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HEAT TRANSFER IN HORIZONTAL TUBES AT SUPERCRITICAL PRESSURES FOR ORGANIC RANKINE CYCLE APPLICATIONS

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ABSTRACT

Today process industry has to deal with a lot of waste heat. Very often this waste heat is dumped to the environment, while it could be efficiently used. Low grade heat is also available in solar thermal systems, geothermal systems, biomass combustion and etc. This heat can be converted into electricity using an organic Rankine cycle (ORC). This thermodynamic cycle is similar to the well-known Rankine steam cycle, but it works with an organic working fluid instead of water/steam. As a consequence, the temperature of the heat source at the evaporator is typically lower compared to the steam cycles.

To increase the cycle efficiency, supercritical cycles seem very promising. The advantage of supercritical ORCs is a better thermal match between the heat source and the working fluid temperature profiles in the supercritical heat exchanger. Consequently, the overall system efficiency improves.

In the current literature, research work on heat transfer mechanisms under supercritical conditions in ORC is very limited. Therefore, it is an essence to study and investigate the relatively unknown heat transfer phenomena at supercritical working conditions for organic fluids in the temperature and pressure ranges relevant for ORCs. Heat exchanger design and heat transfer coefficients are factors that have great influence on the heat transfer and the overall cycle efficiency. Furthermore, special attention must be drawn to the choice of appropriate working fluid, which is a factor of great importance because the organic fluid properties also affect the overall efficiency of the cycle.

In this paper, relevant researches are reviewed regarding Supercritical Organic Rankine Cycle applications, selection of working fluids and description of newly designed test facility is presented.

INTRODUCTION

Research activities about heat transfer at supercritical pressure started from the early 50's and in the early 70's [1], [2], [3], attention had been paid to Organic Rankine Cycles due to it was considered as promising technology for low grade heat recovery. However theoretical and experimental research activities for waste heat recovery systems based on Supercritical Organic Rankine Cycle technology appeared two decades later.

In the literature database there is a gap of more than 20 years since the first published paper is in 1981 [4], regarding the Supercritical Organic Rakine Cycle. During a period of 20 years no investigation for supercritical ORC applications had been done because of concerns about safety and economic feasibility. More active theoretical and experimental research starts from 2009 [5], [6], [7]. In current SC ORC studies attention has been paid to appropriate working fluid selection for a better thermal match between the heat source and the fluid characteristics. Furthermore, determining corresponding heat transfer correlations is also an issue in order to improve the heat exchanger surface and the design algorithms.

This paper reviews prior work carried out on supercritical heat transfer process and heat transfer coefficients, where attention has been paid to horizontal flow taking into consideration the characteristics that influence the heat transfer. The research work was conducted to gather knowledge available from the literature and to define the further research needs.

In the recent years many researchers have showed interest on heat transfer of fluids at supercritical pressures. A motivation for experimental and theoretical studies had been raised by the need of a better understanding of the heat transfer under supercritical conditions and for designing practical systems. Heat transfer of supercritical fluids is important to a number of technological applications, presented in modern power plants where heat is transferred to supercritical water heat exchangers. Furthermore, highly charged machine elements such as gas turbine blades, supercomputer elements, magnets and power transmission cables are cooled with supercritical fluids.

Moreover, in the current literature, related research work on heat transfer mechanisms under supercritical conditions in ORCs is very limited. This conclusion is also applicable for experimental work for heat transfer in horizontal flow.

The Supercritical Organic Rankine Cycle is a not a new technology. It has been derived from the classic Rankine cycle that has been widely deployed in power plants. SC ORC technology has a great potential for industrial waste heat recovery, taking into account the fact that today's process industry has to deal with enormous amount of waste heat. The waste heat that is applicable and moreover it has the highest potential for SC ORC technology is between the temperature 90 °C and 400 °C. Working conditions of the SC ORC plant are tightly related with the selection and the thermo-physical properties of the organic fluid. Moreover, in our case a selection of three refrigerants was done and the temperature range that is of interest is between 90 °C and 120 °C. Figure 1 presents T-s diagram and schematic of SC ORC with the cycle's basic components such as supercritical feed pump, heat exchanger, expander and condenser.

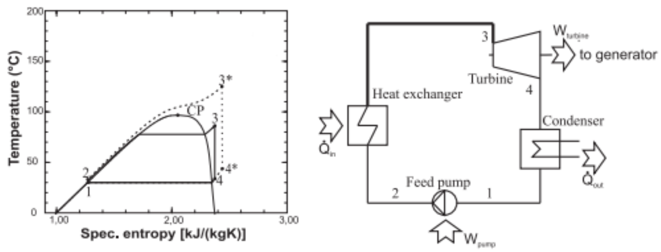


Figure 1 (a) T-s diagram; (b) Schematic overview of SC ORC

Furthermore, a lot of papers can be found related to heat transfer in the critical region for a variety of fluids such as carbon dioxide, water, oxygen, hydrogen, helium. Nevertheless, there are very few experimental data for heat transfer of organic fluids such as refrigerants that are used in supercritical ORC's. Therefore, it is an essence to study and investigate the relatively unknown heat transfer mechanisms at supercritical working conditions for organic fluids in the temperature and pressure ranges relevant for SC ORCs.

Another important parameter that has influence on the heat transfer is the working fluid flow direction. Tests had been done mainly in vertical tubes, but there are also several in horizontal ones with various diameters of the test tubes. The aim of our research is development of supercritical ORC technology, more particularly design of heat exchanger positioned horizontally. Consequently, a need for thorough understanding of heat transfer phenomena has been raised. Therefore, first a literature survey is performed and then a description of new test setup is presented. As general conclusion derived from the relevant literature study is that there is a lack of data regarding heat transfer in horizontal flow relevant for SC ORCs. In many of the tests, less attention has been paid to the buoyancy effect and the derived correlations usually have limited application to specific conditions and dimensions. However, these studies have significant importance and can serve as a baseline for further research.

NOMENCLATURE

p	[bar]	Pressure
Q	[MW/m^2)]	Mass flux
D	[mm]	Diameter
Tb	[°C]	Bulk temperature
G	[Mg/m^{2}s]	Heat flux
L	[mm]	Length (tube)
SCORC	[-]	Supercritical Organic Rankine Cycle
Nu	[-]	Nusselt number
Re	[-]	Reinolds number
Pr	[-]	Prandtl number

OVERVIEW OF SUPERCRITICAL HEAT TRANSFER EXPERIMENTAL STUDIES

Research activities about heat transfer at supercritical pressure started in the early 1930s. Schmidt and his associates [1] did an investigation of free convection heat transfer to fluids at near-critical point, applied in a new effective cooling system for turbine blades in jet engines. In the 1950s [8], [9], [10] the concept of using supercritical water appeared to be an attractive option for increasing the total thermal efficiency in the steam generators/turbines in the fossil-fuel power plants. The difference of working in supercritical conditions than in subcritical pressure is that no liquid – vapor phase transition occurs and therefore there is no such phenomenon occurring like critical heat flux and dry-out. Moreover, in the period of 1950s until 1980s [11], [12], [13], researchers from the former USSR and USA conducted several studies to investigate the potential of using supercritical water as a coolant in nuclear reactors. The primary objectives were firstly increasing the thermal efficiency of modern nuclear power plants from 30% to 45% or even higher, and secondly decreasing the operational and capital cost.

Despite the intense research work related to supercritical heat transfer in heating conditions found in relevant literature data sources there are significant gaps in understanding the supercritical heat transfer mechanisms directly related to SC Organic Rankine Cycle applications.

So far, in the literature the main focus is on supercritical heat transfer on fluids such as water/steam and carbon dioxide. Carbon dioxide is an easier fluid to handle because of its relatively low critical temperature and pressure. Therefore most of the experiments are based on using supercritical CO_2. Moreover, most of the data collected for forced convection heat transfer near the critical point has been obtained for pipes and channels with uniform cross section in vertical flow.

Heat transfer in the near critical region is influenced by significant physical and transport variations of the fluid across temperature changes. At supercritical pressure the fluid is in quasi-single phase flow. Below pseudo-critical temperature the fluid has more liquid-like properties while above it behaves more like vapor. The pseudo-critical temperature is a state defined as a temperature where the specific heat capacity reaches a peak for a given pressure. It should be noted that the peak in specific heat leads to a maximum in the local heat transfer coefficient in the vicinity of the region. Moreover, besides the changes of specific heat capacity with pressure and temperature variations there are also changes in density, viscosity, Prandtl number and thermal conductivity. The density drop leads to increased flow velocities, thereby increasing the pressure drop.

Figure 2 presents the evolution of thermo-physical properties of CO_2 and the effect of temperature on specific heat c_p, thermal conductivity λ, the dynamic viscosity μ and the mass density ρ at a constant pressure of 80 bar.

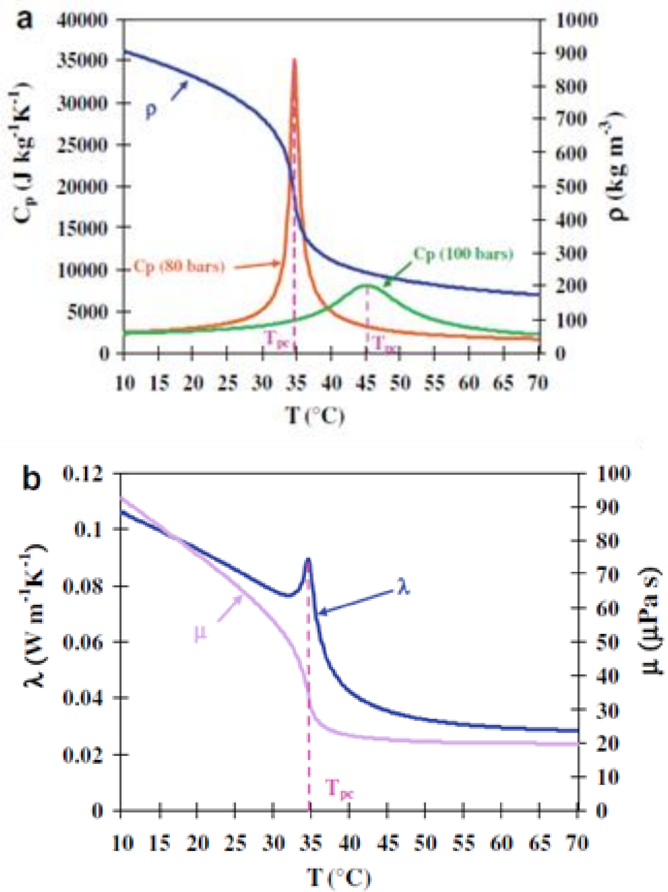


Figure 2 Evolution of thermo-physical properties CO_2
(a) specific heat and density
(b) viscosity and thermal conductivity [2]

These figures are obtained by using the Span and Wagner equations for the density and specific heat, while viscosity and thermal conductivity are plotted according to Vesovic and Wakeham equations [2]. These studies are considered as key references of evaluation of CO_2 properties in the supercritical region.

There are several general characteristics that have an influence on heat transfer to a supercritical fluid and are enlisted hereinafter:

- influence of the heat flux
- influence of the mass flux
- influence of the tube diameter,
- influence of the flow direction
- influence of the buoyancy and
- heat transfer enhancement
- heat transfer deterioration.

Several review studies related to forced convection heat transfer at supercritical pressure have been conducted. Namely, Petukhov [3] made in 1970 a review of experimental works and correlations for heat transfer and pressure drop for supercritical water and CO_2. Furthermore, Jackson and Hall [14-16] (1975 and 1979) investigated the heat transfer phenomena at supercritical pressure, comparing several correlations with test data and a new semi-empirical correlation was proposed. This correlation incorporates the effect of buoyancy on the heat transfer at supercritical pressure. Polyakov [17] based his survey on previous work done and added in a review with numerical analysis back in 1991. His focus was on the heat transfer mechanism and the trigger of heat transfer deterioration was discussed in his review. In 2000, Kirillov [18] reviewed the research done in Russia about heat and mass transfer at supercritical parameters of water and a new correlation was presented. Prioro [19] made a literature survey in 2004, giving an overview of almost all already existing correlations.

Many tests were performed in research centers in the USSR on supercritical water, carbon dioxide and oxygen [8, 20]. The phenomenon of heat transfer deterioration was first observed by Shitsman [20] at low mass fluxes. During the tests, pressure pulsation occurred when the bulk temperature approached the pseudo-critical value. Based on the test data several correlations were developed for predicting heat transfer coefficient, onset of heat transfer deterioration and friction pressure drop.

The experiments of Dickinson (1958) [9], Ackermann (1970) [21], Yamagata (1972) [11] and Griem (1995) [22] were mainly related to the design of supercritical pressure fossil power plants. The tube diameter ranges from 7.5 mm up to 24 mm. A good agreement was obtained between the test data of Dickinson [9] and the Dittus-Boelter equation ($Nu = CRe^m Pr^n$) at a wall temperature below 350 °C. On the other hand, large deviation was obtained at a wall temperature between 350 °C and 430 °C. In both the experiments of Domin (1963) [23] and of Dickinson [9], no heat transfer deterioration was observed, whereas heat transfer deterioration occurs in the tests of Yamagata [11] and of Ackermann [21]. It was shown by Yamagata [11] that at low heat fluxes, heat transfer is enhanced near the pseudo-critical line. Heat transfer deterioration happened at high heat fluxes. Ackermann [21] observed boiling state like noise at the onset of heat transfer deterioration, which was, therefore, treated as a similar phenomenon like boiling crisis under sub-critical pressures. The test data indicated that pseudo-critical heat flux (CHF), at which heat transfer deterioration occurs, is increased by the increasing pressure, increasing mass flux and decreasing tube diameter.

The experimental approaches of Bishop (1964) [13] and Swenson (1965) [24] were performed in the frame of designing supercritical light water reactors. In the work of Bishop [13], small diameter tubes were used, whereas in the work of Swenson [24], circular tubes of a larger diameter 9.4 mm were applied. In addition to smooth circular tubes, whistled circular tubes and annular channels were also used by Bishop [13]. Nevertheless, no experimental data in annular channels are available in the literature. Performed tests show the entrance effect on heat transfer coefficient. In the experiments of

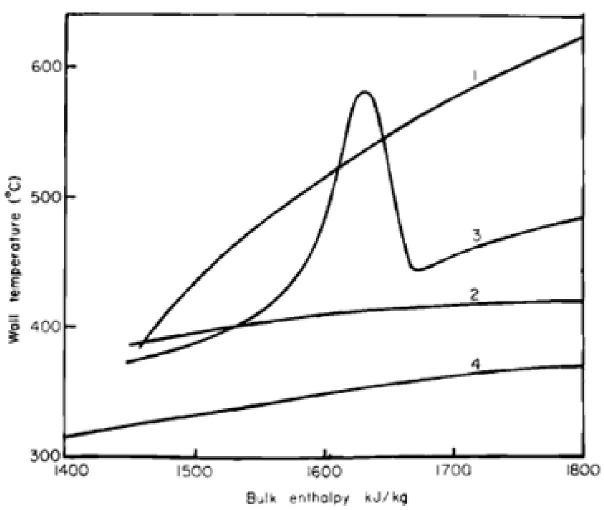
Swenson [24], a heat transfer deterioration was not observed. Empirical correlations were derived based on the test data achieved.

In 1964, Vikrev and Lokshin conducted one of the earliest major studies of heat transfer to supercritical water in horizontal flow [12]. This also was the first attempt to quantitatively formulate deterioration of heat transfer coefficients in supercritical conditions.

The deterioration in horizontal pipes is less prompt than in vertical upward flow pipes. In a horizontal setup, the temperature difference occurring between the upper and lower surface of the pipe is caused by the buoyancy. This temperature variation leads to a reduction in the heat transfer coefficient at the upper surface compared to the one obtained in the lower surface.

For a downward heated flow there is a continuous enhancement in heat transfer as buoyancy becomes relatively stronger. This behavior has been found with many other fluids at supercritical pressure. Not only the heat transfer is improved, but also wall temperatures are less sensitive to the heat flux.

Miropolsky and Shitsman [25] measured the temperature distribution for supercritical water around a horizontal and vertical 1.6 cm diameter pipe presented in Figure 3. The temperature difference between the bulk temperature and the upper surface is a lot bigger than the difference between the lower surface and the bulk temperature. In the conditions presented in figure 3, this leads to a reduction in the heat transfer coefficient about a factor of 4 comparing the upper surface to the lower one.



(1) Horizontal pipe – upper surface
(2) Horizontal pipe – lower surface
(3) Vertical pipe – upward flow
(4) Bulk fluid temperature

Figure 3 Temperature distribution as a function of local bulk enthalpy along heated vertical and horizontal pipes (1.6 cm diameter) for water at 245 bar (= 1.11 pcrit): $\dot{m}/A$=60 g/scm^2 and q=52 W/cm^2 [25].

In 1966, Shitsman was one of the first to investigate the effect of density changes (buoyancy) during horizontal flow [26]. Therefore, he measured temperatures at top and bottom along the tube section over which heat transfer took place. Temperature differences up to 250 °C were observed, denoting the large possible impact of buoyancy forces. In order to quantify the significance, the product of the Grashof and Prandtl number was taken. However, no similar experiments had been conducted before. This meant that Shitsman's correlation for buoyancy could not be verified against any existing data but his own. Nevertheless, his study and novelty in his research had a major impact on further work.

The focus on the effects of buoyancy and local acceleration due to density variation more closely started in 1976. Adebiyi and Hall [27] measured the wall temperature at top and bottom surface of a tube for a wide range of heat fluxes and mass flow rates. Free convection could occur due to the relatively large diameter of 22.1 mm of the horizontal tube. They noticed that the buoyancy effect does not only cause heat transfer deterioration at the top surface of the tube, but also an improvement of it at the bottom surface. Furthermore, buoyancy-free and buoyancy-dependent cases were distinguish-ed.

Hall and Jackson [14] proposed a mechanism for which buoyancy will affect the heat transfer. The dominant factor is the modification of the shear stress distribution across the pipe, with a consequential change in turbulence production.

As mentioned before, buoyancy effects are present in horizontal flows due to a stratification of the flow. The hotter (less dense) fluid can be found in the upper part of the pipe. Also, there may be an effect on the heat transfer near the upper surface of the pipe due to damping effect of the stabilizing density gradient on turbulence. At the lower surface heat transfer is frequently better than upper one for forced convection alone, suggesting that there may be some amplification of turbulence by destabilizing density gradient in this region.

Belyakov [28] performed some measurements for heat transfer of supercritical water in horizontal pipes. The deterioration of the upper surface occurs progressively along the pipe and does not show the sharp peaks that are obtained with upward flow. As the ratio of the heat flux to the mass flow flux increases, the wall temperature and thus deterioration at the upper surface increases.

HEAT TRANSFER CORRELATIONS

Over time, several heat transfer correlations have been developed. Most of these are for water, helium, CO_2 or a selection of mentioned foregoing fluids. Nevertheless, these correlations can be widely used in design procedures. Below, four different correlations are presented. They are applicable for horizontal flow and are of great importance for our further experimental research.

Firstly, the correlation of Petukhov, Krasnochekov and Protopopov [10] for water and CO_2 at supercritical conditions has proven to show a good predicting performance. About 85% of their data and data of experiments by other researchers match their correlation within a deviation of 15%.

$$Nu_b = Nu_{0,b}\left(\frac{\bar{c}_p}{c_{p,b}}\right)^{0.35}\left(\frac{\lambda_b}{\lambda_w}\right)^{-0.33}\left(\frac{\mu_b}{\mu_w}\right)^{0.11}$$

$$Nu_{0,b} = \left(\frac{\frac{f_b}{8} Re_b \overline{Pr}}{12.7 \left(\frac{f_b}{8}\right)^{0.5} \left(\overline{Pr}^{\frac{2}{3}} - 1\right) + 1.07} \right)$$

$$f = (1.82 log_{10}(Re_b) - 1.64)^{-2}$$

Valid within:

$$2x10^4 < Re_b < 8.6x10^5$$
$$0.85 < \overline{Pr}_b < 65$$
$$0.90 < \frac{\mu_b}{\mu_w} < 3.60$$
$$1.00 < \frac{k_b}{k_w} < 6.00$$
$$0.07 < \frac{\bar{c}_p}{c_{p,b}} < 4.50$$

Secondly, the correlation of Swenson, Carver and Kakarala [24] is applicable for water in horizontal tubes:

$$Nu = \frac{hD}{k_w} = 0.00459 \left(\frac{DG}{\mu_w}\right)^{0.923} \left(\frac{H_w - H_b}{T_w - T_b} \frac{\mu_w}{k_w}\right)^{0.613} \left(\frac{\rho_w}{\rho_b}\right)^{0.231}$$

Thirdly, the proposed one by Shitsman [29] is applicable for helium, CO2 and water in horizontal tubes and is of the Dittus-Boelter form:

$$Nu_b = 0.023 Re_b^{0.8} Pr_{min}^{0.8}$$

In the above expression, Pr_{min} is the smaller one of Pr_b and Pr_w.

Moreover, the correlation of Jackson and Fewster [30] is based upon the Dittus-Boelter form for constant properties convection:

$$Nu_b = 0.0183 Re_b^{0.82} \overline{Pr^{0.5}} \left(\frac{\rho_w}{\rho_b}\right)^{0.3}$$

The buoyancy effect is neglected in the equation.

These correlations have been used for designing the new test setup. From the existing data the pressure is between the range of 22.0 – 41.7 MPa, and all of the correlations are applicable for horizontal flow which is of major importance for our case.

It is generally agreed that the correlations do not show sufficient agreement with experiments to justify their use except in very limited conditions.

At bulk temperatures well above the critical temperature, the heat transfer resembles more to a normal single phase heat transfer to a gas, which can be predicted with a conventional Dittus-Boelter type of correlation.

NOVEL TEST FACILITY FOR HEAT TRANSFER AND PRESSURE DROP MEASSUREMENTS

In order to do an investigation of the heat transfer characteristics of an organic fluid working at supercritical pressure, a new test set up was designed. Therefore, a selection of appropriate fluids is performed. This is very important for maximum recovery of low-grade heat source. For our experimental work, three refrigerants R125, R134a and R1234yf are selected as working fluids and their performances on supercritical states are going to be tested in the setup. Table no. 2 presents the thermo-physical and environmental impact properties of the selected fluids.

Table 2 Thermo-physical and environmental impact properties of the selected fluids

Parameters	**R125**	**R134a**	**R1234yf**
Chemical formulae	C2HF5	CF3CH2F	C3F4H2
Mol. Weight (g/mol)	120	102,03	114,04
Boiling point (°C)	-48,5	-26,15	-30,15
Critical temperature (°C)	66,02	101,06	93,85
Critical pressure (bar)	36,18	40,56	33,75
Ozone Depletion Potential (ODP)	0	0	0
Global Warming Potential (GWP) (within 100 years)	3400	1300	4

The aim of doing measurements in the new test setup is to determine correlations for heat transfer and pressure drop phenomena in supercritical flow regimes in ORCs. Local heat transfer and pressure drop measurements over the test section are performed under supercritical conditions of temperature and pressure, with mass flux and heat flux changes. The results will give better understanding of the flow regimes and heat transfer behavior and the derived correlations are going to be used for a heat exchanger design suitable to work in supercritical ORC application - conditions.

The setup consists of three different fluid loops: cooling loop, heating loop and supercritical refrigerant loop. In addition a general overview of the test section is discussed.

SUPERCRITICAL REFRIGERANT LOOP – EXPERIMENTAL TEST SECTION

The supercritical refrigerant loop is the main experimental test section and it is used to carry out heat transfer measurements to supercritical fluids in horizontal flow. Namely, as mentioned in the previous chapters there is a lack of experimental data available.

The refrigerant from the storage container is pressurized with a supercritical circulation pump into the circulation system. A filter is installed before the circulation pump in order to prevent particles to penetrate inside it and to avoid damage of the pump. The circulation pump from www.Verder.be model: “Hydra-Cell sealless pump” G15EDBTHFEHH has multiple diaphragm design that provides pulse less flow and there is no mix between the oil and the refrigerant.

In the circulation tube line there is a pressure control accumulator, positioned before the inlet of the pump with the aim to stabilize and control the system pressure level. The accumulator is connected to a nitrogen gas cylinder, and by adjusting the pressure of N_2 the working fluid can be set and maintained to the desired value. The refrigerant is heated up in the electrical and tube-in-tube preheaters to the required inlet temperature. The electrical preheater is mainly used during the start-up of the testing facility, due to the refrigerant is at ambient temperature level. By circulating it through the preheaters and the test section, the refrigerant can be brought up to the desired inlet temperature within an acceptable amount of time. An electrically operated heat exchanger is used because of its ability to be controlled more precisely and directly. This enables a more precise control of the test section inlet temperature.

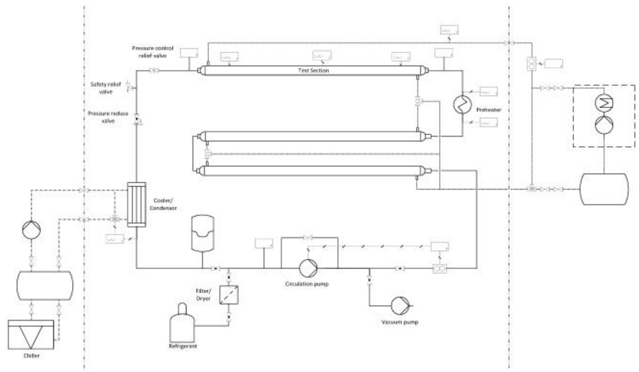


Figure 4 Schematic overview of experimental test section

From the preheater the working fluid flows through the test section where the heat transfer from the thermal oil to the organic fluid takes place. The test section is made of horizontal copper alloy tube CuFe2P. The selected inner diameter of the tubes is 28.5 mm, thickness of 1.9 mm and heating length of 4 m. Moreover, for fully-developed flow in the tube, the test section tube is connected with unheated tubes (adiabatic section before and after the heating section) of 0.5 m on the both sides of the tube. Main reason for using these copper alloy tubes and not standard copper tubes is the working condition of the test setup, and that is p=50 bar and T=120 °C. The mechanical properties of the cooper alloy tubes show good performance on the desired/set conditions.

To reduce the exchange of heat with the environment the tubing of the system is insulated. Measurements on the test section in the setup are performed in horizontal tube and are done with a large number of sensors. The pressure drop over the test section and in the setup itself is measured with pressure sensors (denoted by a p-symbol in the figure). The mass flux and the pressure in the experimental section are precisely controlled by a control valve system. Set of thermocouples denoted by t-symbol are soldered on the outer surface of the test section of the tube to measure the outside wall temperature on the bottom, top and from the both sides of the tube. Thanks to the thermocouples layout a buoyancy effect can be determined. Also thermocouples are inserted in the inlet and outlet of the tube to measure the bulk temperature, and set of thermocouples are inserted to measure the inflow temperature. They are electrically insulated to prevent the feed current from passing through. The individual thermocouple channels are calibrated.

All signals are collected by a data acquisition system. They are going to be used to determine the heat transfer and pressure drop correlations, necessary for designing the heat exchanger.

HEATING LOOP

The heating loop consists of thermal oil heating block, insulated vessel and additional parts such as valves, mass flow meters (denoted as symbol F), sensors etc. The schematic diagram of the heating loop is shown in Figure 5.

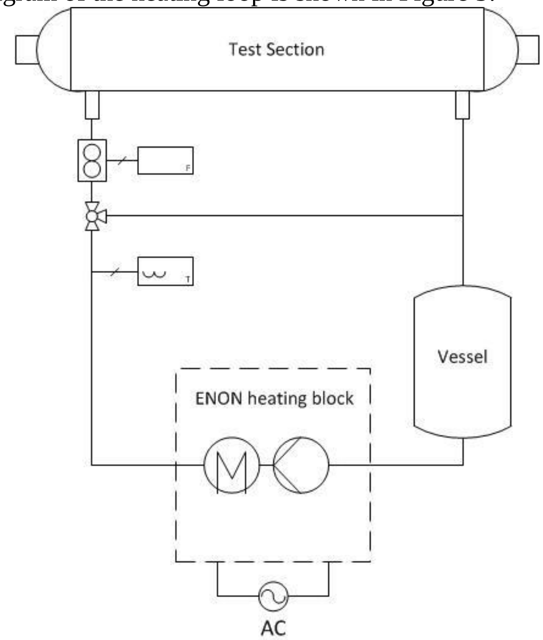


Figure 5 Schematic overview of heating loop and thermal oil heater unit

The heating block is a thermal oil heater unit with a capacity of 20 kW. This unit represents a compact block with the following components: centrifugal pump 10 m³/h, electrical heater with power of 20 kW; automatic venting; safety thermostat; temperature sensor for the oil; monitor max. temperature; expansion vessel 19 ltr.; an oil-level monitoring; drain valve; speed braking box; manometer; PID temperature controller, a junction box: 3x400 V + PE, power control with thyristors, Pt100 temperature sensor in exhaust oil circuit.

The thermal oil is electrically heated by a current stabilized 3-phase power source and pumped to the test section where the actual heat transfer to the refrigerant takes place. At the inlet of the test section the oil temperature is controlled by temperature sensor Pt100. The test section is a double tube exchanger with a counter flow where the oil flows in the annulus and afterwards it flows back to the heating block and it is collected in the expansion vessel.

In the heater unit, the thermal oil Therminol ADX10 is used. The physical, chemical and thermal properties of Therminol ADX10 meet the working condition requirements of the test set up. Namely, the thermal oil Therminol ADX10 is a low viscosity synthetic organic heat transfer fluid particularly recommended for indirect liquid phase process heating at medium temperature up to 250 °C. Moreover, the excellent low temperature pumpability and heat transfer coefficient lead to a number of benefits when this fluid is used as a single unit in heating systems.

COOLING LOOP

This loop consists of a cold vessel with capacity of 900 litre that uses a mixture of water/glycol (70%/30% by volume) as a fluid. The cold vessel is used as cold source for the refrigerant condenser and the pressure control circuit. A 3-way valve is installed in front of the condenser in order to control the water flow rate to the condenser. For controlling the cold source temperature a 37 kW chiller is installed outside of the building and it is switched on and off by a thermostatic device inside the cold vessel. A cooler is a small heat exchanger that cools down the fluid to a temperature just below the entrance conditions of the test section.

The schematic diagram of the cooling loop is shown in Figure 6.

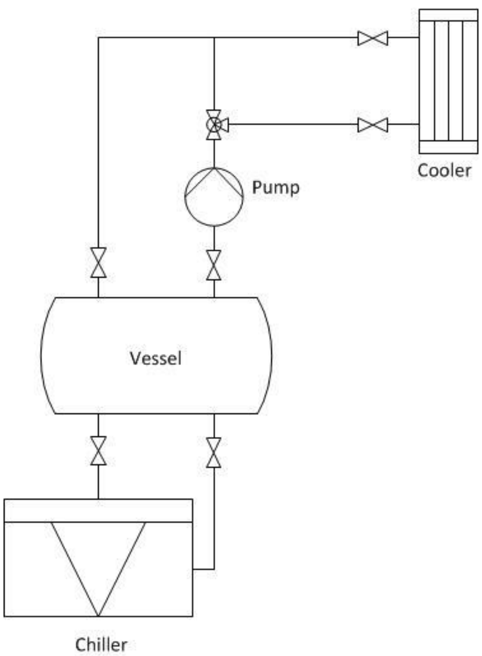


Figure 6 Schematic overview of cooling loop

CONCLUSION

This paper presents a comprehensive review of supercritical heat transfer in horizontal flow and review of the experimental boundary conditions. In the current literature a significant amount of research work can be found regarding deterioration and enhancement of heat transfer in supercritical regimes, while to the buoyancy effect has not been paid much attention. Generally, most of the experiments were done in small diameter tubes, the fluid flow was mainly vertical in downward and upward flow and the employed working fluids were mostly CO_2, water, helium and etc. Furthermore, a large parameter range is covered for pressure p (22.0-44.1 MPa), heat flux G (0.1 – 5.1 Mg/m²s), mass flux Q (0.0 – 4.5 MW/m²), diameter D (2.0 – 32.0 mm) and bulk temperature Tb ≤ 575 °C. Thereinafter, there are several empirical correlations derived based on the test data and the equations are mainly of Dietus-Boelter type.

As a conclusion that can be derived from the literature study is that the parameter range and the selected working fluids in the existing experimental data differ from our conditions, working fluids and applications of interest. Moreover, the existing correlations do not show sufficient agreement with experiments to justify their use except in very limited conditions. Therefore, in order to obtain more accurate heat transfer coefficients suitable for supercritical heat transfer of refrigerants in a horizontal flow a development of new correlations is an essence. Also, significant improvement for predicting the supercritical heat transfer behavior is important for further development of practical engineering designs. This is going to be done experimentally in the new test setup and the derived correlations would be used for designing a heat exchanger suitable to work on supercritical application – conditions.

Hence an experimental measurements and more methodical approach for determining heat transfer correlations for organic fluids suitable to work under supercritical conditions in SC ORC is going to be done and the results from the measurements are expected to be presented in the Conference.

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