EXPERIMENTAL EVALUATION ON LOCAL HEAT TRANSFER BY FORCED CONVECTION IN VARIOUS GEOMETRIES

Antonio Carlos Ferreira, antonio.ferreira@unoesc.edu.br
Cristiano Meneghini, cristiano.meneghini@unoesc.edu.br
Daniela Pintro, dani9pintro@yahoo.com.br
Universidade do Oeste de Santa Catarina, Rua Getúlio Vargas nº 2125, Joaçaba – Brazil

Abstract. *In the present scenario the most varied forms of the heat transfer are studied for optimization of industrial processes involved. In this study forced airflow at 2.34m/s up until 7.11m/s across a steel pipes with diferent geometries on diameter 0.033m were performed to determined the local heat transfer coefficient by forced convection, h. The investigation of the local heat transfer coefficient were performed in cylindrical, square and hexagon cylinders. With this purpose it is used an experimental apparatus, which allowed us to calculate the coefficient by the empirical correlations given by Hilpert(1933), Zukauskas (1972) and Churcill & Bernstein (1977). Subsequently, some simulations were performed using commercial software in order to take a comparison of the results obtained experimentally. The h values were obtained using the empirical correlations form literature, that were in accordance through the empirical correlations comparision. The experimental and simulation results show us that the h values it is in agreement with the expected values, and both grow linearly to not cylindrical geometries.*

Keywords*: Heat Transfer Coefficient by Forced Convection, cylindrical geometries, square cylinders, hexagon cylinders.*

1. INTRODUCTION

The various heat transfer mechanisms are used in various industrial and commercial applications, and currently many studies on such forms of heat transfer are studied and improved for better efficiency in your application. Knowledge of heat transfer coefficient h is necessary for heat transfer design and calculation, in manufacturing processes, oil and gas flow processes, air conditioning and refrigeration systems. The heat transfer coefficient is critical and very important for developing and designing better flow process, that resulting in reduced energy consumption. With the design complexity and modernization of process, the study of forced convection over various geometries has become interesting (Jisheng and Tarasuk, 1992).

Numerous works have been made for modeling and simulation the fluid flow and heat transfer in a single circular and square cylinder. Furthermore, there are many papers that study the flow and heat transfer over multiple circular cylinders and several arrangements. Zdravkovich (1997) presents an extensive review of the hydrodynamic studies that context.

Morgan (1987) conducted an extensive review of literature on convection from a heated pipe in circular cylinder and proposed an correlation that depends on Reynolds and Prandtl number. In the context of flow over of circular cylinder, Hilpert (1933) was one of the earliest authors in the area of forced convection from heated pipe surfaces with air. The correlation proposed was done using integrated mean temperature values, not mean film temperature, and with inaccurate values for the thermo physical properties of air. Fand and Keswani (1972) reviewed the work of Hilpert (1933) and recalculated the values of the constants C and m of equation, using more accurate values for the thermo physical properties of air.

Kolar et al. (1997) shows the measurements on a pair of square cylinders through lases Doppler velocimetry at *Re*=23100, where the authors examined the strengths of the vortices both near the gap. Mizushima and Akinga (2003) investigated numerically and experimentally the interactions of wakes for the flow in square and circular bars. Kumar et al. (2008) conducted a two-dimensional simulation to understand the dynamics of vortices on a flow around nine square cylinders placed in a side-by-side arrangement for *s/d* ratio ranging from 0.3 to 12. They deployed lattice Boltzmann method in the low Reynolds number, *Re*=80.

From de above discussion, it follows that there are many reported work on the hydrodynamic aspects, and the thermal discussions are almost unavailable. Therefore, our aim of the present work is to investigate experimentally and through Computational Fluid Dynamic commercial software, the local heat transfer coefficient by convection, h, on a single heated horizontal circular, square and hexagon cylinder at high Reynolds number.

2. EMPIRICAL CORRELATIONS

Heat transfer by forced convection generally use a fan, blower or pump to provide high velocity fluid (gas or fluid). This high velocity fluid results in a decreased thermal resistance across the boundary layer from the fluid to the surface.

This, in turn, increases the amount of heat that is carried away by the fluid. Actually, there are many correlations to calculate the local heat transfer coefficient by forced convection from heated vertical and horizontal cylinders, in both situations, forced and natural convection. The coefficients obtained in the present work were based basically on empirical correlations proposed by three authors. The accuracy of correlations with which the heat transfer coefficient can be calculated depends on the correlation used. Forced convection is characterized by the dimensionless numbers of Nusselt, Prandtl and Reynolds. Nusselt number represents the ratio between the heat transfer by convection and by conduction. The ratio of the momentum diffusivity and the thermal diffusivity is the well-known Prandtl number. Reynolds number is a measure of the relative importance between the momentum flux by advection and by diffusion in the same direction. The dimensionless numbers are determined by equations below:

$$Nu_D = \frac{hD}{k} \tag{1}$$

$$\mathrm{Pr} = \frac{\nu}{\alpha} \tag{2}$$

$$\mathrm{Re}_D = \frac{\rho \overline{V} D}{\mu} \tag{3}$$

In the equations above, D is the characteristic dimension of the cylindrical, h is heat transfer coefficient by convection, k is the thermal conductivity of the fluid, ν is the kinematic, α is the thermal diffusivity of the fluid, ρ is the density of the fluid, $\overline{V}$ is the mean velocity of the fluid, and μ is the dynamic viscosity of the fluid.

Hilpert (1933) proposed a correlation to describe forced convection in horizontal heated circular cylinder, shown by Eq. (4), where the values of *C* and *m*, are given on Table 1. This correlation was done using integrated mean temperature values, and can be used for other geometries.

$$Nu_D = C\,\mathrm{Re}_D^m\,\mathrm{Pr}^{1/3} \tag{4}$$

Table 1. Hilpert constants for forced convection.

Re_D	C	m
0.4 – 4	0.981	0.33
4 – 40	0.911	0.385
40 – 4000	0.683	0.446
4000 – 400000	0.193	0.618
400000 – 40000000	0.027	0.805

Another correlation proposed by Zukauskas (1972) for convection heat transfer over a circular cylinder heated was given by Eq. (5), where the values of *c* and *m* are given by Table 2, and if Pr<10, *n*=0.37, or if Pr>10, *n*=0.36. With exception of the Pr_S, all calculations were done at the mean film temperature.

$$Nu_D = c\,\mathrm{Re}_D^m\,\mathrm{Pr}^n\left(\frac{\mathrm{Pr}}{\mathrm{Pr}_S}\right)^{0.25} \tag{5}$$

Table 2. Zukauskas constants for forced convection from Eq. (5).

Re_D	c	m
1 – 40	0.75	0.4
40 – 1000	0.51	0.5
1000 – 2x10^5	0.26	0.6
2x10^5 – 10^6	0.076	0.7

Churchill and Bernstein (1977) proposed a single correlation that covers all range of Re_D, as well as a wide range of Pr, shown by Eq. (6). The empirical correlation was recommended for all $\mathrm{Re}_D.\mathrm{Pr}$ > 0.2, and all properties were calculated at the film temperature for circular cylinder.

$$Nu_D = 0.3 + \frac{0.62\,\mathrm{Re}_D^{1/2}\,\mathrm{Pr}^{1/3}}{\left[1+\left(0.4/\mathrm{Pr}^{2/3}\right)\right]^{1/4}}\left[1+\left(\frac{\mathrm{Re}_D}{282000}\right)^{5/8}\right]^{4/5} \tag{6}$$

3. EXPERIMENTAL PROCEDURE

The experimental apparatus illustrated in Fig. (1) was designed and built to experimentally determine the coefficient h for various kinds of cross section on cylinders. The test body has an inner hole to insert the electrical resistance and the ends of the cylinder are fixed through a support. The arrangement of test bench not allowed the inclination of the test bodies. The support is isolated to prevent heat transfer between the support and the test body. The fan test section has 200 mm in diameter and 300 mm away from the specimen. Two thermocouples were used in the experimental apparatus and one thermocouple to measure and control the environmental temperature. The thermocouples were connected to a data acquisition system, and the test body surface temperature was controlled with the HH-25KC thermometer via k-type thermocouples. The k-type thermocouples has a resolution of 0.1°C and the accuracy of 0.5°C over the temperature range -85 °C to 1100°C. The thermocouple was installed in frontal of the specimen. In order to obtain the uniform surface temperature and stability preliminary heating tests were performed to check the test arrangement. The equilibrium conditions were obtained within 20 minutes of heating and were controlled by monitoring the thermocouple at 10 seconds time intervals during 15 minutes.

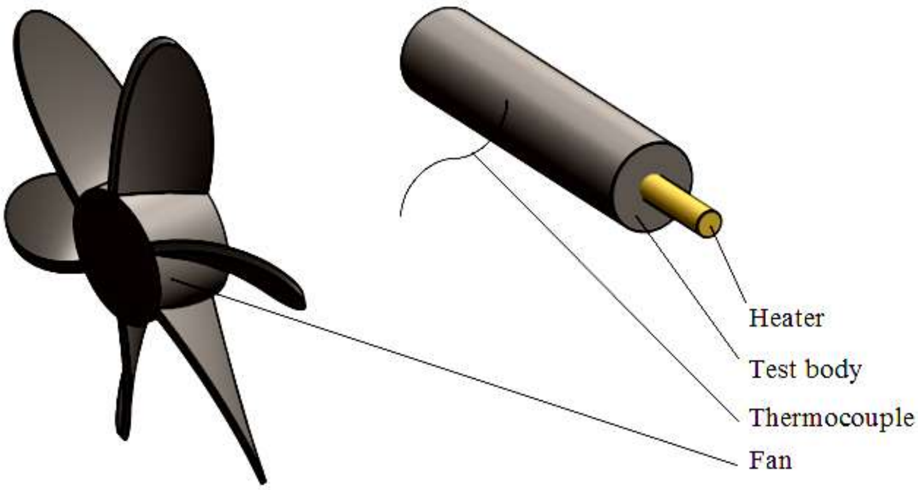

Figure 1. Experimental apparatus.

It was manufacturing three different test specimens, an cylindrical format, an square and an hexagon format as shown in Fig. (2). The specimens were machined from a homogeneous 1020 carbon steel bar at the Machining Laboratory at UNOESC. 1020 carbon steel was chosen due to its widely used in commercial heat exchangers and the fact of high thermal conductivity, which is great for a heat exchanger. All three test bodies has 33 mm of hydraulic diameter, 140 mm of length and a diameter of 10 mm in inner hole. The electrical resistance has 150 W of power and 10 mm of external diameter.

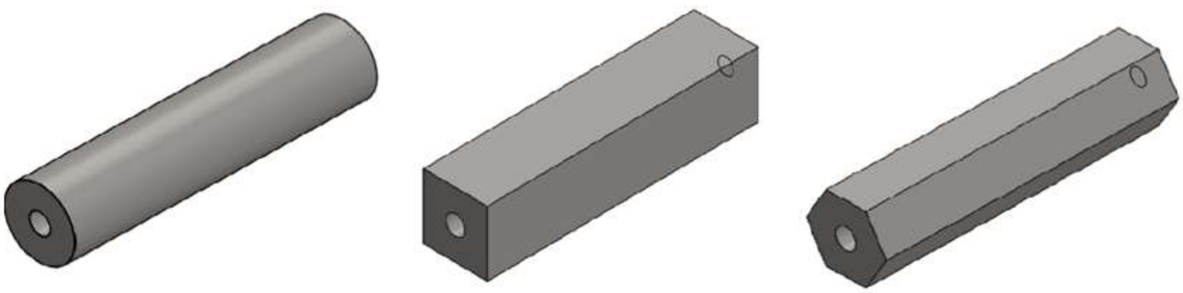

Figure 2. Test bodies used in the experimental apparatus.

The volumetric flow rate is controlled by a frequency inverter where it is possible obtain a wide range of air velocity over the body test. The test bench was continuously monitored to obtain a uniformly heated surface, and mainly constant airflow over the body test with an anemometer that has an resolution of 0.01 m/s. Finished the measurements in the body test the electrical resistance was switched off and until to cool to room temperature. This procediment was repeated twice for each test body, and the average of two tests was calculated and used to calculate the heat transfer coefficient by forced convection, h.

For the three test bodies the tests were conducted at horizontal inclination for air flow velocities of 2.34 m/s, 3.25 m/s, 4.53 m/s, 5.72 m/s and 7.11 m/s. After obtained the equilibrium conditions data for the cylinder surface temperature, environmental temperature and wind velocity of air over the test specimen were recorded. The uncertainty in experimental results is 6%.

4. RESULTS

Table (3) shows the local heat transfer coefficients by forced convection calculated using the empirical correlations presented above only in circular cylindrical. Each correlation has its specificity in terms of dimensionless numbers, and therefore, the results tend to be different from each other. The values on Tab. (3) indicate that as air velocity increased, the differences between values of h for circular cylinder also increased. Zukauskas correlations given all values of h bigger than other correlations presented in this work, and the coefficient h grows linearly with increasing air velocity. For all tests conditions the Churchill and Bermstein (1977) and Hilpert (1933) correlations yielded very near values of coefficient h. The largest difference occurred with the Zukauskas (1972) and Churchill and Bermstein (1977) correlations to 7.11 m/s.

Table 3. Coefficients h (W/m².°C) calculated by the empirical correlations in circular cylindrical.

Correlations	2.34m/s	3.25m/s	4.53m/s	5.72m/s	7.11m/s
Zukauskas (1972)	29.92	36.36	44.31	50.95	58.06
Churchill and Bernstein (1977)	27.92	32.93	39.98	43.93	49.17
Hilpert (1933)	24.99	30.57	37.53	43.35	49.62

Figure (3) presents the comparison between Nusselt and air velocity values for the heat transfer in the circular cylindrical in forced convection. The experimentally Nusselt number for the air velocities used was compared with the empirical correlations presented above. The results on Fig. (3) indicate that as air velocity increased, the experimentally Nusselt number values obtained from empirical correlations also increased.

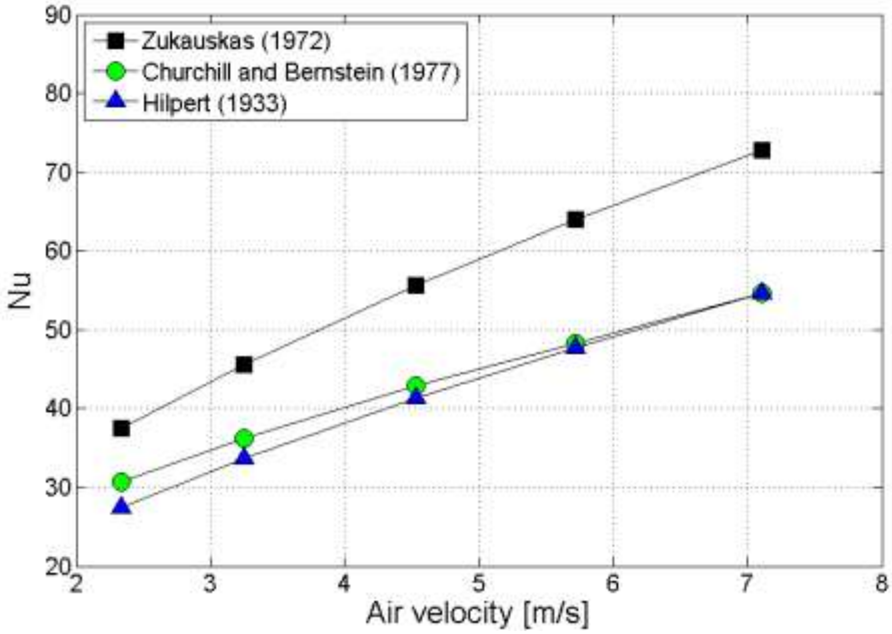

Figure 3. Calculated Nusselt number versus air velocity in circular cylinder and forced convection.

The values of surface temperature for the circular cylinder in forced convection are illustrated in Fig. (4) for all values to the air velocity. This figure shows how is the distribution of surface temperatures in relation to the local coefficient of hear transfer by forced convection. For smaller coefficient values h, it is clearly notable that the surface temperature is high, and lower Reynolds numbers result in lower heat transfer by advection of the flow. It is noted that the maximum relative error found among the three correlations used was 13.8%, which can be considered reasonable in this cases of empirical correlations.

Figure (5) shows the results of the local heat transfer coefficient by forced convection in a square and hexagon cylinder as a function of Reynolds number. In both tests the flow regime was turbulent and was used only the Hilpert (1933) empirical correlation for obtained the coefficient h. It was demonstrated in this figure that there is a noticeable difference in the values of h by changing the square cylinder to the hexagon cylinder, and when Reynolds numbers increases the discrepancy of local coefficient h between the geometries used grows linearly. This may be observed through a linear fit of the two cases studied, which showed values of R^2 very close to 1, meaning that there is a strong correlation between Reynolds number and the local coefficient heat transfer by forced convection h. Moreover, the hexagon cylinder showed higher values of coefficient h, which means bigger heat transfer when compared to square cylinder in the same Reynolds number.

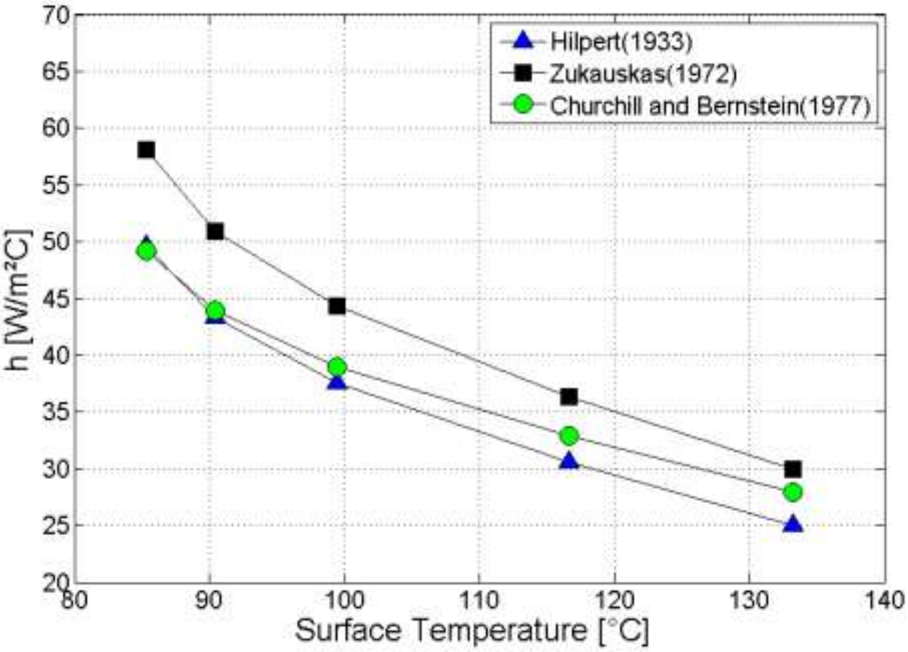

Figure 4. Results of surface temperature with the coefficient heat transfer by forced convection on circular cylinder.

Figure (6) shows the results of coefficient distribution h by using Computational Fluid Dynamic software, Solid Works Flow Simulation, on a square cylinder in forced convection at air velocity on 7.11 m/s. The dashed line represents the measure point, and agrees very well to the value determined from empirical correlations. It is clearly that coefficient h are precisely higher on the front face of the cylinders compared to the top, bottom and rear faces as demonstrated in Fig. (6). This indicates a higher heat transfer characteristic at the front face. The effect of flow separation showed in previous works is responsible for less heat transfer in the rear faces of the cylinders that means a smaller value of local coefficient h.

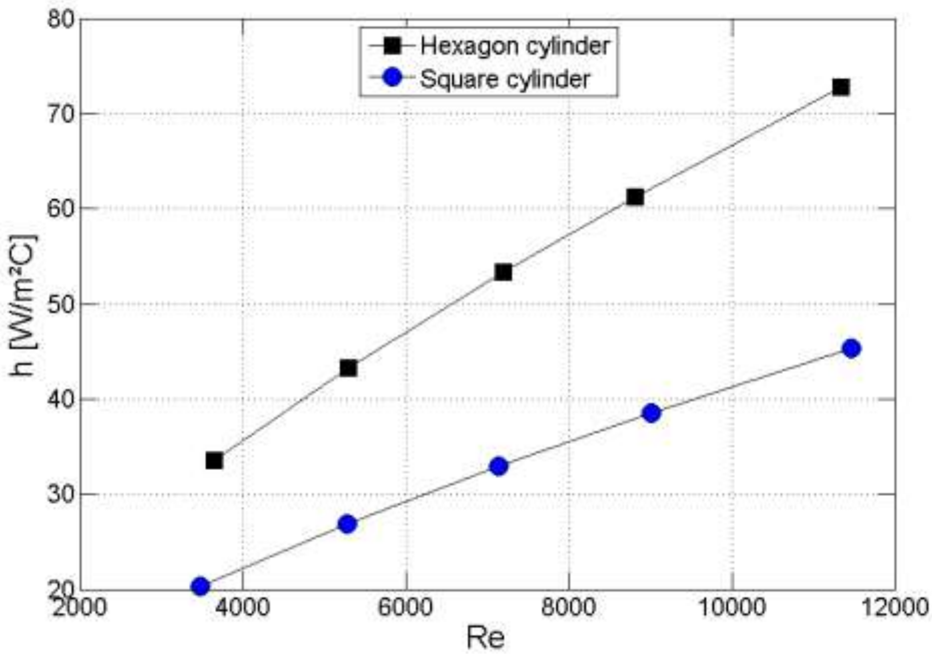

Figure 5. Results of coefficient heat transfer by forced convection on square and hexagon cylinder using Hilpert´s correlation.

The development of the boundary layer varies the local coefficient h by forced convection on a square cylinder as seen in Fig (6). Also, the separation and vortex from the cylinder surface into its wake can result in variation of local Nusselt number, and the maximum value occurs where the thickness of the boundary layer is smallest and in the corners. This happened because the cold fluid impinges directly on the front faces of the cylinders which results a greater heat transfer at those faces. It is noted too that the distribution is symmetric on the front surface with respect to the mid longitudinal plane. The conclusion is that the heat transfer in the surface depends to the flow field, and the local heat transfer is minimum where the air velocity magnitudes are very small. In the top surfaces of the square cylinder, heat transfer decreases along the flow direction.

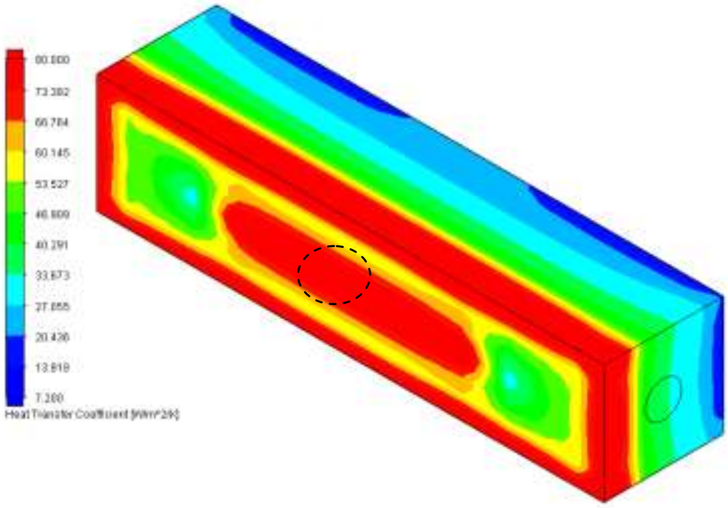

Figure 6. Distribution of local heat transfer coefficient by forced convection on a square cylinder at air velocity 7.11m/s.

In order to compare the values for hexagon cylinders, Fig. (7) shows the simulation on distribution of local heat transfer coefficient in this case too. It is clearly that all situations described above in the square cylinder, happened in the hexagon cylinder.

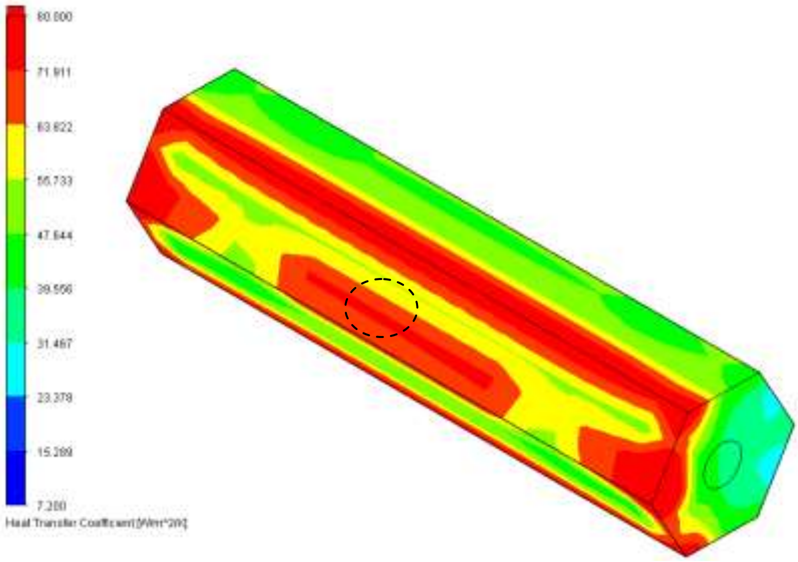

Figure 7. Distribution of local heat transfer coefficient by forced convection on hexagon cylinder at air velocity 7.11m/s.

5. ACKNOWLEDGEMENTS

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