ANALYSIS OF THE EFFECT OF THE USE OF A CONVERGER IN AN AUTOMOTIVE HEAT EXCHANGER PERFORMANCE

Gusmão Dutra, getuliodutra@hotmail.com
FIAT Automóveis S.A., Departamento Experimentação Chassi-Térmica
Rodovia Fernão Dias, km 429 – CEP 32530-000 – Betim – MG

Rudolf Huebner, rudolf@ufmg.br
Departamento de Engenharia Mecânica, Universidade Federal de Minas Gerais
Campus Pampulha, Av. Antônio Carlos, 6627, 31270-901, Belo Horizonte - MG – Brasil

Abstract. *One problem regarding the heat exchanger project for automotive systems is doing it in an optimized way. The external air flow is not uniform along the heat exchanger surface due to the frontal vehicle design and the use of fans, with or without a converging section, used to drive the air. In the present work a test bench was developed to analyze the flow field and temperature distribution along the heat exchanger surface. The surface temperature was measured using a thermal camera and air temperature was measured using thermocouples. Water and air temperatures were measured to perform the energy balance. The flow velocity was measured using a thermal anemometer. All tests were performed at different wind conditions inside a climatic chamber with a roller dynamometer. The experimental results were compared with theoretical calculations and shown the flow field influence on the radiator efficiency. These results can be used as guidelines to modify the frontal vehicle design or the fan structure.*

Keywords*: Heat Exchanger, Energy Balance, Flow Distribution, Experimental*

1. INTRODUCTION

Heat exchangers are devices that provide the flow of thermal energy between two or more fluids at different temperatures (Kakaç, 1998). Compact heat exchangers such as plate-fin types and fin-and-tube are widely used for gas-gas or gas-liquid applications. Compact heat exchangers are employed in many industrial processes in chemical and petroleum engineering, refrigeration, air conditioning, aeronautics and automotive (Xie et al., 2008 and Dong et al., 2007).

The automotive industry is continuously involved in a strong competitive career to obtain the best automobile design in multiple aspects. The air cooled heat exchangers found in a vehicle have an important role in its weight and also in the design of the front-end module, which also has a strong impact on the car aerodynamic behavior (Oliet et al., 2007). Other factor to be considered is the increasingly restrictive laws regulating pollutant emissions from all kinds of automobiles, together with the continuous rising in petroleum price justify the enormous research effort devoted to optimize the efficiency of internal combustion engines (Lozano et al., 2008 and Wagner at al., 2001).

Compact heat exchangers have the advantage of being compacts, need a small volume; have a low weight, a high effectiveness and low cost. Very often these heat exchangers are used in cooling systems, with air as a cooling agent in cross flow over a tube bank. The form of the plate fins has been developed by numerous companies and widely investigated. Despite their long-term usage in industry, heat transfer and pressure drop are still investigated (Jacimovic et al, 2006).

Wang et al. (1997) and Wang et al. (2002) made experiments on the heat transfer and pressure drop characteristics of fin and tube heat exchangers. Junqi et al (2007) studied 11 cross-flow heat exchangers to get correlations for heat transfer and pressure drop evaluation. Rathod et al. (2007) use the classical effectiveness and number of unit transference (NUT) to evaluate the performance of flat finned tube fin heat exchanger with different fin surfaces but the results are limited to gas and water heat exchangers. Bougeard (2007) uses infrared thermography to investigate the local heat transfer coefficient in a plate fin and two-tube rows assembly.

The aim of this work is to evaluate the temperature distribution along the surface of a heat exchanger and to correlate it to its effectiveness. An experimental methodology was proposed to characterize the heat exchanger leading to a semi-empirical model. This model was used in conjunction with the thermography to evaluate the performance of heat exchangers using fans with or without convergers at different frontal wind velocities. Results show the converger influences the heat exchanger performance in a significant way as the wind velocity increases.

2. METHODOLOGY

2.1. Heat Exchanger Characterization

All tests were performed in laboratory inside a room, denominated climatic chamber, with controlled air temperature, humidity and velocity. Figure 1 shows a car inside the chamber.

Figure 1. Climatic chamber with a roller dynamometer (Fiat Automóveis S.A.,2008)

The radiator is a compact heat exchanger type plate fin and tube which has a set of pipes with a "U" shape and plate fins contacting every pipe. This radiator uses air as coolant fluid and a mixture of water and ethyl-glycol flowing inside the tubes. No information was provided by the heat exchanger manufacturer thus tests to characterize the device were performed. The heat exchanger was placed inside the climatic chamber. In this test the air velocity was measured using a thermal anemometer, the air temperature distribution along the heat exchanger surface was measured using thermocouples (type K) installed upstream and downstream the heat exchanger and a thermal camera was used to measure the radiator surface temperature. The mixture mass flow rate was evaluated through the energy balance between the air and the mixture because no device to measure this parameter was available.

Figure 1 shows a frontal view of the heat exchanger, the thermocouples positions and velocity measuring points.

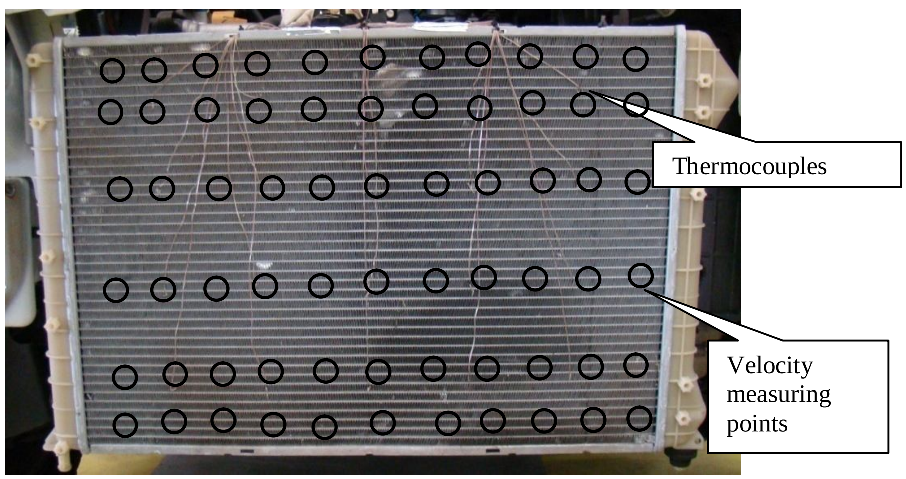


Figure 2 – Radiator frontal view

Eleven tests were performed to get the mixture mass flow rate as a function of the engine speed. The air temperature was 25°C, engine speed used were 900, 1400, 3000 and 4000 rpm. The resistance provided by the dynamometer was set in order to get an engine thermal rejection which provides a mixture inlet temperature between 50 and 90°C in the heat exchanger. All data were measured after the radiator inlet temperature stabilization which was characterized by a maximum variation of 0,5°C in two minutes.

The amount of energy transferred to the air is given by equation 1

$$q_{air} = \dot{m}_{air}\, c_{p-air} (T_{o-air} - T_{i-air}) \quad (1)$$

where m_{air} is the air mass flow rate (kg/s), $T_{i\text{-}air}$ represents the air inlet temperature (°C), $T_{o\text{-}air}$ is the air outlet temperature (°C) and $C_{p\text{-}air}$ is the air specific heat at constant pressure (J/kg.°C). The air mass flow rate was evaluated considering the summation of the air mass flow through every region where the velocity was measured.

The energy released by the water-ethyl-glycol mixture is given by equation 2

$$q_f = \dot{m}_f \, c_{p-f} (T_{i-f} - T_{o-f}) \qquad (2)$$

Where $\dot{m}_f$ represents the mixture mass flow rate (kg/s), $T_{i\text{-}f}$ is the mixture inlet temperature (°C), $T_{o\text{-}f}$ is the mixture outlet temperature (°C) and $c_{p\text{-}f}$ represents the mixture specific heat (J/kg.s). The mixture mass flow rate was evaluated considering the heat released by the mixture is completely transferred to the air.

Basically there are two distinct methodologies to design heat exchangers. The first uses the concept of log-mean temperature difference which is very effective when all inlet and outlet temperatures are available; otherwise a trial-error procedure is necessary. The second methodology is the method of number of transfer units (NUT) based on the concept of heat exchanger effectiveness (Kakaç, 1998).

The effectiveness (ε) is defined as the ratio of the actual heat transfer rate in a heat exchanger to thermodynamically limited maximum possible heat transfer rate. The actual heat transfer rate considering the effectiveness is given by equation 3.

$$q = \varepsilon C_{\min} (T_{i-f} - T_{o-air}) \qquad (3)$$

Where C_{min} represents the minimum capacity rate given by equation 4.

$$C = \dot{m} \; c_p \text{ (air ou water-ethyl-glycol)} \qquad (4)$$

The effectiveness is a function of the capacity rate ratio and NUT as expressed by equation 5

$$\varepsilon = f\left(\frac{C_{\min}}{C_{\max}}, NUT\right) \qquad (5)$$

The number of transfer units is given by equation 6

$$NUT = \frac{AU}{C_{\min}} \qquad (6)$$

Where A is the heat transfer area and U represents the overall heat transfer coefficient.

The actual heat transfer can also be given by equation (7)

$$q = UA(\Delta T_{lm}) \qquad (7)$$

where the log-mean temperature difference is defined below.

$$\Delta T_{lm} = \frac{\Delta T_i - \Delta T_o}{\ln(\Delta T_i / \Delta T_o)} \qquad (8)$$

The data collected in the first test was used to evaluate the heat exchanger effectiveness.

On a second test, the characterization of the heat exchanger was verified considering 14 tests performed inside the climatic chamber in accordance the FIAT normative.

2.2. Case Study – Converger Effect

The fan can be considered one of the most important components in the engine cooling system after the heat exchanger (radiator). The fan can be used with or without a component denominated converger. Figure 3 shows a fan with and without a converger.

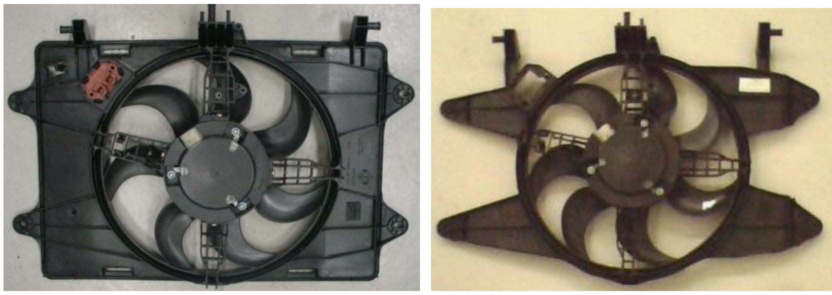


Figure 3 – Fan with converger (left) and without converger (right).

The converger is used to guide the air intake on conditions of little or none frontal wind. The drawback of its use is that the heat exchanger performance can deteriorate due to the fact that the converger acts as barrier at high speed wind. A study was performed using the effectiveness method considering the use of fan with and without the converger. Tests were performed inside the climatic chamber considering no frontal wind, a 30 km/h frontal wind and finally 72 km/h frontal wind. The same instrumentation used in the heat exchanger characterization were used in this test.

3. RESULTS AND DISCUSSION

3.1. Heat Exchanger Characterization

Figure 4 shows the mixture mass flow rate as a function of the velocity speed considering the results from the first experiment. The mass flow rate varies linearly with the engine speed and data shows a small dispersion around the linear adjusted profile.

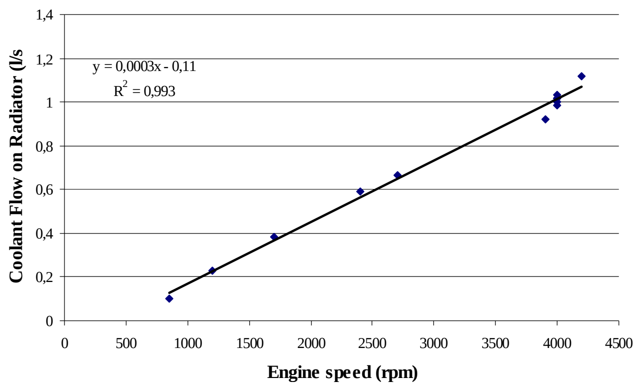


Figure 4 – Mixture mass flow rate as a function of engine speed

The effectiveness was also evaluated using data from the first test. The NUT value for each test is shown in figure 5 and a small dispersion around an average value of 1,2 can be observed, suggesting that effectiveness can be considered a function of the capacity rate ratio only.

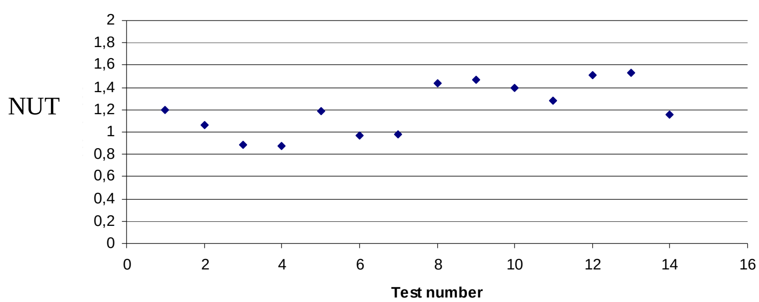


Figure 5. NUT variation along the tests.

Figure 6 shows the effectiveness as a function of the capacity rate ratio. Experimental data was adjusted by a non-linear equation. One equation was adjusted considering the air with the minimal capacity rate while the other considers the mixture with the minimal capacity rate. In the case of air the dispersion is bigger due to the amount of data using in the regression.

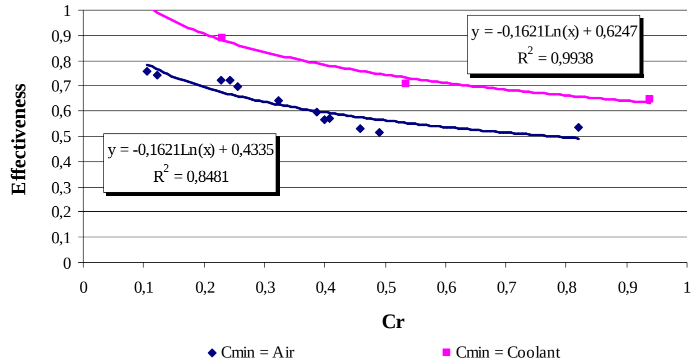
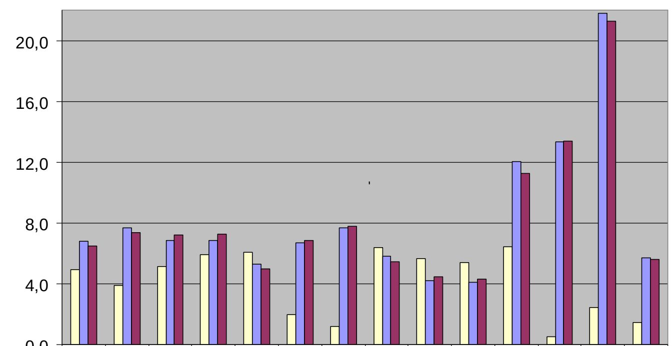


Figure 6. Effectiveness as a function of capacity rate ratio.

The model was validated considering different engine operation conditions. Figure 7 shows a comparison between experimental and theoretical results. The maximum error was of 6,4% which could be reduced using a more robust regression curve considering also the NUT effect.



	1	2	3	4	5	6	7	8	9	10	11	12	13	14
error (%)	4,9	3,9	5,2	5,9	6,1	2,0	1,2	6,4	5,7	5,4	6,4	0,5	2,5	1,4
T_{in}- T_{out} (Theoretical)	6,8	7,7	6,8	6,9	5,3	6,7	7,7	5,8	4,2	4,1	12,0	13,3	21,8	5,7
T_{in}- T_{out} (Measured)	6,5	7,4	7,2	7,3	5,0	6,8	7,8	5,5	4,4	4,3	11,3	13,4	21,3	5,6

Figure 7 – Comparison between experimental and theoretical data

3.2. Case Study – Converger Effect

Figure 8 shows the air velocity profile and radiator surface temperature considering a fan with converger operating without a frontal wind. The velocity distribution reveals a highly non uniform velocity profile. The velocity has a maximum value near the fan and tends to decrease near the radiator border and fan axis. The thermal image shows the temperature variation along the radiator surface. The temperature is higher near the water inlet and minimum at the outlet, but there is a temperature peak in the region on the center of the fan axis. Similar results are presented in figure 9 which corresponds to a fan without a converger and frontal wind.

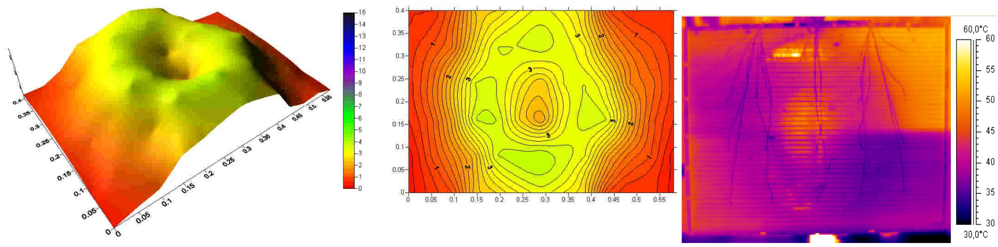


Figure 8 – Air velocity profile and surface temperature – Fan with converger operating without frontal wind.

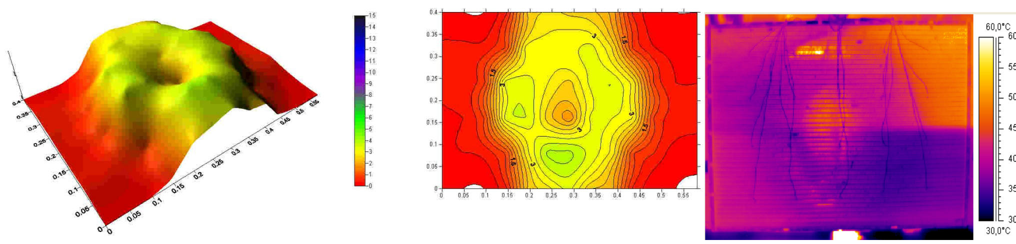


Figura 9 – Air velocity profile and surfece temperature - Fan witout converger operating without frontal wind

Figures 10 and 11 show the air velocity profile and radiator surface temperature considering a frontal wind of 30 km/h. The velocity distribution reveals a highly non uniform velocity profile but the fan without converger tends to induce the flow in a larger frontal area. In both cases the temperature distribution is more uniform due to the convection provided by the air velocity.

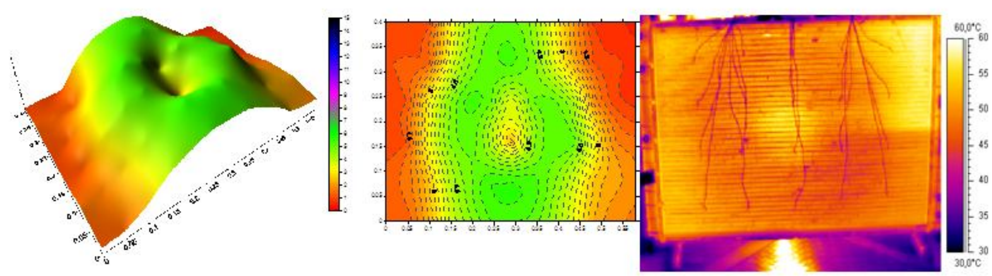


Figura 10. Air velocity profile and surface temperature – Fan with converger operating with frontal wind of 30km/h.

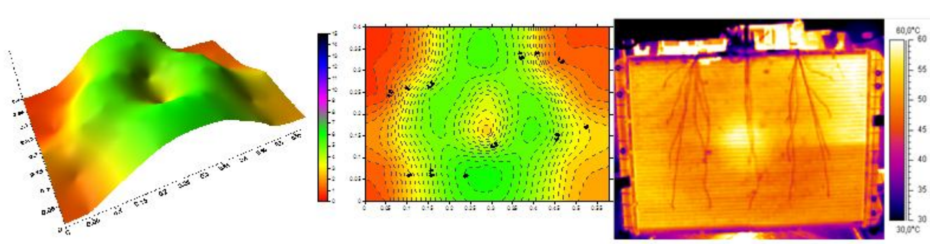


Figura 11. Perfil Air velocity profile and surface temperature – Fan without converger operating with frontal wind of 30km/h.

Similar results are presented in the case of frontal wind of 72 km/h, as show in figures 12 and 13.

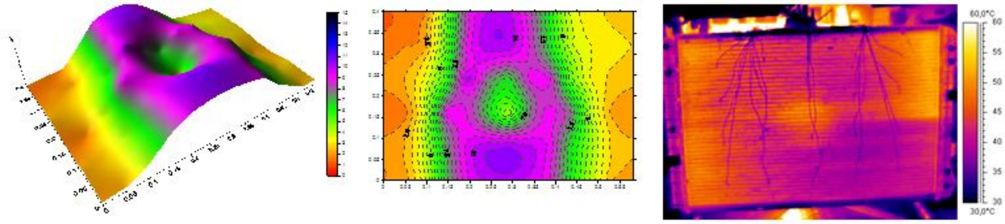


Figura 12. Air velocity profile and surface temperature – Fan with converger operating with frontal wind of 72 km/h

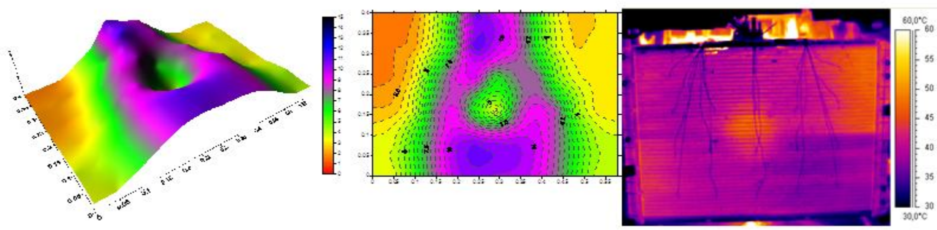


Figura 13. Perfil Perfil Air velocity profile and surface temperature – Fan without converger operating with frontal wind of 72 km/h

The mixture temperature variation was evaluated considering the experimental data and the effectiveness model developed previously Tables 1, 2 and 3 summarize the results.

Table 1. Results without frontal wind.

	Experimental		Calculated	
	With Converger	Without Converger	With Converger	Without Converger
Volumetric air flow rate (m³/s)	0,51	0,4	0,51	0,4
Inlet Temperaturer (°C)	60,6	60,5	60,6	60,5
Engine Speed (rpm)	900	900	900	900
Outlet Temperature (°C)	38,85	43,0	39,8	43,8
Temperature Varation (°C)	21,71	17,5	20,5	16,5
Error (%)			4,6	4,5

Table 2. Results with frontal Wind of 30km/h

	Experimental		Calculated	
	With Converger	Without Converger	With Converger	Without Converger
	Prova 8	Prova 1		
Volumetric air flow rate (m³/s)	0,7394	0,7837	0,7394	0,7837
Inlet Temperaturer (°C)	60,36	63,37	60,33	63,37
Engine Speed (rpm)	4000	4000	4000	4000
Outlet Temperature (°C)	54,53	56,55	54,3	56,19
Temperature Varation (°C)	5,82	6,81	6,5	5,5
Error (%)			4,9% (-)	6,4% (-)

Table 3. Results with frontal Wind of 72 km/h

	Experimental		Calculated	
	With Converger	Without Converger	With Converger	Without Converger
Volumetric air flow rate (m^3/s)	1,4	1,32	1,32	1,32
Inlet Temperaturer (°C)	52,27	57,65	54,47	57,65
Engine Speed (rpm)	4000	4000	4000	4000
Outlet Temperature (°C)	45,43	49,96	47,8	49,84
Temperature Varation (°C)	6,83	7,69	7,2	7,8
Error (%)			5,2% (+)	1,2% (+)

In condition without frontal wind both configurations present the same performance. The performance difference can be noted as the frontal velocity increases as show in tables 2 and 3. The heat exchanger using the fan without converger tends to present higher temperature variation thus more heat is removed. In the condition of no frontal wind the influence of the converger is higher than in the condition of frontal wind.

4. CONCLUSION

A heat exchanger used in the cooling system of an internal combustion engine was characterized. The effectiveness was expressed as a function of capacity rate ratio and was used to evaluate radiator performance at different conditions. Results show a good agreement between experimental and semi-empirical results. The performances of fans operating with and without a converger were analyzed. At low frontal velocities there is no difference in the heat exchanger performance, but as the frontal wind velocity increases the system without converger tends to be more effective. The influence of the converger is higher with no frontal wind, so its application is more necessary when cooling system has poor performance in idle condition. Thermal images were used to map the radiator surface temperature and the temperature profile reveals regions of high temperature due to constraints in the flow.

5. ACKNOWLEDGEMENTS

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