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Analysis of Heat Transfer using Radiation in Circular Finned-Tube Heat Exchanger

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ABSTRACT

Transferring heat using conduction and convection requires the presence of a temperature gradient in some form of matter ,while the heat transfer by thermal radiation requires no matter. It is extremely important process, and from physical sense it is perhaps the most interesting form of the heat transfer modes. It is relevant to many industrial heating, cooling, and drying processes, as well as to energy conversion methods that involves fossil fuel combustion and solar radiation. In this paper, our objective is studying the radiation in Circular Finned-tube heat exchanger. Finned-tube heat exchangers are predominantly used in space conditioning systems, as well as other applications requiring heat exchange between two fluids. One important widespread use is in cooling applications. The results show that the heat transfer in the mode of the radiation is just about 2% compared with forced convection , but when there is natural convection in the system the term of radiation is considerable.

Introduction

Compact heat exchanger , including two types of heat exchangers such as plate-fin heat exchanger and fin-and-tube heat exchangers. Finned-tube heat exchangers have been used for heat exchange between gases and liquids for many years. In a gas-to-liquid exchanger, the heat transfer coefficient on the liquid side is generally one order of magnitude higher than that on the gas side. Therefore, to have balanced thermal conductances on the both sides for a minimum-size heat exchanger, fins are used on the gas side to increase the surface area. Figure of (1) shows three important finned-tube heat exchanger construction type. Figure 1.(a) shows circular finned-tube geometry, Figure1.(b) shows the plate finned-tube geometry and Figure1.(c) shows the plate-fin flat-tube geometry. All of them are widely used in the air-conditioning, refrigerator , and automotive industry. Some examples are cooling towers, evaporators, condensers, and radiators.

Finned-tube exchanger can with stand high pressure on the tube side. The highest temperature is again limited by the type of bonding materials employed, and materials thickness. Finned-tube exchangers usually are less compact than plate-fin exchangers. Finned-tube exchangers with a surface area density of about 3300 m^2/m^3 are available commercially. On the fin side, the desired surface area can be achieved through the proper fin density and fin geometry. Typical fin densities for plate fins vary from *250 to 800* fins per meter, fin thickness vary from *0.08 to 0.25 mm* , and fin flow lengths vary from *25 to 250 mm.* Nevertheless, heat exchanger design is a traditional and classical issue. It can be based on previous design experiences and industrial requirements. The design of heat exchangers, in general, including geometrical parameters and operating specifications, optimization process and cost estimation. Firstly, the geometrical parameters are chosen for a specified heat duty. Then, these parameters are changed manually by a trial-and-error process in order to satisfy the conditions on allowable pressure drops. Subsequently, the heat transfer coefficients and friction factors are recalculated until both the heat duty and pressure drop meet specified requirements. So, the design task is a complex trial-and-error process and there is always the possibility that designed results are not the optimum.

An extensive number of both numerical and experimental studies have been conducted on finned-tube heat exchanger performance. Liang et.al, [1] made a comparison of a one-dimensional analytical method, a one-dimensional numerical model, and a two-dimensional numerical model of a plate fin-and-tube heat exchanger. They gave instructions on how to strike a balance between accuracy and complexity , which are crucial problems when using computational fluid dynamics. 12 three-row plate fin-and-tube heat

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exchangers with four different types of fin configurations was investigated by Kang [2]. He presented a wide range of useful correlations for heat transfer analysis by Reynolds and Nusselt numbers. Jang and Chen [3]studied the effects of different geometrical parameters, including tube row number, wavy angles and wavy heights on Reynolds number. Tao et.al,[4] studied a three dimensional numerical simulation for laminar flow of a herringbone fin-and-tube heat exchanger using a body-fitted coordinate method with fin efficiency taken into account.

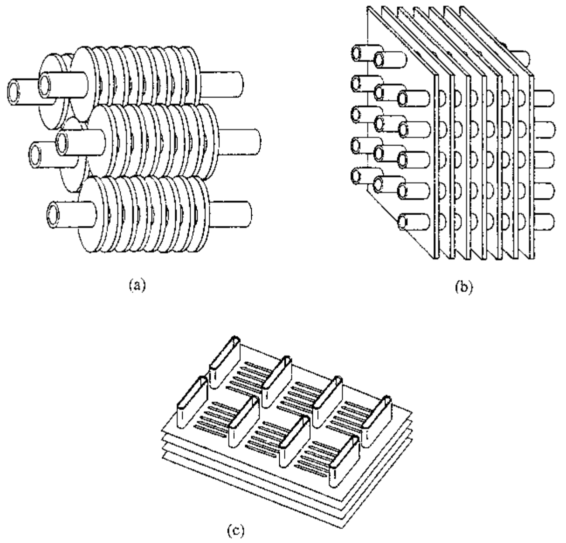

Figure 1. Finned-tube heat exchanger (a) Circular finned-tube type.(b) Plate finned-tube type. (c) Louvered plate-fin flat-tube type

Fluid flow and heat exchange on the air side of a multi-row fin-and-tube heat exchanger was analyzed by Carija and Frankovic [5]. They showed a comparison between a fin-and-tube heat exchanger with flat and Louvered fins in a wider range of operating conditions defined by Reynolds number[6].

Lin and Jang [7] studied about conjugate heat transfer and fluid flow in fin-tube heat exchangers with Wavy-Type vortex generators. Wang and Lo [8-9] conducted several studies about flow visualization. They performed a experimental study on flow visualization of annular and delta winglet vortex generators in fin-and-tube heat exchanger application. In another study, they focused on flow visualization of Wavy-Type vortex generators having inline fin-tube arrangement. Lee et.al, [10-11] numerically analyzed the heat transfer characteristics of fin-and-tube evaporators for two-dimensional air flow maldistributions. The simulation results for the uniform air flow case were compared to test data in order to validate the simulation program. Analysis of fin efficiency in enhanced fin-and-tube heat exchangers when there are dry conditions in the system was performed by Perrotin and Clodic [12]. Joen and Willockx [13] conducted a research about performance prediction of compact fin-and-tube heat exchangers in maldistributions air flow. They carried out an experimental study in two different configuration, single and three-screen mode. They investigated a wide range of Reynolds numbers on the air side, and eventually a test rig was constructed to evaluate the heat transfer capacity on the air and water side. Pongsoi et.al, [14] investigated on the air-side performance of a multipass parallel and counter cross-flow L-Footed spiral fin and tube heat exchanger. Their results showed that the number of tube rows has no significant effect on the air-side heat transfer or on friction characteristics at high Reynolds numbers. Lee and kang [15] conducted a research on air-side heat transfer characteristics of spiral-type circular fin-and-tube heat exchangers. In a similar research, Srisawad and Wangwises [16] analyzed the heat transfer characteristics of a new helically coiled crimped spiral finned-tube heat exchanger. An analysis of laminar heat transfer and fluid flow in a Wavy fin-and-tube heat exchanger has been carried out by Glazar and Anica [17]. Their results showed that there is an optimal fin pitch for each air velocity, which gives the best heat exchanger performance from the heat transfer point of view.

This paper analyzed heat transfer in circular finned-tube heat exchanger. This type of the heat exchanger is used typically for air-to-liquid in the cooling applications. In these cases a hot gas flows on the tubes with circular fins, and its heat is transferred to the liquid. According to the high temperature of gas, it is reasonable that the term of heat transfer by radiation be considerable. Here, we analyzed transferring heat in the modes of radiation and forced convection for eight circular finned-tube heat exchangers.

Thermal analysis

The basic geometry for a finned-tube heat exchanger is shown in Figure of (2). The total number of tube can be calculated in terms of tube pitches P_t and P_c and the dimensions of the exchanger, where P_t is the transverse tube pitch and P_c the longitudinal tube pitch. Note that the number of tubes in the first row perpendicular to the flow direction is $L_{3/}P_t$ and the number of tubes in the second row is $(L_{3/}P_t - 1)$. Also, the number of rows with the same pattern as the first row is $(L_{2/}P_c + 1)/2$ and the number of rows with the same pattern as the second row is $(L_{2/}P_c - 1)/2$. Therefore, the total numbers of tubes is:

$$N_t = \frac{L_3}{P_t}\frac{\frac{L_2}{P_c}+1}{2} + \left(\frac{L_3}{P_t}-1\right)\frac{\frac{L_2}{P_c}-1}{2} \quad (1)$$

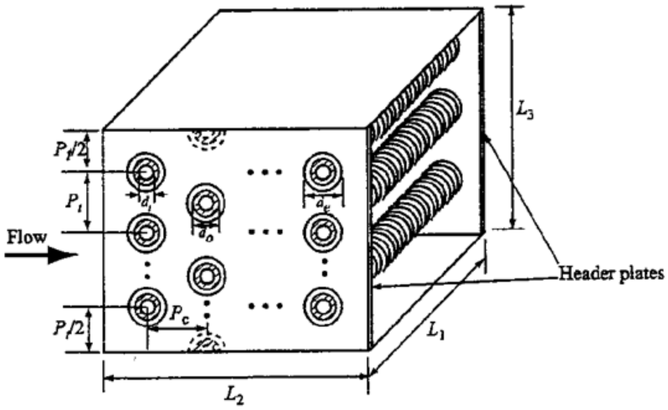

Figure 2. Circular finned-tube heat exchanger

The total heat transfer area A_t is composed of the primary surface area consists of the tube surface area A_P and the fin surface area A_f. The primary surface area is:

$$A_P = \pi d_o \left(L_1 - \delta N_f L_1\right) N_t + 2\left(L_2 L_3 - \frac{\pi d_o^2}{4} N_t\right) \quad (2)$$

The fin surface area is:

$$A_f = \left(\frac{2\pi\left(d_e^2 - d_o^2\right)}{4} + \pi d_e \delta\right) N_f L_1 N_t \quad (3)$$

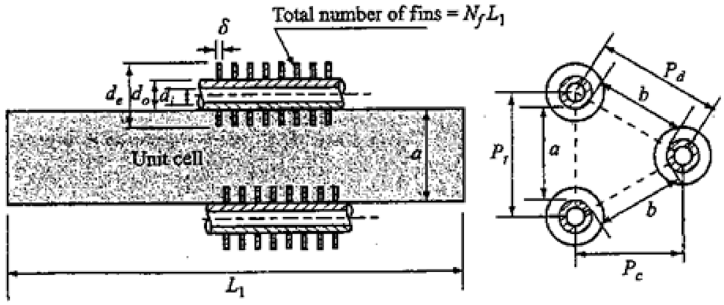

Figure 3. Unit cell of a staggered finned-tube arrangement

Where N_f is the number of fins per unit length. The total minimum free-flow area can be calculated by multiplying the unit cell area by the number of unit cells (L_3/P_t). The minimum free flow area occurs either at a plane through *a* or two planes through *b* in Figure of (3) , whichever the smaller area takes the minimum free-flow area. However, since two planes through *b* are always greater than one plane through *a* for the equilateral triangular arrangement, the minimum free-flow area must occur at the plane through *a*. in other words, the flow through *a* is not restricted by the diagonal planes through *b* because $2b \succ a$. Consequently, the minimum free-flow area is calculated using the dark area in figure of (3).

$$A_c = \frac{L_3}{P_t}\left[(P_t - d_o)L_1 - (d_e - d_o)\delta N_f L_1\right] \tag{4}$$

The hydraulic diameter is introduced using equation of (5),

$$D_h = \frac{4A_c L_2}{A_t} \tag{5}$$

Also, the mass velocity and pin pitch are respectively:

$$G = \frac{\dot{m}}{A_c} \tag{6}$$

$$P_f = \frac{1}{N_f} \tag{7}$$

In compact heat exchanger there are some dimensionless number that help to calculate heat transfer coefficients and pressure drops. Some of them are:

$$St = \frac{Nu}{Re.Pr} \tag{8}$$

$$Nu = \frac{h.D_h}{K} \tag{9}$$

$$Pr = \frac{C_p.\mu}{K} \tag{10}$$

It is customary to use the Colburn factor to represent the thermal characteristics of the compact heat exchangers. It is defined and expressed using equation (11) as,

$$j = St.Pr^{2/3} \tag{11}$$

Beriggs and Young [6] introduced some functions for the Colburn and friction factors,

$$j = 0.134\,\mathrm{Re}^{-0.319}\left(\frac{P_f - \delta}{d_e - d_o}\right)^{0.2}\left(\frac{P_f - \delta}{\delta}\right)^{0.11} \tag{12}$$

So, the convective heat transfer coefficient arising forced convection is:

$$h = j.G.C_p.Pr^{-2/3} \tag{13}$$

About transferring heat using Radiation the profile function is:

$$f(r) = \frac{\delta_a}{2} + \frac{R}{2}(\delta_b - \delta_a) \tag{14}$$

Where

$$R = \frac{r_a - r}{r_a - r_b} \tag{15}$$

When $\delta_a = \delta_b$ and $\delta_a = 0$,the specific cases of the fins of rectangular and triangular profile occur. The terminology and coordinate system for the radial fin of trapezoidal profile is shown in Figure of (4). Observe that the radial height coordinate has its origin at the center of the fin and is positive in the outward direction. Energy conservation law denotes that the heat dissipation from the faces of the element *dr* is equal with the difference in heat conduction into and out of the element *dr*. The main assumption is that the radiation occurs from both faces of the fin.

$$K\frac{d}{dr}\left[2\pi r\left(2.f\left(r\right)\right)\frac{dT}{dr}\right]dr = 2\pi r\left(K_1T^4\text{-}K_2\right)dr \tag{16}$$

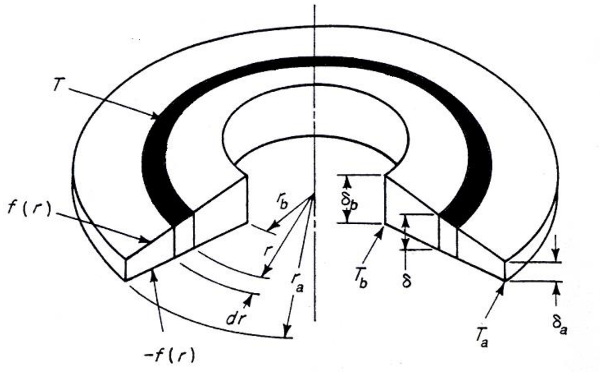

Figure 4.Coordinate system for a radiating radial fin of trapezoidal profile

The taper and radius ratios have been defined respectively,

$$\lambda=\frac{\delta_a}{\delta_b} \tag{17}$$

$$\rho=\frac{r_b}{r_a} \tag{18}$$

Eventually the governing differential equation for the temperature profile is:

$$\left[\frac{1}{\rho}\text{-}\lambda+\xi\left(\lambda\text{-}1\right)\right]\frac{d^2y}{d\xi^2}+\left[\frac{1\text{-}\lambda\rho}{\rho\xi}\text{-}2\left(\lambda\text{-}1\right)\right]\frac{dy}{d\xi}\text{-}\frac{\zeta\left(1\text{-}\rho\right)}{\rho}\left(y^4\text{-}\frac{K_2}{K_1T_b{}^4}\right)=0 \tag{19}$$

Here $y=T/T_b$ and $\xi=r/r_b$, and the profile number for the radial radiating fin is:

$$\zeta=\frac{K_1T_b{}^3r_b{}^2}{K\delta} \tag{20}$$

Equation of (20) is nonlinear and must be solved numerically for the dimensionless temperature as a function of the dimensionless ratio ξ for assumed values of the parameters ζ, ρ, λ and $K_2/K_1T_b{}^4$. The solution is subject to the condition that $dy/d\xi = dT/dr$ is equal to zero at $\xi = r_a/r_b = 1/\rho$.

The heat transferred from the fin may be evaluated by applying Fourier's law at the fin base,

$$q_b=\text{-}2\pi K\delta_b r_b\left.\frac{dT}{dr}\right|_{r=r_b} \tag{21}$$

Or in terms of the dimensionless parameters,

$$q_b=\text{-}2\pi K\delta_b T_b\left.\frac{dT}{d\xi}\right|_{\xi=1} \tag{22}$$

And fin efficiency will be this actual heat flow divided by the ideal heat flow with no heat gained from the surroundings:

$$\eta=\text{-}\frac{2\rho^2\left.\frac{dy}{d\xi}\right|_{\xi=1}}{\zeta\left(1\text{-}\rho^2\right)} \tag{23}$$

For friction factor, Robinson and Briggs [7] recommended this equation:

$$f = 9.465\,\mathrm{Re}^{-0.316}\left(\frac{P_t}{d_o}\right)^{0.927}\left(\frac{P_t}{P_d}\right)^{0.515} \tag{24}$$

For a circular finned-tube heat exchanger, the tube outside flow in each tube row experiences a contraction and an expansion. Thus, the pressure losses associated with a tube row within the core are of the same order of magnitude as those at the entrance with the first tube row and those at the exit with the last tube row. Consequently, the entrance and exit pressure drops are not calculated separately, but they are generally lumped into the friction factor for individually finned tubes. By eliminating the entrance and exit terms, the total pressure drop associated with the core is:

$$\Delta P = \frac{G^2}{2\rho_i}\left[2\left(\frac{\rho_i}{\rho_o}-1\right)+\frac{4fL}{D_h}\rho_i\left(\frac{1}{\rho_m}\right)\right] \tag{25}$$

Where $\left(\frac{1}{\rho_m}\right)$ was defined by,

$$\left(\frac{1}{\rho_m}\right)=\frac{1}{2}\left(\frac{1}{\rho_i}+\frac{1}{\rho_o}\right) \tag{26}$$

Results and discussions

To analysis heat transfer coefficient, a case study has been done. A cold air is heated by hot wastewater. The cold air enters at 278 K with a flow rate of 124 kg/s and will leave at 303 K . The hot water enters at a flow rate of 50 kg/s and 365 K . The allowable air pressure drop is 250 Pa and the allowable water pressure drop is 3 Kpa. The thermal conductivies of the aluminum fins and the copper tubes are 200 W/m.K and 358 W/m.K , respectively. The tubes of the heat exchanger have an equilateral triangular arrangement. The tube pitch is 4.976 cm and the fin pitch is 3.46 fins/cm. The fin thickness is 0.305 mm. the three diameters are given as d_i =2.2098 cm for the tube inside diameter, d_o =2.601 cm for the outside diameter and d_e =4.412 cm for the fin outside diameter.

Table 1. Designing information

Description	Value
Air outlet temperature	303 K
Water outlet temperature	350.1 K
Pressure drop for water flow	3 Kpa
Pressure drop for air flow	250 Pa
Fan power	0.258 hp
Pump power	40.938 hp

Table 2. Thermal quantity results

Description	Value
Number of transfer unit	0.389
Effectiveness	0.287
Heat transfer rate	3.122×10^6 W
Convective coefficient for coolant side	8442 W/m^2.K
Convective coefficient for air side	194.456 W/m^2.K
UA Value	4.862×10^4 W/K

Table 3. Hydraulic quantity results

Description	Value
Hydraulic diameter	4.851 mm
Reynolds number for water	7.648×10^4

Reynolds number for air	5.697×10^3
Air velocity at exit	18.173 m/s
Water velocity	1.267 m/s

Table 4. Geometry quantity results

Description	Value
Tube inside diameter	22.1 mm
Tube outside diameter	26.01 mm
Fin outside diameter	44.12 mm
Transverse tube pitch	49.76 mm
Longitudinal tube pitch	43.09 mm
Fin thickness	0.305 mm
Number of fins per centimeter centimeter	3.46 cm^{-1}
Number of tubes	105.6
Total heat transfer area at air side side	366 m^2
porosity	0.439
Surface area density	394.118 m^2/m^3

To analysis heat transfer using Radiation it is assumed that the radial dins have a base temperature of 363.8 K and is fabricated of Aluminum having thermal conductivity 200 W/m.K. The fins have an emissivity of o.809, and solar energy is received by the dissipating face at a rate of 1303.7 W/m^2, and the fin material has an absorptivity (The absorptance is the amount of radiation absorbed by a surface compared to that absorbed by a black body) of 0.231 at the wavelength of solar radiation. Table of (5) show the result of radiation analysis for this finned tube heat exchanger.

Table 5. Heat transfer using Radiation

Description	Profile number	Radius ratio	Environmental parameter	Total fin number	Heat transfer(Watt)
Value	0.024	0.589	0.443	162610.37	130088

Although the rate of heat transfer by radiation is significant compared to transferring heat using forced convection, but heat transfer coefficient arising radiation is just 2.1% of heat transfer coefficient due to forced convection. Also, based on Asadi and Khoshkhoo research [18] , one of the important factor in transferring heat by radiation is arrangement of flow. In Parallel flow arrangement there is the maximum value of heat transferring arising radiation, while in Counter flow is minimum. To analysis precisely heat transferring by radiation finned-tube heat exchanger, eight types of circular finned-tube heat exchanger has been studied . Table of (7) shows its results.

Table 6. Surface properties of different type of circular finned-tube heat exchanger

Surface designation	**CF-7.0-5/8J**	**CF-8.7-5/8J(a)**	**CF-8.7-5/8J(b)**	**CF-9.05-3/4J(a)**	**CF-9.05-3/4J(b)**	**CF-9.05-3/4J(c)**	**CF-9.05-3/4J(d)**	**CF-9.05-3/4J(e)**
Tube arrangement	Staggered	Staggered	Staggered	Staggered	Staggered	Staggered	Staggered	Staggered
Tube diameter(10^{-3}m)	16.38	16.38	16.38	19.66	19.66	19.66	19.66	19.66
Fin outside diameter(10^{-3}m)	28.5	28.5	28.5	37.2	37.2	37.2	37.2	37.2
Transverse tube spacing(10^{-3}m)	31.3	31.3	46.9	39.5	50.3	69.2	69.2	50.3
Longitudinal tube spacing(10^{-3}m)	34.3	34.3	34.3	44.5	44.5	44.5	20.3	34.9
Fins/in	7.0	8.7	8.7	9.05	9.05	9.05	9.05	9.05
Fin thickness(10^{-3}m)	0.25	0.25	0.25	0.31	0.31	0.31	0.31	0.31

Free-flow/frontal area	0.449	0.443	0.628	0.455	0.572	0.688	0.537	0.572
Heat transfer area/total volume	269	324	216	354	279	203	443	354
Fin area/ total area	0.830	0.862	0.862	0.835	0.835	0.835	0.835	0.835
Hydraulic diameter(10^{-3}m)	6.68	5.48	11.67	5.13	8.18	13.59	4.85	6.43

Table 7. Radiation information for different types of circular finned-tube heat exchanger

Description	CF-7.0-5/8J	CF-8.7-5/8J(a)	CF-8.7-5/8J(b)	CF-9.05-3/4J(a)	CF-9.05-3/4J(b)	CF-9.05-3/4J(c)	CF-9.05-3/4J(d)	CF-9.05-3/4J(e)
Profile number	0.002	0.002	0.002	0.0034	0.0034	0.0034	0.0034	0.0034
Radius ratio	0.547	0.547	0.547	0.528	0.528	0.528	0.528	0.528
Fin efficiency(Rectangular Profile)	0.58	0.58	0.58	0.57	0.57	0.57	0.57	0.57
Fin efficiency(Trapezoidal Profile)	0.57	0.57	0.57	0.56	0.56	0.56	0.56	0.56
Fin efficiency(Triangular Profile)	0.58	0.58	0.58	0.57	0.57	0.57	0.57	0.57
Total fin number	28484	48122	48122	44812	35763	25959	62870	45795
Heat transfer(Watt)	9684	16361	16361	14896	22531	16354	39608	28851

Three types of profile , Rectangular, Trapezoidal and Triangular, are investigated for heat exchangers. Trapezoidal profile has the lowest efficiency among others. The fin of *CF-9.05-3/4J(d)* has the best performance, typically due to the lowest Longitudinal tube spacing (P_c). Of course, heat transfer in the above table is ideal heat transfer. The results denote that although the rate of heat transfer using forced convection is very high compared with Radiation, but when the temperature of hot fluid is high this rate is a considerable value. Of course, in this situation , as it is mentioned previously, the flow arrangement play a key role.

Conclusion

On the common and popular types of the heat exchanger in cooling applications is circular finned-tube heat exchanger. Generally, transferring heat occur in three modes, conduction, convection, and radiation. When the temperature of working fluid is high the term of radiation will be important, because the rate of heat transfer is a function of " T^4 ". In this paper we analyzed eight types of circular finned-tube heat exchangers for this term. The results showed that when there is forced convection in the system transferring heat in the mode of radiation is negligible, and only when this term is important that there is natural convection or the temperature of hot stream is so high as well as the arrangement of the heat exchanger is parallel.

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