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3-D Numerical Simulation of Scraped Surface Heat Exchanger with Helical Screw

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***Abstract*—**Surface scraping is a passive heat transfer enhancement technique that is directly used in scraped surface heat exchanger (SSHE). The scraping action prevents the accumulation of the product on the inner wall, which intensifies the heat transfer and avoids the formation of dead zones. SSHEs are widely used in industry for several applications such as crystallization, sterilization, freezing, gelatinization, and many other continuous processes. They are designed to deal with products that are viscous, sticky or that contain particulate matter. This research work presents a three-dimensional numerical simulation of the coupled thermal and hydrodynamic behavior within a SSHE which includes Archimedes' screw instead of scraper blades. The finite volume Fluent 15.0 was used to solve continuity, momentum and energy equations using multiple reference frame formulation. The process fluid investigated under this study is the pure glycerin. Different geometrical parameters were studied in the case of steady, non-isothermal, laminar flow. In particular, attention is focused on the effect of the conicity of the rotor and the pitch of Archimedes' screw on temperature and velocity distribution and heat transfer rate. Numerical investigations show that the increase of the number of turns in the screw from five to seven turns leads to amelioration of heat transfer coefficient, and the increase of the conicity of the rotor from 0.1 to 0.15 leads to an increase in the rate of heat transfer. Further studies should investigate the effect of different operating parameters (axial and rotational Reynolds number) on the hydrodynamic and thermal behavior of the SSHE.

***Keywords*—**ANSYS-Fluent, hydrodynamic behavior, SSHE, thermal behavior.

I. Introduction

SCRAPED surface heat exchangers (SSHEs) are commonly used in the food, chemical, and pharmaceutical industries. They are ideally suited for products that are viscous, sticky, or that contain particulate matter.

During operation, the product is brought in contact with a heat transfer surface that is rapidly and continuously scraped, thereby exposing the surface to the passage of untreated product.

The product film continually scraped from the heat transfer wall induces high heat transfer rates. SSHEs process viscous products far more efficiently than conventional plate or tubular heat exchangers.

The deposition of solid at the exchanger wall a phenomenon well known as fouling problem leads to a decrease the heat transfer coefficient. Fouling problem is an undesired phenomenon from thermal viewpoint and from nutritional view point because it can reduce the quality and the safety of the treated product. For sticky and viscous foods such as heavy salad dressings, margarine, chocolate, peanut butter, fondant, and ice cream, heat exchange is only possible by using SSHEs that can ensure a uniform and continuous production. Increasing the thermal efficiency of a SSHE is a concern of many researches. Works have been focused especially on the effect of different operator and geometrical parameters on hydrodynamic and thermal behavior of SSHE.

The heat transfer coefficient increased greatly when using scrapers in vessels instead of stirrers for viscous fluid [1]. Hence, the scraping action requires a higher power input rate.

Researches [2], [3] studied the heat transfer in a SSHE based on a mathematical model supported by the penetration theory which is based on transient conduction into a semi-infinite solid. The heat convection of fluid is neglected, and a stagnant film at the heat transfer wall is assumed that immediately and completely mixes with the bulk after a scraper action. The distribution of temperature before and after a scraped action is presented in Fig. 1.

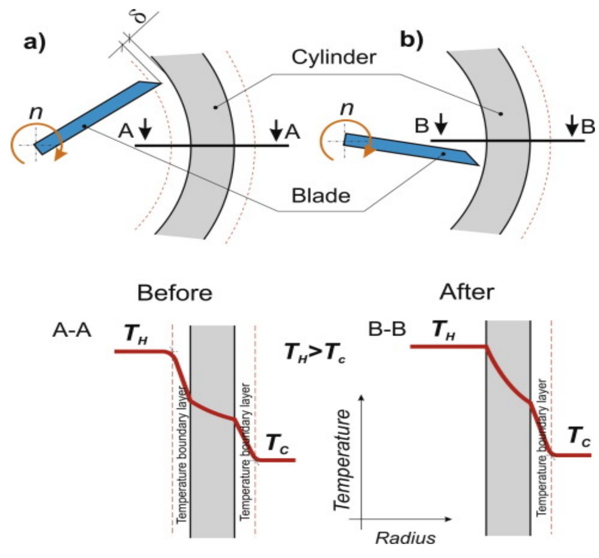

Fig. 1 Distribution of temperature before and after a scraping action [4]

The empirical model estimating the heat transfer coefficient based on the penetration theory is presented as follows [5]:

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$$Nu = \frac{2}{\sqrt{\pi}} Re^{0.5} Pr^{0.5} nb^{0.5} \quad (1)$$

Several limitations of this model included heat transfer independence of viscosity and velocity of liquid through the exchanger. Another limitation of the penetration theory is the over prediction of heat transfer for high viscous fluid due to the assumption of the complete mixing. In relation to this, Harriott [3] reported that the value of heat transfer predicted for water and fluids with moderate viscosity by (1) is optimistic. However, for highly viscous fluids, the heat transfer coefficients would be less than those predicted by the penetration theory due to incomplete mixing of the fluid in annular space with the fluid scraped from the wall.

For low values of rotational Reynolds number, the actual value of heat transfer would be lower than that predicted by the theoretical formula [5]. However, for high value of rotational Reynolds number, as a result of turbulence flow, the opposite tendency would be expected.

A correction factor has been introduced to the expression [5] in order to predict real SSHE behavior [6]. The correction factor accounts for the incompleteness of temperature equalization in the boundary layer, the effect of radial dispersion and the decrease in driving force for heat transfer due to axial dispersion.

The effect of the back-mixing in the inlet bowl is characterized by a sharp temperature increase (if heating) or a sharp temperature decrease (if cooling) of the product when it enters the exchanger (in the inlet bowl) and before it flows along the exchange surface itself [7]. The back mixing could be ignored at high Peclet numbers and low Stanton numbers [8].

A 3D CFD study was developed in order to analyze the hydrodynamic and thermal behavior of a SSHE operating in turbulent regime by varying geometrical (effect of number of blades) and operating parameters (effect of rotational and axial Reynolds number) [9]. As a result, an empirical model was developed in order to estimate the overall heat transfer.

A heat correlation model was established numerically by using a 3D-CFD code FLUENT for Newtonian and non-Newtonian shear thinning fluids [10].

In order to characterize the hydrodynamic and thermal behavior of SSHE equipped with helical ribbon, an empirical correlation that predicts the heat transfer coefficient was proposed [11].

The innovation in this work is to study the hydrodynamic and thermal behavior of a SSHE which, instead of using scraper blades or helical ribbon, includes an Archimedes' screw that can provide simultaneous mixing and agitation, maintains a good and uniform heat exchange, and reduces the back-mixing phenomenon.

II. Governing Equations

The physical assumptions used during our investigation are:

- Non isothermal, laminar flow.
- Newtonian, incompressible fluid.
- Constant speed of rotation of the agitator.
- Rotating reference frame formula is used to render our problem which is unsteady in the stationary (inertial) frame steady with respect to the moving frame.

The continuity equation which expresses the principle of the mass conservation in a flow is written as:

$$\mathrm{div}(\rho \vec{V}) = 0 \quad (2)$$

The momentum equation is written as follows:

$$\rho\left[\frac{\partial\vec{V}}{\partial t} + \overrightarrow{\overrightarrow{\mathrm{grad}}}\vec{V}.\vec{V} + \vec{\omega}\wedge(\vec{\omega}\wedge\overrightarrow{OM}) + 2\,\vec{\omega}\wedge\vec{V}\right] = -\overrightarrow{\mathrm{grad}}\,\mathbf{P} + \mathrm{div}\,\vec{\vec{\tau}} + \rho\,\vec{g} \quad (3)$$

The momentum equation contains two additional acceleration terms: the Coriolis acceleration ($2\,\vec{\omega}\wedge\vec{V}$) and the centripetal acceleration $\left(\vec{\omega}\wedge(\vec{\omega}\wedge\overrightarrow{OM})\right)$.

The thermal energy equation is written as follows:

$$\frac{\partial T}{\partial t} + \mathrm{div}\left(\vec{V}\ T - \frac{2}{\pi}\left(\frac{d}{D}\right)^2 \frac{1}{Re_r}\frac{1}{Pr}\ \overrightarrow{\mathrm{grad}}\ T\right) = 0 \quad (4)$$

III. Computational Domain

The buoyancy ratio of the SSHE considered in this work is the same as that studied experimentally by Trommelen and Beek [12] and numerically by Ali and Baccar [11]. Hence, the ratio of rotor shaft radius to internal exchanger radius is equal to 0.6 and the ratio of the exchanger length to its internal radius is equal to 6. However, in the present work, we consider a SSHE equipped with helical screw.

The different geometries were designed with SOLIDWORKS 3D CAD tool. The geometrical variable chosen for the numerical simulation study are the screw pitch (geometry 1 and 2 presented respectively in Figs. 2 and 3) and the conicity of the rotor (geometry 3 and 4 presented respectively in Figs. 4 and 5). Details of screws design are resumed in Table I.

TABLE I
Geometrical Characteristics of Screws

	Length H(m)	Number of turns	Rotor diameter dr(m)	Pitch p(m)	Conicity
Geometry 1	0.76	5	0.152	0.148	0
Geometry 2	0.76	7	0.152	0.106	0
Geometry 3	0.76	6	0.152	0.123	0.1
Geometry 4	0.76	6	0.152	0.123	0.15

Fig. 2 Geometry 1

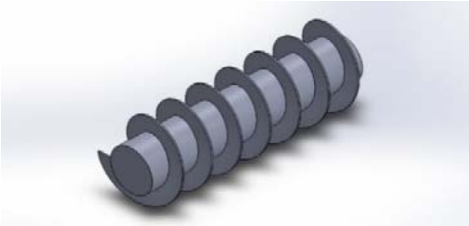

Fig. 3 Geometry 2

Fig. 4 Geometry 3

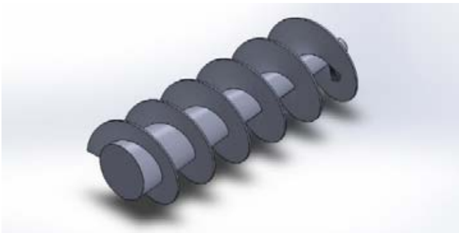

Fig. 5 Geometry 4

The stator diameter D was considered equal to 0.253 m. The clearance between the tip of the screw and the stator wall was taken equal to 0.001 m.

IV. Grid Structure

The different geometry was meshed using ANSYS R15.0, details about grid structure are resumed in Table II.

TABLE II
DETAILS OF GRID MESH

	Number of elements	Number of nodes
Geometry 1	1356059	281673
Geometry 2	5497800	1163147
Geometry 3	1837331	375838
Geometry 4	1844973	377264

V. Boundary Conditions

ANSYS-Fluent 15.0 code was used to solve continuity, momentum and energy equation in SSHE geometry using multiple reference frame formulation (MRF). The boundary conditions considered in this work are specified as follows:

- The no-slip condition was applied at the walls.
- Outflow: Zero velocity gradient was applied for the exchanger outlet.
- The velocity of the fluid at the inlet of the exchanger was taken equal to 0.317 m/s.
- The rotor is rotating at an angular velocity equal to 1.72 rad/s.
- The temperature of the fluid introduced at the SSHE was taken equal to 300 K
- The heating temperature at the external wall of the SSHE was assumed to be constant and equal to 700 K.
- Adiabatic condition was assumed for the rotor.

VI. Result and Discussion

A. The Effect of Number of Turns in the Screw

In this section, we studied the effect of number of turns in the screw (the effect of the pitch of the screw) on the hydrodynamic and thermal behavior of the SSHE.

Figs. 6 and 7 illustrate the distribution of velocity vectors in (r-z) plane respectively, for five and seven turns in the screw for an axial Reynolds number equal to 1 and radial Reynolds number equal to 10. Between two successive turns, we distinguish the formation of a rotating vortex structure. These vortex structures lead to improve the mixing of the scraped fluid with the bulk fluid. Moreover, the fluid flowing in the annular space between the external screw edge and the heated wall is relatively intensive. Hence, this axial flow rate improves the removal of the thermal boundary layer.

We distinguish also that the recirculation zones between two successive turns constituting the screw are more extended and more intensive for the case of seven turns. In fact, the mixing of heated fluid after a scraping action, contributes to a substantial increase of the local driving force for heat transfer, corresponding to temperature difference between the heated wall and the bulk fluid which leads to increasing the heat transfer rate. This result is confirmed by the higher surface heat transfer along the SSHE illustrated in Fig. 11 for the case of seven turns compared with the heat transfer illustrated in Fig. 10 which corresponds to the case of five turns. In fact, we can distinguish an amelioration of heat transfer for the geometry with high number of turns due to increasing the scraping length effect.

Figs. 8 and 9 represent the distribution of temperature in (r-z) plane respectively, for five and seven turns screw. We can distinguish that the highest temperatures are localized at the vicinity of the heated wall. Outside the thermal boundary layer, we notice a uniform distribution of temperature. This is due to the presence of vortices which lead to the homogenization of fluid.

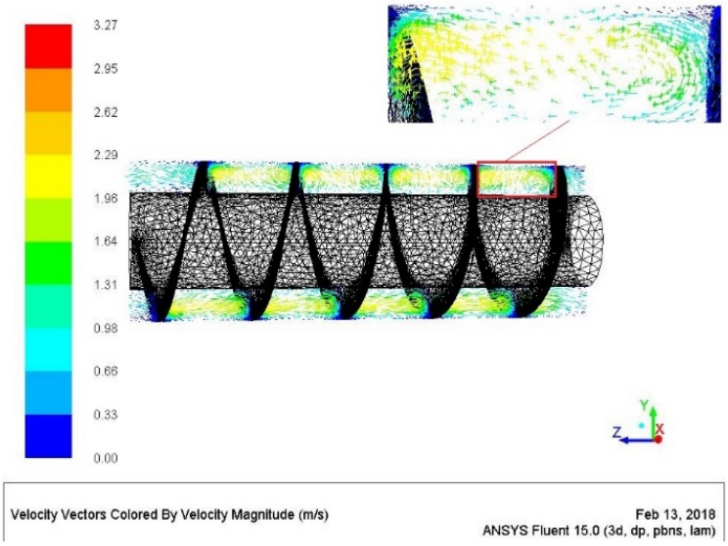

Fig. 6 Velocity vectors in the r-z plane (geometry 1)

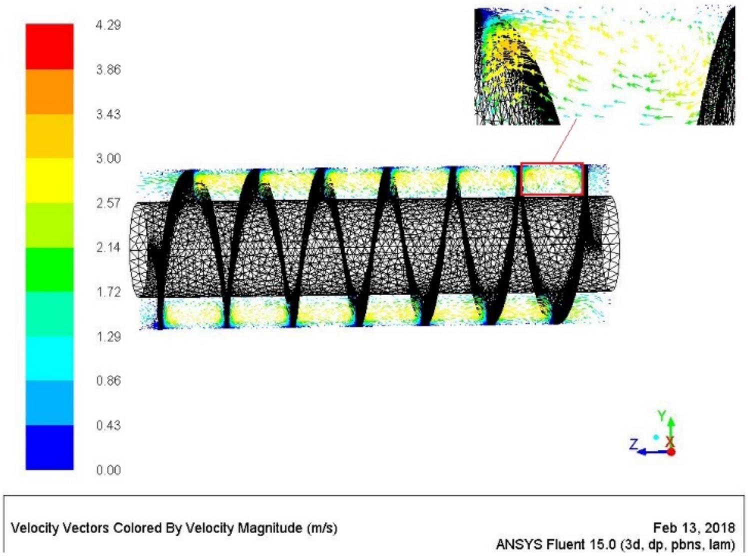

Fig. 7 Velocity vectors in the r-z plane (geometry 2)

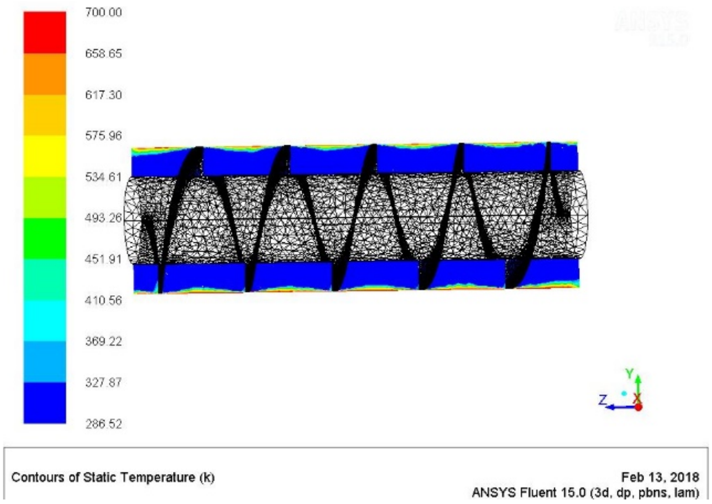

Fig. 8 Temperature profile in the r-z plane (geometry 1)

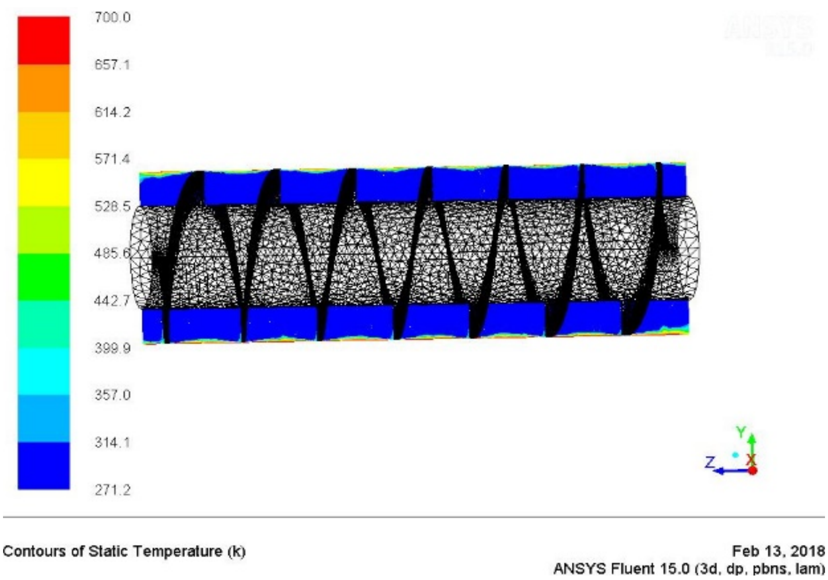

Fig. 9 Temperature profile in the r-z plane (geometry 2)

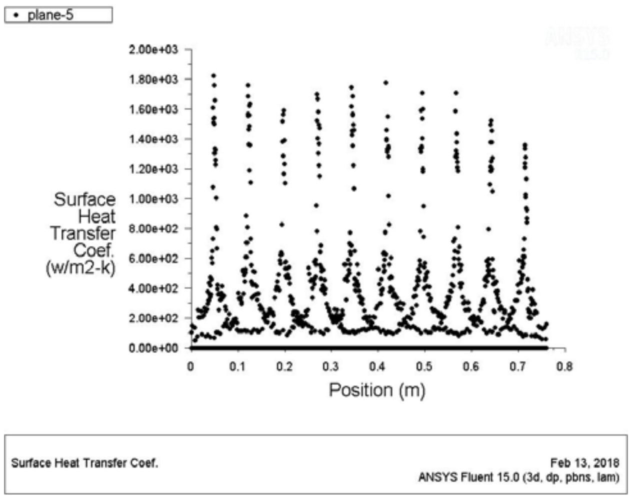

Fig. 10 Profile of heat transfer coefficient (geometry 1)

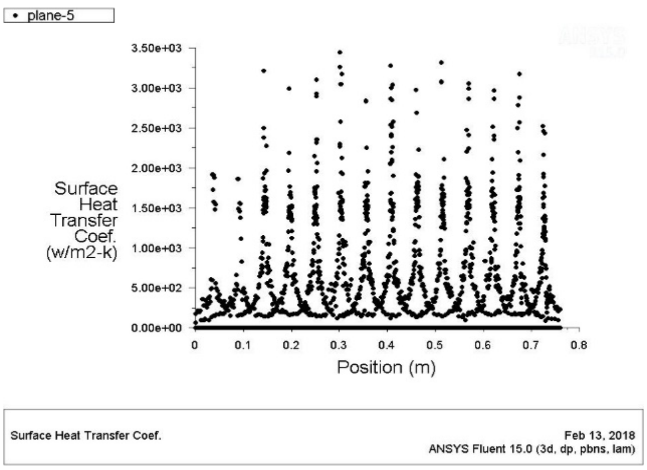

Fig. 11 Profile of heat transfer coefficient (geometry 2)

B. The Effect of the Conicity of the Rotor

In this section, we studied the effect of the conicity of the rotor on the hydrodynamic and thermal behavior of the SSHE.

Figs. 12 and 13 illustrate the distribution of velocity in (r-z) plane corresponding respectively to a conicity equal to 0.1, 0.15 for an axial Reynolds number equal to 1 and radial Reynolds number equal to 10. We identified the presence of a rotating vortex structure between all successive turns. Hence, we figure out the increase of the velocity along the SSHE due to the decrease of the cross section between the rotor and the heated wall. For the geometry of the rotor with a conicity equal to 0.15, we distinguish the presence of more intensive vortex at the entrance of the exchanger.

Figs. 16 and 17 presented the variation of heat transfer along the SSHE respectively for a conicity equal to 0.1 and 0.15. We distinguish a higher heat transfer coefficient for a geometry with higher conicity, this could be explained by the fact that the increase of the annular space between the rotor and the heated wall leads to the presence of a higher volume of bulk fluid that mixes with the scraped fluid which leads to an increase of the driving force of the heat transfer. We distinguish also that the heat transfer increases globally along the SSHE. This result was expected due to the increase of the velocity along the SSHE which leads to a higher heat transfer rate.

Figs. 14 and 15 reproduce temperature field in (r-z) plane for a conicity equal to 0.1 and 0.15. Comparing the two figures, we could distinguish a larger gradient of temperature corresponding to the thermal boundary layer localized in the vicinity of the heated wall for the geometry with a conicity equal to 0.1. This result is confirmed by the lowest surface heat transfer rate obtained for a conicity equal to 0.1 compared with surface heat transfer obtained for a conicity equal to 0.15.

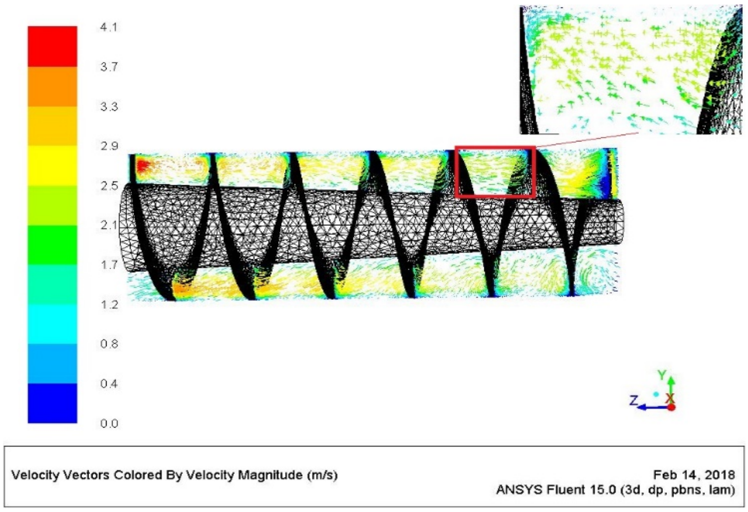

Fig. 12 Velocity vectors in the r-z plane (geometry 3)

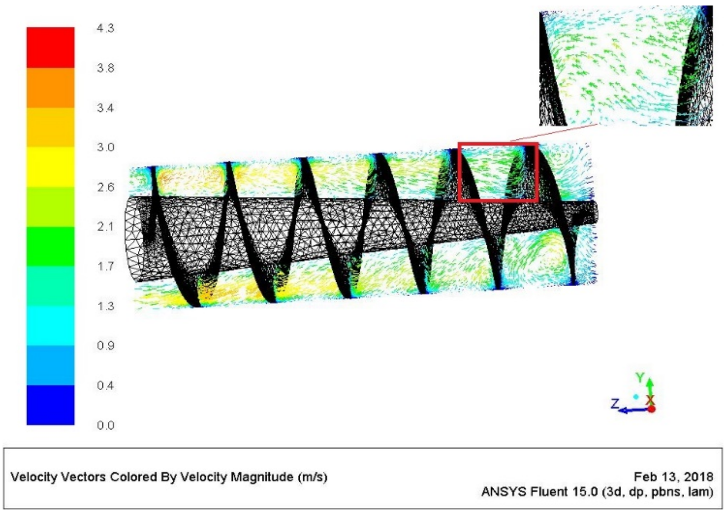

Fig. 13 Velocity vectors in the r-z plane (geometry 4)

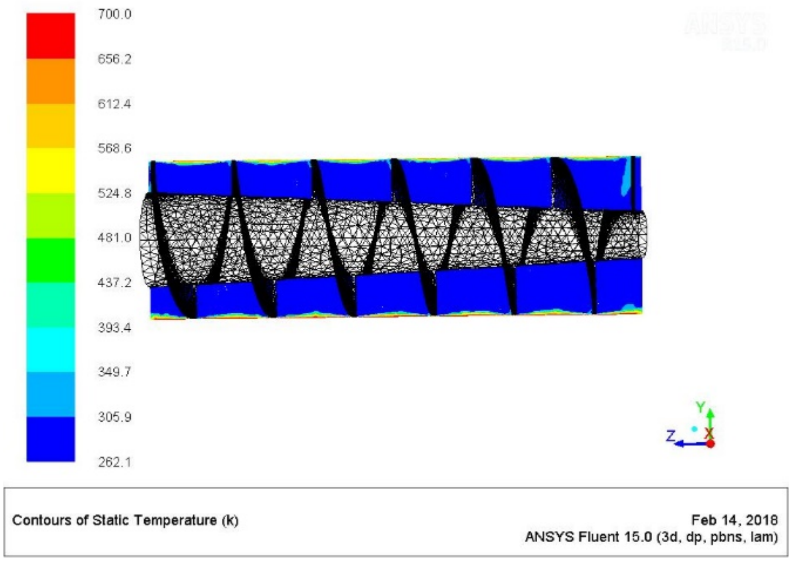

Fig. 14 Temperature profile in the r-z plane (geometry 3)

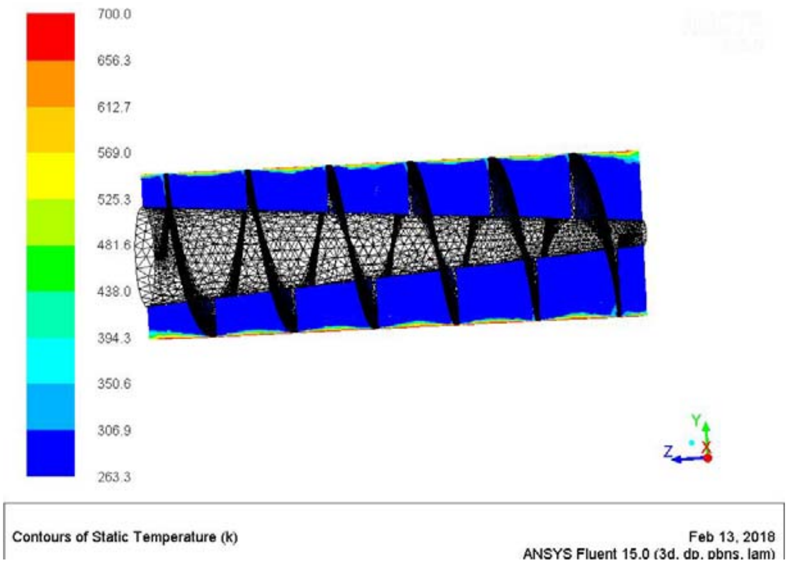

Fig. 15 Temperature profile in the r-z plane (geometry 4)

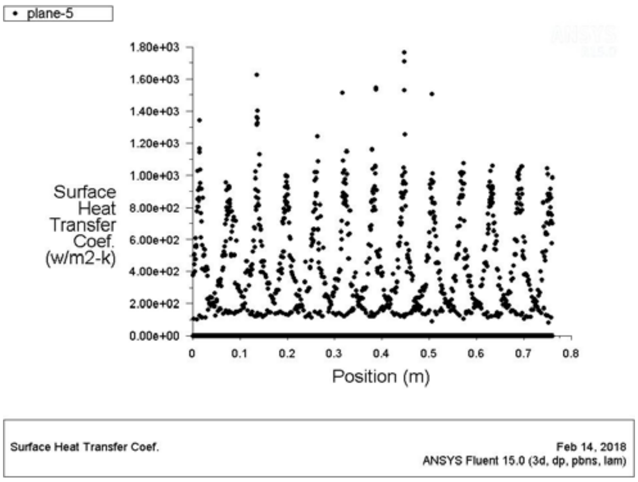

Fig. 16 Profile of heat transfer coefficient (geometry 3)

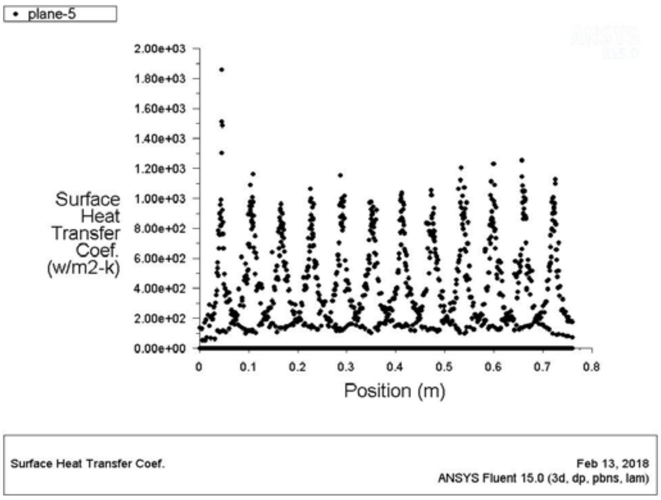

Fig. 17 Profile of heat transfer coefficient (geometry 4)

VII. Conclusion and Perspective

A numerical simulation of the hydrodynamic and thermal behavior of the SSHE was achieved for the case of laminar non isothermal Newtonian fluid. The effect of two geometric parameters (the conicity of the rotor and the number of turns screw) on velocity distribution and heat transfer was investigated under this study. The results of simulation show that increasing the number of turns in the screw leads to increase the thermal efficiency of the SSHE. This could be explained by the high volume of fluid scraped from the stator wall by the rotor with seven turns screw compared with the rotor with five turns. The increase of the conicity of the rotor from 0.1 to 0.15 leads to an amelioration of the heat transfer coefficient. As a perspective, we may study the effect of different operator parameters (axial and rotational Reynolds number) on the hydrodynamic and thermal behavior of the SSHE.

Nomenclature

D	internal diameter of the exchanger (m)
d	agitator diameter (m)
d_r	rotor shaft diameter (m)
g	gravity acceleration (m s^{-2})
H	height of the exchanger (m)
N	rotational speed (rev/s)
nb	number of blades
P	pressure (Pa)
p	pitch of the helical ribbon
T	temperature (K)
V	Velocity (m /s)
W_m	mean axial velocity ($m.s^{-1}$)
Greek symbols	
ρ	fluid density ($kg.m^{-3}$)
μ	dynamic viscosity (Pa.s)
Dimensionless numbers	
$Re_a = \frac{\rho(D - d_r)W_m}{\mu}$	axial Reynolds number

$Re_r = \frac{\rho \, N d^2}{\mu}$ rotational Reynolds number

$Pr = \frac{\mu \, Cp}{\lambda}$ Prandtl number

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