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Numerical investigation of heat transfer in air channels for thermal energy storage applications in buildings

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Abstract

Energy demand has significantly increased in building sectors over the past decade because of population growth, increased indoor activities, and increased energy requirements for controlling the internal environment due to global climate change. The residential sector accounts for 32% of the total energy consumption. Space and water heating account for most of the fossil fuel use in residential buildings. Several studies have been done to optimize the design of heat transfer fins and channels in space and water heating systems to improve their efficiency. In this study the effects of the cross-sectional shape of air-channels on heat transfer from the channels to the air is investigated. Air channels with circular, triangular, rectangular and sinusoidal shaped cross sections are considered. Further, heat transfer to air flowing through beds comprising these air channels are also investigated. Numerical analyses is performed to investigate the dependency of the outlet temperatures to optimize the shape of the air channels and beds. Results show the sinusoidal channel and bed have better performance in relation to the outlet temperature. Considering channels with a length of 100 mm, a surface area of 1770 m^2, and a wall-to-air heat transfer rate of 790 W/m^2, the air outlet temperature for the sinusoidal channel is about 57°C higher than that of the circular channel. Considering beds with a height of 5 mm and wall-to-air heat transfer rate of 790 W/m^2, the outlet temperature of the sinusoidal bed is about 10°C higher than the outlet temperature for a rectangular bed with similar surface area. The results are applicable to the design of thermal energy storage units that could be used directly or indirectly (such as adsorbent beds for thermal energy storages) to supply heat to air within HVAC systems.

1. Introduction

Energy demand has significantly increased in building sectors over the past decade because of population growth, increased indoor activities, and more energy required to control the internal environment due to global climate change [1]. In Ontario, about 20% of the total energy supply is consumed in the form of electricity, most of which comes from non-emitting sources such as nuclear and hydro, while the other 80% is derived from burning fossil fuels for heating, transportation and industry. [2]. According to the Internal Energy Agency, in most countries the residential sector accounts for ~30% of the total energy consumption, however in terms of primary energy this value increases to around 40% [3]. Space heating, water heating, and cooking account for most of the fossil fuel use in residential buildings [1]. The energy required for space heating of residential buildings can be decreased by improving insulation, minimizing air leakage, using heat recovery from ventilation air, and optimizing the design of the components that are responsible for heating and cooling such as vents and ducts [4]. In a previous study, a prototype duct system was used to measure and evaluate the ability for smart booster fans and dampers to control airflow to different vents for the purpose of increasing the efficiency of HVAC systems. The results from both the experimental and numerical evaluation show that the smart booster fan and dampers can significantly improve the airflow at a vent that is underperforming [5].

An important approach to minimizing energy consumption in the building sector is through life cycle analysis [6]. The life cycle of a building includes the production of building materials, construction, the use phase (which includes operation and maintenance), disassembly, and disposal. All these phases can be considered in order to minimize the life cycle energy use and CO_2 emissions. Several studies have shown that for buildings constructed in temperate or cold regions, most of the energy consumption over the building's life cycle occurs during the use phase [7-9]. Thus, integrating and optimizing the design of HVAC components that reduce energy consumption during the use phase in buildings is a sensible strategy.

Decarbonization of electricity generation followed by electrification in the buildings sector is known to be one of the promising pathways to achieve a low carbon future [10]. It is estimated that eliminating the use of fossil fuels in the building sector can support reductions in CO_2 emissions by 31% by 2050 [11] There are currently four technologies for electrification of space heating: air source heat pumps (HPs), air source HPs in conjunction with other heating sources, ground source HPs, and electric resistance thermal storage heating. Air source HPs are a common technology used to transport heat from outdoor to interior space during the heating season and from indoors to

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outdoors during the cooling season. HPs can be ducted or ductless. Ground source HPs transfer the heat to and from earth instead of the outside air. Electric resistance storage heaters, mostly in the form of elements encased in heat storing ceramic, have been used for decades [12]. HPs can produce equivalent space heating for as little as one-quarter the cost of conventional heating or cooling which operate based on fossil fuels [13]. Thermal energy storage (TES) systems are mainly used to merge renewable energy in the electricity production mix, and also proved to be useful for storing the electrical energy from peak to off peak hours, becoming useful for demand-side management (DSM) [14]. Using simple heat storages including tanks for the stratified storage of hot and cold water have been studied widely in building applications [15-17]. Ioan Sarbu et al. [18] reviewed TES technologies and described several energy storage methods and the calculation of storage capacities such as sensible, latent , and thermo-chemical storages. In another study, Dieter Patteeuw et al. [19] have coupled TES systems to electric heating systems (HPs and resistance heaters). They showed that only these integrated systems can overcome the constraints on both the supply and demand side of electric power systems. Arteconi et al. [14] analyzed HPs with radiators coupled with TES to show how the HP system operates and affects the thermal comfort of occupant base on demand side management (DSM). They showed that the HPs integrated with TES systems can achieve good control of the indoor temperature, even after three hours of turning off the HPs.

In order to improve the efficiency of these systems, several studies have been done to reach optimal design of heat transfer fins and channels. Each one of them had suggested various shapes and designs base on their systems [20-22]. In another study, A.Mohan et al. [23] gave both theoretical and simulation analysis for the duct size. They also compared the pressure loss in rectangular and circular ducts, concluding that the circular duct has minimal friction loss. Also, Vishal shah et al. [24] compared rectangular and circular ducts and showed that rectangular ducts cause more turbulence compared to circular ducts.

In this study the effects of the cross-sectional shape of air-channels on heat transfer from the internal surface of channels to air flowing through the channels is investigated. Air channels with circular, triangular, rectangular and sinusoidal shaped cross sections are considered. Further, heat transfer to air flowing through beds comprising these air channel shapes are also investigated. The beds have a greater cross-sectional area than the channels and can be used in the design of ducts that transport larger volumes of air in HVAC systems. Numerical analyses is performed to investigate the outlet temperatures to optimize the shape of the air channels and beds. Results show the sinusoidal channel and bed have better performance in relation to the outlet temperature. Considering channels with a length of 100 mm, a surface area of 1770 m^2, and a wall-to-air heat transfer rate of 790 W/m^2, the air outlet temperature for the sinusoidal channel is about 57°C higher than that of the circular channel. Considering beds with a height of 5 mm and wall-to-air heat transfer rate of 790 W/m^2, the outlet temperature of the sinusoidal bed is about 10°C higher than the outlet temperature for a rectangular bed with similar surface area. The results are applicable to the design of TES units that could be used directly or indirectly (such as adsorbent beds for TES) to supply heat to air passing through HVAC systems within buildings.

2. Materials and methods

2.1 Numerical Analysis Methods for Determining Heat Transfer within Channels

Numerical analyses was performed to compare the amount of heat transferred to air that passes through channels with different cross-sectional geometries. Four different cross-sectional geometries were considered: 1) circular, 2) triangular, 3) rectangular, and 4) sinusoidal. These channels are shown in Figure 1 and their dimensions are given in Table 1. The length, L, and internal surface area, A, for all channels is 100 mm, and 1770 mm^2, respectively. The characteristic parameter, a, and cross-sectional area for each channel differs as shown in Figure 1 and Table 1. All four channels have the same internal surface area of 1770 mm^2 which is assumed to provide a constant heat flux of 790 W/m^2. For each of the four channels, air enters the inlet with a velocity of 1 m/s, a temperature of 20 °C, and a constant density of 1.225 kg/m^3. For each channel, the outlet is assumed to have zero Pascal gauge pressure. Also, a no slip boundary condition has been applied at the contact surface between the air and the channels. Computational Fluid Dynamics (CFD) analysis was performed using Ansys software to determine the heat transferred to the air as it passes through the channels. The mesh properties used in the simulation for determining the heat transfer within channels are given in Table 2. Figure 2 shows the specified mesh for each case study. Also, the dependency of the solution from the grid for all cases is studied. For instance, the number of elements used to simulate the sinusoidal channel is 298240 and reducing this number by 37,690 causes a change in the results of only 3%. Considering the properties of air at 20°C such as density (1.2047 kg/m^3), dynamic viscosity ($1.8205*10^{-5}$ kg/m.s) and also calculating the hydraulic diameters of the channels as 5.63 mm, 3.40 mm, 4.42mm and 3.02mm for the circular, triangular, rectangular and sinusoidal channel shapes, respectively, the Reynolds numbers for the circular, triangular, rectangular and sinusoidal channel shapes are 373, 225, 292 and 200, respectively, showing laminar flow for all channel shapes.

Table 1. Dimensions of the duct channels.

Shape	a (mm)	L (mm)	A (mm^2)	Cross-sectional area (mm^2)
Circular	5.63	100	1770	24.96
Triangular	5.90	100	1770	15.09
Rectangular	4.42	100	1770	19.58
Sinusoidal	5.00	100	1770	12.50

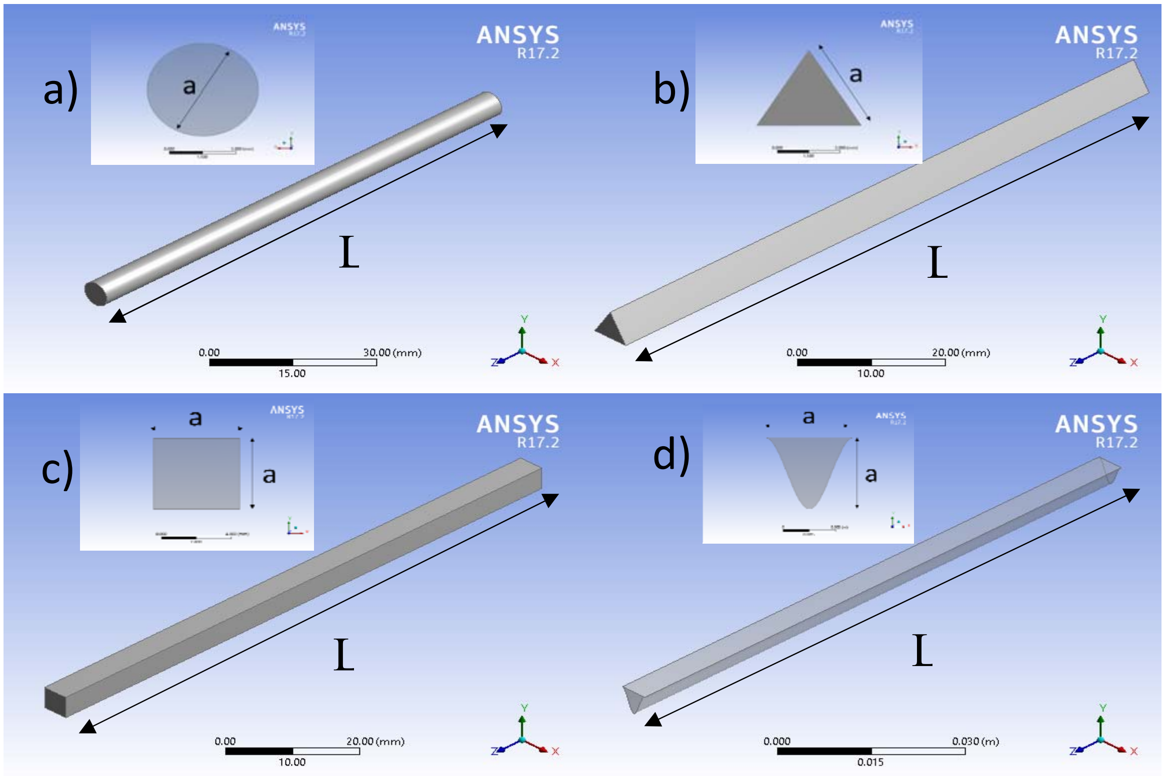

Figure 1. Geometry used in the simulation (a) Circular (b) Triangular (c) Rectangular (d) Sinusoidal.

Table 2. Mesh properties used in the simulations for the channels.

Shape	Min size (m)	Aspect ratio	Skewness	Growth rate	Number of elements
Circular	1.46e-5	1.64	0.240	1.2	4536
Triangular	1.46e-5	3.9	0.183	1.2	6768
Rectangular	1.46e-5	3.1	0.003	1.2	6390
Sinusoidal	1.46e-5	3.2	0.150	1.2	298240

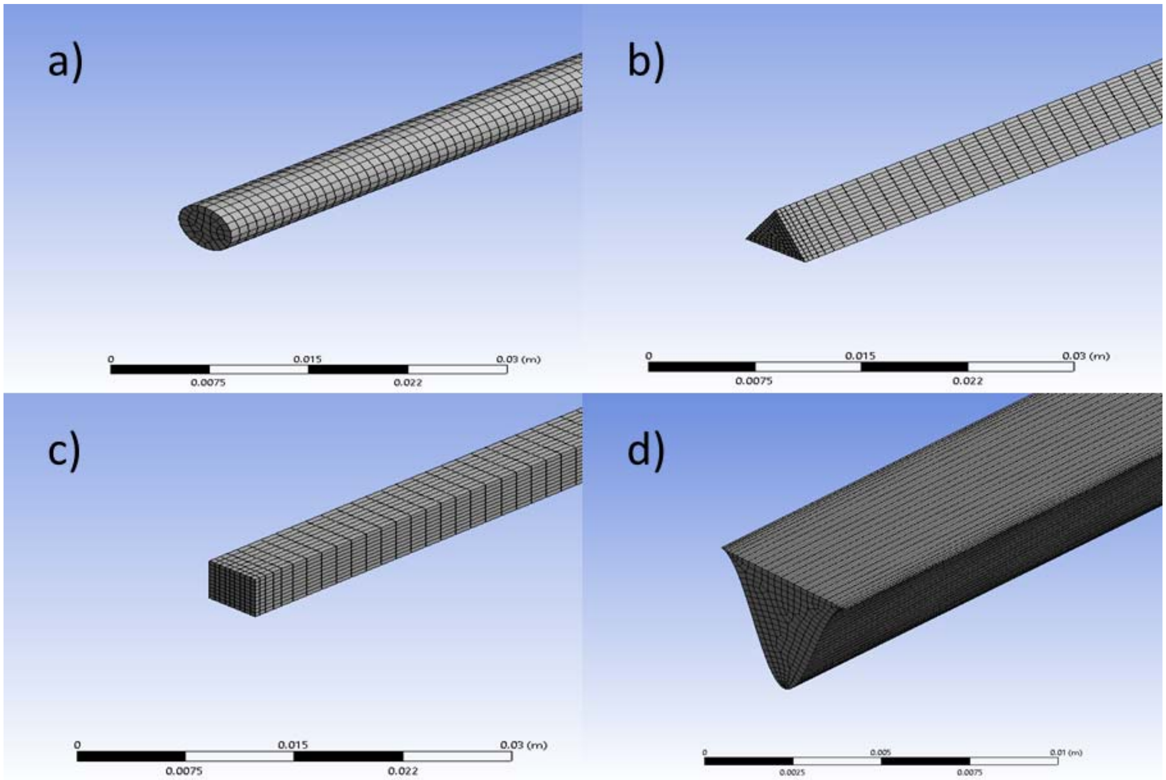

Figure 2. Specified mesh used in the simulation (a) Circular (b) Triangular (c) Rectangular (d) Sinusoidal.

2.2 Numerical Analysis Methods for Determining Heat Transfer within Beds

The heat transferred to air flowing through five bed designs with different shapes is numerically determined. The bed designs are based on five different shapes, which are: 1) Sinusoidal, 2) Triangular 1 (same width, d, and length, L, as the sinusoidal bed), 2) Triangular 2 (same internal surface area, A, as the sinusoidal bed), 3) Rectangular 1 (same width and length as the sinusoidal bed) and Rectangular 2 (same surface area as the sinusoidal bed). The geometry of the beds is shown in Figures 3 and 4 and the corresponding values for the dimensional parameters are given in Table 3. The height of all beds is assumed to have the same value of h = 5 mm. The internal surfaces of all beds is assumed to supply a constant heat flux of 790 W/m^2. The boundary conditions are similar to those used in simulating the channels. That is, air enters the inlet of the beds with a velocity of 1 m/s, a temperature of 20 °C, and a constant density of 1.225 kg/m^3. For each bed the outlet is assumed to have zero Pascal gauge pressure. Also, the outer side walls are assumed to be perfectly insulated. Computational Fluid Dynamics (CFD) analysis was performed using Ansys software to determine the heat transferred to the air as it passes through the beds. The mesh properties used in the simulation for determining the heat transfer within beds are given in Table 4. Figure 5 shows the specified mesh for each case study. Also, the dependency of the solution from the grid for all cases is studied. For instance, the number of elements used to simulate the sinusoidal duct is 575,865 and reducing this number by 13,401 causes a change in the results of only 0.2%. Considering the properties of air at 20°C such as density (1.2047 kg/m^3), dynamic viscosity (1.8205*10^{-5} kg/m.s) and also calculating the hydraulic diameters of the beds as 7.94 mm, 9.25 mm, 9.30 mm, 9.59 mm and 9.66 mm for the sinusoidal, triangular 1, triangular 2, rectangular 1 and rectangular 2 bed shapes, respectively, the Reynolds numbers for the sinusoidal, triangular 1, triangular 2, rectangular 1 and rectangular 2 bed shapes are 525, 612, 615 , 635 and 639, respectively showing laminar flow for all bed shapes.

Table 3. Dimension of beds.

Bed shape	H (mm)	D (mm)	L (mm)	A (mm^2)	Cross-section area (mm^2)
Sinusoidal	5	118	100	14341	590
Triangular 1	5	118	100	12064	590
Triangular 2	5	137.94	100	14341	689.7
Rectangular 1	5	118	100	11800	590
Rectangular 2	5	143.41	100	14341	717.05

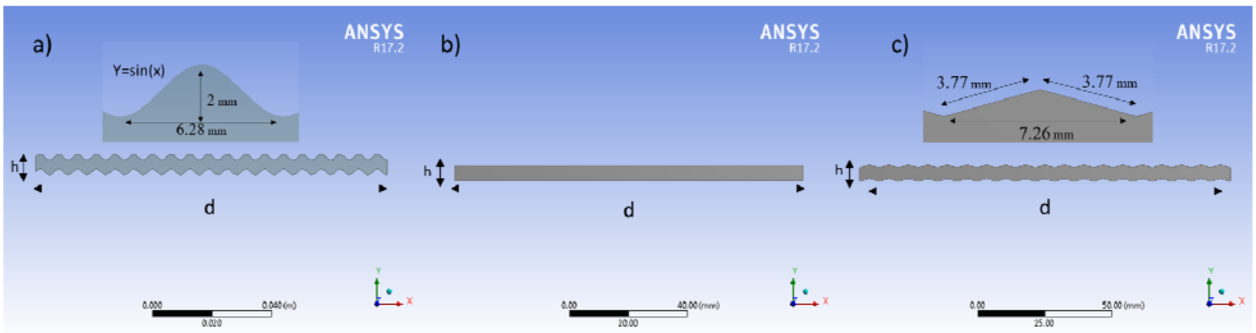

Figure 3. Cross-section schematic of modelled beds. (a) Sinusoidal (b) Rectangular (c) Triangular.

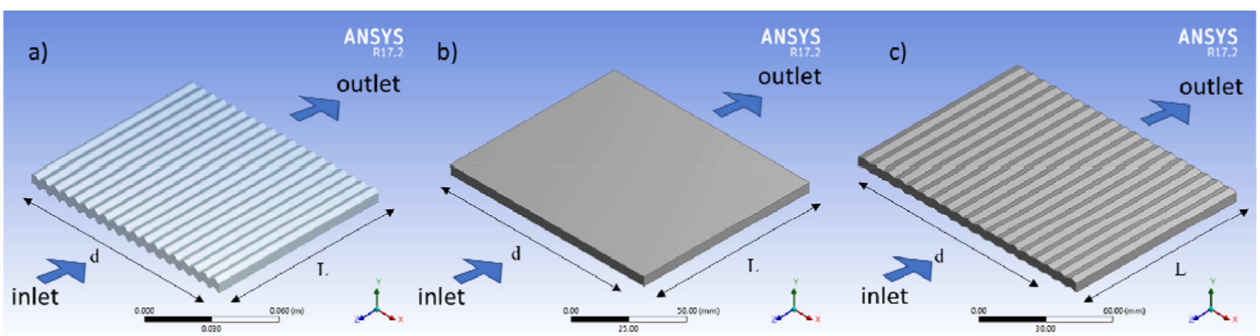

Figure 4. Geometry used in the simulation (a) Sinusoidal (b) Rectangular (c) Triangular.

Table 4. Mesh properties used in the simulations for the beds.

Shape	Min size (m)	Aspect ratio	Skewness	Growth rate	Number of elements
Sinusoidal	1.46e-5	1.68	0.342	1.2	575865
Triangular 1	1.46e-5	2.3	0.177	1.2	14740
Triangular 2	1.46e-5	2.5	0.180	1.2	15600
Rectangular1	1.46e-5	2.25	0.019	1.2	12420
Rectangular 2	1.46e-5	2.56	0.015	1.2	11890

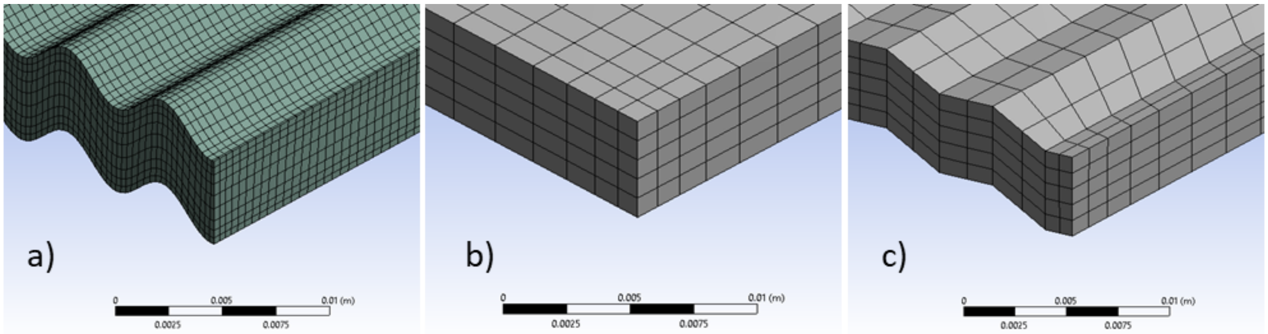

Figure 5. Specified mesh used in the simulation of the (a) sinusoidal (b) rectangular, and (c) triangular beds.

3. Results

Figure 6 shows the average outlet temperature for the four different channel shapes studied. The results show that the sinusoidal shape has the highest outlet temperature of 136.3 °C. Accordingly, the average outlet temperatures for circular, triangular, and rectangular shapes are 79.3 °C, 122.9 °C, and 95.9 °C, respectively.

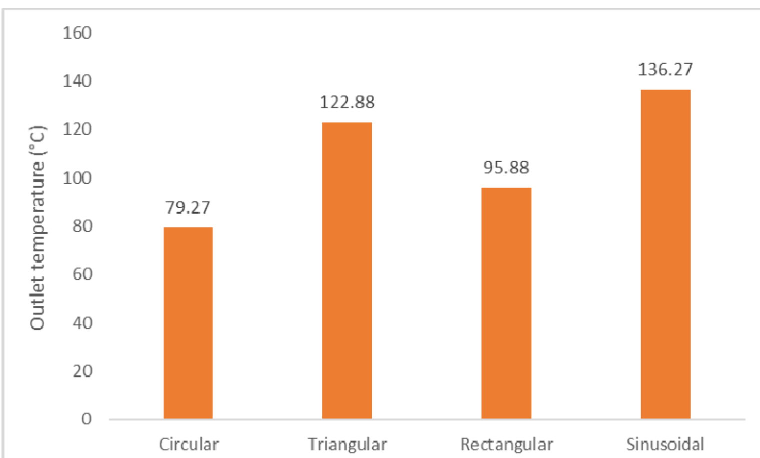

Figure 6. Average outlet temperature for different channel shapes.

Figure 7 shows the average outlet temperature for the five different bed shapes studied. The results show that the sinusoidal shaped bed has the highest outlet temperature of 65.9 °C. Accordingly, the average outlet temperatures for triangular 1 (same dimension as the sinusoidal), triangular 2 (same surface area as the sinusoidal), rectangular 1 (same dimension as the sinusoidal) and rectangular 2 (same surface area as the sinusoidal) are 58.3 °C, 58.2 °C, 56.3 °C and 56.3 °C, respectively.

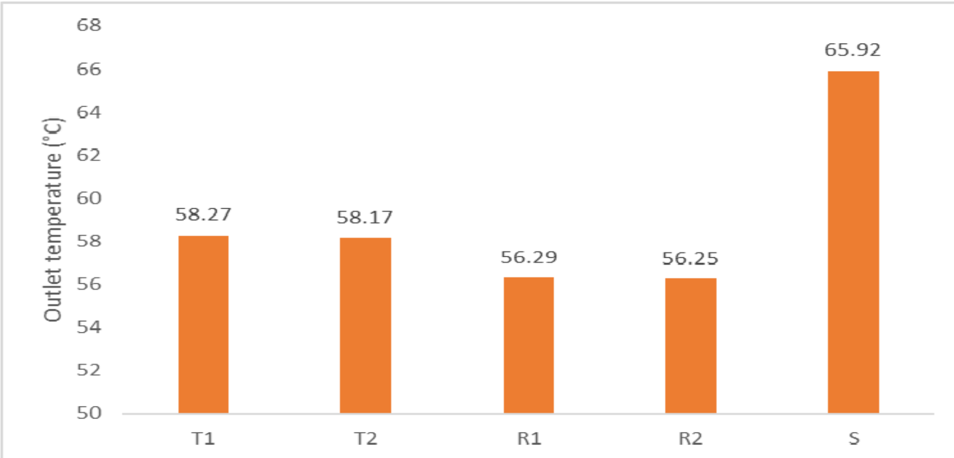

Figure 7. Average outlet temperature for triangular 1 (T1, same dimensions as the sinusoidal), triangular 2 (T2, same surface area as the sinusoidal), rectangular 1 (R1, same dimensions as the sinusoidal) and rectangular 2 (R2, same surface area as the sinusoidal).

Table 5. Inlet pressure of the channels.

Channels	inlet pressure (Pa)
Sinusoidal	7.71
Circular	2.29
Rectangular	3.27
Triangular	4.82

Table 6. Inlet pressure of the beds.

Beds	inlet pressure (Pa)
T1	1.19
T2	1.18
R1	1.13
R2	1.13
S	1.67

Table 5 and Table 6 show the inlet gauge pressure of the air for the channels and beds, respectively. Since the outlet pressures at the boundry conditions were set to zero-gauge pressure, the pressure drop for each case is equal to the inlet gauge pressures. It can be seen that for both channels and beds the sinusoidal shape results in a higher inlet pressure (7.71 Pa for the channel and 1.67 Pa for the bed) compared to all other cross-sectional shapes considered.

Tables 7 and 8 compare the wall-to-air heat transfer rate to the power loss due to the pressure drop along the channels and beds, respectively. It can be seen that the power lost for the sinusoidal channel and bed are higher than the other shapes.

Table 7. Power calculation of the channels from the heat transfer and drop pressure.

Channel shapes	Wall-to-air heat transfer (KJ/s)	Power loss due to pressure drop (KJ/s)
Circular	1.82E-03	5.62E-05
Triangular	1.91E-03	7.15E-05
Rectangular	1.83E-03	6.30E-05
Sinusoidal channel	1.79E-03	9.48E-05

Table 8. Power calculation of the beds from the heat transfer and drop pressure.

Bed shapes	Power gained from the heat transfer (KJ/s)	Power lost from the drop pressure (KJ/s)
Sinusoidal bed	2.78E-02	9.69E-04
Triangular 1	2.78E-02	6.91E-04
Triangular 2	3.08E-02	8.00E-04
Rectangular 1	2.64E-02	6.56E-04
Rectangular 2	4.06E-02	7.97E-04

4. Discussion

As shown in Figures 4 and 5 the temperature of the air flowing through the sinusoidal-shaped channels and beds increases to the greatest extent. These results can assist in the design of ducts with integrated TES. For example, phase change materials or adsorbent-based thermochemical energy storage materials may be applied at the surfaces of the ducts. Further, TES integrated into the ducting network of HVAC systems can potentially be coupled with heat pumps to efficiently utilize surplus electricity during low-demand periods. Future work is required to investigate the potential of such systems and the optimal duct design for their implementation.

5. Conclusion

This paper demonstrates a study on determining the heat transfer within differently shaped duct channels and beds using numerical analysis (CFD) and compares the outlet temperatures for four cases studied. Circular, triangular, square, and sinusoidal duct shapes are considered. In a heating application, wherein heat is transferred from the internal surfaces of the ducts to the air, the sinusoidal channel and bed increased the temperature of the air more than the other cross-sectional shapes considered. For a set heating rate and surface area, the outlet temperature of the sinusoidal channel is about 57 °C higher than the circular channel, whereas the outlet temperature of the sinusoidal bed is about 10 °C higher than that of a rectangular bed. Energy losses due to the pressure drop in the channels and beds with a sinusoidal-shaped cross-section are greater than those in channels and beds with triangular or rectangular cross-sections. However, the energy losses due to pressure drop are small in comparison to the heat gained from the internal surfaces of the channels and beds for the conditions considered in this work (a wall-to-air heat transfer rate of 790 m/s). Thus, for the conditions considered in this work, the sinusoidal cross-sectional shape shows better performance than the triangular, rectangular, or circular cross-sectional shapes for both channels and beds.

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