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An application of infinite regular polyhedrons geometry to design heat exchangers

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ABSTRACT

Paper presents an idea of utilization infinite regular polyhedrons for designing heat exchange devices. The investigated prototype of the heat exchanger was consisted of twenty identical solids. Each of them was composed from eight regular hexagons. CFD simulations, reported in this paper, were used to obtain an influence of the flow domain modification on the intensification of the forced convection heat transfer in the developed device. Computational analyses clearly showed that the experimental testing of the heat exchanger prototype should be realized with include of cuboid turbulence inserts. Heat transfer performance and pressure drop obtained by experimental investigations are presented, too. The average Nusselt number and Fanning friction factor were expressed in terms of the Reynolds number.

1. Introduction

One of the problems, that solid geometry solves lie in complete fill any space with identical Archimedes or Platon polyhedrons. The final solid is constructed by moving a base element parallel to axis of Cartesian coordinate system. The author proposes using the geometry of infinite regular polyhedrons for designing heat exchange devices. There are three types of solids filling the space in the form of two congruent labyrinths. The basic structural component of the first solid consists of squares. As shown in Fig. 1, six of them have a common vertex. The second solid, which is presented in Fig. 2, has four hexagons with a common vertex. In the third case (Fig. 3), six hexagons meet in the common vertex.

All regular polyhedrons, presented here, can be applied in a wild verity of engineering applications as parallel, cross and counter flow heat exchangers. Furthermore, it is possible to design short term thermal energy storage (TES) unites based on the referred solids. In this variant of construction, phase change material (PCM) should fill out one of labyrinths.

The available literature review shows that there are no reports of selected geometry among the heat transfer applications. Only some projects are loosely related to investigated heat exchanger.

Lu used analytical models to evaluate the efficiency of microcell aluminum honeycombs heat exchanger. The fluid flow in a honeycomb core was laminar. In Ref. [1] author reported that the optimal cell morphology is not constant but depend upon the geometry and heat transfer condition. It was concluded that the optimal relative density of the honeycombs is about 0.1. Construction geometry tested by Lu is illustrated in Fig. 4.

In Ref. [2] laminar and fully developed flow trough single- and double-trapezoidal ducts using a finite-difference method were only theoretically analyzed by Sadasivam, Manglik and Jog. Fig. 5 shows the geometry of investigated heat exchanger. Simulations of velocity and temperature fields were obtained for a wide range of duct aspect ratios and with four different trapezoidal angles. The influence of a duct aspect ratio on *Re* number and *Nu* number was presented in the form of polynomials. A strong dependence of thermal and pressure drop characteristics on duct geometry was reported.

Yeh et al. [3] experimentally investigated thermal contact resistance of the aluminum honeycomb sandwiched by two aluminum blocks. The example of a tested structure is showed in Fig. 6. It was experimentally observed that effective thermal conductivity in the axial direction of honeycombs is larger than that in the lateral direction and an axial total conductance of honeycombs increases with a decrease of cell size and specimen height.

The friction factor and thermal performance of a heat exchanger having hexagonal fins were tested by Yakut et al. [4]. The effects of hexagonal fins geometry (Fig. 7) and distance between fins on the thermal and pressure drop characteristics were experimentally investigated.

2. Description of the tested heat exchanger

The investigated prototype of the heat exchanger was consisted of twenty identical elements. Each component part, shown in Fig. 8, was composed by eight regular hexagons connected at an angle equal to 109°28′16″ with 20 mm side length. Fig. 9 presents

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Nomenclature

$\dot{C}$	flow-stream capacity rate (W/K)
c_p	fluid specific heat at constant pressure (J/(kg K)
D_H	hydraulic diameter (m)
f_F	Fanning friction factor
F	heat transfer surface area (m^2)
$F(z)$	experimental parameters
$\dot{G}$	mass flow rate (kg/s)
h	convective heat transfer coefficient (W/(m^2 K))
k	thermal conductivity (W/(m K))
L	heat exchanger length (m)
Nu	Nusselt number
P	pressure (Pa)
$\dot{Q}$	heat rate (W)
Re	Reynolds number
U	overall heat transfer coefficient (W/(m^2 K))
v	air velocity (m/s)
$\dot{V}$	volume flow rate (m^3/s)

Greek symbols

Δ	difference between pressure or temperature (Pa or K)
δ	thickness of the heat exchanger separate wall (m)
θ	fluid temperature (K)
ρ	density (kg/m^3)
$\sigma(x)$	uncertainty of variables

Subscripts

a	air
b	brass
C	cold fluid
H	hot fluid
in	inlet boundary
m	measurements
out	outlet boundary

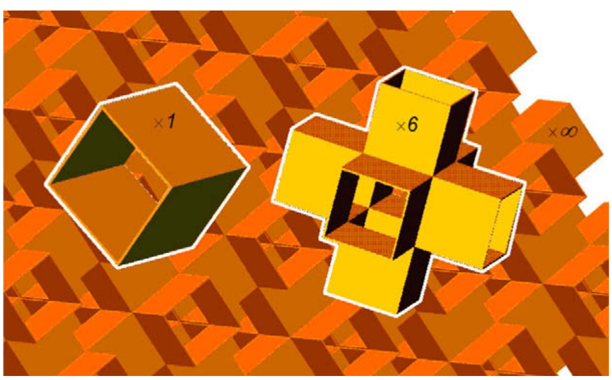

Fig. 1. The first infinite solid.

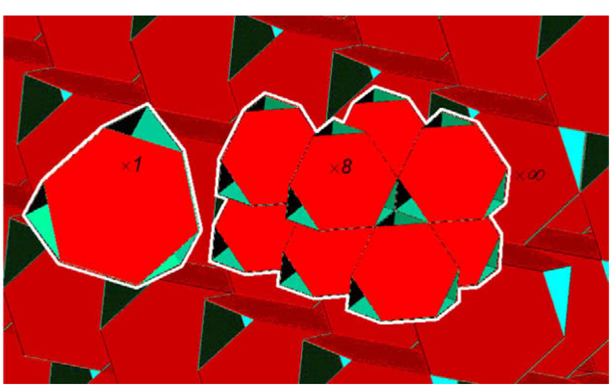

Fig. 2. The second infinite solid.

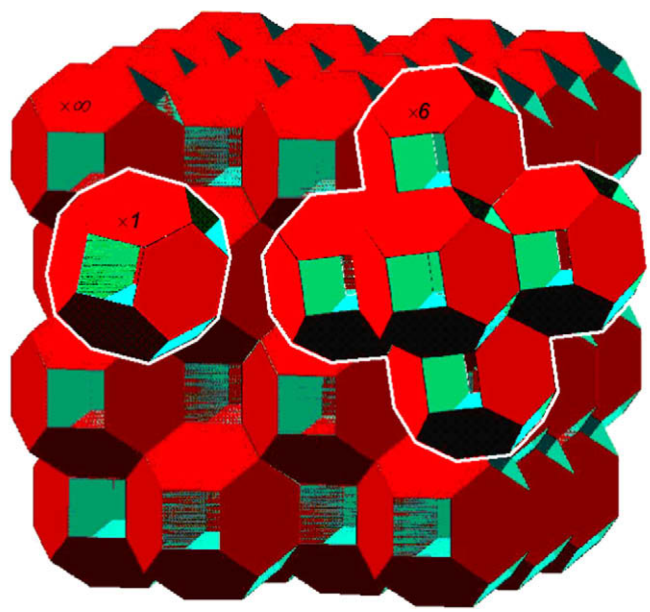

Fig. 3. The third infinite solid.

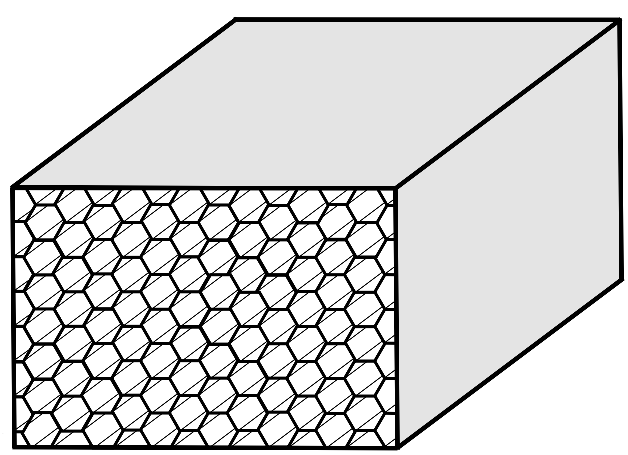

Fig. 4. Heat transfer geometry analyzed in Ref. [1].

the complete heat exchanger which was made of 0.2 mm thickness sheet brass. Before the experimental test the device was insulated with 50 mm thickness polyethylene foam sheets for reducing heat losses from the casing.

Depends on modification inlets and outlets of the base module, the heat exchanger can operates on the cross-, counter-, parallel- or any mixed-flow principles. This is one of the main advantages of the developed construction.