 297K and 101644Pa. The initial air velocity is 0.0, and air volume fraction is 1.0 in the chamber. The physical properties given in Table 3 are used. The number of grid cells is 26,000 for the full-insert cases, and 44,485 for the partial-insert cases. The time step size is 0.0002s for the full insert cases with the smaller inlet velocity. The time step size decreases as the number of nodes and the velocity increase.

### 4.2. Grid-Independence Verification

CFD Run1, which has a relatively small number of grid cells and large time step size compared to other runs, is tested for grid independence. The grid-independence run is done on the mesh that has 65,600 cells, using a time step size of 0.0001s. The average air pressure and temperature at the final compression time calculated from Run1 and its grid-independence run are, respectively: 1,317,934Pa and 1,318,324Pa, and 345.8K and 345.6K. The temperature contour plots of the two at the final compression time also display identical features. Therefore, grid independence is satisfied.

### 4.3. Validation

Two validation methods have been done. The first validation is a comparison with experimental results. Since CFD Run1 simulates the same experimental conditions as the 10PPI run shown in Table 2, the bulk air pressure rise and temperature rise calculated from CFD can be compared to the experimental results (shown in Fig. 7). In the CFD simulation, the total compression time is slightly longer than in the experiment. In the experiment, the flow rate slows at the end due to the valve behavior, causing the air temperature to drop slightly. Overall, the two agree.

![Image](a) 10PPI metal foam insert

![Image](b) 40PPI metal foam insert

Fig. 6. Comparison of experimental results to Zero-D model solutions obtained by using different heat transfer correlations (Variables are non-dimensionalized by their initial values).

<table>
<thead>
<tr>
<th>Cases</th>
<th>Insert</th>
<th>$t_f$ (s)</th>
<th>$U_0$ (m/s)</th>
<th>No. of Grid Cells</th>
<th>$\Delta t$ (s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Run1</td>
<td>10PPI, Full</td>
<td>2.6</td>
<td>0.103</td>
<td>26000</td>
<td>0.0002</td>
</tr>
<tr>
<td>Run2</td>
<td>40PPI, Full</td>
<td>2.6</td>
<td>0.103</td>
<td>26000</td>
<td>0.0002</td>
</tr>
<tr>
<td>Run3</td>
<td>10PPI, Full</td>
<td>1.3</td>
<td>0.206</td>
<td>26000</td>
<td>0.0001</td>
</tr>
<tr>
<td>Run4</td>
<td>40PPI, Full</td>
<td>1.3</td>
<td>0.206</td>
<td>26000</td>
<td>0.0001</td>
</tr>
<tr>
<td>Run5</td>
<td>10PPI, Partial</td>
<td>2.6</td>
<td>0.103</td>
<td>44485</td>
<td>0.00005</td>
</tr>
<tr>
<td>Run6</td>
<td>40PPI, Partial</td>
<td>2.6</td>
<td>0.103</td>
<td>44485</td>
<td>0.00005</td>
</tr>
<tr>
<td>Run7</td>
<td>10PPI, Partial</td>
<td>1.3</td>
<td>0.206</td>
<td>44485</td>
<td>0.00025</td>
</tr>
<tr>
<td>Run8</td>
<td>40PPI, Partial</td>
<td>1.3</td>
<td>0.206</td>
<td>44485</td>
<td>0.00025</td>
</tr>
</tbody>
</table>

In the second validation, the problem of case Run 1 is simulated with several modifications. The first change is that thermal boundary conditions are changed from isothermal to adiabatic:

$$\frac{\partial T}{\partial t} |_{r=R} = \frac{\partial T_s}{\partial x} |_{x=L} = \frac{\partial T_s}{\partial x} |_{x=L} = 0$$  \hspace{1cm} (24)

In order to make the water adiabatic, the thermal conductivity is manipulated to $10^{-6}W/(mK)$. Additionally, the heat transfer between solid and fluid is made zero by setting the interfacial heat transfer coefficient, $h_v$, to zero. As expected, the CFD simulates adiabatic compression of air. In theory, the pressure vs. volume trajectory of adiabatic compression has a polytropic exponent of 1.4. The CFD result shows that the bulk pressure
verse volume follows the same adiabatic trend as shown in Fig. 8. At the end of compression, the pressure calculated from CFD is slightly smaller than that from the theory. This is because the liquid piston still conducts a small amount of heat for its conductivity is not exactly made to zero for numerical stability reasons.

The computed fluid temperature field and streamlines are shown in Figs. 10 – 12. An arrow with a note “Interface” indicates the instantaneous location of water-air interface. In addition to the full scale contour plots, enlarged air region plots at later compression times are shown on the right of each figure.

The velocity streamlines in the fully occupied chambers appear to be straight as shown in Fig. 10; very small and weak secondary flows are present near the top corner of the chamber. The pressure drop of the flow through the metal foam is small compared to the absolute pressure of air, yet it is large enough to overcome small local pressure instabilities that could otherwise result in vortices. In the partially occupied chambers, vortices are formed outside the porous region, yet, inside the porous region, the flow is stable, as shown in Figs. 11 and 12. With the same metal foam, smaller compression speeds result in larger vortex sizes. The effects of the two different metal foams (10PPI vs. 40PPI) are shown to be insignificant on the size of the vortices formed outside the metal foam regions.

No entries in the literature were found that simulate liquid piston compression with porous media inserts. The closest reference, to the best knowledge of the authors, is a solid piston compression problem with partial insert of porous media [20]; yet, this paper formulates the problem with a one-energy equation model and single phase flow. No pressure drop data are given.

5. CFD RESULTS

The eight CFD cases in Table 4 are simulated. The bulk air temperature during compression is calculated and shown in Fig. 9. In most cases, the metal foam does a good job of suppressing the air temperature rise during compression. The smallest bulk air temperature rise during compression is 35K (Run2), which is compressed by the smaller liquid-piston speed and has a chamber that is fully occupied by the finer-mesh metal foam. For the same porous insert, higher compression speeds result in larger temperature rises. For the same compression speed, the full insert case has an advantage over the partial insert case, and 40PPI foam has an advantage over the 10PPI foam for reducing the temperature rise. In the partial insert situations, since the foam is inserted in the upper half of the chamber region, the bulk air temperature rises fast in the beginning. As the air is being pushed from the non-porous region into the porous region, its temperature cools due to heat transfer with the metal foam. This causes the bulk air temperature to drop by a few degrees in the middle of the compression process.
At the end of compression, the air temperature distribution becomes highly non-uniform, as volume decreases. Relatively high local temperature values are seen in regions very close to the top cap. In fully-occupied chambers, the high local air temperature values at the end of compression are mainly due to the stagnant flow. Since Nusselt numbers between the flow and the porous medium depend on Reynolds number, stagnant flow impedes heat transfer. In partially occupied chambers, the high local air temperature values at the end of compression are high due to lack of heat transfer surfaces; even though, the local flow is relatively active, as shown by a vortex that has been formed since the earlier stage of compression.

The computed solid temperature fields at the end of compression for various cases are shown in Fig. 13. In the fully occupied chambers, the largest local temperature rise in the solid is less than 1K. In the partially occupied chambers, the largest local temperature rise is 57.4K, which occurs at the top surface of the metal foam. The bulk of the solid material is maintained at a low temperature. Great differences in local temperature values between the air and the solid are found in certain regions. This confirms the advantage of using the two-energy equation formulation instead of the one-energy, equilibrium formulation for this kind of problem.

6. CONCLUSIONS

The present study has numerically investigated liquid piston compression chambers with metal foam inserts. Two aluminum foams, 10PPI and 40PPI, are studied. The permeability and the coefficient for the Forchheimer term are obtained by measuring the pressure drop values. Compression experiments using a liquid piston and metal foam inserts are conducted to provide data which are compared to Zero-D solutions based on different heat transfer correlations from references. From the comparison, the Kamiuto and Yee heat transfer correlation is found to be more suitable for the metal
Fig. 11. Temperature field and velocity streamline at different times during compression, partially occupied chambers, low compression speed.

Based on the models validated by experiments, CFD simulations on the compressor are done by combining the VOF and the two-energy equation formulations. Fluid field and temperature distributions are obtained from the CFD. The air flow in the fully occupied chamber is straight, parallel to the chamber’s axial direction due to resistance of metal foam. In the partially occupied chambers, secondary flow structures are present during compression in the air regions outside the metal foam. The dimensions of vortices generated are related to compression speed; smaller speeds induce larger vortices, and larger speeds induce smaller vortices. The air is most effectively cooled in fully occupied chambers. The temperature distributions of the metal foam in the partially occupied chambers are non-uniform at the end of compression. High temperature values are found at the top surface of the metal foam. In fully occupied chambers, the maximum local temperature rise for the metal foam is less than 1K. The very different local temperatures between the metal foam and the adjacent fluid confirms the appropriateness of using the two-energy equation modeling (instead of the one-energy equation, equilibrium modeling) for CFD simulations of flow being compressed through porous media.

The formation of vortical structures is enhanced when a partial insert is used. These vortical structures have the potential to reduce the maximum air temperature by mixing the stagnant high temperature air at the top of the chamber with the relatively cool air near the walls and porous insert. This study shows that, although vortical structures are enhanced when a partial insert is used, the maximum air temperature is still higher than the full insert case. This suggests that an optimum $L_1$ distance may exist such that mixing is enhanced without diminishing too much the heat-absorbing solid surface. This could be investigated in future studies.
(a) Run 7, 10PPI, Partial Insert, $U_0 = 0.206 \text{m/s}$, $t_f = 1.3 \text{s}$

(b) Run 8, 40PPI, Partial Insert, $U_0 = 0.206 \text{m/s}$, $t_f = 1.3 \text{s}$

Fig. 12. Temperature field and velocity streamline at different times during compression, partially occupied chambers, high compression speed

Fig. 13. Temperature distributions in the metal foam at the end of compression, $t = t_f$
NOMENCLATURE

\( a_v \) Area per unit volume of porous medium
\( b \) Coefficient for the Forchheimer term
\( c_p \) Constant-pressure specific heat
\( c_v \) Constant-volume specific heat
\( d \) A characteristic length based on filament diameter
\( d_f \) Filament diameter
\( d_m \) Mean pore diameter
\( g \) Gravitational acceleration
\( h_{sf} \) Surface heat transfer coefficient
\( h_v \) Volumetric heat transfer coefficient
\( K \) Permeability
\( \kappa \) Thermal conductivity
\( L \) Chamber length
\( L_x \) Length of the upper region without insert
\( L_{ins} \) Length of the insert region
\( Nu \) Nusselt number
\( P \) Average pressure
\( p \) Local pressure
\( Pr \) Prandtl number
\( R \) Radius of chamber
\( r \) Radial coordinate
\( \mathcal{R} \) Ideal gas constant
\( Re_d \) Reynolds number based on characteristic length \( d \)
\( \dot{m} \) Mass flow rate
\( \dot{q}_m \) Momentum source term
\( T \) Local air temperature
\( T_0 \) Initial temperature; wall temperature
\( T_{air} \) Average air temperature in the chamber
\( T_s \) Local solid temperature
\( T_{solid} \) Average temperature of solid in the chamber
\( t \) Time
\( U_{in} \) Liquid piston velocity
\( V \) Instantaneous volume of chamber
\( x \) Axial coordinate

Greek Symbols
\( \alpha \) Volume fraction
\( \epsilon \) Porosity
\( \mu \) Dynamic viscosity
\( \rho \) Density

Subscripts
0 Initial value of variable
1 Air phase
2 Water phase
D Darcian velocity
f Values at the end of compression
s Solid

Superscripts
* dimensionless variable

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