Modeling of Cold Plates for Power Electronic Cooling

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Abstract

Cold plates are used to cool components with heat fluxes of around 100 W/cm^2. The power electronic cooling library contains models for cold plates and heat dissipation models for some power electronic components. The designed numerical models can be used to pre-design cold plates using different structures and finned areas to be able to reach the required junction temperature. In this paper simulation results are compared with test rig data. The test rig is designed to investigate cold plates using different coolants at changing heat fluxes, fluid mass flow rates and inlet temperatures.

Keywords: cooling; cold plate; simulation; water propylene glycol; power electronics

1 Introduction

Thermal management for power electronics is essential in order to provide efficient and durable components. Different cooling technologies are used to remove the dissipated heat from the component depending on the heat flux and the level of power electronics integration.

Power conversion modules which consists for example of several IGBTs (Insulated Gate Bipolar Transistors) and diodes become more and more integrated so that high heat fluxes up to 200 W/cm^2 are the result [1].

Forced convection air cooling can be used for heat fluxes of around 50 W/cm^2 using finned cooling elements to spread the heat. The cooling element needs space to be mounted so that an integrated design can often only be achieved using a heat pipe to transport the heat from the heat source to the heat sink.

Increasing the possible heat flux has two effects to the power module, the junction temperature can be reduced and the output current of a power module can be increased using the same quantity of silicon. To achieve this aim cold plates with a liquid coolant can be used. The achievable heat flux for cold plates with baseplate design is about 100 W/cm^2, integrated cold plates which have no baseplate and therefore no thermal grease layer achieve nearly 150 W/cm^2. With even more integrated micro heat exchanger and water jet impingement devices fluxes up to 250 W/cm^2 are possible. Higher fluxes of about 1000 W/cm^2 can be achieved with phase change [2].

The cold plate models described in this paper focuses on cold plate designs shown in figure 1. The power electronics is mounted on the drill side of the cold plate with a thin layer of thermal grease as interface material. The fluid part of the cold plate can have different designs. Two important designs, an integrated tube and thinned channel design are included in the library. The heat loss of some power electronics can be calculated using simplified models.

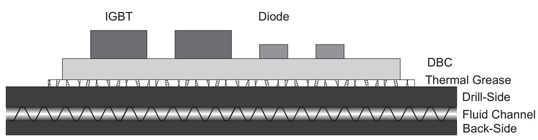

Figure 1: Schematic configuration of power electronic on the cold plates

The cold plate model can be used in the pre-design phase to predict the junction temperature distribution of several power modules placed on one cold plate. The important factors to determine the temperature distribution are the heat conduction in the solid part of the plate and the convective heat transfer in the tube or channel. Correlations for heat transfer coefficients in tubes and finned channels are used. Therefore it is required that the geometric design of the channel and the flow pattern are known.

The thermal spreading effect in the solid can only be taken into account using for example a finite volume approach with a sufficient discretization. Nevertheless these cold plate models cannot replace analysis of the fluid flow in the cold plate especially at the in- and outlet and deflections. These questions have to be an-

swered using CFD tools and measurements.
The optimisation of the temperature distribution can be performed using different solid thickness and channel designs combined with a change of the power module position.
The used cold plate for the model validation is an off the shelf component, see figure 2.

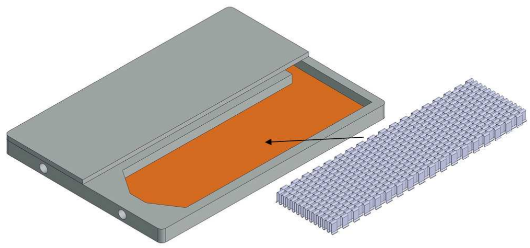

Figure 2: Cold plate which is used at the test rig

2 Cold plate test rig

The numerical model of the cold plate is simplified to a certain extent. Not every physical process can be simulated. The heat transfer is based on correlations, the bend in the cold plate for example is not taken into account and the overall approach is based on finite volumes for the fluid channels. Therefore the made assumptions and simplifications have to be checked using test rig data.
A test rig has therefore been erected at the Institute of Thermo-Fluid Dynamics to investigate different types of cold plates, see figure 3. It is possible to use different one-phase liquids at atmospheric pressure. The cooling power of the used chiller is 10 kW at 20 °C, its outlet temperature can be adjusted by an internal PI controller. The mass flow, which is measured with a coriolis mass flow meter (accuracy of 0.05 % of the defined end value and 0.1 % of the actual measured value), has to be manual adjusted using a valve. The cold plate can be bypassed using a 3-way valve.
At the in- and outlet of the cold plate the pressure and the temperature are measured using a difference pressure sensor (accuracy of 0.1 % of the end value, range=0...10 bar) and thermocouples type T. All thermocouples have been calibrated to achieve a accuracy of $\pm$0.1 K.
The cold plate used for gathering data for this paper is equipped with six thermocouples type T in the fluid stream, six thermocouples type T at the bottom surface of the plate right below the fluid sensors and six thermocouples type T in grooves on the cold plate top located above the fluid sensors, see figure 4.

To get a well-defined and homogeneous heat flux, the power electronic is replaced by heat dummies with different base areas. The required heat flux can be adjusted with heater elements which are located at the top part of the heat dummy. The used heater elements can have a heating power up to 250 W, which can be adjusted with a power control. The homogeneous heat flux is achieved by using a massive aluminium block. The inertia is therefore not comparable with real power electronic components. For transient tests a real power electronic component has to be used.
The two main problems which have to be solved is on the one hand the temperature measurement with an accuracy of $\leq$0.1 K and on the other hand the correct positioning of the sensors to get as accurate test data as possible. The problems which occur from these challenges have not been solved completely.
A further problem is the use of thermal grease to fill the gaps between heat dummy and the cold plate. The temperature distribution is directly dependent on the thermal grease distribution and thickness. A different interface material has to be used for areas of this magnitude.

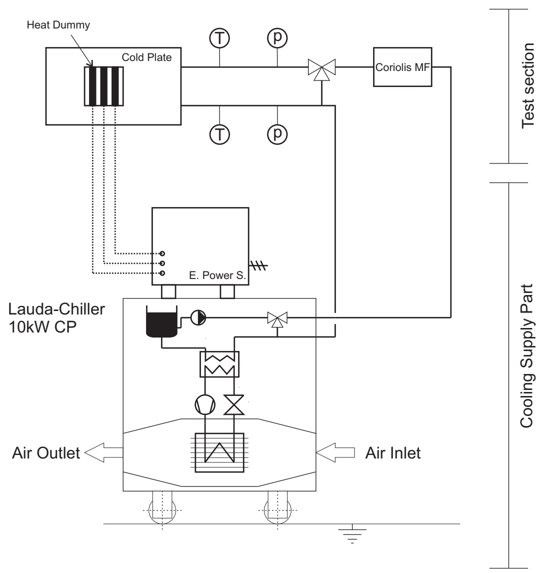

Figure 3: Test rig schematic

3 Power Electronic Cooling-Library

3.1 Structure of the Library

The PEC-Library is basically divided into two parts: one part contains models for the heat loss of power electronics and one part models of the cold plates.

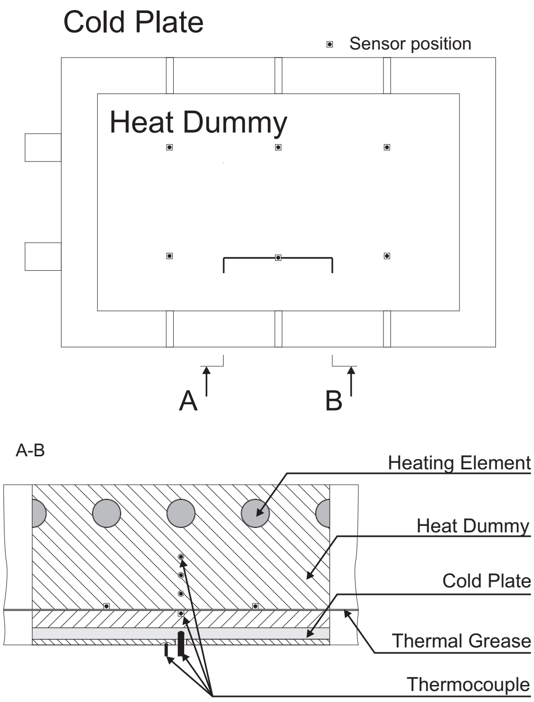

Figure 4: Schematic of measurement locations and heat dummy

3.2 Models of the Power Electronics

The aim of the power electronic models is to calculate the power loss of power electronic devices. However, one basic problem when modelling the cooling of power electronics are the significantly different time constants between the cooling devices and the power electronics itself. Whereas the thermal time constants of the cooling devices are several seconds, the time constants of voltage and electrical current in the power semiconductors are rather μs, due to the high switching frequencies of several kHz. Instead of using detailed simulation models calculating the transient loss of each semiconductor device, as it was done for example in [4], a simpler approach is used here, based on the characteristics available in data sheets. The average power loss of one IGBT implemented in a three-phase inverter using pulse-width modulation control based on the comparison of sine-waves and sawtooth waves can then be calculated for one operating point depending on the power consumption of the load and the switching frequency using an approximation presented in [3], eq. 1-3.

$$\begin{aligned} P_{loss,on-state} &= \frac{1}{2}\cdot\left(\frac{U_{CE0}}{\pi}\cdot\widehat{I}_L+\frac{R_{CE}}{4}\cdot\widehat{I}_L^2\right) \\ &\quad +m\cdot cos(\phi)\cdot\left(\frac{U_{CE0}}{8}\cdot\widehat{I}_L+\frac{R_{CE}}{3\pi}\cdot\widehat{I}_L^2\right) \end{aligned} \tag{1}$$

$$P_{loss,switching} = f_s\cdot(W_{on}+W_{off})\cdot\frac{U_{cc}}{U_{ref}} \tag{2}$$

$$P_{loss} = P_{loss,on-state}+P_{loss,switching} \tag{3}$$

Hereby is U_{CE0} the collector-emitter threshold voltage, $\widehat{I}_L$ the peak value of the load current, R_{CE} the resistance between collector and emitter, m the modulation depth, ϕ the phase angle between voltage and current, f_s the switching frequency, W_{on} and W_{off} the switching losses and U_{ref} a reference voltage. Similar approaches are used for MOSFET-based semiconductor devices and diodes implemented in rectifiers.

Detection of hot spots inside the power electronics is therefore not possible, since the temperature distribution within the device is not modelled. The junction temperature of the transistors is obtained using the thermal resistance between junction and case, delivered in data sheets.

The power electronic models exchange information with the cold plate models via a connector called PowerLossPort. Since a discretized cold plate model requires not only the heat losses of the power electronics but also their dimensions and positions on the cold plate, the dimension and position vectors relative to the fixed coordinate system of the cold plate have to be handed over from the power electronic models to the cold plate models which is done using the connector definition below:

```
connector PowerLossPort_a
flow Modelica.SIunits.HeatFlowRate heatFlowRate;
Modelica.SIunits.Temperature temperature;
input Modelica.SIunits.Length dimension[2];
input Modelica.SIunits.Length position[2];
end PowerLossPort_a
```

The corresponding connector PowerLossPort_b is defined analogue using the prefix output instead of input.

3.3 Cold Plate Models

Basically two different types of models exist: lumped models and discretized models of the cold plates. The simple lumped models of the cold plates are intended to be implemented in large systems. The aim is to get a rough idea of the case and the junction temperature and to correctly model the time delay of the coolant as well as the capacitance of cold plate and power electronics, which is important in order to develop control strategies.

However the lumped approach is for instance not suitable if there is a large temperature increase of the coolant in the cold plate, resulting in an inhomogeneous surface temperature distribution or if there is a high heat load with a small contact area. For the latter the temperature spreading in the cold plate material becomes relevant for the calculation of the surface temperature of the cold plate and the junction temperature of the power electronics. Therefore the library also comprises more detailed models of the cold plates, based on finite volumes, allowing to discretize the cold plates in all three coordinate directions.

There are basically two types of cold plates: either a pipe coil or a finned channel is embedded in metal. Therefore the base elements comprise one ConductivityElement, modelling the thermal conduction in the cold plate material and one ConvectivityElement, modelling the fluid control volume. The ConductivityElement describes the transient heat conduction in an isotropic cuboidal element in all three dimensions, using one capacitance connected via six thermal resistances to the heat ports of the model (see figure 5). The ConvectivityElement describes the mass and energy balance of the coolant. The ConvectivityElement used for embedded coils also comprises a wall model, such allowing to have different materials for the coils and the cold plate itself. All convectivity elements are based on the ModelicaFluid model PartialDistributedFlow_pLumped [ModelicaFluid]. Each element has two fluid ports and two heat ports, so that heat transfer can only occur in one direction perpendicular to the flow direction.

Heat transfer and pressure drop models are implemented for convective flow in pipes and in finned channels. The implemented Nusselt correlation for finned channels is based on the following form

$$\mathrm{Nu} = C \cdot \mathrm{Re}^{x} \cdot \mathrm{Pr}^{1/3} \tag{4}$$

The medium models used are compressible fluids (extending from PartialLinearFluid from the Modelica Standard Library 2.2). Even though the model also runs with other liquids, i.e. extending from Modelica.Media.Incompressible.TableBased, it is far more efficient to use compressible fluids, especially with higher disretization and when using the models in a closed loop.

Model PowerLossDistribution

The heat transfer to the cold plate elements (conductivity and connectivity elements) is realised using the connector HeatPort with the connector definition

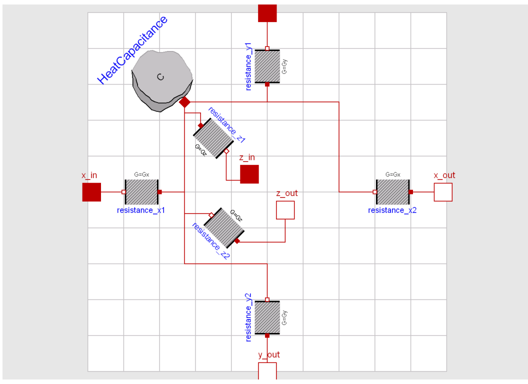

Figure 5: ConductivityElement in Modelica

below:

```
connector HeatPort
flow Modelica.SIunits.HeatFlowRate heatFlowRate;
Modelica.SIunits.Temperature temperature;
end HeatPort
```

Since the power electronic models use the PowerLossPort as an interface, a model is needed, which maps the connector PowerLossPort to the connector HeatPort. Since an arbitrary number of power electronics can be connected to the cold plate, this model is also responsible for the distribution of the heat losses of the different power electronic components to the cold plate elements. Due to the possibility of arbitrarily positioning the power electronic models on the cold plate, it is possible that some cold plate elements are only partly covered by one or more power electronics. So for every cold plate element a contact area has to be calculated which is done using the information in the position and dimension vectors. Besides the information committed by the interface the model requires two arrays (dx_Coldplate and dy_Coldpate), storing the size of the elements in x-direction and in y-direction respectively.

The contact area is obtained by first determining the position of the electronic. It is checked whether the starting or the end point is between the starting and the end point of one element. The result of this query is stored in a matrix with boolean indices. If the starting point of the electronics is located in the middle of one element, the starting point of the element is marked false, and the end point of the element is marked true. Is one element completely covered, starting and end point are marked true. Using this boolean matrix it is possible to determine the contact area for each power electronic component with respect to one cold plate element. Using the specific heat load of each power

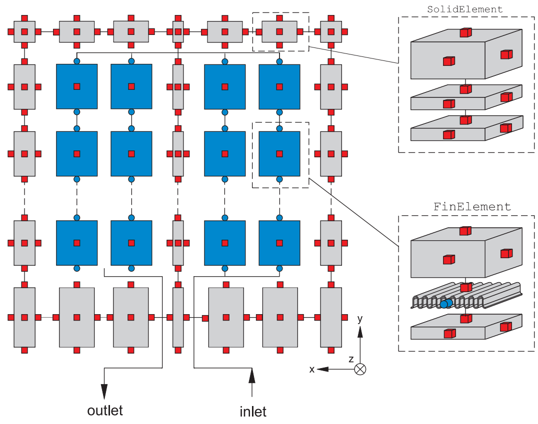

Figure 6: Discretized model of example cold plate

electronic the heat transferred to each cold plate element can be calculated by superposition of the heat loads from different power electronic components [7]. The model PowerLossDistribution is used in every cold plate model.

Example cold plate

In this library up to now three commercial available cold plates are implemented: one tube cold plate and two finned cold plates. Exemplary the implementation of one of the finned cold plate is presented here. The cold plate consists of two parallel finned channels, connected in series (see figure 1) and embedded in an aluminium plate. The external dimensions are known (298 mm x 179 mm x 13 mm), as is the channel height and the width of the the drill-through area. All other geometric parameters are unknown, especially the design of the in- and outlet areas and of the deflection at one end.

In the model, the plate is discretized three times in z-direction (corresponding to the height of upper and lower plate and the channel); the discretization of the channels in x- and y- direction can be chosen arbitrarily (see figure 6). The model is built using SolidElements (containing an array of ConductivityElements) and FinElements (containing one ConvectivityElement and two ConductivityElements). The arrays of SolidElements are then connected with the arrays of FinElements using for-loops.

4 Results

Simulations were performed in order to see the effect of the discretization on the result. Two heat loads (750 W each) were applied (for size and position see figure 7). For discrete elements which are only partly covered by heat loads, the heat load applied on this discrete element is assumed to be homogeneous. Therefore a too rough discretization underestimates the highest surface temperature, as can be seen in figure 7. The number of discrete elements is of course more important for punctual loading than for homogeneous distributed heat loads.

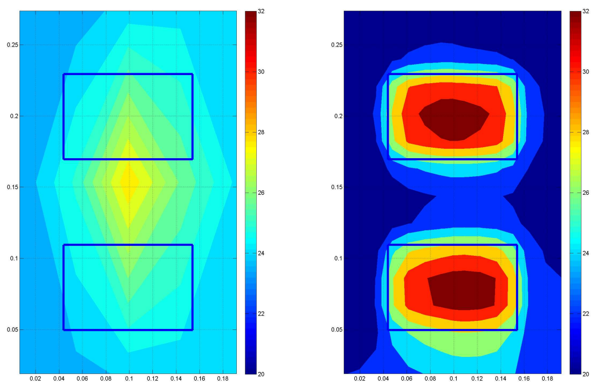

Figure 7: Discretization with x=5 and y=3 (left), x=15 and y=14 (right)

The example cold plate was also tested on a test rig, in order to ensure that the Nusselt correlation, where the parameters *x* and *C* can be adjusted using information from the manufacturer for water as coolant, also hold for other coolants. The measurement results presented in this section were obtained for the following boundary conditions:

- homogeneous heat load of 1550 W,
- mixture of water (40%) with propylene glycol (60%) as coolant,
- different mass flow rates, from 30 g/s to 350 g/s,
- four different inlet temperatures, from -8°C to 60°C
- all results are stationary results.

Figure 8 shows the pressure loss in the cold plate for different inlet temperatures and mass flow rates. The pressure loss increases with decreasing temperature due to the higher viscosity. For 60°C the relation $\Delta p \propto \dot{m}^2$ holds, indicating a complete turbulent flow in the cold plate. For low temperatures the flow in the channels itself is probably laminar, but the flow in the inlet and outlet parts as well as in the 180° deflection may still be turbulent, resulting in a total pressure loss with laminar and turbulent parts (the

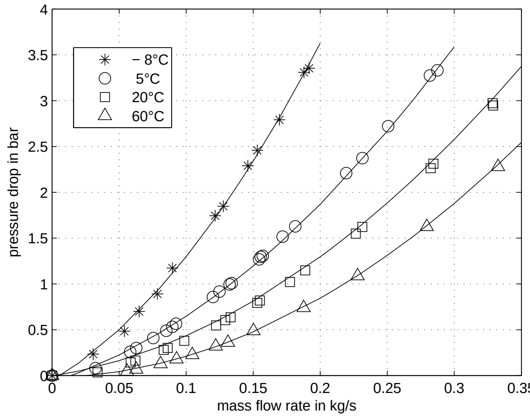

Figure 8: Pressure loss in the cold plate

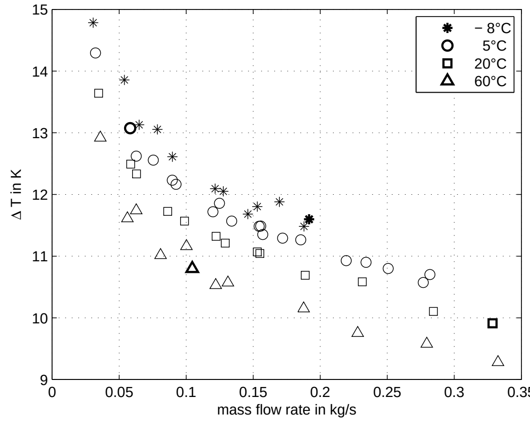

Figure 9: Difference between surface and fluid temperature for different mass flow rates and inlet temperatures

quadratic interpolation for -10°C inlet temperature indicates $\Delta p = 49 \cdot \dot{m}^2 + 8.6 \cdot \dot{m}$).

In figure 9 the temperature difference between the surface and the fluid temperature is plotted for different mass flow rates and inlet temperatures. The surface temperature increases (i.e. the heat transfer coefficient decreases) for decreasing mass flow rates and decreasing temperatures. For low temperatures the heat transfer coefficient decreases due to the decreasing thermal conductivity and due to the high viscosity, causing a larger boundary layer of the fluid. It can be also stated, that even for very low Reynolds numbers the heat transfer coefficient depends significantly on the Reynolds number.

Simulations were performed using the cold plate model corresponding to the tested cold plate under the same boundary conditions. The discretization was x = 6 and y = 7, the factors C and x of eqn. 4 were 0.9 and 0.7 respectively. The results for the surface-fluid temperature difference are plotted together with the measurement results in figure 10. First a significant offset can be observed between measurements and simulations. However this is due to the fact, that with the method used to measure the surface temperatures, not the real surface temperature is measured but a temperature between the upper surface temperature of the cold plate and the lower surface temperature of the heat loads. But even when shifting the simulation results (figure 11) there is a discrepancy between measurements and simulations for the combination of small mass flow rates and low temperatures. This is due to the fact that the same Nusselt-correlation is used for all Re-numbers, however for low temperatures the Reynolds numbers are very low (below 50), indicating a laminar flow, despite the staggered grid. [6] suggest a lower x for laminar flow. It is even possible that for Reynolds number below 10, convection is no longer the dominating factor, but conduction (which is not considered in the model) becomes more important.

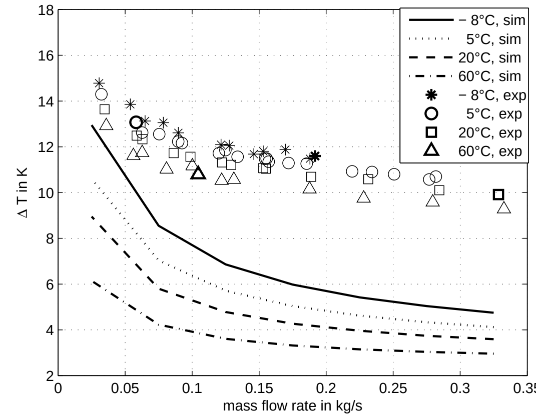

Figure 10: Difference between surface and fluid temperature for different mass flow rates and inlet temperatures

5 Summarising and outlook

The presented numerical model is suitable to predesign cold plates. Based on the first results for the example cold plate one can state that the used modeling approach is suitable. Although the geometric structure of the flow channel is not completely known, the implemented heat transfer correlations for staggered finned channel show a good agreement for a wide mass flow and temperature range. However experiments with well-known geometries will be carried out in or-

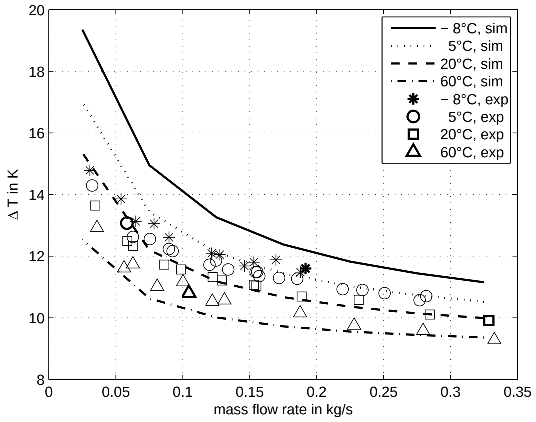

Figure 11: Difference between surface and fluid temperature for different mass flow rates and inlet temperatures, offset of the simulated temperatures of 6.5 K

der to better verify the correlations, especially for very low Reynolds numbers. Additionally more different cold plate layouts will be implemented into the library.

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