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ENERGY OPTIMIZATION OF EXPERIMENTAL BENCH TEST FOR STUDY OF INTERNAL HEAT TRANSFER COEFICIENT IN QUENCHING BY IMMERSION OF TUBES

Kássio Nogueira Cançado
Ricardo Poley Martins Ferreira
Luiz Machado
Programa de Pós Graduação em Engenharia Mecanica - Universidade Federal de Minas Gerais - Av. Pres. Antônio Carlos, 6627 - Pampulha, Belo Horizonte - MG, 31270-901
kancado@hotmail.com
poley@demec.ufmg.br
luizm@demec.ufmg.br

Juan José Garcia Pabon
Instituto de Engenharia Mecânica – Universidade Federal de Itajubá
langa27@hotmail.com

Lis Nunes Soares
Vallourec Soluções Tubulares do Brasil
Lis.Soares@vallourec.com

Abstract. *The quenching process is fundamental to the final quality of steel tubes. Experiments on quenching processes are costly and time-consuming. Numerical simulations allow to reduce the time and costs of these studies. However, they are limited by the uncertainty arising from the heat transfer coefficient between the water and the tube surface. The internal heat transfer coefficient can be determined experimentally, however the size of the industrial quenching processes prevent the realization of a real scale test. A test bank can be used to determine the heat transfer coefficient by similarity (Buckingham's Theorem). The main limitation of this type of test bench is power required by the system. This problem can be solved with the use of organic fluids. This work aims to select an organic fluid that allows the maximization of the dimensions of the tube diameter used in the bench test with the minimization of the energy consumed by the same. An optimization by genetic algorithm type will determine the fluid that best adapt to proposed experiment. The correct selection of the working fluid will allow the use of the test bench for a greater number of conditions, increasing the results accuracy. The best performing organic fluid was the pentane which a pressure drop of 3 bar and a power demand of 21.4 kW. Compared to R236fa, which presented the worst result (7.7 bar and 24 kW), the optimization of the working fluid allowed a reduction of 156% in the pressure drop and 12% in the power demand. When compared to water (8 bar and 192 kW), the use of optimized organic fluid allowed a reduction of 166% in the pressure drop and 797% in the power demand, enabling assembly and reducing the cost of equipment*

Keywords: *quenching, heat transfer coefficient, optimization energy, similarity analysis.*

1. INTRODUCTION

Quenching is a thermal process that aims to allow the steel acquire values of hardness and mechanical resistance appropriate to a particular application. This process consists in heating the steel to the temperature of the austenitic phase followed by a rapid cooling, this forms martensite. The most common means of cooling are: water, oil, aqueous solutions of polymers and inert gases (Santos, 2010).

A quenching process widely used in the manufacture of seamless tubes is the cooling in a water tank. In this method the tube is heated and then immersed in a water tank. To assist cooling and ensure homogeneity, the tube is supported on rollers causing it to rotate while receiving water sprays distributed evenly in the longitudinal direction. To promote cooling of the inner surface a jet of water is applied to one side of the tube. Figure 1 illustrates this quenching process.

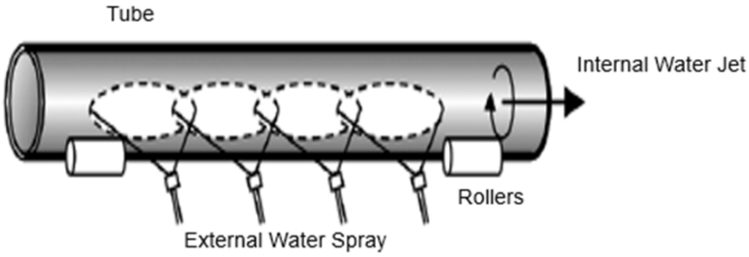

Figure 1. Cooling system of a tube during quenching in water tank.
(Sakamoto *et al.*, 2016)

The immerse quenching process described is governed by five basic parameters that influence significantly the final quality of the steel: (1) speed water jet incident on the tube (2) immersion time, (3) tube rotation, (4) water temperature and (5) tube temperature. In general, these parameters are related to the heat transfer between steel and water. Studies on tempering systems are costly and time-consuming. Mathematical models allow simulate different geometries as well as different cooling conditions. One of the objectives of these models is to predict the cooling profile of the tubes, so that it is possible to optimize the process variables, reducing rework costs and preventing defects, such as mechanical properties outside of the desired range, warping and cracking. The more efficient control of the process variables contributes to the final quality of the product and represents an important step towards integration with other production systems.

However, one of the limitations of mathematical models stems from the high uncertainty of the heat transfer coefficient between the tube and the water.

The heat transfer coefficient between the outer wall of the tube and the water is approximately constant along the length, since all parts of the tube are exposed to water jets as it rotates (uniformity of the thermo-hydraulic conditions). The heat transfer coefficient between the inner wall of the tube and the water flowing in its interior on the other hand is more complex. This cannot be considered constant, since it varies along the flow water for two reasons: (1) the heating of the water along the flow inhibits the heat transfer, causing a reduction of the heat transfer coefficient in the longitudinal direction; (2) the hot wall of the tube induces the formation of vapor along the tube, which reduces the specific mass of the liquid / vapor mixture in the longitudinal direction, causing an increase in the fluid velocity and, consequently, an increase in the heat transfer coefficient. Depending on adjustments to the temperature and velocity of the water of the internal jet, the effects (1) and (2) can compensate for, implying an approximately uniform internal heat transfer coefficient along the tube. (Sakamoto *et al.*, 2016)

The determination of reliable heat transfer coefficient allows mathematical models which more accurate and represents a major step for computer simulations and be used by technicians and metallurgical scientists, helping to improve the final quality of the product and reducing production costs.

A test bench able to reproduce the water flow inside the tube allowing the experimental determination of the heat transfer coefficient between the inner surface and the water is heavily constrained by the dimensions of industrial quenching process. The objective of this work is to develop a mathematical model that represents a small-scale test bank in order to optimize the design parameters, making it possible to build a bench capable of reproducing all the conditions of flow providing accurate and reliable results.

2. METHODOLOGY

2.1 Dimensional analysis

The dimensions of the operating system used in the industrial quenching process make it impossible to carry out real-scale (1:1) experiments, so the experimental determination of the heat transfer coefficient will be carried out using the concept of similarity (dimensional analysis) with help from a small-scale test bench. In it, the diameters of the tubes tested will be lower than those of the actual tubes as well as the flow of the cooling fluid.

There are two major advantages of working with dimensionless groups. The first is that the number of dimensionless groups is less than the number of dimensional variables, so that the number of experimental tests to be carried out is significantly lower. The other advantage is that the use of dimensionless groups enables to extend the results to other situations. For example, results obtained from tests with a tube of a certain diameter may be extended to a tube of different diameter, regardless of whether it is greater or less than the diameter tested. On it addition, of course, the use of similarity allows testing on systems with greater accessibility (Fox and McDonald, 2018).

Analyzing the convective heat transfer (Incropera, et al., 2014) and applying the Buckingham's Theorem (Fox and McDonald, 2018), three dimensionless groups are associated with the problem presented: (1) Nusselt number, (2) Reynolds number, 3) Prandtl number.

To ensure similarity between the real process (Prototype) and that reproduced on the test bench (Model), the Reynolds number of the water flow in the tube will be reproduced for flow in the test bench. As in Eq. (1).

$$Re_{prototype} = Re_{model} \tag{1}$$

2.2 Test bench

The test bench has the function to simulate the flow in the different sections of the tube, this means it is necessary to simulate different fluids temperatures, from environment temperature (beginning of flow) to saturation temperature (end of the flow). The simplified diagram of the test bench as in Fig. 2.

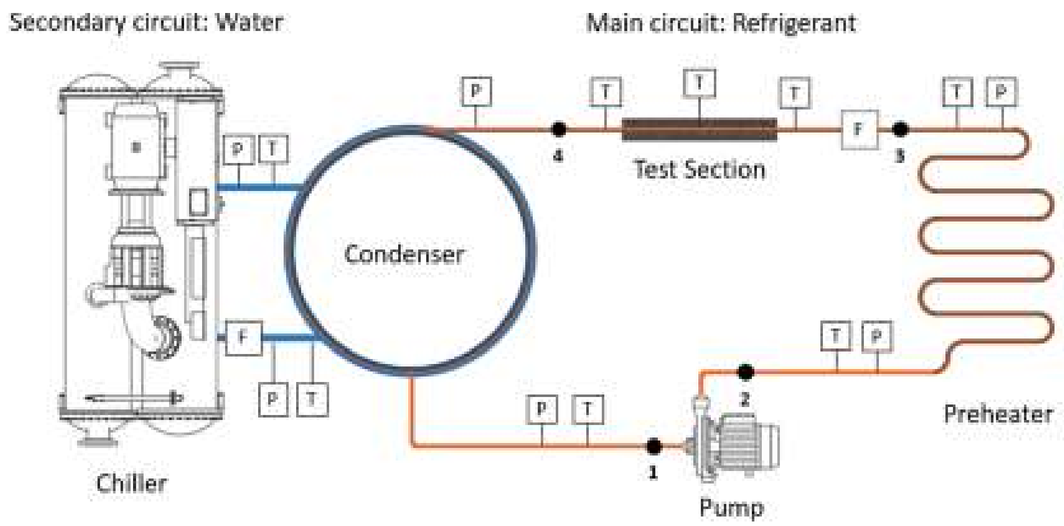

Figure 2. Simplified diagram of the test bench for the study of convective boiling

The working fluid enters the pump in a liquid state (point 1) and comes out as pressurized liquid (point 2). Under these conditions, the liquid enters the preheater, which is constituted by a tube (where the working fluid circulates), surrounded by a high-power electric resistance, fed by high electric power. This resistance heating fluid until the mixture liquid / vapor state with vapor quality x1 (point 3), which value can be set by energy balance in the preheater. Measurement of the fluid flow is essential for this balance and therefore the use of a precision flow meter (in this case turbine type meter) is indispensable. Then the liquid / vapor mixture enters the test section, which consists of a straight tube surrounded by a medium-power electric resistance. Thus, the Two-phase fluid in the test section receives heat, so that the vapor quality x_2 at the output of that section (point 4) is greater than the x1 vapor quality at the input. The value of x2 can be obtained by energy balance in the tests section. Finally, the fluid enters in the condenser, which is constituted by a tube wound in a propeller shape, surrounded by an envelope tube. The working fluid circulates in the central tube, while cold water flows counter currently in the annular space between the central tube and the envelope tube. Thus, the working fluid provides heat to the water, so that it leaves the condenser in a liquid state (point 1), restarting the cycle.

The Test Bank consists of two circuits: main circuit where the working fluid circulates and the secondary circuit where cold water circulates. The main circuit consists of the following components: pump, preheater, test section, condenser, temperature (T) and pressure (P) sensors and flow meters (F). The secondary circuit consists of chiller, pumps, temperature (T) and pressure (P) sensors and flow meters (F).

The heat transfer coefficient by convective boiling h_{eb} between the working fluid and the inner wall of the tube of the test section can be determined as in Eq. (2).

$$q = h_{eb}\, S\, (T_p - T_{sat}) \tag{2}$$

where q is the heat transfer rate in the test section, S is the area of the heat transfer surface, T_{sat} is the saturation temperature of the working fluid at boiling point and T_p is the temperature of the inner wall of tube in the test section.

The value of q is given by Eq. (3).

$$q = V_{cd}\, I \tag{3}$$

where V_{cd} is the potential difference and I is the electric current in the electrical resistance in the test section.

The value of S is given by Eq. (4).

$$S=\pi\ d_{int}\ L \tag{4}$$

where d_{int} is the inner diameter and L is the test section tube length.

The basic idea of the test bench is to generate a cloud of data points. This cloud will allow the determination of a convective heat transfer correlation involving ratios between the Nusselt and Prandtl numbers as a function of the Reynolds number.

2.3 Mathematical model

The proposed mathematical model allows to represents the main circuit described above. Thus, all basic components are characterized and the results reproduce the real behavior of the system.

The mathematical model of the test bank has 6 input variables: Reynolds of the prototype; Working fluid; Tube diameter in the test section; Fluid temperature and pressure; Vapor quality at the evaporator outlet.

The pipe diameter on the test bench was determined to suit the appropriate speed range (<2.5 m/s), except in the test section where we used the condition of similarity was used, thus the refrigerant circuit has a larger diameter with a reduction in the test section.

With these variables it is possible to use the similarity condition of Eq. (1) to determine the volumetric flow rate of the system Eq. (5).

$$\dot{V}=\left(\frac{Re_{prototype}\ \mu_{fluid}}{\rho_{fluid}\ D_{test\ tube}}\right)\ S_{test\ tube} \tag{5}$$

where $Re_{prototype}$ is the prototype Reynolds number, μ_{fluid} is the viscosity of the working fluid, ρ_{fluid} is the working fluid density, $D_{test\ tube}$ is the tube diameter of the test section and $S_{test\ tube}$ is the cross-sectional area of the tube.

The mass flow rate of the working fluid can be determined as in Eq. (6).

$$\dot{m}=\dot{V}\ \rho_{fluid} \tag{6}$$

where: $\dot{V}$ is the volumetric flow rate of the working fluid and ρ_{fluid} is the density of the working fluid.

The required power in the evaporator is determined by energy balance Eq. (7).

$$\dot{Q}_{evap}=\dot{m}\left[(x\ h_{lv}+h_l)-h_{evap\_in}\right] \tag{7}$$

where $\dot{m}$ is the working fluid mass flow rate, h_{evap_in} is the enthalpy of the fluid at the evaporator inlet and h_l is the enthalpy of the saturate liquid, hl_v is the enthalpy of vaporization of the fluid and x is vapor quality at the evaporator outlet.

The state of the fluid at the outlet of the tube in the test section is determined by energy balance Eq. (8).

$$h_{tube\_test\_out}=h_{tube\_test\_in}+\frac{\dot{Q}_{tube\_test\_out}}{\dot{m}} \tag{8}$$

where $\dot{Q}_{tube\_test\_out}$ is the power supplied to the test tube, $\dot{m}$ is the working fluid mass flow rate and $h_{tube\_test\_in}$ is the enthalpy of the fluid at the input of the test section.

The reject power in the condenser is determined by energy balance Eq. (9).

$$\dot{Q}_{cond}=\dot{m}\ (h_{cond\_out}-h_{cond\_in}) \tag{9}$$

where $\dot{m}$ is the working fluid mass flow rate, h_{cond_in} is the enthalpy of the fluid at the chiller inlet and h_{cond_out} is the enthalpy of the working fluid at the chiller outlet.

2.4 Preheater sizing

The preheater consists of a tube wound by an electric resistance of the helical microtubular type as in fig. 3. In this resistance, the electric power supplied is a function of the contact area between the resistance and the tube wall, therefore the preheater must be sized so that the power of the electric resistance is equal to or greater than that required by the working fluid. The length of the preheater can be determined by Eq. (10).

Figure 3 - Microtubular helical resistance
Quality-Up catalog

$$L_{preheater} = \frac{L_{res}\, B_{res}}{P_{tube}} \tag{10}$$

where L_{res} is the length of the resistance given by Eq. (11); B_{res} is the width of the resistance and P_{tube} is the perimeter of the tube given by Eq. (12):

$$L_{res} = \frac{\dot{Q}_{evap}}{W_{\mathrm{res}}} \tag{11}$$

where $\dot{Q}_{evap}$ is the power required by the preheater in W and W_{res} is the power of resistance in W/m

$$P_{tube} = \pi * D \tag{12}$$

where D is diameter of the tube.

2.5 Condenser sizing

An extension of Newton's Law of Cooling with the global coefficient of heat exchange (U) in place of the convective coefficient of heat exchange (α) and considering a temperature difference (∆T) varying with the position of the heat exchanger, can be used to determine the area of a double tube heat exchanger, as in Eq. (13).

$$S_{evap} = \frac{\dot{Q}_{evap}}{U\, DTML} \tag{13}$$

where U is the global heat transfer coefficient; DTML is the logarithmic temperature difference; given by Eq. (14).

$$DTML = \frac{(T_{H\,in} - T_{C\,in}) - (T_{H\,out} - T_{C\,out})}{\ln\left(\frac{(T_{H\,in} - T_{C\,in})}{(T_{H\,out} - T_{C\,out})}\right)} \tag{14}$$

where $T_{H\,in}$ e $T_{H\,out}$ are the inlet and outlet temperatures of the hot fluid; $T_{C\,in}$ e $T_{C\,out}$ are the inlet and outlet temperatures of the cold fluid.

The global heat transfer coefficient, U is given by Eq. (15).

$$U = \frac{1}{\frac{1}{\alpha_H} + \frac{1}{\alpha_C}} \tag{15}$$

where α_H e α_C are the convective heat exchange coefficients of hot and cold fluid respectively.

The flow of both fluids (hot and cold) can be considered monophasic (the amount of vapor is small), so the coefficient of heat transfer can be calculated through Dittus-Boelter correlation expressed in Eq. (16).

$$\alpha = \frac{(0{,}023\ Re^{0.8}\ Pr^{0{,}3})\ k}{\mathrm{D}} \tag{16}$$

where Re is the Reynolds number; Pr is the Prandtl number; k is thermal conductivity of the fluid; D the diameter of the tube.

The length of the condenser can be calculated according to Eq. (17).

$$L_{condenser} = \frac{S_{evap}}{\pi\ D} \tag{17}$$

2.6 Pressure Drop Calculation

The drop pressure in the system is approximated by the drop pressure of a single-phase flow, since the amount of vapor is small. This is determined by Eq. (18).

$$\Delta P = f\,\frac{L^{*}\upsilon^{*}\ G^{2}}{2^{*}D} \tag{18}$$

where L is the pipe length of the main circuit (Fig. 1), D is the tube diameter, f is the friction factor, G is the mass velocity and υ is the specific volume of the fluid.

2.7 Organic Fluid Selection

The optimization of a process or system is to obtain the best solution for the process considering certain restrictions and using specific methods. (Andrade, 2014)

In terms of optimization there is no optimum fluid, there is an optimal evaporation temperature of the fluid that maximizes its efficiency (pinch point). (Quoilin *et al.*, 2011) Thus, the fluid must be chosen so that the working temperature is close to the pinch point.

Aboelwafa *et al.*, (2018) reviews the work done on the solar Rankine cycle systems for power generation and focuses on the working fluids investigated in the literature and the application of these systems in water pumping and water desalination.

To quantify the best solution obtained in the optimization process it is necessary to define an objective function $f(x)$, which should be maximized or minimized, according to the need of the problem. The end result of the objective function is determined by the manipulation of adjustable (or manipulated) variables, which can physically represent the size of the equipment or the operating conditions. Therefore, in an optimization it is defined:

- Maximum or minimum of:
  - $f(x)$ objective function
- Subject to:
  - $h(x) = 0$ equality constraints
  - $g(x) \geq 0$ inequality constraints

where x is a vector of n adjustable variables (x_1, x_2, ..., x_n), $h(x)$ is a vector of constraint equations and $g(x)$ is a vector of constraint inequalities.

In the case of the proposed problem it is desired to choose the organic fluid that minimize the pressure drop. Thus:

$$f(x) = min(\Delta P) \tag{11}$$

The manipulated variables are: Working Fluid; Tube Diameter

The constraints involved in the process are: Saturation temperature > Environment temperature; Saturation temperature < 100 °C; Power required in the preheater < 25 kW; Reynolds of the model = Reynolds of the prototype; The mathematical model and physical term of the test bench must be respected.

With the pre-established parameters a multi-objective optimization will be performed with the use of a genetic algorithm. Three values of diameters will be used: 0.5; 1 and 1.5 inches (all commercial diameters) and thirty organic fluids. Table 1 contains the fluids used and some of their properties.

Table 1. Fluids used and some properties
Properties of the liquid phase at room temperature (25 ° C)

Fluid	Density [kg/m³]	viscosity [kg/ms]	specific heat [kJ/kgK]
Water	997,00	8,90E-04	4,18
Ammonia	602,76	1,32E-04	4,78
Ethane	314,95	3,70E-05	8,05
Propylene	506,28	9,50E-05	2,67
Propane	492,36	9,71E-05	2,72
IsoButane	550,65	1,51E-04	2,43
Pentane	620,78	1,81E-04	2,32
Isopentane	614,97	1,94E-04	2,27
Cyclopentane	740,29	3,09E-04	1,81
n-Heptane	679,50	3,90E-04	2,24
Benzene	873,44	6,02E-04	1,74
n-Octane	698,18	5,09E-04	2,23
n-Nonane	714,01	6,53E-04	2,21
Toluene	862,16	5,52E-04	1,70
o-Xylene	875,96	7,56E-04	1,76
EthylBenzene	862,46	6,11E-04	1,75
Methanol	786,52	5,44E-04	2,53
Ethanol	785,05	1,08E-03	2,43
R32	961,01	1,12E-04	1,94
R134a	1206,71	1,95E-04	1,42
R152A	899,47	1,61E-04	1,80
R227EA	1387,67	2,39E-04	1,18
R236FA	1359,75	2,86E-04	1,24
R236EA	1426,28	3,83E-04	1,26
R245fa	1338,46	3,95E-04	1,32
R22	1190,65	1,27E-04	1,26
R123	1463,89	4,18E-04	1,02
R1234yf	1091,91	1,45E-04	1,39
R1234ze(E)	1163,11	1,88E-04	1,39

3. RESULTS

The analysis of simulations performed using the mathematical model showed that the system design has two critical points in different operating situations. When the fluid is cold, the higher volumetric flow rate is required to maintain the flow similarity condition. On the other hand, the higher power demand in the preheater occurs when the fluid reaches the saturation temperature. The graph of Fig. 4 shows that the volumetric flow decreases with increasing temperature of the flow, this is because the viscosity and the density of the fluid are sensitive to the temperature increase as observed in Fig. 5, since the Reynolds number depends of these properties, the increase in temperature causes a lower volumetric flow rate to maintain the similarity condition. The Fig. 6 shows that the peak in energy demand occurs at the saturation temperature of the fluid, this is because latent heat is required for the fluid change phase. Can be concluded that the power of the pump must be determined for the situation of lower temperature and the electric power of the preheater for the situation where the formation of vapor in the flow occurs. The Figs. from 4 to 6 refers to water.

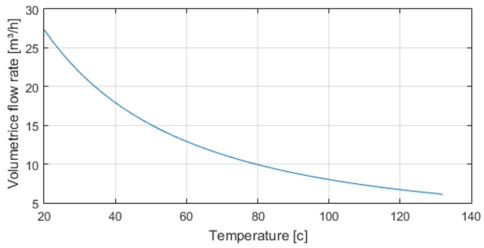

Figure 4 – Water volumetric flow rate variation with increasing temperature

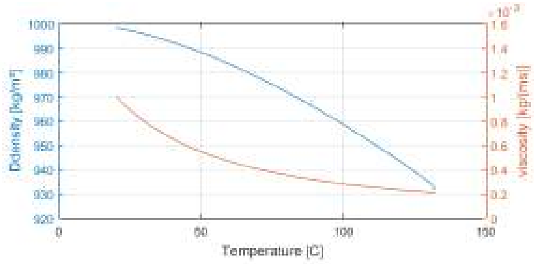

Figure 5 - Variation of water properties with increasing temperature

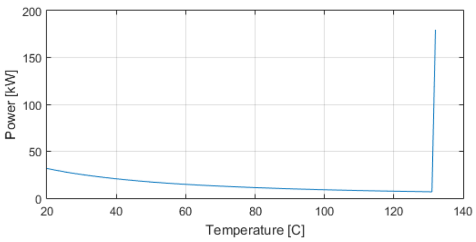

Figure 6 - Variation in power demand with increasing temperature

The fluid circuit was originally designed to use the same diameter in all sections of the test bench, however the early results showed that only the optimization of fluid would not be enough to keep the pressure drop within the usual values. Using pentane (optimized fluid) as a reference, the system pressure drop is 65 bar, which is high for this type of application. The main cause for such high pressure drop is the fluid velocity (about 13 m / s). This speed is much higher than recommended, which is usually less than 2.5 m / s. To solve this problem, it was proposed a configuration with two bands of diameter, one larger in the refrigerant circuit and one smaller in the test section. The larger diameter allowed the fluid to circulate most of the time with lower speeds (approximately 1.5 m / s), while the reduction in the test section allowed speed to increase therein, which is essential to maintain the similarity. The load loss with this new setting is significantly lower than the previous setting, approximately 3 bar.

The result of the optimization showed that the fluid with the lowest pressure drop was the Pentane with a pressure drop of approximately 3 bar and a power demand in the preheater of approximately 21.4 kW. Among the analyzed fluids R236fa presented the worst result with a pressure drop of approximately 7.7 bar and a power demand of 24 kW.

The replacement of the water by an organic fluid enabled the assembly of the proposed test bench while the optimization allowed a more compact assembly and of lower cost. Fig.7 shows the proposed layout for assembly. The optimization allowed to work with a temperature range of 25 to 75 ° C, with the capacity to generate fraction fractions of up to 5%. The Reynolds range in the test section is of the order of 7.5E5 bringing speeds of approximately 13 m / s to a 0.5-inch diameter. The circuit tubing is 1.5 inches inching at speeds 1.5 m / s

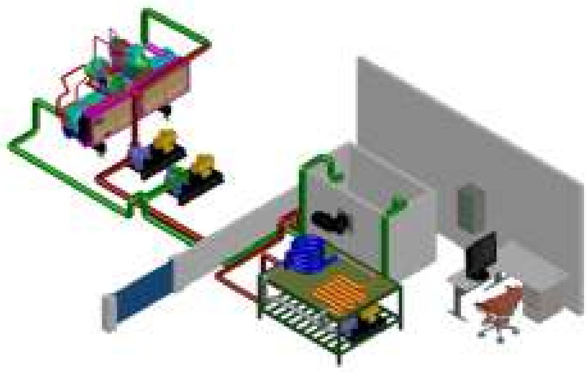

Figure 7 – Layout of test bench

4. CONCLUSIONS

The mathematical model allowed to simulate in a practical way the operation of the test bench. It was possible to identify the critical operating points of the system and select the working fluid that optimized the operation.

The fluid with the lowest pressure drop among the analyzed was Pentane, 3 bar and power demand of 21.4 kW. Compared with the R236fa which had the largest pressure drop 7.7 bar and power demand of 24 kW, the optimization of the fluid allowed a 156% reduction in the pressure drop and 12% of reduction in the power demand of the preheater. Compared to water, 8 bar pressure drop and 192 kW of power demand the reduction was 166% in the pressure drop and 797 % in the power demand.

5. ACKNOWLEDGEMENTS

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