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STUDY OF CONVECTIVE CONDENSATION IN A THERMOSIPHON LOOP

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ABSTRACT

In this article, we focus on the condenser of a loop thermosiphon designed to cool power electronic component. The objective here is to condense Novec 649, our chosen working fluid for this loop. It is a fluid recently developed by 3M, which is known for having low environmental impact and non-flammable. We first present a theoretical analysis with the calculation and the hypotheses leading to the design of the multi-tubular condenser. Then we present a full size thermosiphon built for experimental validation. A discussion then addresses some of the design hypotheses.

Three main parameters are studied : the tilting angle of the condenser (from horizontal to vertical orientations), the temperature of the coolant and finally the mass flow effect at different saturation temperatures. In our setup, we dissipate up to 2.4 kW at the evaporator level. The produced vapor is then condensed in the heat exchanger using cold water flowing at countercurrent. A number of measurements are made via thermocouples and pressure sensors located at both ends of the condenser to measure the average heat exchange coefficient.

1 INTRODUCTION

Condensation is an important process that is being found in many industrial applications, such as refrigeration, air conditioning and energy conversion in power plants. Its growth is fueled by the demand for better efficiency especially now that air cooling technologies have reached their limits. For example, regarding power electronics cooling, the miniaturization and densification of electronic components in confined spaces lead to an increase in heat power fluxes that are becoming more and more difficult to extract. Two-phase cooling provides a better and more efficient alternative for future systems.

However, condensation is a very complex phenomena, and although many studies have produced analytical solutions that fit very well in some special conditions, most of the understanding comes from experimental observation through empirical or phenomenological models. This is due to the intrinsic nature of two-phase flows, the interaction and dependency of the liquid and vapor phases on flow conditions, such as angle of inclination, mass flux, temperature, thermo-physical characteristics, all

NOMENCLATURE

A	[m ²]	Section tube area
c	[m]	Tube circumference
D	[m]	Diameter
f	[-]	Friction factor
g	[m s ⁻²]	Gravity constant
G_k	[kg m ⁻² s ⁻¹]	Surface mass flow of phase k
H	[W m ⁻² K ⁻¹]	Heat transfer coefficient
h_{LV}	[J kg ⁻¹]	Latent heat of evaporation
j	[m s ⁻¹]	Superficial phase velocity
J_G	[-]	Dimensionless gaz velocity
J_G^T	[-]	Transition dimensionless gaz velocity
L	[m]	Condenser length
li	[m]	Interface line length
\dot{M}	[kg s ⁻¹]	Mass flow rate across the interface
P	[Pa]	Pressure
Pe	[-]	Peclet number
Pr	[-]	Prandtl number
\dot{Q}	[W]	Heat dissipation per tube
Q_{V_k}	[m ³ s ⁻¹]	Volumetric flow of phase k
Re	[-]	Reynolds number
Re_{LO}	[-]	Liquid only Reynolds number = $\frac{GD}{\mu_l}$
S	[m]	Heat exchange surface
T	[K]	Temperature
U	[m s ⁻¹]	Phase velocity
x	[-]	Mass fraction
z	[-]	Axis direction of the flow

Special characters

α	[-]	Void ratio
ρ	[kg m ⁻³]	Fluid density
δ	[m]	Liquid thickness in annular flow
θ	[-]	Inclination angle
τ	[Pa]	Friction
μ	[Pa.s]	Dynamic viscosity
λ	[W m ⁻¹ K ⁻¹]	Thermal conductivity
ϕ	[]	Two phase

Subscripts

l	Liquid
v	Vapor
i	Interfacial
w	Water
sat	Saturation
int	Internal
ext	External
SS	Stainless steel
p	Wall

of which affect the efficiency of the condensation process.

Most studies on the matter address horizontal and vertical condensation. Yet, of the few papers which present experimental investigations on inclined condensation[1-8], almost all show that

it is much more efficient to condense at tilted angle compared to a purely horizontal or vertical orientation. Thus, the objective will be to check these findings in the frame of a complete thermosiphon, for a specific fluid.

1.1 Previous work

Meyer et al. [2] and Lips and Meyer [3] studied condensation of R134a with a mass flux ranging from 100 to 300 kg m⁻² s⁻¹. They tilted a 8.34 mm tube between +90° and -90°. Maximum heat transfer was found hovering around -15° and -30°. The lowest heat transfer was reached for the vertical downward position. It was also found that heat transfer increased with decreasing mass flux.

Mohseni et al. [4] also experimented with R-134a. Compared to the previous study, Mohseni made tests with tubes similar to that of Meyer et al. [2], the angle range was also similar and the mass flux was of the same order of magnitude (from 53 to 170 kg m⁻² s⁻¹). Yet, the maximum heat transfer was found to be very different from that in the Lips and Meyer study, at +30°. It should be noted that they both agreed on the minimum heat transfer coefficient at -90° and that Mohseni made measurement every 30° while Lips and Meyer made measurement every 15°.

Emiw et al. [11] studied the effect of the temperature difference and inclination angle at low mass fluxes. They used a circular tube and varied the angle between +90° and -90°. While temperature difference was not found to have an effect in the upward or the downward vertical position, it was inversely correlated to the heat transfer coefficient regarding downward inclined flow and had no major effect when it came to upward inclined flow.

Del Col et al. [5] used R-134a and R-32 in their experiments. However, they used a square channel of an equivalent hydraulic diameter of 1.23 mm. The inclination angle was also varied between +90° and -90°. The major findings being that heat transfer decreased with inclination below a certain mass flow rate threshold. They produced a numerical method to predict and calculate this threshold value.

Lyulin et al.[12] studied the condensation of ethanol in a 4.8 mm circular tube, with a laminar flow. At 58 °C and with a mass flux ranging from 0.24 and 2.04 kg m⁻² s⁻¹ they found that the heat transfer coefficient was maximum between -15° and -35°, and that the heat transfer was inversely correlated to the temperature difference between the vapor and the cold source.

Wang and Du [6] condensed water in 4 tubes with diameters of 1.94, 2.8, 3.95, and 4.98 mm. The inclination angles ranged from 0° to -45°. They found that tube diameter affected largely the efficiency of the condensation. Heat transfer coefficient was maximum at 45° in larger tubes and minimum in the smaller ones.

Tepe and Mueller[7] experimented with methanol. Inclinations were limited to three values (0°, -15° and -90°). At low heat flux, maximum heat transfer was also found at -15°. Minimum heat transfer was found at the downward vertical position.

Finally, Xing et al. [8] condensed R134a with a mass flux ranging from 199 to 699 kg m⁻² s⁻¹. They tilted a 14.81 mm diameter tube between +90° and -90°. Maximum heat transfer was

found at both -15° and +30°. The effect of the inclination angle was also found to increase with decreasing mass flux. Based on the Froude number, they produced a formula that can calculate the variation of the heat transfer coefficient with the inclination angle, knowing its value in the horizontal position.

1.2 Application case

The design and making of the thermosiphon loop is part of a larger project in power electronics. Converters are designed to handle up to 800 kW of electric power each. They are based on inverter structures which can dissipate up to 8 kW of heat. Each inverter is made up of two half-bridge structures that can dissipate 4 kW each. The thermosiphon is evaluated as a replacement for the current water cooling system. Some key aspects of the half bridge structures are summarized below.

Efficiency of "half-bridge" (worst case): 99 % (98 %)
 Power to be dissipated (worst case): 1.8 kW (4 kW)
 Baseplate setpoint temperature : 70 °C
 heat flux (worst case): 4.3 W cm⁻² (9 W cm⁻²)

2 Design of the condenser

2.1 Two-phase flow configuration

The condenser will be subject to a two-phase flow. The behavior of both phases and their interfaces can take multiple forms, depending on flow conditions. The main way to identify patterns still rely on "flow maps". Overall, seven main flow patterns have been identified. They are indexed in "flow maps" such as the example given in Figure 1. They have been developed by observing the vapor and liquid flow distribution in a purely experimental approach, using common fluids and small tubes. They are used to characterize the transition regions between flow patterns and to get an idea about the flow pattern before any attempt of making an analytical model. The flow maps use logarithmic scales and either use dimensionless coordinates or the superficial flow velocity j_k for each phase defined as:

$$j_k = \frac{Qv_k}{A} = \frac{G_k}{\rho_k} \quad (1)$$

where : $G_l = G(1 - x)$ and $G_v = Gx$

In our case, the condenser is cooled using water in a counter-current flow system. With our chosen multi-tubular condenser structure, two main design parameters are to be defined: the number of tubes and the internal diameter of each one. The length of the condenser L is arbitrarily set at 1 m. We assume that the vapor flow is equally distributed among all tubes in a permanent flow hypothesis. Regarding the internal diameter, 5 common values are evaluated: 12, 16, 18, 20 and 30 mm. For each of them, we calculate both liquid and vapor superficial velocities (eq. 1) in order to guess the flow pattern. The use of a map made for air\water flow to predict a two phase flow of a new refrigerant is due to lack of such maps for refrigerants. Figure 2

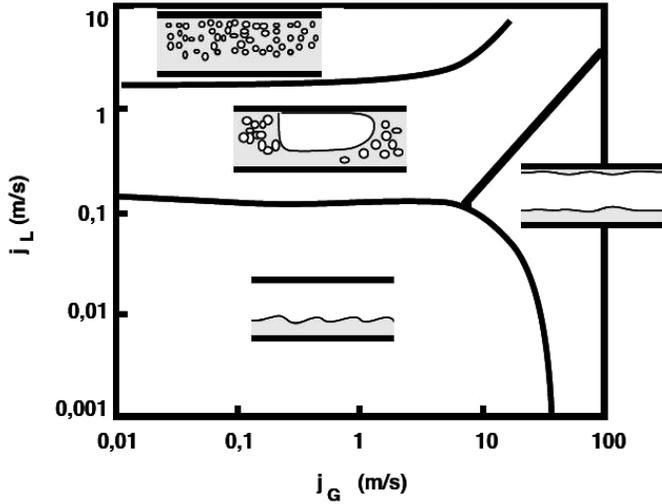


Figure 1. Air/water flow map in a horizontal tube [13]

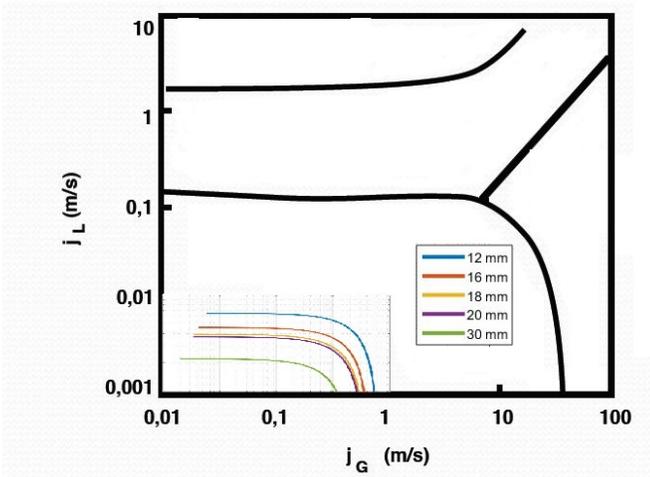


Figure 2. Superficial velocities indicating that flow will be stratified

predicts a stratified flow for the horizontal position for all chosen diameters.

The next step is to solve the equations of conservation of mass, momentum and energy. This will allow us to calculate the maximum condensed power per tube and thus the number of tubes required in the condenser. The results being angle-dependent, we will focus the theoretical developments in this paper to the horizontal position (only for simplification purposes). We will use a two-fluid system since it is expected to have a stratified flow. The goal here is to primarily calculate the void and mass fraction. The equations of conservation are written as follows:

$$\frac{d\rho_v \alpha U_v}{dz} = \dot{M}_l \quad (2)$$

$$\frac{d\rho_l(1-\alpha)U_l}{dz} = -\dot{M}_l \quad (3)$$

$$\begin{aligned} \frac{d\rho_l(1-\alpha)U_l^2}{dz} = & -(1-\alpha)\frac{dP}{dz} + \frac{\tau_{il}li}{A} \\ & + \frac{\tau_{pl}c_{pl}}{A} + \rho_l(1-\alpha)g \sin \theta - \dot{M}_l U_l \end{aligned} \quad (4)$$

$$\frac{d\rho_v \alpha U_v^2}{dz} = -\alpha \frac{dP}{dz} + \frac{\tau_{iv}li}{A} + \frac{\tau_{pv}c_{pv}}{A} + \rho_v \alpha g \sin \theta + \dot{M}_l U_l \quad (5)$$

$$\frac{\partial [\rho_v \alpha H_v U_v + \rho_l(1-\alpha)H_l U_l]}{\partial z} = GhLV \frac{dx}{dz} = -\frac{\dot{Q}_p S_p}{A} \quad (6)$$

We use equation 3 and 2 to eliminate the pressure gradient, this will give us a void fraction equation to solve :

$$\begin{aligned} \frac{d\alpha}{dz} G^2 \left(\frac{x^2(1-\alpha)}{\rho_v \alpha^2} + \frac{(1-x)^2 \alpha}{\rho_l(1-\alpha)^2} \right) = \\ \frac{dx}{dz} G^2 \left(\frac{x(1-\alpha)}{\rho_v \alpha} + \frac{(1-x)\alpha}{\rho_l(1-\alpha)} \right) - \alpha(1-\alpha)(\rho_l - \rho_v)g \sin \theta \\ + (1-\alpha) \frac{\tau_{pv}c_{pv}}{A} + \alpha \frac{\tau_{pl}c_{pl}}{A} - \frac{\tau_{iv}li}{A} \end{aligned} \quad (7)$$

Heat is being transferred from condensing NOVEC649 vapor to the water coolant through internal tubes made from stainless steel. Dissipated heat can be calculated as follows :

$$\dot{Q}_p = \frac{\Delta T L}{\left(\frac{1}{H_\phi \pi D_{int}} + \frac{1}{H_{water} \pi D_{ext}} + \frac{\log \frac{D_{ext}}{D_{int}}}{2\pi \lambda_{SS}} \right)} \quad (8)$$

For the outer flow of the water coolant, we use the Gnielinski [10] correlation. As for the inner two phase flow condensation, we will use the heat transfer set proposed by Cavallini et al. [9]. The set of equations defined in the paper and needed to calculate heat transfer are defined as follows :

$$\begin{aligned} \text{if } J_G \leq J_G^T, H_\phi = H_D \\ \text{else } H_\phi = H_{Annular} \end{aligned}$$

with the following definitions :

J_G and J_G^T are two dimensionless parameters defined in [9] as :

$$J_G = \frac{xG}{(gD\rho_V(\rho_L - \rho_V))^{0.5}}$$

$$J_G^T = \left[\left(\frac{7.5}{(4.3\chi_{it}^{1.111} + 1)} \right)^{-3} + C_T^{-3} \right]^{-\frac{1}{3}}$$

Where $C_T = 2.6$. It should be noted that this value is given for refrigerants like CFC's, HCFC's and HFC's. NOVEC649, being an HFO and a new fluid on the market, some caution should be taken upon the calculated results.

$$H_{Annular} = 0.023Pr_l^{0.4}Re_{LO}^{0.8} \left(\frac{x(1-\alpha)}{\rho_v\alpha} + \frac{(1-x)\alpha}{\rho_l(1-\alpha)} \right)$$

$$\left[1 + 1.128x^{0.817} \left(\frac{\rho_l}{\rho_v} \right)^{0.3685} \left(\frac{\mu_l}{\mu_v} \right)^{0.2363} \left(1 - \frac{\mu_v}{\mu_l} \right)^{2.144} Pr_l^{-0.1} \right] \quad (9)$$

$$H_{Strat} = 0.725 \left[1 + 0.741 \left(\frac{1-x}{x} \right)^{0.3321} \right]^{-1}$$

$$\left[\frac{\lambda_l^3 \rho_L (\rho_l - \rho_v) g h_{LV}}{\mu_l D \Delta T} \right]^{0.25} + (1-x)^{0.087} 0.023 Pr_l^{0.4} Re_{LO}^{0.8} \frac{\lambda_l}{D} \quad (10)$$

$$H_D = \left[H_{Annular} \left(\frac{J_G^T}{J_G} \right)^{0.8} - H_{Strat} \right] \left(\frac{J_G}{J_G^T} \right) + H_{Strat} \quad (11)$$

Finally the Lockhart Martinelli coefficient is calculated as [9]

$$\chi_{it} = \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\rho_V}{\rho_L} \right)^{0.5} \left(\frac{\mu_L}{\mu_V} \right)^{0.1}$$

2.2 Numerical Results

The goal was to determine the minimal number of tubes that would allow to condense all the vapor coming from the evaporator. The maximum coolant temperature is set to 313 K, while the maximum saturation temperature of NOVEC649 when dissipating 1.8 kW is set at 323 K which gives us a maximum of 10 °C temperature difference upon condensation. Finally, Thermodynamic data relative to NOVEC649 were acquired thanks to the NIST REFPROP 10.0 Database.

Figure 3 gives an example of solving the void fraction equation over the 12 mm tube diameter, compared to the Lockhart-Martinelli correlation for void fraction[14]. The difference does not exceed 5 % in absolute value.

Figure 4 shows the minimum number of tubes required, as a function of their diameter. As expected, the diameter is inversely

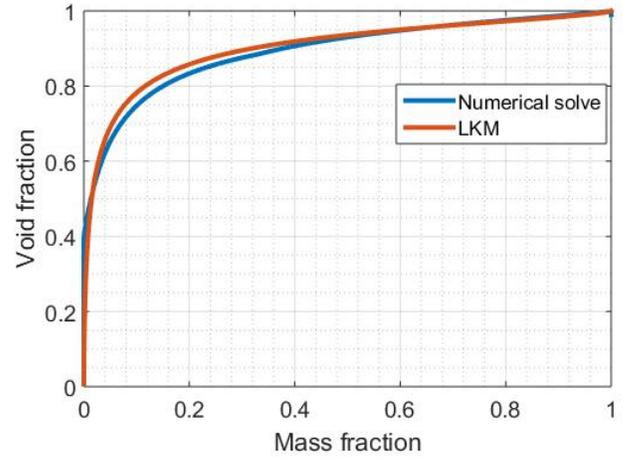


Figure 3. Comparison of the void fraction as a function of mass as predicted by the Lockhart-Martinelli correlation and as calculated with the numerical model.

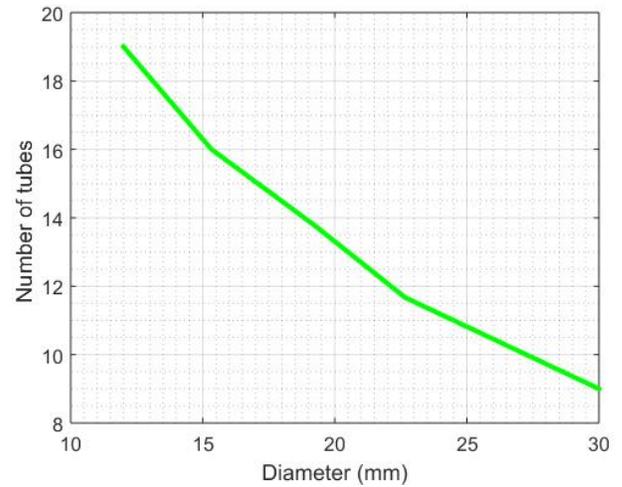


Figure 4. Number of necessary tube to have a complete condensation

correlated with the number of tubes. Figure 5 shows the volume taken by the condenser, also considering the volume of the water coolant. Here, the volume is directly correlated with the diameter and the smallest condenser is the one with the smallest diameter. Combining the two criteria, we choose the condenser with the smallest tube diameter (12 mm) to be built.

3 Early experimental results

In our thermosiphon loop, the criteria used to evaluate the efficiency of the condenser is the temperature difference between the outlet of the condenser and the outlet of the water coolant. A smaller value translates to a high heat transfer coefficient and thus a more efficient condensation. We measure this value in a permanent state defined by the stabilization of temperatures for 30 min within the margin of error which are 0.2 °C for our T type thermocouple. For each angle of inclination, we dissipate from 300 W to 2.4 kW in our evaporator. The corresponding vapor

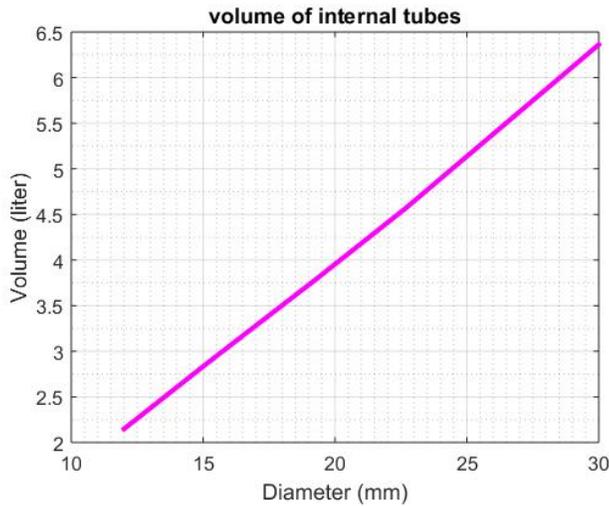


Figure 5. Total volume of the internal condenser tubes

	Mass flux g/s	G per tube kg/m ² /s	Inlet vapor Re
300	3,33	1,55	795
600	6,67	3,1	1591
900	10	4,65	2387
1200	13,34	6,21	3183
1500	16,68	7,76	3979
1800	20	9,31	4775
2100	23,35	10,86	5570
2400	26,7	12,42	6388

mass flow for each case is listed in the table below. These values were calculated based on the thermodynamic properties of the fluid :

3.1 Angle and mass flow effect

The figure 6 shows the temperature difference measured for different angles and different mass flows. The figure clearly show that raising the mass flow through dissipating more power in the system raises the temperature difference needed to condense the fluid for all angles. As more mass flow is passing through the tubes, the flow is more turbulent and the exchange is enhanced but the energy being dissipated also raises which explains the increase in the temperature difference.

As for the inclination effect, the angle that were tested are 0°, 15°, 30°, 45°, 60° and 90° in the downward flow position. Analysing figure 6 shows that the inclination has no effect at laminar flow (up to 600 W dissipated), as we raise the power and thus the mass flux, the inclination effect comes into play as the vertical position has the most temperature difference and thus the worst dissipation rate compared to other angles. With a difference of 1.8 °C compared to the horizontal position when dissipating 2.4 kW. This corresponds to almost 20 % increase

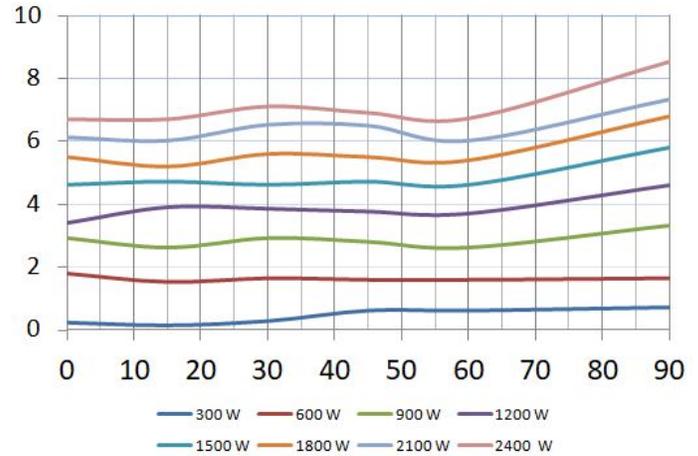


Figure 6. Temperature difference between the outlets of the fluid and the coolant as function of angle of inclination for different dissipated powers in the system.

compared from the minimum.

4 Conclusion

We have made a loop thermosiphon as part of a solution used to cool down electronics, we are currently studying all aspects of the loop. This paper focuses only on the condenser and the condensation process. We have showed here the calculation that have been made as a first step to get an idea about the dimensions in order to make the condenser. Figure 7 shows the condenser that's been made for the tests, NOVEC649 coming from the evaporator is condensed and the heat is dissipated by the counter-current external flow of water. Analysing the first results, we have seen that increasing mass flow increases the temperature difference needed for the process. the effect of inclination increase with mass flux and the worst angle corresponds to the vertical position. This result constitute the early measured results for this test bench. More tests will be done to verify these observations.

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Figure 7. Tilted condenser as part of the thermosiphon device

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