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COMPARATIVE THERMAL PERFORMANCE OF FLAT PLATE FINS AND INLINE STRIP FINS HEAT SINKS

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Abstract. Comparative results for the convective thermal resistance of continuous plate fins and inline strip fins heat sinks are presented based on correlations from the literature for the Nusselt number for the fins of both heat sink configurations. The purpose of this investigation is to compare the thermal performance of both heat sinks with the same base and fins height and thickness, under similar forced airflow cooling conditions. The Nusselt number heat transfer correlations from the literature considered the heat transfer from the fins, but did not include the convection from the bare heat sink base (not covered by the fins). Thus, this additional heat transfer from the heat sink base was evaluated and included in a new more realistic model and the results were compared with those from the literature models. The results showed that when compared to the plate fins heat sinks, the inline strip fins convective heat transfer coefficients were larger enough to compensate for their reduced heat transfer area, resulting in smaller convective thermal resistances. It was also verified that the convective thermal resistance decreases with the strip length, due to larger convective heat transfer coefficients on the fins. When the heat transfer from the heat sinks base was considered, there was an additional reduction of the heat sinks convective thermal resistance to the cooling fluid.

Keywords: Heat Sinks, Convective Thermal Resistance, Inline Strip Fins, Plate Fins

1. INTRODUCTION

Heat sinks are widely used in electronic equipment, with the purpose to enhance the convective heat transfer from a component with high heat flux to a forced flow of a cooling fluid (usually ambient air). They must be designed so that the components temperatures always remain within their reliable limits, as specified by their manufacturer. In spite of their large use in the electronics industry, the heat sinks thermal characteristics remain a topic of investigation due to their variety of geometry, size, operating conditions, and also due to distinct design goals (WEBB, 2007).

Two of the most investigated heat sink configurations are the parallel plates heat sinks and the aligned strip fins heat sinks, as indicated in Fig. 1, with the cooling flow parallel to the heat sink base. Along the years, many analytical, experimental and numerical investigations were performed to obtain the thermal resistance and the head loss of the cooling flow through the heat sink channels. The cooling fluid is usually the ambient air, mainly due to its availability, handling facility and the insulating electrical properties. Several analytical investigations were performed for the parallel plates heat sinks, mostly with laminar flow in the channels and the results were conveniently presented in the form of compact models, expressed as algebraic equations. Experimental and numerical investigations were also performed for these heat sinks and all the results have been compared in the literature, considering both laminar and turbulent flow in the heat sink channels. The strip fins heat sinks have been investigated mostly through experiments and numerical simulations for both laminar and turbulent flow in the interfin channels, but the results presented in the literature are not as conveniently grouped in the form of compact models as those for the plate fins.

Heat transfer results for the parallel plates heat sinks were presented for example by Sparrow *et al*. (1978), Kadle and Sparrow (1986), Sata, *et al.,* (1997), Jonsson and Moshfegh (2001) and Henríquez *et al*., (2007). These experimental and numerical investigations took into account the effects of distinct conditions of the flow entrance in the heat sinks, fins spacing, thickness and height, as well as the interfins cooling flow in the laminar and turbulent regimes.

Considering now inline strip fins heat sinks, the heat transfer was also reported by several investigators. Sparrow *et al.* (1977) reported numerical results for laminar flow in the fins channels and indicated that the partition of each fin into smaller strips reduces the flow and thermal boundary layers growth in each fin strip, thus reducing their convective thermal resistance. Ozturk and Tari (2008) investigated numerically the cooling of a CPU by means of aligned strip fins heat sinks, considering the complete computer chassis as the calculation domain. They investigated the effects of the number of fins and their distribution, the fin material and the base plate thickness. Al-Sallami *et al*. (2016) reported numerical results for the convective heat transfer from strip fins heat sinks, with inline and staggered arrangement of the fins. Their results included the effect of perforated fins to enhance the convective heat transfer and indicated that the

staggered arrangement provides larger heat transfer rates from the heat sinks, at the expense of larger pressure losses. Hong and Cheng (2009) presented numerical results for the convective cooling of water cooled offset strip fins heat sinks and investigated the effect of the fins geometry on the heat sink performance. They indicated that there is an optimum strip fin size to minimize the flow pressure drop and that it depends on the heat flux input and the maximum wall temperature.

The numerical investigations, as those reported previously, are useful to predict the heat sink performance from their detailed information about the flow and temperature fields, for a specific heat sink configuration and operating conditions. Additional numerical results may also be easily obtained to represent the effects of changes of the heat sink size and operating conditions. Then, a heat sink prototype, considering distinct design purposes, may be selected from the numerical simulations. The actual thermal behavior of the prototype may then be measured in laboratory experiments.

In order to reduce the number of numerical simulations, they may start from a heat sink configuration selected from compact thermal models usually expressed by algebraic correlations, as the Nusselt number for the heat sink cooling flow. These compact models are usually obtained from analytical or approximate solutions to models of the convective heat transfer and the flow in the heat sink channels. Thus, these models are useful in the first stages of a heat sink thermal design.

The purpose of the present work is to compare, by means of compact thermal models, the thermal resistance of two of the most common heat sink configurations: plate fins and inline strip fins, considering the same heat sink base, fins spacing and thickness. For each fin configuration, initially Nusselt number correlations were obtained from the literature (Teertstra *et al*. (1999, 2000)) for the cooling flow forced convection. These correlations considered only the convective heat transfer from the fins and were originally employed to obtain the thermal resistances from the heat sinks. The present work included an estimative to the heat transfer from the bare heat sinks base and a new value for the heat sink thermal resistance was obtained. Thus, the results from the literature and from the modified model were employed to compare the thermal resistances of both heat sink configurations.

2. COMPACT THERMAL MODELS

Compact thermal models were employed to compare the thermal resistance of two heat sink configurations indicated in Fig. 1: plate fins and inline strip fins. Forced convection airflow was assumed parallel to the base of both Aluminum heat sinks with the same base dimensions (LxW), fins height (H), thickness (t) and pass (b), as indicated in Tab. 1. For the inline strip fins, the ratio (S/P) of the slot width (S) to the slot pitch (P) was assumed equal to 0.5 in the present analysis.
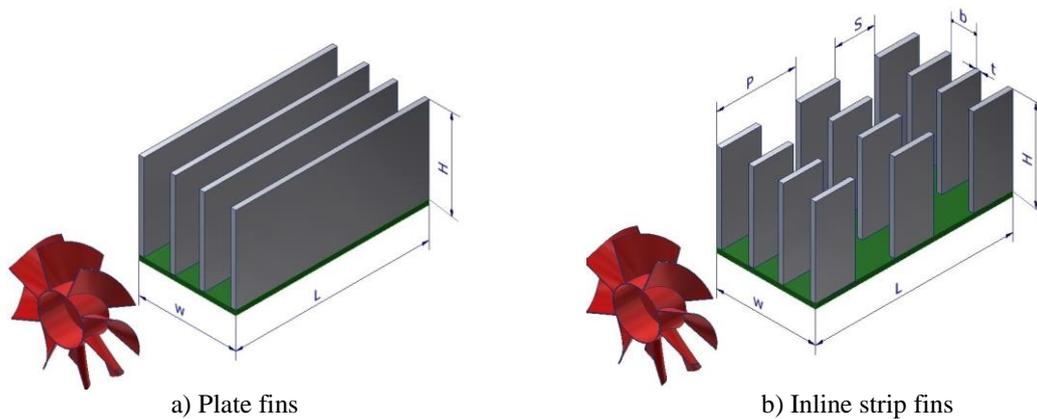

a) Plate fins   b) Inline strip fins

Figure 1. Geometry of the two heat sink configurations

Table 1. Geometric parameters and thermal conductivity

Parameter	Values
L	115 mm
H	49 mm
b	5,7 mm
t	1,25 mm
w	65 mm
K	200 W/m.K

2.1 Nusselt number for the plate fins heat sinks

For the plate fin heat sinks, the Nusselt number was obtained from the model developed by Teertstra, *et al.,* (2000). They presented a composite Nusselt number correlation based on the asymptote values for fully developed and for developing flow in isothermal channels. It was based on the formula presented by Churchill and Usagi (1972), as expressed by

$$Nu_{bi} = \left[\left(Nu_{fd} \right)^{-3} + \left(Nu_{dev} \right)^{-3} \right]^{-1/3} \tag{1}$$

In Eq. (1), the Nusselt number Nu_{bi} is defined by

$$Nu_{bi} = \frac{qb}{k(2LH)(T_w - T_i)} \tag{2}$$

The Nusselt number Nu_{bi} was based on the interfins spacing b and the analysis performed by Teertstra *et al.,* (2000) in a fins channel considering only the heat transfer from the fins area: A = (2LH). On the right side of Eq. (1), Nu_{fd} and Nu_{dev} are respectively the asymptote solutions for the fully developed and for the developing flow Nusselt numbers in the interfins channel, both based on the temperature difference between the isothermal fins surface (T_w) and the incoming cooling flow (T_i). They were expressed as

$$Nu_{fd} = \frac{1}{2} \text{Re}_b \text{ Pr} \tag{3}$$

$$Nu_{dev} = 0.664\sqrt{\text{Re}_b} \text{ Pr}^{\frac{1}{3}} \sqrt{\left(1 + \frac{3.65}{\sqrt{\text{Re}_b}} \right)} \tag{4}$$

In Eqs. (3) and (4), *Pr* is the cooling fluid Prandtl number and Re_b is a modified Reynolds number, defined by

$$\text{Re}_b = \frac{Ub}{v} \frac{b}{L} \tag{5}$$

Replacing Eqs. (3) and (4) into Eq. (1), the Nusselt number for ideal isothermal fins may be expressed as in Eq. (6).

$$Nu_{bi} = \left[\left(\frac{\text{Re}_b \text{ Pr}}{2} \right)^{-3} + \left(0.664\sqrt{\text{Re}_b} \text{ Pr}^{1/3} \sqrt{1 + \frac{3.65}{\sqrt{\text{Re}_b}}} \right)^{-3} \right]^{-1/3} \tag{6}$$

Assuming plate fins with adiabatic fins tip, their efficiency η_{pf} may be expresed as in Eq. (7):

$$\eta_{pf} = \frac{\tanh \sqrt{2Nu_i \frac{k_f}{k} \frac{H}{b} \frac{H}{t} \left(\frac{t}{L} + 1 \right)}}{\sqrt{2Nu_i \frac{k_f}{k} \frac{H}{b} \frac{H}{t} \left(\frac{t}{L} + 1 \right)}} \tag{7}$$

The actual convective thermal resistance for plate fins heat sinks may then be expressed as in Eq. (8):

$$R_{th} = \frac{b}{k(2LH) Nu_{bi}\eta_{pf}} \tag{8}$$

2.2 Nusselt number for the inline strip fins heat sinks

These heat sinks replace the single plate of a heat sink row by a number of smaller strip fins. They are used due to a reduction of the thermal boundary layer growth at each strip fin, thus increasing the heat sink average heat transfer coefficient (Webb, 2007). When compared to the plate fin, the strip fins in a row have however smaller heat transfer area. These two effects have opposing effects on this heat sink thermal resistance, as compared to the corresponding plate fin heat sink. The larger average convective coefficient contributes to smaller thermal resistance, but the corresponding smaller heat transfer area points in the opposite direction. Thus, one main purpose of this work is to compare the thermal performance of plate fins and strip fins heat sinks. The model for the strip fins heat sinks was based on a second work of Teertstra *et al.* (1999), where they considered that the average convective heat transfer coefficient for this heat sink was limited by a lower bound and an upper bound. The lower bound was associated to a channel length L_{LB} equal to the total length of all strips in a single row. Due to an approximate geometric expression used for the lower bound length L_{LB} in the mentioned reference, a small correction to this length was introduced in the present work, so that it was expressed by

$$L_{LB} = (L+S)\left(1 - \frac{S}{P}\right) \qquad (9)$$

The lower bound modified Reynolds number Re_{LB}, associated to the lower bound length L_{LB}, was related to the plate fin modified Reynolds number Re_b, defined by Eq. (5), as follows:

$$Re_{LB} = \frac{Re_b}{\left(1 - \frac{S}{P}\right)\left(1 + \frac{S}{L}\right)} \qquad (10)$$

A new ideal lower bound Nusselt number $Nu_{i,LB}$, associated to isothermal strip fins, was then evaluated as in Eq. (6), with the modified Reynolds Re_{LB}. The fins efficiency in this case was also evaluated assuming adiabatic fins tips with the lower bound length L_{LB} defined by Eq. (9).

$$\eta_{sLB} = \frac{\tanh\sqrt{2Nu_{i,LB}\frac{k_f}{k}\frac{H}{b}\frac{H}{t}\left(\frac{t}{(L+S)\left(1-\frac{S}{L}\right)}+1\right)}}{\sqrt{2Nu_{i,LB}\frac{k_f}{k}\frac{H}{b}\frac{H}{t}\left(\frac{t}{(L+S)\left(1-\frac{S}{L}\right)}+1\right)}} \qquad (11)$$

The upper bound length was equal to the length of a single strip fin: $L_{UB} = (P - S)$. The modified Reynolds number associated to this length, Re_{UB}, was also related to the plate fin modified Reynolds number Re_b, defined in Eq. (5):

$$Re_{UB} = \frac{Re_b}{\left(\frac{P}{L}\right)-\left(\frac{S}{L}\right)} \qquad (12)$$

The corresponding ideal Nusselt number $Nu_{i,UB}$ for the upper bound limit was evaluated from Eq. (6) with the value obtained for Re_{UB}. Assuming adiabatic fins tips, the fins efficiency for the upper bound limit, η_{sUB}, was obtained from

$$\eta_{sUB} = \frac{\tanh\sqrt{2Nu_{i,UB}\dfrac{k_f}{k}\dfrac{H}{b}\dfrac{H}{t}\left(\dfrac{t}{(P-S)}+1\right)}}{\sqrt{2Nu_{i,UB}\dfrac{k_f}{k}\dfrac{H}{b}\dfrac{H}{t}\left(\dfrac{t}{(P-S)}+1\right)}} \qquad (13)$$

Teertstra *et al.* (1999) presented experimental results for the average Nusselt number for strip fins heat sinks, considering *(S/P) = 0.5* and within the ranges 0.11 < *(P/L)* < 0.44 and 40 < Re_b < 180 and showed that they were within 12% of the mean value for the lower and the upper bound limits:

$$Nu_{bs} = 0.5\left(\eta_{sLB}Nu_{iLB} + \eta_{sUB}Nu_{iUB}\right) \qquad (14)$$

This prediction for Nu_{bs} was then used to evaluate the strip fin heat sink convective thermal resistance similar to Eq. (8), but using the actual heat transfer area of the strip fins channel:

$$R_{th} = \frac{b}{kNu_{bs}}\frac{1}{2(L+S)\left(1-\dfrac{S}{P}\right)H} \qquad (15)$$

2.3 Convection from the heat sinks base

In the present work, the convective heat transfer rate from the heat sinks base not covered by the fins to the cooling flow was also estimated and added to that of the fins in each heat sink channel. Following a procedure adopted by Webb (2007), the average Nusselt number at the base of each channel was assumed as that for the parallel flow to a flat plate with a length *L* equal to that of the heat sink:

$$Nu_L = 0,664\sqrt{Re_L}\,Pr^{1/3} \qquad (16)$$

In this equation, the Reynolds number Re_L and the Nussel number are based on the heat sink length L. The convective heat transfer from the channel base was then evaluated from the resulting heat transfer coefficient h_L as follows:

$$q_L = h_L(bL)(T_w - T_i) \qquad (17)$$

The total heat transfer rate from a heat sink channel was obtained from the sum of q_L and the convection heat transfer rate from the fins, evaluated from the previous correlations for the Nusselt number in each heat sink configuration. For the plate fin heat sinks, h_{bi} was obtained from Eqs. (2), and (6), while η_{pf} was obtained from Eq. (7):

$$q_{tpf} = q_f + q_L = \left[h_{bi}(2LH)\eta_{pf} + h_L(bL)\right](T_w - T_i) \qquad (18)$$

The thermal resistance associated to the total rate of convective heat transfer for the plate fins heat sinks was

$$R_{th,pf} = \frac{1}{h_{bi}(2LH)\eta_{pf} + h_L(bL)} \qquad (19)$$

For the strip fins heat sinks, the heat transfer coefficient h_{bs} was obtained from Eq. (14), where the lower bound and the upper bound fins efficiencies are already included.

$$q_t = q_f + q_L = \left[h_{bs}2(L+S)\left(1-\frac{S}{P}\right) + h_L(bL)\right](T_w - T_i) \qquad (20)$$

Thus, the total thermal resistance for the strip fins heat sinks, including the heat transfer from its base, may be expressed by

$$R_{th,sf} = \frac{1}{h_{bs} 2(L+S)\left(1 - \dfrac{S}{P}\right) + h_L(bL)}$$

(21)

Comparative results will be presented next for the two heat sink configurations (plate fins and strip fins), using the models presented by Teertstra *et al.* (1999, 2000), where just the heat transfer from the fins is evaluated, and also the modified model, taking into account the convection from the heat sinks base.

3. RESULTS AND DISCUSSION

Numerical tests were performed for heat sinks with the same geometry and material as that tested by Teertsra *et al.* (1999), in order to compare the present results with their experimental data for strip fins. Their heat sink had a length $L = 0,115$ m, fins spacing $b = 5,7 \ 10^{-3}$ m, height $H = 49 \ 10^{-3}$ m, and thickness $t = 1,25 \ 10^{-3}$ m. The heat sink thermal conductivity was $k = 200$ W/m.K and the number of rows of fins was $N = 7$. For the strip fins, a fixed ratio $(S/P) = 0.5$ was used in their experiments and in the present simulations, and the predictions were made with two distinct numbers of inline strips (N_S) in each fin row: $N_S = 5$ and $N_S = 10$. The numerical tests were performed for an average velocity in the interfins channels (U) in the range from 1 to 10 m/s, associated to the plate fins modified Reynolds number Re_b within the limits of validation of the correlations for the Nusselt number ($40 < Re_b < 180$).

The results for the average Nusselt number Nu_b at the fins for both heat sink configurations are presented in Fig. 2 as functions of the modified Reynolds number Re_b. For the plate fins heat sink, Nu_b represents the product of Nu_{bi} and η_{pf}, obtained respectively from Eqs. (5) and (6) and only the prediction results are presented, since there are no experimental data for this configuration. For the strip fins heat sinks, Nu_b represents the values obtained from Eq. (11a). All the numerical results show an increase of Nu_b with the Reynolds number Re_b, but for the plate fins the average Nusselt number is below the values obtained for the strip fins, as expected, according to the mentioned works of Sparrow *et al.* (1977) and Webb (2007). For the strip fins, the predicted Nusselt number also increases with the Reynolds number Re_b and with the number of strips ($N_S = 5$ and 10) in each fin row, due to the corresponding shorter strip lengths and the associated reduction of the thermal boundary layer growth in each strip. The experimental results presented by Teertstra *et al.* (1999) for the same numbers of strips are also included in this figure to give a comparison to the results of numerical simulations. It should be kept in mind that for the strip fins, the numerical predictions for the Nusselt number are the average of the upper and lower bounds, indicated by Eq. (11a). Fig. 2 shows that most of the experimental results were above the predictions for both numbers of strips ($N_S = 5$ and 10) but the relative deviations were within 12%.
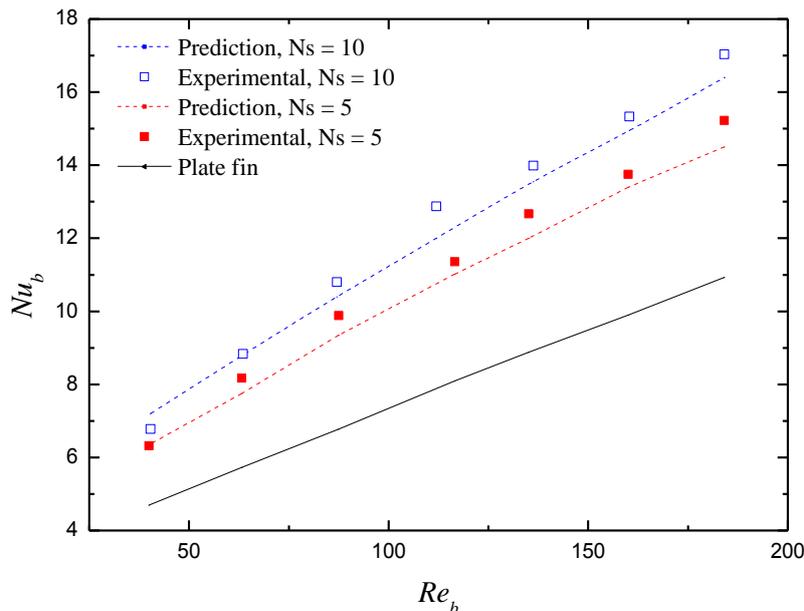

Figure 2. Average Nusselt number for the flat plate and the inline strip fins

The heat sinks convective thermal resistances were obtained from Eqs. (6a) and (11b) respectively for the plate fins and the strip fins when the heat transfer was evaluated only from the fins surfaces. When the heat transfer from the base

of the interfins channels was added, the plate fins thermal resistance was obtained from Eq. (15) and for the strip fins heat sinks it was obtained from Eq. (17). The results presented in Fig. 3 include both models for the plate fins and the strip fins heat sinks, showing that all thermal resistances decrease as the interfins flow average velocity increases. They also show that the strip fins have smaller thermal resistances than the plate fins, indicating that the strip fins higher heat transfer coefficients observed in Fig. 2 have a larger effect than their decrease of the heat transfer area, so that their thermal resistance decreases. The results follow the trend observed in Fig. 2 of an increase of the heat transfer coefficient with the number of strips (N_S), thus imposing smaller heat sink thermal resistance as N_S increases. Comparing the results from both models, it is observed that the inclusion of the heat transfer from the interfins base channel always decreases the heat sinks thermal resistance.
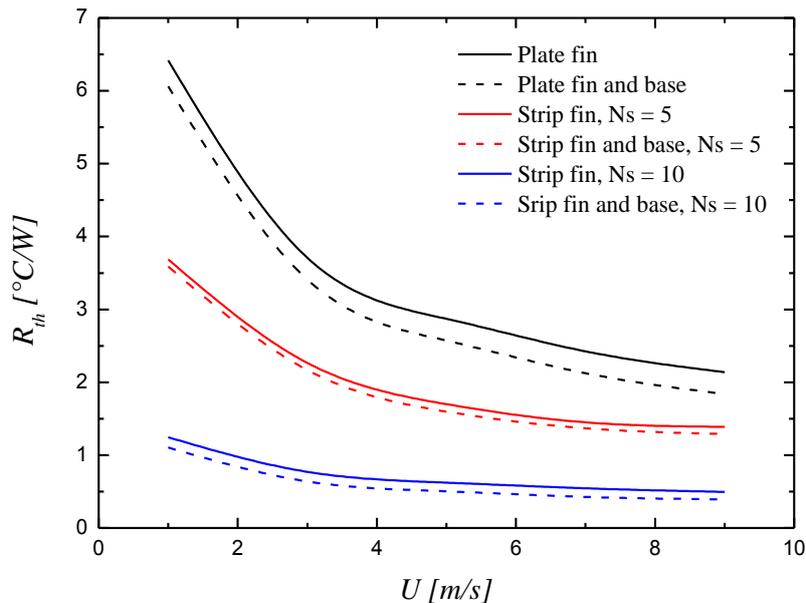

Figure 3. Effect of the interfin flow average velocity on the heat sinks thermal resistance

4. CONCLUSIONS

The thermal performance of a flat plate heat sink was initially compared to a similar strip fin heat sink by means of correlations for the fins heat transfer obtained from the literature, considering a range of the average flow velocity in the interfins channels. The results indicated that the convective heat transfer coefficients for both heat sinks increase with the interfins channels flow Reynolds number. The convection coefficients were smaller for the plate fins than for the strip fins, as expected from the analyses of Sparrow *et al.* (1977) and Webb (2007). The results also indicated smaller thermal resistances for the strip fins heat sink, indicating that their larger heat transfer coefficient had a more pronounced effect than the reduction of their heat transfer area, when compared to the plate fins of a similar heat sink. The inclusion of the convective heat transfer from the base of the interfins channels in the previous model indicated a decrease of both heat sinks thermal resistance, with the same trends. Thus, strip fins heat sinks have a better thermal performance than a similar plate fins heat sink under similar operating conditions. To have a broader picture for a heat sink selection, these results should however be compared with the power needed to pump the flow through the heat sink and the final selection should also be based on the design goals for a heat sink.

5. ACKNOWLEDGEMENTS

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