

Design of an Experimental Test Bench for Flow Boiling Heat Transfer Studies in ORC-Relevant Operating Conditions

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Abstract:

Flow boiling heat transfer studies have traditionally focused on parameters such as fluid type, mass flux, heat flux, saturation temperature, and tube diameter. However, most available data and correlations were developed for refrigeration applications and do not reflect the operating conditions typically encountered in Organic Rankine Cycle (ORC) systems. As ORC technology becomes increasingly relevant for small-scale power generation and cogeneration, growing evidence shows that widely used correlations for flow boiling in horizontal circular tubes often fail to predict heat transfer accurately under ORC-relevant conditions, particularly at high heat fluxes and saturation temperatures. Additional comparisons between independent datasets obtained under nominally similar operating conditions also reveal that discrepancies between experimental heat transfer coefficients may range from small to substantial, highlighting the limited availability of inter-facility reproducibility studies in flow boiling research. Another source of uncertainty is the heating methodology used in the experiments, as electrically and fluid-heated test sections produce different thermal boundary conditions, which may affect the measured heat transfer coefficient. To address these challenges, the present work proposes a test bench design specifically conceived for ORC-relevant flow boiling conditions. The facility incorporates two interchangeable test sections, one of which is electrically heated and the other is fluid-heated by thermal oil. This configuration enables direct comparison of heating boundary conditions under matched operating conditions. The test bench was designed to operate over a broad range of mass velocity, heat flux, and saturation temperature, and to support both validation against existing literature data and future studies with low-global-warming-potential working fluids. It is anticipated that the proposed test bench will enhance the physical interpretation of flow boiling behavior and contribute to the development of more reliable heat transfer correlations for ORC evaporators.

Keywords:

Flow Boiling Heat Transfer; Heat Exchanger; Heat Transfer Coefficient; Organic Rankine Cycle; Smooth Tube.

1. Introduction

Two-phase flow processes, such as boiling and condensation, are fundamental to the operation of thermodynamic systems in applications such as refrigeration and power generation. Flow boiling is particularly important because it enables high heat fluxes to be transferred at relatively small temperature differences, leading to heat transfer coefficients that are typically much higher than those observed in single-phase convection [1]. For this reason, flow boiling plays a central role in the design of compact, high-performance evaporators. Most previous studies in horizontal circular tubes have focused on the influence of fluid type, mass flux, heat flux, saturation temperature, and tube diameter on the two-phase flow boiling heat transfer coefficient (h_{tp}) [2]. However, most experimental databases and predictive correlations reported in the literature were developed and validated under operating conditions representative of refrigeration systems [3]. These conditions differ significantly from those encountered in other applications in which flow boiling is equally relevant, particularly in Organic Rankine Cycle (ORC) evaporators, where higher saturation temperatures, higher heat fluxes, and different geometric requirements are often involved [4].

This limitation is particularly important because ORC systems provide an environmentally and economically attractive means of converting low- and medium-temperature heat into useful power [5]. Typical heat sources include solar thermal, geothermal, biomass, and a wide range of waste heat streams from industrial processes and engine exhaust gases [6]. In addition, ORC systems can be integrated into combined heat and power configurations, supplying both electricity and useful thermal energy. In all these applications, the evaporator is one of the most critical components, and its design depends strongly on the accurate prediction of flow boiling

heat transfer coefficients [7]. Nevertheless, several studies have shown that widely used correlations for horizontal circular tubes can exhibit significant deviations when applied to ORC-relevant operating conditions, particularly for fluids such as R245fa at high reduced pressures [8] and high heat fluxes [9].

Figure 1 illustrates this issue using experimental results obtained under conditions that are more representative of ORC evaporators. In Figure 1(a), R245fa experimental data [4] show that increasing heat flux does not simply shift the magnitude of h_{tp} , but also modifies its variation with vapour quality, indicating a change in the underlying physics between nucleate and convective boiling mechanisms. In Figure 1(b), R1233zd(E), a low-global-warming-potential alternative to R245fa, exhibits a similarly strong sensitivity to saturation temperature in the high-temperature range [10]. In both cases, the Liu and Winterton (1991) [11] correlation is unable to reproduce the experimental trends with acceptable accuracy, as reflected by the large mean relative absolute deviations (*MRAD*) shown in Figure 1. Although only one correlation is presented here, this behaviour is consistent with the broader difficulty reported in the literature when refrigeration-based correlations are extended to ORC-relevant conditions [12].

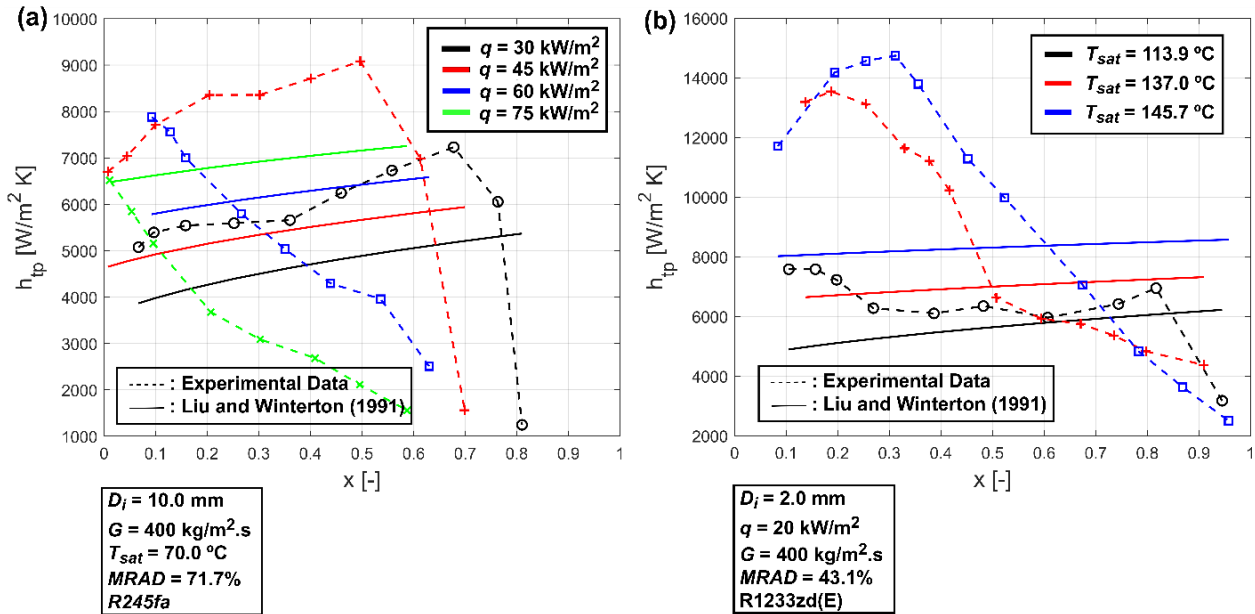


Figure 1. Experimental h_{tp} versus vapour quality (x) compared with Liu and Winterton (1991) correlation: (a) R245fa at $D_i = 10$ mm, $G = 400$ kg/m².s, $T_{sat} = 70$ °C, and varying heat flux; (b) R1233zd(E) at $D_i = 2$ mm, $G = 400$ kg/m².s, $q = 20$ kW/m², and varying saturation temperature. Dashed lines denote experimental data and solid lines denote correlation predictions.

From a physical standpoint, these discrepancies can be explained by the change in dominant heat transfer mechanisms across the operating range. At high reduced pressures and high heat fluxes, nucleate boiling tends to become increasingly important because the latent heat of vaporization decreases, surface tension is reduced, and bubble nucleation and detachment are promoted [13]. However, as the saturation temperature approaches the critical temperature, the liquid viscosity decreases, and the liquid film adjacent to the wall becomes more easily disrupted, which can promote early dryout and a subsequent reduction in h_{tp} [12]. By contrast, under typical refrigeration conditions, lower saturation temperatures and lower heat fluxes favour a stronger contribution from convective boiling, with the increase in vapour quality enhancing the forced convection of the two-phase mixture. These differences help explain why correlations developed predominantly from refrigeration datasets often fail to capture both the magnitude and the shape of the heat transfer coefficient curves observed in ORC applications.

Moreover, to examine the variability of experimentally determined h_{tp} values, a focused review was conducted to identify datasets reported by different authors and obtained in independent experimental facilities under nominally comparable flow boiling conditions. Two sets of experimental data were found for which such a comparison is feasible. Figure 2 presents these comparisons for R245fa together with the values predicted by the Liu and Winterton (1991) correlation [11], which is included only as a reference benchmark.

In Figure 2(a), both datasets were obtained in stainless-steel electrically heated test sections. The internal diameters were 2.3 mm and 3.0 mm, while the total heated lengths were 464 mm [14] and 900 mm [15], respectively. In addition to these geometric differences, the mass velocity was not exactly the same in the two studies. Despite these differences, the experimental h_{tp} values remain relatively close over the entire vapour-quality range, yielding a mean relative absolute deviation (*MRAD*) of 5.5% between the two datasets. By

contrast, the Liu and Winterton (1991) correlation was substantially more sensitive to those differences, with the mean absolute relative deviation increasing from approximately 5% to 15%.

The comparison shown in Figure 2(b) concerns two datasets obtained in electrically heated stainless-steel tubes with the same internal diameter of 10 mm. The total heated lengths employed were 2000 mm [16] (red colored case) and 800 mm [9] (black colored case), respectively, and the main difference between the experimental conditions was the imposed heat flux and saturation temperature. These data span a wide range of vapour quality, including regions above $x = 0.7$, where dryout-related effects are expected and where conventional correlations are known to fail in capturing the abrupt decrease in h_{tp} . In this case, the *MRAD* between the experimental datasets reaches 45%, indicating a substantially larger inter-study discrepancy.

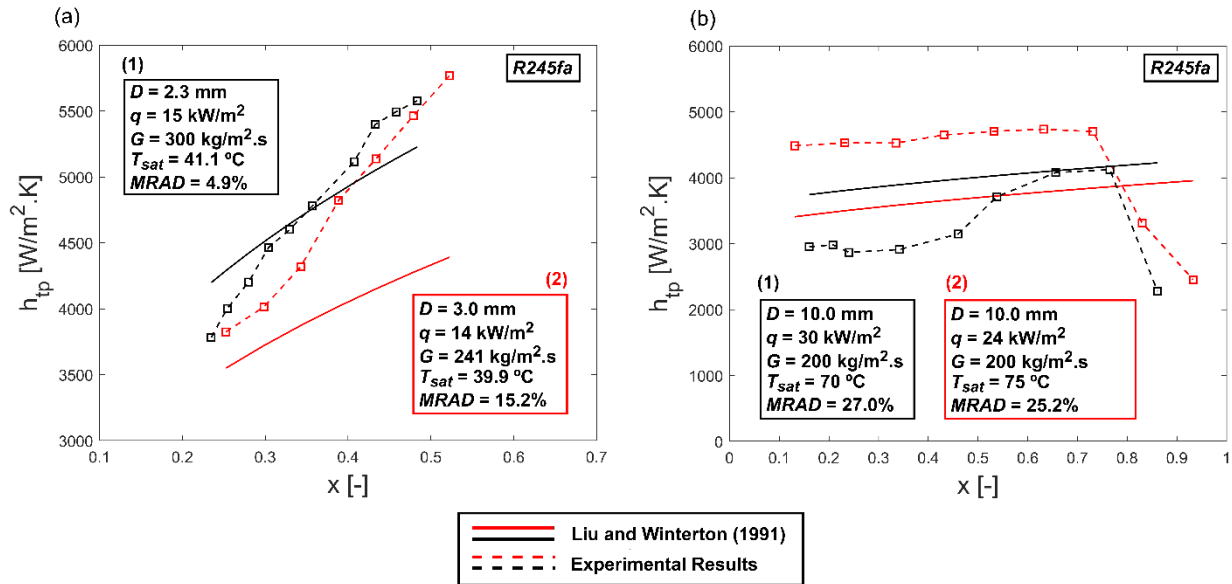


Figure 2. Comparison between the experimental h_{tp} reported by independent test benches for R245fa under nominally comparable flow boiling conditions, with Liu and Winterton (1991) [11] correlation included for reference: (a) comparison between 2.3 mm and 3.0 mm electrically heated stainless-steel tubes; (b) comparison between two electrically heated stainless-steel tubes of 10 mm internal diameter.

These comparisons demonstrate that discrepancies between h_{tp} datasets obtained under nominally comparable conditions can vary from minor to substantial, depending on the experimental facility. In the case of Figure 2(b), the observed differences appear larger than might be expected from small variations in the reported operating conditions alone, as evidenced by the Liu and Winterton (1991) [11] correlation results. However, the available information does not allow to identify the origin of these deviations with certainty. Factors such as tube surface roughness, thermocouple placement, local heat flux distribution, and data reduction procedures may all contribute to these differences. What can be stated more confidently is that the flow boiling literature contains very few inter-facility comparisons or reproducibility studies under matched conditions. Consequently, discrepancies between independent datasets are rarely quantified in detail, despite their potentially significant impact on the development, validation, and predictive accuracy of heat transfer correlations.

Additionally, a further, and often neglected, source of uncertainty concerns the heating boundary condition adopted in experimental studies. In the literature, flow boiling databases frequently combine results obtained in electrically heated and fluid-heated test sections without explicitly distinguishing their influence on the measured h_{tp} [2]. Yet these two approaches impose different thermal boundary conditions and may lead to different wall temperature distributions, local heat flux profiles, and boiling responses, as observed in pool boiling experiments [17]. Consequently, part of the experimental scatter embedded in existing correlations may be associated not only with fluid properties and operating conditions, but also with the heating methodology itself.

In this context, the present work proposes an experimental facility specifically conceived to address these challenges. The setup incorporates two interchangeable test sections, one heated by hot oil and the other by electrical resistances, enabling direct comparison of flow boiling heat transfer coefficients under matched operating conditions representative of ORC evaporators. By generating new data in high-temperature and high-heat flux conditions, while also isolating the influence of heating method, the proposed approach is expected to improve the physical interpretation of flow boiling results and support the future development of more reliable heat transfer correlations for ORC systems.

2. Experimental test bench

2.1. Operating principle

As experimental flow boiling data under ORC-relevant operating conditions is scarce and the influence of heating methodology on the measured heat transfer coefficient is not well understood, an experimental facility was designed to investigate these aspects under controlled conditions [18]. As illustrated in Figure 3, the facility comprises a primary loop for the organic working fluid, two independent thermal-oil loops, and a water-cooling loop. One oil loop is dedicated to the preheating stage, while the other is used exclusively in the fluid-heated test section. In the primary loop, the working fluid is circulated by a variable-speed pump. Upstream of the pump, a liquid receiver is installed to ensure that liquid only enters the pump and to reduce pressure fluctuations in the circuit. The working fluid is first sent to a preheater, where it transitions from the compressed-liquid state to the prescribed inlet condition of the test section, corresponding to a known vapor quality (x).

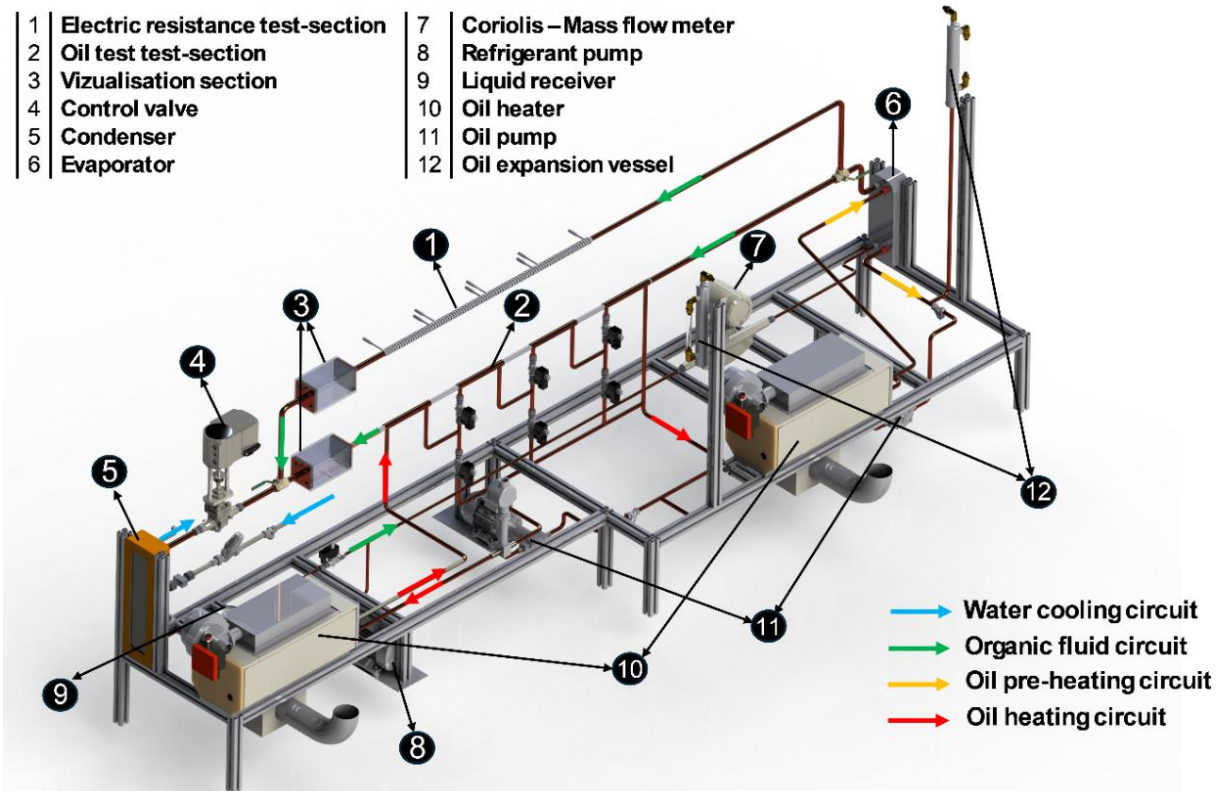


Figure 3. 3D model of the experimental setup to measure the h_{tp} of organic fluids during the boiling process [18].

After passing through the preheater, the fluid enters a calming section before moving on to the measurement region. This section is designed to minimise the impact of hydrodynamic disturbances generated upstream, as previous studies have demonstrated that geometric perturbations, such as bends and U-tubes, can affect the local heat transfer coefficient over significant distances downstream. In particular, disturbances have been reported to persist for up to 48 times the internal tube diameter in two-phase flow [19]. Accordingly, the calming section was designed in line with these findings.

Additionally, depending on the selected operating mode, the working fluid then flows through either an electrically heated section, which approximates a constant heat flux boundary condition, or a fluid-heated section, in which heat is transferred from thermal oil. Downstream of the test section, a pressure control valve regulates the pressure in order to maintain steady-state saturation conditions, given the inherent pressure fluctuations associated with boiling flow. Moreover, an adiabatic visualisation section equipped with a high-speed camera is installed downstream of the heated region to enable observation of the two-phase flow patterns and support the interpretation of the heat transfer mechanisms, which is essential for improving correlation development [20]. Finally, the working fluid condenses in a water-cooled plate heat exchanger and is returned to the liquid receiver, thus closing the loop.

2.2. Test bench operating envelope

Figure 4 illustrates the operating envelope of the test bench, mapped against typical conditions reported for ORC evaporators in the literature [21–24]. The facility was designed to cover a broad range of operating conditions representative of ORC applications, including mass velocities (G) from 100 to 700 kg/m².s, heat

fluxes (q) from 10 kW/m², up to 166 kW/m², and saturation temperatures (T_{sat}) from 60 to 150 °C. This wide operating window enables the investigation of flow boiling mechanisms over a substantial portion of the ORC-relevant domain. The upper limit of the saturation temperature was intentionally selected to approach the critical temperature of *R245fa*, which was chosen as the reference working fluid for the initial validation campaign. This design target guided the pressure rating, sealing specifications, and overall mechanical design of the facility, leading to a maximum operating pressure of 32 bar(a), consistent with the expected thermodynamic range of *R245fa*. It should be noted that the maximum value of $T_{sat} = 155^\circ\text{C}$ shown in the ORC operating envelope in Figure 4 corresponds specifically to the critical temperature of *R245fa*. Thus, although the initial design is centered on this fluid, the facility retains sufficient flexibility for future studies with new-generation low-GWP fluids, such as *R1233zd(E)*, *R1336mzz(Z)*, *R1234ze(Z)*, and hydrocarbons, such as cyclopentane, thereby supporting the development of sustainable ORC technologies [25] over a broad range of ORC-relevant conditions. Finally, the test section was initially designed with an internal diameter (D_i) of 20 mm, a value selected due to its potential relevance to practical ORC evaporator design [24].

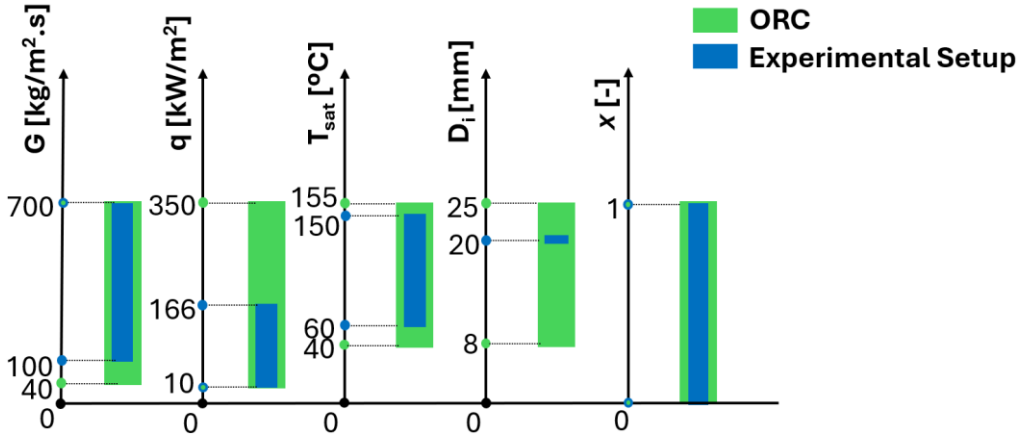


Figure 4. Operating envelope of the test bench mapped against the typical ORC operating range.

2.3. Test Section – Electrically Heated

As previously mentioned, due to the lack of studies comparing fluid-heated and electrically-heated test sections, a methodology is presented to investigate these differences and clarify them for the scientific community. Therefore, in the test bench, the electrical resistance test section consists of four independent resistances coiled around the tube (see Figure 3), each powered by its own AC source. As can be seen from Newton's law of cooling in Eq. (1), the internal wall temperature ($T_{w,i}$ [°C]) is required to measure the h_{tp} . However, it is challenging to measure this directly as it would disturb the fluid flow.

$$h_{tp} = \frac{q}{T_{w,i} - T_{sat}} \quad (1)$$

Therefore, the outside wall temperature ($T_{w,e}$ [°C]) is recorded instead. At each axial measurement location, three thermocouples are installed around the tube circumference, spaced by 90°, in order to capture possible circumferential temperature gradients associated with non-uniform boiling phenomena, such as dryout, as can be observed in Figure 5. This is especially relevant in horizontal flow boiling, where dryout tends to occur preferentially near the upper part of the tube due to the asymmetric distribution of the liquid film. The outer wall temperature at each section is taken as the arithmetic mean of the three thermocouple readings.

In addition, axial conduction along the copper wall was assessed using Fourier's law and considered negligible under the present operating conditions, owing to the high thermal conductivity of the tube material and the steady-state nature of the experiments [1]. Consequently, the h_{tp} between each electrical resistance is calculated according to Eqs (2) and (3).

$$\frac{1}{h_{tp}} = \frac{T_{w,i} - T_{sat}}{q''_e} - \frac{\ln\left(\frac{D_o}{D_i}\right) \cdot D_i}{2 \cdot k_{tube}} \quad (2)$$

$$q_e = \frac{V \cdot I}{A_s} \quad (3)$$

Where D_o [m] and D_i [m] are the outer and inner diameters of the tube, respectively, k_{tube} [W/m.K] is the thermal conductivity of the tube material, V [volt] and I [A] are the applied voltage and current, and A_s is the external

heated surface area associated with the active electrical resistance. Finally, q_e [W/m²] is the heat flux coming from each electrical resistance.

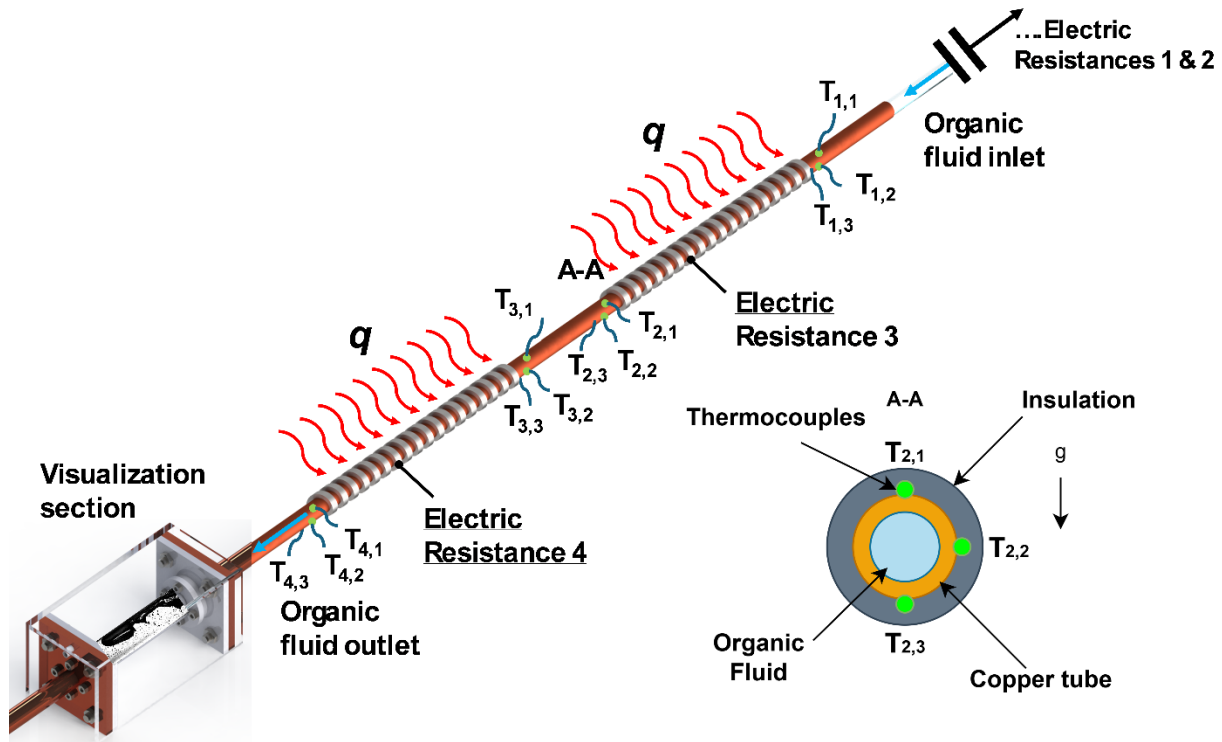


Figure 5. Scheme of the electrically heated test section

To relate the measured heat transfer coefficient to the corresponding thermodynamic state of the working fluid, the local vapour quality is determined from an energy balance. For each heated segment, the vapour quality (x) is calculated with Eq (4).

$$x = x_{in} + \frac{q_e \cdot A_s}{\dot{m}_{ref} \cdot h_{fg}} \quad (4)$$

Where \dot{m}_{ref} [kg/s] is the organic fluid mass flow rate, and h_{fg} [kJ/kg] the latent heat of vaporization of the organic fluid. For the first electrical resistance, x_{in} [-] is obtained from the energy balance in the oil preheater. For the subsequent electrical resistances, x_{in} corresponds to the outlet vapor quality calculated from the preceding heated section. Therefore, it is important to note that, since both q_e and A_s depend on the number of active electrical resistances, this configuration provides substantial flexibility in defining the thermal loading of the test section.

Thus, an important feature of this design is that the effective heated area can be changed independently of the total electrical power supplied. For example, a given increase in vapor quality may be achieved either by applying a high power over a small heated area or by distributing a similar total power over a larger area, thereby reducing the local heat flux. If only a single resistance were available, any change in heat flux would have to be achieved exclusively through power variation, which would simultaneously alter the vapor-quality evolution along the test section. By contrast, the four-resistance configuration decouples, to a significant extent, the thermal power input from the effective heat transfer area, allowing a broader range of controlled experimental conditions and a more systematic investigation of heat flux effects on flow boiling behavior.

2.4. Test Section – Fluid Heated

Following the same design principle adopted for the electrically heated test section, the fluid-heated section was conceived to vary the effective heat transfer area while maintaining control over the total thermal power supplied to the organic fluid. As shown in Figure 3, this section consists of four concentric counterflow heat exchangers, through which thermal oil transfers heat to the organic fluid flowing inside the test tube. The number of active heat exchangers can be selected according to the desired operating conditions, enabling flexible adjustment of the heating area and thermal power input. This setup enables operating conditions comparable to those in the electrically heated test section to be reproduced, and vice versa, allowing direct comparison between the two heating methodologies.

To determine the h_{tp} , the main difference between the electrical resistance test section and the oil test section is in the calculation of the heat flux. In the oil heat exchanger configuration shown in Figure 6, both the $T_{w,e}$

and the oil temperature (T_{oil} [°C]) are measured at the inlet and outlet of each concentric heat exchanger. As the oil cools continuously along the test section, the local heat flux is not uniform. However, this effect is mitigated by dividing the heated length into four independent counterflow heat exchangers, enabling each segment to be treated separately and associated with a more uniform average heat flux. Additionally, oil temperatures are measured as close as possible to the top and bottom of the tube to capture any radial temperature differences related to flow-pattern-dependent heat transfer asymmetries [26]. This strategy is consistent with the circumferential wall-temperature measurements adopted in the electrically heated section. Hence, under these assumptions, the heat flux imposed by the oil in each heated segment (q_{oil} [W/m²]) is calculated with Eq (5).

$$q_{oil} = \dot{m}_{oil} \cdot c_{p,oil} (T_{oil,in} - T_{oil,out}) \cdot \left(\frac{1}{A_s}\right) \quad (5)$$

Where \dot{m}_{oil} [kg/s] is the oil mass flow rate, $c_{p,oil}$ [J/kg.°C] is the oil specific heat, $T_{oil,in}$ [°C] and $T_{oil,out}$ [°C], are the arithmetic averages of the oil temperature measurements at the top and bottom of the tube in the inlet and outlet of each concentric heat exchanger (see Figure 6), respectively, and A_s [m²] is the surface area of the outer wall of the tube immersed in oil. Finally, the calculation of the vapor quality (x) between each concentric heat exchanger is analogous to the electrically heated test section.

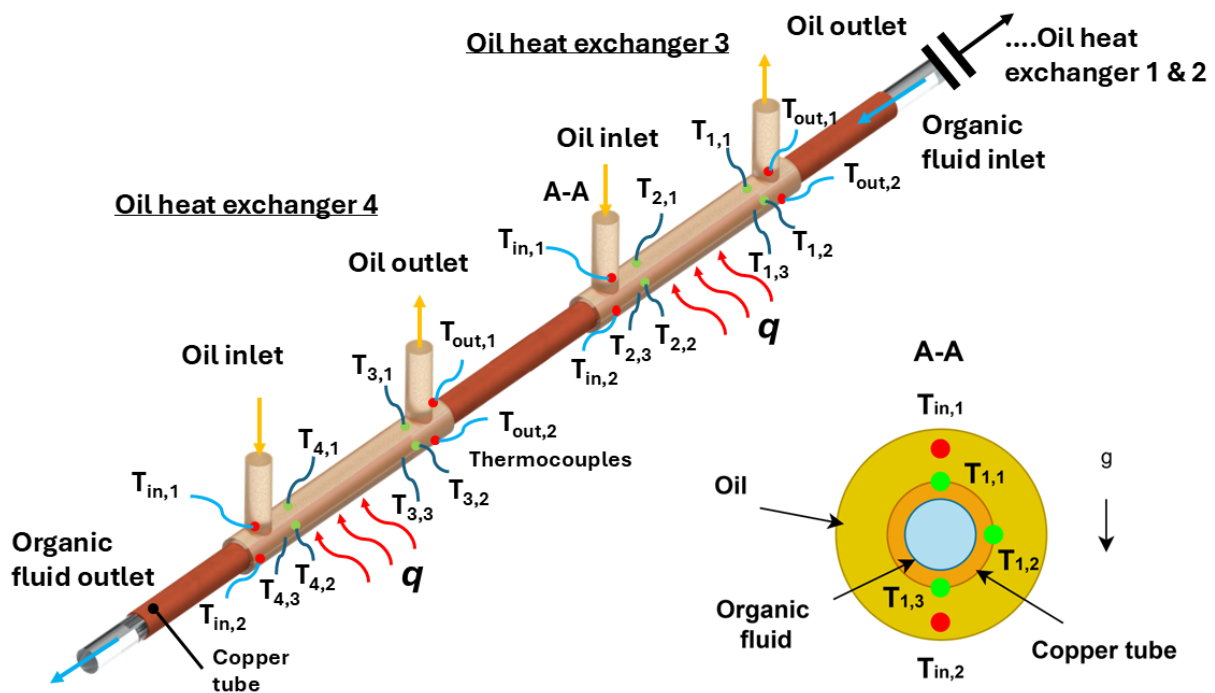


Figure 6. Scheme of the oil-heated test section

3. Conclusions

This work has highlighted two major limitations in the current flow boiling literature when applied to ORC evaporators. First, most existing correlations were developed and validated under refrigeration-oriented conditions and are therefore unable to accurately describe the heat-transfer behavior observed at the higher saturation temperatures and heat fluxes typical of ORC applications. The comparison with experimental data for R245fa and R1233zd(E) shows that a widely used benchmark correlation, Liu and Winterton (1991), fails not only in terms of prediction error but also in reproducing the behavior of the flow boiling heat transfer coefficient in the vapor quality range. Second, the focused comparison of nominally similar datasets obtained in different facilities revealed that experimental discrepancies may vary from modest to significant, while inter-facility reproducibility studies remain scarce in the literature. This indicates that part of the scatter embedded in current databases and correlations may be linked to differences in experimental methodology that are rarely quantified explicitly.

To address these issues, an experimental test bench was designed to operate over a wide ORC-relevant range of mass velocity, heat flux, and saturation temperature, with initial validation centered on R245fa. The facility incorporates two interchangeable heated test sections: an electrically heated section with four independently powered resistances and a fluid-heated section based on four concentric counterflow heat exchangers. This dual configuration enables matched operating conditions to be imposed under different heating boundary conditions, while also allowing future comparison with literature datasets obtained in independent facilities. In

this way, the proposed test bench provides a basis for generating new flow boiling data under ORC-relevant conditions and clarifying the influence of heating methodology and reproducibility on the measured heat transfer coefficient. These capabilities are expected to support the future refinement and development of more reliable flow boiling heat transfer correlations for ORC systems.

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Nomenclature

Letter symbols

A	Area, m ²
c_p	Specific heat, J/kg.°C
h	Heat transfer coefficient, W/m ² .K
I	Current Intensity, A
k	Thermal conductivity, W/m.K
$MRAD$	Mean relative absolute deviation, [%]
q	Heat flux, W/m ²
T	Temperature, °C
V	Voltage, volt
x	Vapor quality, -

Subscripts and superscripts

e	electric resistance
i	inner surface
in	inlet of test section
o	outer surface
oil	oil heating fluid
s	surface
sat	saturated state
$tube$	tube material of the working fluid
tp	two-phase
w	wall

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