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S e c t i o n 2
HEAT AND MASS TRANSFER

PERFORMANCE EVALUATION OF LAMINAR FLUIDLY DEVELOPED FLOW THROUGH TRAPEZOIDAL AND HEXAGONAL DUCTS SUBJECTED TO H1 BOUNDARY CONDITION. PART 1

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Abstract
Extended performance evaluation criteria (ExPEC) have been used to assess the performance characteristics of single-phase fully developed laminar flow through bundle of tubes with trapezoidal and hexagonal ducts. The bundle with circular tubes has been used as a reference heat transfer unit. The H1 boundary condition has been selected as thermal boundary condition. The performance characteristics of the heat unit with non-circular tubes have been evaluated and compared to those of the reference unit for different objectives and constraints imposed. As a common constraint, the hydraulic diameter of the non-circular duct has been specified. The results showed that only in the case VG-2a the benefit can be obtained for the values of the irreversibility ratio ϕ_{o} □ 1.

Keywords: performance evaluation criteria, single-phase laminar flow, trapezoidal and hexagonal ducts, entropy generation

Nomenclature

- A heat transfer surface area $[\mathrm{m}^2]$
- c_p specific heat capacity $[\mathrm{J\,kg^{-1}K^{-1}}]$
- D reference circular tube diameter [m]
- D_h hydraulic diameter [m]
- h heat transfer coefficient $[\mathrm{W\,m^{-2}\,K^{1}}]$
- k thermal conductivity $[\mathrm{W\,m^{-1}\,K^{-1}}]$
- L tube length [m]
- $\dot{m}$ mass flow rate in tube $[\mathrm{kg\,s^{-1}}]$
- N_t number of tubes
- P pumping power [W]
- p wetted perimeter [m]
- Δp pressure drop [Pa]
- $\dot{Q}$ heat transfer rate [W]
- $\dot{S}_{gen}$ entropy generation rate $[\mathrm{W\,K^{-1}}]$
- T temperature [K]
- ΔT wall-to-fluid temperature difference [K]
- V volume of tubes $[\mathrm{m}^3]$
- W mass flow rate in heat exchanger $[\mathrm{kg\,s^{-1}}]$
- x axial distance along the tube [m]

Greek symbols

- ϑ temperature difference, $T_w - T$
- μ dynamic viscosity $[\mathrm{Pa\,s}]$
- ρ fluid density $[\mathrm{kg\,m^{-3}}]$

Dimensionless groups

- A_* dimensionless heat transfer surface, $A_w / A_{w,c}$
- D_* dimensionless tube diameter, D_h / D_c
- L_* dimensionless tube length, L / L_c
- f Fanning friction factor
- f_* Fanning friction factor ratio, f / f_c
- Nu Nusselt number
- Nu_* Nusselt number ratio, Nu / Nu_c
- N_S augmentation entropy generation number
- N_* ratio of number of tubes, $N_t / N_{t,c}$
- NTU heat transfer units, $4St\,L/D$
- Pr Prandtl number
- P_* dimensionless pumping power, P / P_c
- Q_* dimensionless heat transfer rate, $\dot{Q} / \dot{Q}_c$
- Re Reynolds number
- Re_* Reynolds number ratio, $\mathrm{Re}/\mathrm{Re}_c$
- St Stanton number
- ΔT_i^* dimensionless inlet temperature difference, $\Delta T_i / \Delta T_{i,c}$

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V_* volume ratio, V/V_c
W_* dimensionless mass flow rate, W/W_c
χ shape factor, p/D_h
χ_* ratio of shape factors, χ/χ_c
ε_* ratio of heat exchanger effectiveness, $\varepsilon/\varepsilon_c$
τ dimensionless temperature difference, $\Delta T/T$
ϕ_o irreversibility distribution ratio

Subscripts
c circular tube
f fluid
i value at x = 0
m mean
o value at x = L
w wall

1. Introduction

There has been considerable work on laminar forced-convective heat transfer in non-circular ducts reported in the literature. Shah and London [1], and Shah and Bhatti [2] give extended reviews of a large number of these studies. In the more recent literature, several different flow cross-section geometries for newer compact heat exchanger applications have been studied. They include double-sine [3], circular segment [4], semi-circular [5] and several other unusual duct shapes.

Duct geometries as single- and double-trapezoidal (hexagonal) represent flow channels of a variety of compact heat exchangers. The double-trapezoidal duct shape is encountered in lamella type compact heat exchanger, which find extensive usage in chemical industry [6-8]. Plate heat exchangers are also used in a wide range of applications including food and chemical processing, refrigeration, and waste-heat recovery [9]. The single-trapezoidal channel is employed in plate-fin heat exchangers [8], and micro-channel electronic cooling modules [10].

Due to smaller system dimensions, the hydraulic diameter of flow channels in such heat exchangers are small and the length-to-diameter ratio, L/D_h is relatively large. Due to these length scales and the viscous nature of the fluids being handled, the flow is usually laminar with fully developed conditions. It is therefore important to investigate the performance characteristics of different ducted flows, particular in the laminar regime.

On the basis of the first-law analysis Webb [11] and Webb and Bergles [12] have proposed performance evaluation criteria (PEC) that define the performance benefits of an exchanger having augmented surfaces, relative to standard exchanger with smooth surfaces subject to various objectives and design constraints. A thermodynamic basis to evaluate the merit of augmentation techniques by second-law analysis has been proposed by Bejan [13,14] who developed the entropy generation minimization (EGM) method.

This method has been used as a general criterion for estimating and minimizing the irreversibilities and optimum-design method for heat exchangers. The coupling between fluid flow and heat transfer irreversibilities suggested that the geometry and operating conditions can be optimized to minimize the overall entropy generation. The method has been extended by Zimparov [15,16] including the effect of fluid temperature variation along the length of a tubular heat transfer unit, and new information has been added assessing two objectives simultaneously. The EGM method combined with the first law analysis provides the most powerful tool for the analysis of the thermal performance of any augmentation technique.

For any duct with non-circular shape, the size is determined by either the hydraulic diameter D_h, or the cross-sectional area A_f, since these parameters are related through the shape factor $\chi = 4A_f/D_h^2$. In this regard, two different common constraints can be imposed – specified cross-sectional area $A_f^* = 1$, or specified hydraulic diameter of the ducts, $D_* = 1$. Performance evaluation of laminar fully-developed flow in ducts with non-circular shapes subjected to H1 boundary condition and common constraint $A_f^* = 1$ have been recently presented in [17,18].

The rationale of the present study is to evaluate the thermal performance of laminar fully-developed flow in a bundle with trapezoidal and hexagonal ducts. Figure 1 presents the geometrical details of trapezoidal and double trapezoidal (hexagonal) duct [19]. The boundary condition is H1 (constant wall heat flux) with a common constraint, $D_* = 1$. In this case, the cross- sectional area of the duct is a consequence, $A_f^* = \chi_*$. The bundle of circular tubes has been used as a reference heat transfer unit. Using the first and second laws simultaneously, the performance characteristics of units with non-circular ducts have been evaluated for different objectives and constraints imposed and compared to those of the reference unit with circular tubes.

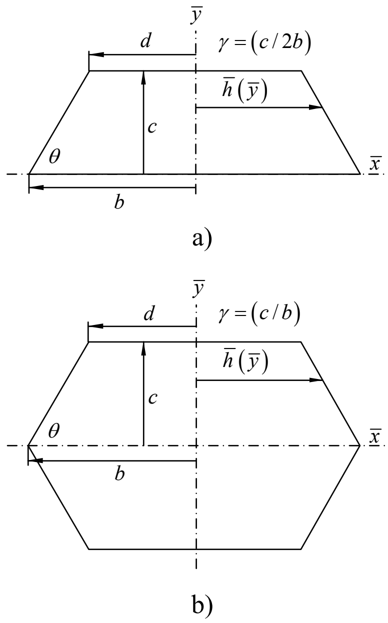

Fig. 1. Coordinate system and geometrical details of: a) trapezoidal duct; b) hexagonal duct

2. Equations based on the entropy production theorem

Consider the energy and entropy balance of the general internal flow configuration, where fluid flows through a duct with a cross sectional area A_f, a perimeter p, and hydraulic diameter $D_h = 4A_f/p$. The shape of the cross section is arbitrary but constant over the entire length of the duct. The flow is single-phase, fully developed, incompressible, and Newtonian. Following the model developed by Zimparov [16], for fully-developed laminar flow in a tube bundle, the rate of entropy generated in the heat unit can be expressed as:

$$\dot{S}_{gen} = \frac{\dot{Q}^2}{N_t^2 k_f Nu \chi L_t T_i T_o} + \frac{8W^2 (f\,\mathrm{Re})\mu L_t}{N_t^2 \rho^2 \chi D_h^4 T_i}, \qquad (1)$$

where: $\dot{Q} = N_t \dot{Q}_t$, $\dot{Q}_t = \dot{m}_t c_p (T_o - T_i)$, $W = \dot{m}_t N_t$, $A = pLN_t$ $T_o = T_i + \frac{hp\Delta T}{\dot{m}_t c_p} L$. (The heated and wetted perimeters are assumed to be the same.) Following Bejan [13,14], the augmentation entropy generation number N_S can be presented as

$$N_S = \frac{\dot{S}_{gen}}{\dot{S}_{gen,circle}} = \frac{N_{S,T} + \phi_o N_{S,P}}{1+\phi_o} = \frac{1}{1+\phi_o}\left(N_{S,T} + \phi_o N_{S,P}\right), \qquad (2)$$

where [15]

$$N_{S,T} = \frac{Q_*^2}{Nu_* N_*^2 \chi_* L_* T_o^*}, \qquad (3a)$$

$$T_o^* = \left[\frac{T_{i,c}}{T_{o,c}} + \frac{Q_*}{W_*}\left(1 - \frac{T_{i,c}}{T_{o,c}}\right)\right], \qquad (3b)$$

$$N_{S,P} = \frac{W_*^2 L_* (f\,\mathrm{Re})_*}{N_*^2 \chi_* D_*^4} \equiv P_*, \qquad (3c)$$

$$\phi_o = \frac{8W_c^2 \mu L_c^2 (f\,\mathrm{Re})_c}{\rho^2 D_c^4} \times \frac{k_f Nu_c T_{o,c}}{\dot{Q}_c^2}. \qquad (3d)$$

The PEC as suggested by Webb [11] and Webb and Bergles [12] characterize nearly all the PEC and some of them will be considered below. The equations are developed for ducts with different cross sectional shape and the equations (in dimensionless form) for inside single-phase laminar flow are, as follows:

$$A_* = p_* L_* N_* = \chi_* D_* N_* L_*, \qquad (4)$$

$$P_* = \frac{W_*^2 L_* (f\,\mathrm{Re})_*}{N_*^2 \chi_* D_*^4}, \qquad (5)$$

$$Q_* = W_* \varepsilon_* \Delta T_i^*, \qquad (6)$$

$$W_* = \mathrm{Re}_* \, p_* N_* = \mathrm{Re}_* \, \chi_* D_* N_*, \qquad (7)$$

where $\chi_* = p_* / D_*$.

3. Performance evaluation criteria

The performance evaluation criteria, as suggested by Webb [11], Webb and Bergles [12] and extended by Zimparov [15] have been considered in this study. These criteria are based on the use of first and second law analyses in the pursuit of two objectives simultaneously. In this study the geometrical and regime parameters of the reference channel (smooth circular tube) are selected to fulfill the requirement of $4L_c/(D_c \mathrm{Re}\,\mathrm{Pr}) = 1$, corresponding to the fully-developed laminar flow in tube.

The values of the shape factor χ, friction factor f and Nusselt number Nu of trapezoidal and hexagonal ducts are taken and calculated from Sadasivam et al. [19]. While obtaining the augmentation entropy generation number, the irreversibility distribution ratio for the circular configuration, ϕ_o, varies in the range $10^{-3} \le \phi_o \le 10^3$.

3.1. Fixed geometry criteria (FG)

These criteria involve a replacement of circular tubes by tubes with non-circular shape of equal length and cross sectional area. The FG-1 cases seek increased heat duty for constant exchanger flow rate and heat transfer area. The FG-2 criteria have the same objective as FG-1, but requires that the non-circular tube design to operate at the same pumping power as the reference circular tube design. The third criterion, FG-3, attempts to effect reduced pumping power for constant heat duty and surface area.

3.1.1 Case FG-1a

The objective functions of the case FG-1a are increased heat rate $Q_* > 1$, decreased entropy generation number $N_S < 1$, and simultaneous effect of the both of them as a general performance criterion $N_S / Q_* < 1$. The constraints imposed are: $W_* = 1$, $\Delta T_i^* = 1$, $A_* = 1$, $L_* = 1$, $D_* = 1$. The consequences of these constraints are $\mathrm{Re}_* = 1$, $P_* > 1$, and $N_* < 1$. The constraint of equal heat transfer surface area, $A_* = 1$, requires $N_* \chi_* = 1$ or $N_* = \chi_*^{-1}$. In this case the total bundle cross-sectional area and the volume ratio are $A_{f,tot}^* = \chi_* N_* D_*^2 = 1$, and $V_* = L_* A_{f,tot}^* = 1$. The constraint of equal mass flow rate $W_* = \mathrm{Re}_* D_* N_* \chi_* = 1$, requires $\mathrm{Re}_* = 1$. The Eqs. (5) and (6) yield

$$P_* = \chi_* (f\,\mathrm{Re})_*, \qquad (8)$$

$$Q_* = \varepsilon_*. \qquad (9)$$

where

$$\varepsilon_* = 1.229 \frac{Nu}{1+Nu}. \qquad (10)$$

The Eqs. (3a-3c) yield

$$T_o^* = 0.981 + 0.019 \varepsilon_*, \qquad (11)$$

$$N_{S,T} = \frac{\chi_* Q_*^2}{Nu_* T_o^*}, \qquad (12)$$

$$N_{S,P} \equiv P_* = \chi_* (f\,\mathrm{Re})_*, \qquad (13)$$

and the augmentation entropy generation number N_S becomes

$$N_S = \frac{1}{1+\phi_o}\left(\frac{\chi_* Q_*^2}{Nu_* T_o^*} + \phi_o \chi_* (f\,\mathrm{Re})_*\right). \qquad (14)$$

The values of Q_* are calculated by Eq. (9), and the variation of Q_* with χ_* and θ are shown in Fig. 1. As seen, the first objective $Q_* > 1$ can be achieved for $\chi_* > 2$.

This benefit increases with the increase of θ and χ_* but it cannot exceed 8%. The bundle with hexagonal ducts performs a little better than the bundle with trapezoidal ducts and the benefit increases slightly with the increase of θ.

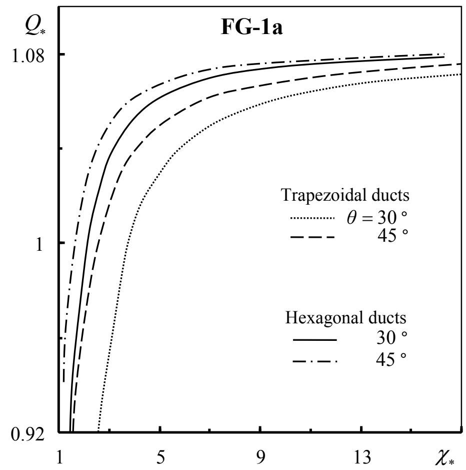

Fig. 1a. The variation of Q_* with χ_* and θ.

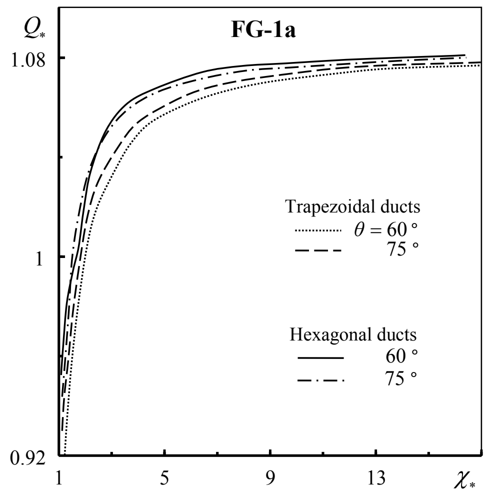

Fig. 1b. The variation of Q_* with χ_* and θ.

Fig. 2 shows how the second criterion N_S/Q_* varies with χ_*, ϕ_o and θ. As seen, for all values of the variables χ_*, ϕ_o and θ, the second objective $N_S/Q_* < 1$ cannot be achieved and any benefit cannot be obtained. That means, that despite some positive results of the first objective $Q_* > 1$, it can be concluded that for the case FG-1a the use of bundle of standard ducts with circular shape is more thermodynamically efficient that the bundle of ducts with trapezoidal or hexagonal ducts. The reason is the great increase of the entropy generated in the process.

3.1.2. Case FG-2a

The objective functions of the case FG-2a are increased heat duty $Q_* > 1$, decreased entropy generation numbers $N_S < 1$, and simultaneous effect of the both of them $N_S/Q_* < 1$. The constraints imposed are: $\Delta T_i^* = 1$, $P_* = 1$, $A_* = 1$, $L_* = 1$, $D_* = 1$. The consequences of these constraints are $W_* < 1$, $N_* < 1$. The constraint $A_* = 1$ requires $N_* = \chi_*^{-1}$, $A_{f,tot}^* = 1$, and $V_* = 1$.

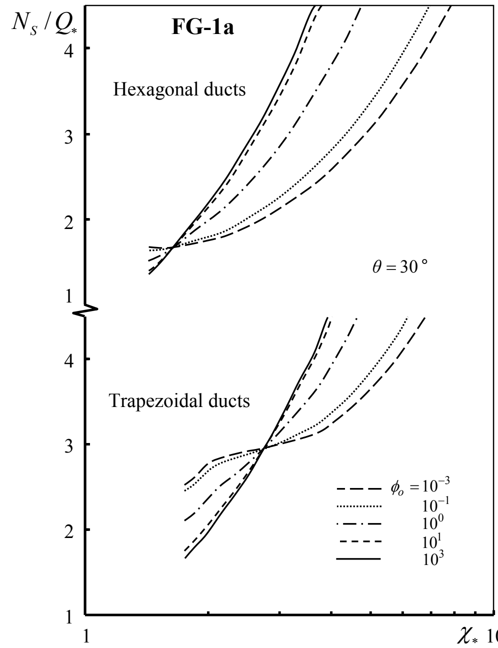

Fig. 2a. The variation of N_S/Q_* with χ_* ($\theta = 30^o$)

The constraint $P_* = 1$, requires

$$W_* = \mathrm{Re}_* = \chi_*^{-1/2} (f\,\mathrm{Re})_*^{-1/2}., \tag{15}$$

and Eqs. (6) and (12) yield

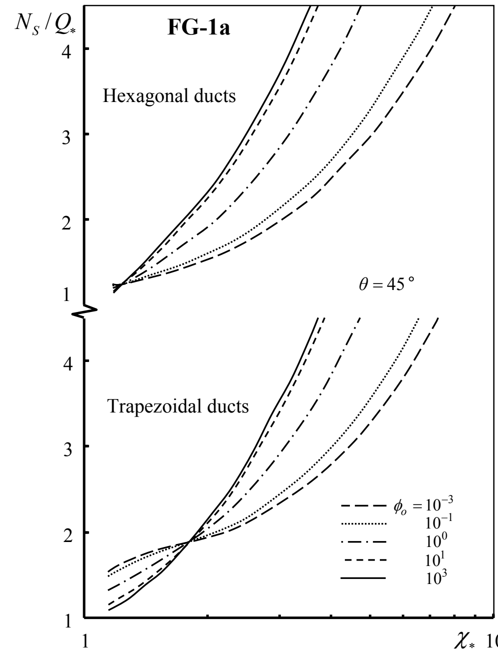

Fig. 2b. The variation of N_S/Q_* with χ_* ($\theta = 45^o$)

$$Q_* = \varepsilon_* W_*, \tag{16}$$

$$N_{S,T} = \frac{\chi_* Q_*^2}{Nu_* T_o^*}, \tag{17}$$

where ε_* and T_o^* are calculated by Eqs. (10) and (11).

The augmentation entropy generation number N_S becomes

$$N_S = \frac{1}{1+\phi_o}\left(\frac{\chi_* Q_*^2}{Nu_* T_o^*} + \phi_o\right). \qquad (18)$$

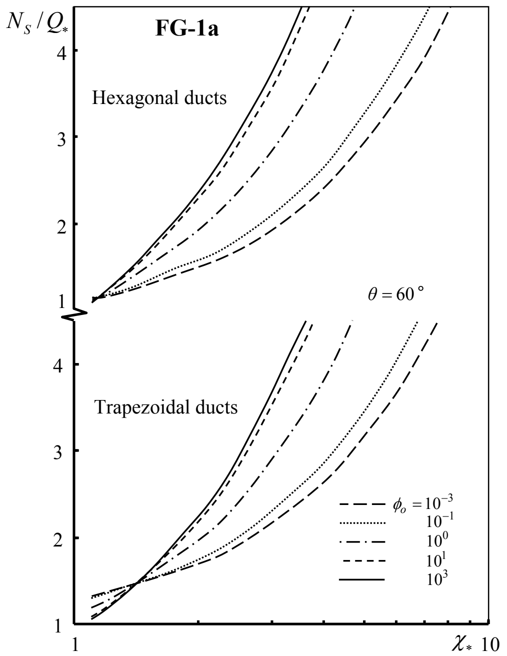

Fig. 2c. The variation of N_S / Q_* with χ_* ($\theta = 60^o$)

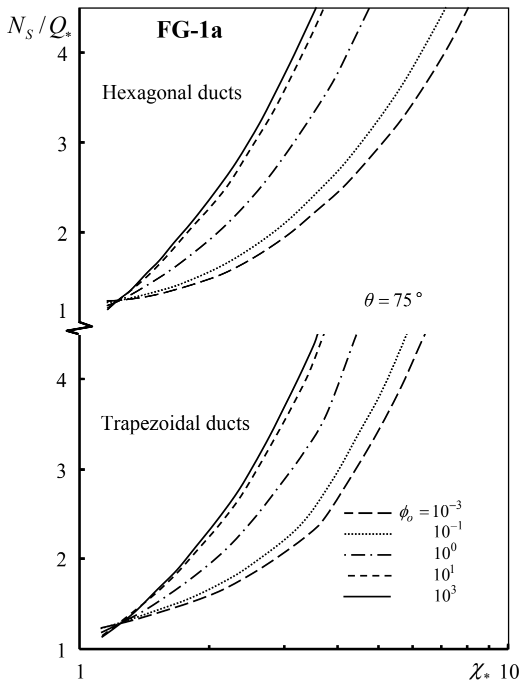

Fig. 2d. The variation of N_S / Q_* with χ_* ($\theta = 75^o$)

The values of Q_* calculated by Eq. (16) are shown in Fig. 3. As seen the first objective $Q_* > 1$ cannot be achieved of bundle neither with trapezoidal nor hexagonal ducts despite the values of χ_* or θ. Also, Q_* decreases rapidly with the increase of χ_* and does not depend on θ. The values of the second objective $N_S / Q_* < 1$ calculated by Eqs. (16) and (18) are presented in Fig. 4.

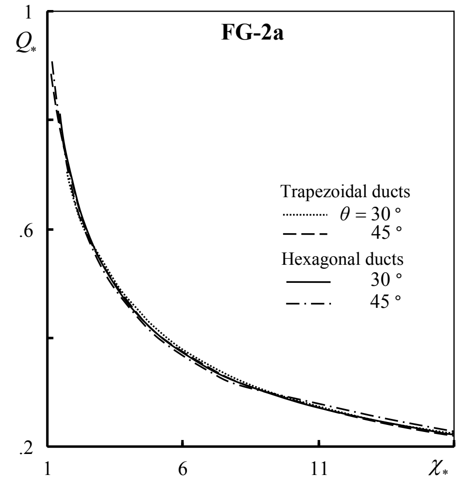

Fig. 3a. The variation of Q_* with χ_* and θ.

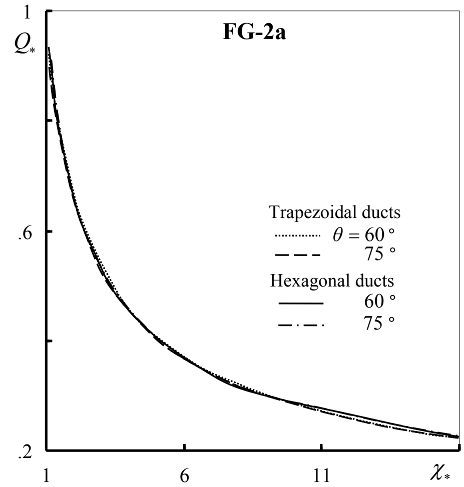

Fig. 3b. The variation of Q_* with χ_* and θ.

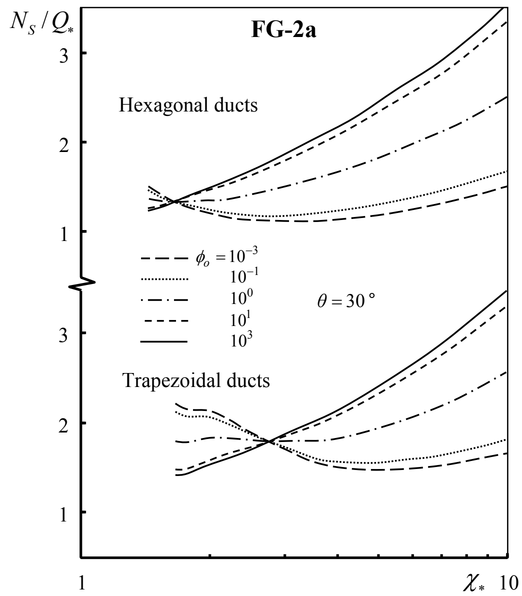

Fig. 4a. The variation of N_S / Q_* with χ_* ($\theta = 30^o$)

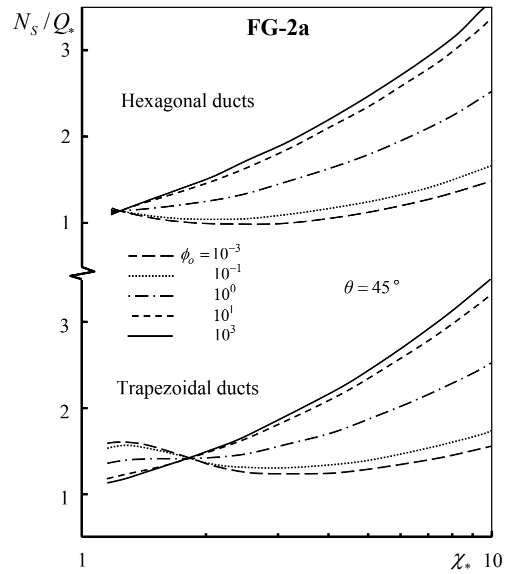

Fig. 4b. The variation of N_S/Q_* with χ_* ($\theta = 45^o$)

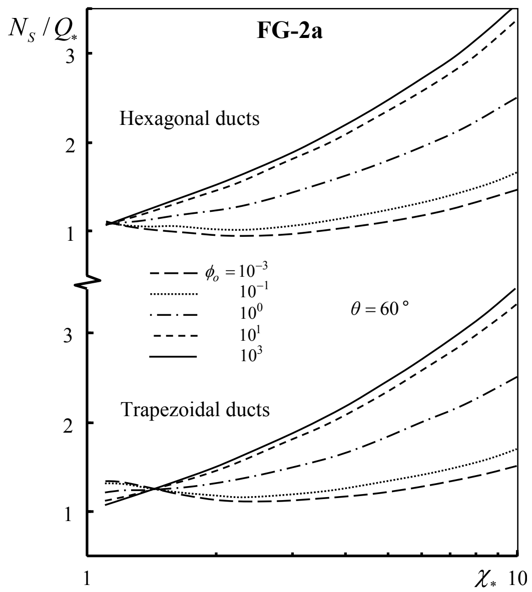

Fig. 4c. The variation of N_S/Q_* with χ_* ($\theta = 60^o$)

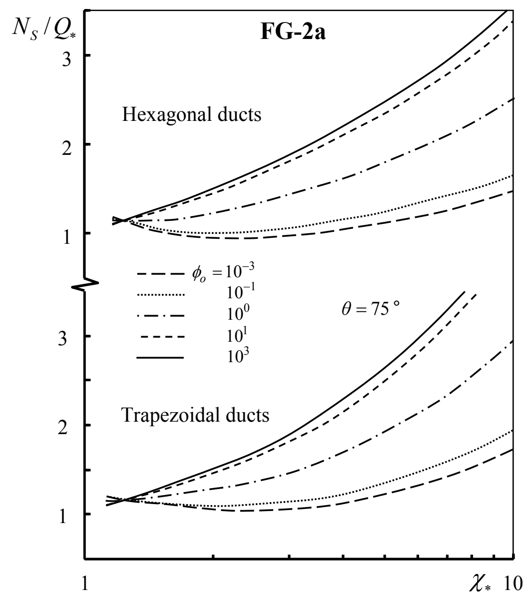

Fig. 4d. The variation of N_S/Q_* with χ_* ($\theta = 75^o$)

As seen, for the bundle with trapezoidal ducts for any values of χ_*, θ or ϕ_o the value of the second criterion is always $N_S/Q_* > 1$. For the bundle with hexagonal ducts this criterion can approach the value of unity with the increase of θ and very small value of ϕ_o ($\phi_o \ll 1$). This means that no benefits can be obtained through the replacement of the bundle with traditional circular ducts with bundle having ducts with no circular shape as trapezoidal or hexagonal.

3.2. Variable geometry criteria (VG)

The criteria VG are applicable when the heat exchanger is "sized" for a required thermal duty with specified flow rate.

3.2.1. Case VG-2a

The objective functions of the case VG-2a are increased heat duty $Q_* > 1$, decreased entropy generation number $N_S < 1$, and simultaneous effect of the both of them $N_S/Q_* < 1$. The constraints imposed are: $W_* = 1$, $\Delta T_i^* = 1$, $P_* = 1$, $A_* = 1$, $D_* = 1$. The consequences of the constraints are, $L_* < 1$, $N_* > 1$, $V_* = 1$. The Eqs. (2-7) yield:

$$L_* = \left[\chi_* (f\,\mathrm{Re})_*\right]^{-1/3}, \tag{19}$$

$$N_* = \chi_*^{-2/3} (f\,\mathrm{Re})_*^{1/3}, \tag{20}$$

$$Q_* = \varepsilon_*, \tag{21}$$

$T_o^* = 0.981 + 0.019\varepsilon_*$, where ε_* is calculated by Eq. (10). $N_{S,P} = P_* = 1$, whereas

$$N_{S,T} = \frac{Q_*^2 \chi_*^{2/3}}{Nu_* T_o^* (f\,\mathrm{Re})_*^{1/3}}. \tag{22}$$

The augmentation entropy generation number N_S becomes

$$N_S = \frac{1}{1+\phi_o}\left\{\frac{Q_*^2 \chi_*^{2/3}}{Nu_* T_o^* (f\,\mathrm{Re})_*^{1/3}} + \phi_o\right\}. \tag{23}$$

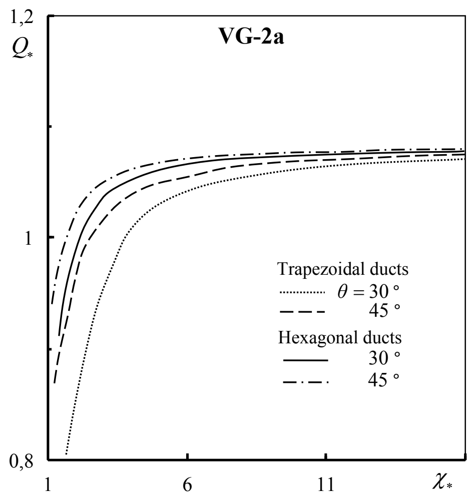

Fig. 5a. The variation of Q_* with χ_* and θ.

The values of Q_*, calculated by Eq. (21) are presented in Fig. 5, as the variation of Q_* with χ_*. The images are similar to those of the case FG-1a where the first objective $Q_* > 1$ can be achieved for $\chi_* > 2$. This benefit increases with the increase of θ and χ_* and reaches the values of the order of 8%. The bundle with hexagonal ducts also performs a little better than the bundle with trapezoidal ducts and the benefit increases slightly with the increase of θ.

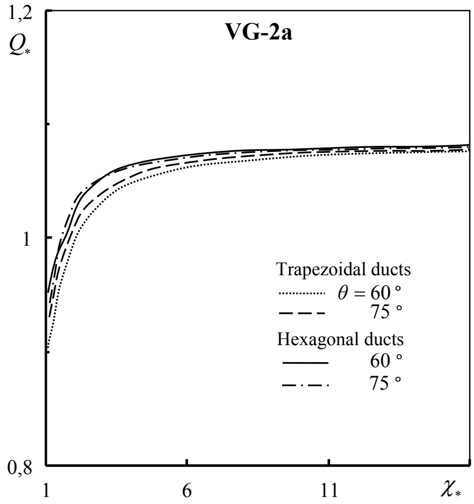

Fig. 5a. The variation of Q_* with χ_* and θ.

The main difference between cases FG-1a and VG-2a can clearly be seen in Fig. 6. In this case, the use of bundle with trapezoidal or hexagonal ducts can really bring about some benefits, since the second objective $N_S / Q_* < 1$, in some cases ($\phi_o \gg 1$) can be achieved. This benefit can be

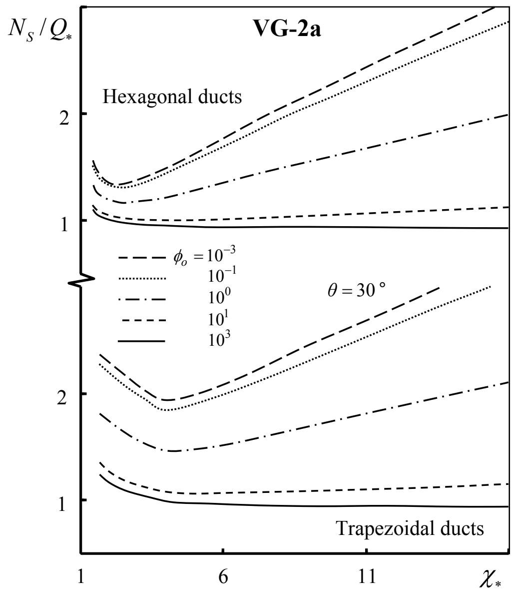

Fig. 6a. The variation of N_S / Q_* with χ_* ($\theta = 30^o$)

obtained for $\chi_* > 2$ and $\phi_o > 10^3$. The bundle with hexagonal ducts performs also a little better that the bundle with trapezoidal ducts and this benefit increases slightly with the increase of θ.

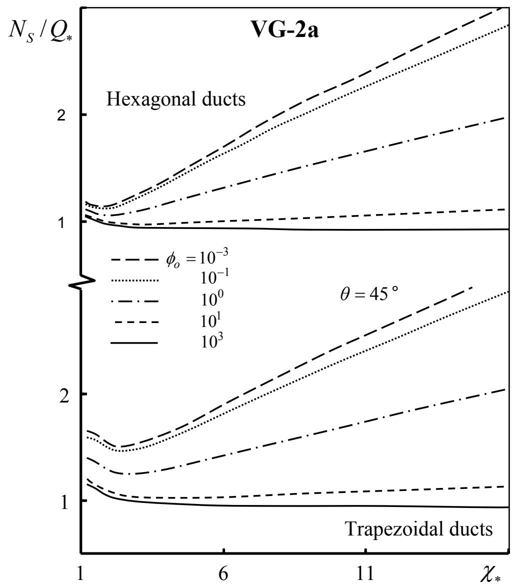

Fig. 6b. The variation of N_S / Q_* with χ_* ($\theta = 45^o$)

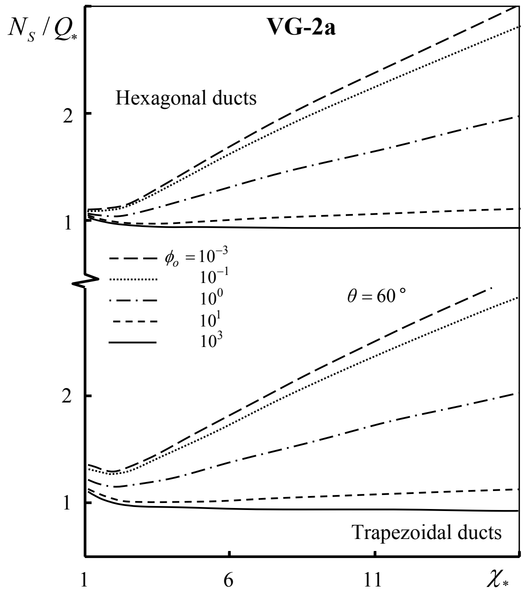

Fig. 6c. The variation of N_S / Q_* with χ_* ($\theta = 60^o$)

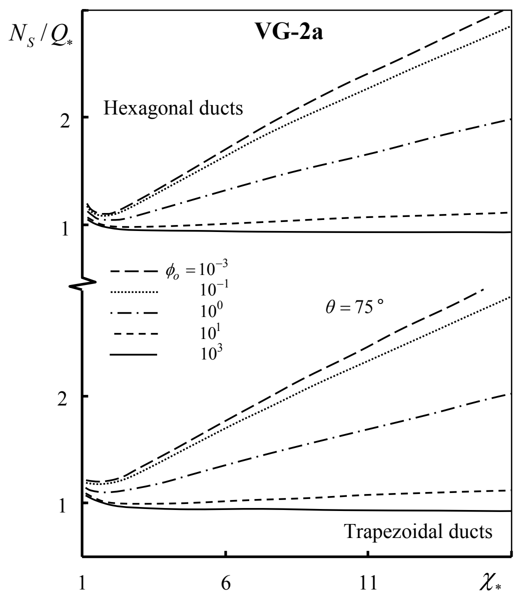

Fig. 6d. The variation of N_S / Q_* with χ_* ($\theta = 75^o$)

4. Conclusions

ExPEC have been used to assess the performance characteristics of single-phase fully developed laminar flow through bundle of tubes with trapezoidal and hexagonal ducts. The bundle with circular tubes has been used as a reference heat transfer unit. The H1 boundary condition has been selected as thermal boundary condition. The performance characteristics of the heat unit with non-circular tubes have been evaluated and compared to those of the reference unit for the cases FG-1a, FG-2a and VG-2a, where the first objective is $Q_* > 1$.

As a common constraint, the hydraulic diameter of the non-circular duct has been specified. The results showed that in the cases FG-1a and FG-2a the replacement of the conventional pipes with ducts with trapezoidal or hexagonal shapes are inefficient, and only in the case VG-2a the benefit can be obtained for the values of the irreversibility ratio ϕ_o □ 1.

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