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EXPERIMENTAL EVALUATION OF WIRE & TUBE CONDENSERS: CONFINING WALLS EFFECTS

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ABSTRACT

This paper presents an experimental investigation into the performance of wire and tube condensers widely used in small refrigeration appliances. The primary purpose of this study was to gather and analyze experimental data on the heat exchanger performance under various operating conditions to establish whether the confining walls play a dominant role in the heat transfer from this unit. The experiments were carried out using a specially developed testing apparatus, consisting of a thermostatic bath, a water pump and a turbine flow meter. A test section to vary the heat exchanger position in relation to the adjacent surfaces was also developed and assembled within a temperature and humidity controlled test room. Some experimental results are presented and discussed with reference to the optimization of the heat transfer rate. In particular the effects of the gaps between the refrigerator and the back, side and bottom walls of the test section were analyzed. The results prove that the condenser performance is strongly affected by its position in relation to the adjacent surfaces.

1. INTRODUCTION

An improvement in the efficiency of household refrigeration systems contributes significantly to a reduction in the world consumption of energy and also to a reduction in the global warming. According to PROCEL (National Program for Electricity Conservation) data from April 2001, 27% of the total Brazilian energy consumption is related to the residential sector and, of this, 32% is spent on refrigeration. In the service sector, which represents 17% of the total, refrigeration is responsible for 17%. Summing the residential and service sector showed that refrigeration is responsible for approximately 11.5% of energy consumption in Brazil.

Refrigerators and freezers are generally sized through the use of basic calculation procedures, applied in an isolated way to each component of the system. In some cases, computational codes, which allow the simultaneous simulation of the behavior of all the system components, are also employed. Regardless of the calculation process used, detailed information on the behavior of each system component is necessary.

In this study the emphasis will be placed on the condenser, more specifically on that of the wire and tube type (Arsego, 2003). The objective is to evaluate experimentally the performance of this component as a function of the geometry, the operating conditions and its position in relation to the adjacent surfaces. This first study will focus only on the aspects related to confinement.

A wire and tube condenser consists of a steel tube bended into a single-passage serpentine shape, with wires spot-welded perpendicularly on both sides. Such condensers may be assembled with tubes in a vertical or horizontal position and the air movement can be forced or natural. In this study only heat exchangers with tubes in the horizontal position and with natural air movement will be considered.

2. PREVIOUS STUDIES

Witzell and Fontaine (1957) carried out the first experimental study on wire and tube condensers. The experimental apparatus consisted on the circulation of hot water through the heat exchangers under controlled conditions. The total rate of heat rejection to the environment was calculated from the measurement of the water mass flow rate and the temperature drop along the heat exchanger. The radiative heat transfer rate was calculated using a simplified process and subtracted from the total heat transfer rate as a way of obtaining the convective component. Since the Prandtl number remained constant during the experiments, the Nusselt number was correlated only with the Grashof number. The geometric parameters which most significantly affected the heat transfer were incorporated into a characteristic diameter, defined as an area weighted average of the tube and wire diameters. Witzell and Fontaine's (1957) experiments were carried out with the heat exchangers in a horizontal position, without rigid control of the environmental conditions and with variations only in the number and diameter of the wires. Also, the average difference between the air and water temperatures at the condenser inlet was kept at around 30^{o}C, this being much higher than usual. Furthermore, they considered only unconfined air flows, which are not typical of real applications. Despite these restrictions, the work of Witzell and Fontaine (1957) introduced an appropriate experimental procedure which has been used in subsequent studies, including the one reported here.

Cyphers et al. (1959) used an experimental apparatus similar to that of Witzell and Fontaine (1957) to investigate the effects of inclination and confinement. They found that in a condenser with horizontal tubes and vertical wires the heat transfer rate is reduced by 25% when the heat exchanger is moved from the horizontal to the vertical position, in the absence of any confinement. The confinement effect was studied using two vertical and parallel plates with the same dimensions as those of the heat exchanger (width = 629 mm, height = 1080 mm). For a certain distance between the plates, the following situations were considered: i) condenser inclined with the upper and lower parts touching the plates, ii) condenser positioned vertically in the middle of the plates and iii) condenser positioned vertically placed at 5, 10 and 15cm from one of the plates. It was found that when the plates are at more than 50cm distance the behavior of the inclined condenser become independent of the confinement. With a reduction in the distance, there is an increase in the heat exchanger inclination and a substantial decrease in the heat transfer. For instance, for a distance of only 10 cm the heat transfer is reduced by 40% in relation to a situation without confinement. It was also found that for a distance of 7.5 cm there is no difference between the inclined condenser and that placed in the middle of the plates. As the distance is increased, however, the heat transfer rate obtained with a heat exchanger positioned vertically in the middle of the plates is always inferior to that obtained with an inclined heat exchanger. In addition it was observed that the more asymmetric the heat exchanger position between the plates, the lower the heat transfer rate. A theoretical model based on heat transfer correlations for vertical and horizontal cylinders was also proposed by Cyphers et al. (1959).

Witzell et al. (1959) and Papanek (1958) carried on the work started by Witzell and Fontaine (1957), introducing improvements to the test section. The condensers were placed inside a wooden box, painted black inside and perforated in the lower and upper parts to allow the passage of air. The idea was to eliminate the ambient air temperature fluctuations and to obtain a better control of the radiative heat exchanges. All the condensers were comprised of tubes with the same diameter and spacing, placed in the vertical position. The diameter and spacing of the wires and the inclination of the heat exchangers were varied during the experiments. Of the tests carried out it was evident that the radiation represented around 30% of the total heat transfer. Like Cyphers et al. (1959), they showed that the heat transfer rate is reduced when the condenser is moved from the horizontal to the vertical position, but that this variation depends on the diameter and spacing of the wires. Correlations for the Nusselt versus Grashof numbers, based on a characteristic diameter and for horizontal and vertical heat exchangers, were also presented.

Collicott et al. (1963) carried on the work of Witzell and Fontaine (1957) and Witzell et al. (1959), seeking to quantify the effective shape factors of the heat exchangers which until then had been considered as unity. To this aim they constructed a test section which consisted of an evacuated chamber, in which the heat exchangers to be tested were placed. Twelve heat exchangers were used, all with the same tube diameter, but with several variations in the diameter and spacing of the wires. The radiative heat transfer rate was determined by measuring the water mass flow rate and the inlet and outlet temperatures. The effective shape factor was then calculated for each heat exchanger and expressed graphically as a function of the diameter to spacing ratio of both the tube and the wires. Another test section, which allowed a variation in the heat exchanger inclination and the measurement of the combined effect of radiation and convection, was also developed. The tests carried out in this section showed that the heat transfer was essentially a function of the wire diameter to spacing ratio, tube diameter to spacing ration and the Rayleigh number. It was found that for a horizontal heat exchanger, the heat transfer is practically independent of the Rayleigh number and of the wire diameter to spacing ratio, and was a function only of the tube diameter to spacing ratio. It was also shown that the heat transfer is reduced with an increase in the heat exchanger inclination, this relation being affected by the Rayleigh number.

Despite the ever increasing use of wire and tube condensers, practically nothing else was published regarding the use of such heat exchangers until 1997, when Tanda and Tagliafico (1997) and Tagliafico and Tanda (1997) published their work. The experimental facility used by these researchers was very much like the one used by Witzell and Fontaine (1957). The water mass flow rate was measured by a turbine flow meter and the average surface temperature of the wires by means of infrared thermography. The test section was placed in a large isolated room without a rigid control of the environmental conditions and did not take into account any type of confinement. Forty two heat exchangers were tested in an attempt to identify the effects of the geometry and the operating conditions. The radiative component was calculated analytically and subtracted from the total heat transfer rate. In order to reduce the effects of possible errors associated with this calculation, the heat exchangers were painted with a low emissivity paint. The convective component was correlated with the difference between the tube wall and air temperatures, with the heat exchanger height and with the diameter and spacing of both the tube and wires. The proposed correlation reproduced the experimental data with an average deviation of ± 6%. It should be pointed out that the experiments carried out by Tanda and Tagliafico (1997) employed temperature differences between the tube wall and the surrounding air ranging from 17 to 48^{o}C, these values being substantially higher than the ones usually found in tropical countries like Brazil.

Although not directly related to the present study, the contributions of Hoke et al. (1995) and Hoke et al. (1997) who studied forced convection wire and tube condensers, should be mentioned. They carried out 1700 tests, using seven heat exchangers and different air velocities and angles of attack. The test section consisted of a wind tunnel which allowed the control of the air velocity and direction. The total heat transfer rate was also evaluated from the water mass flow rate and temperature drop measurements. An analytical model for the computation of the effective shape factor was also presented, this being adopted in this study with minor modifications.

3. EXPERIMENTAL APPARATUS

The experiments were carried out with the experimental apparatus, shown schematically in Figure 1. Such apparatus is composed by the following basic elements: i) thermostatic bath, ii) rotary vane pump, iii) turbine flow meter, iv) control valves, v) filters, vi) mixers and vii) data acquisition system. The test section was assembled inside a temperature and humidity controlled room, constructed according to the specifications of the ISO 7371 (1985) standard.

The total heat dissipation rate was also obtained from the water mass flow rate and from the water temperature difference between the inlet and outlet of the heat exchangers. The water mass flow rate was measured by a turbine flow meter and the temperature difference by a differential thermocouple installed between two mixers (Arsego, 2003). The mass flow rate and the temperature difference were measured with a maximum uncertainty of ± 0.3 kg/h and ± 0.2 ºC, respectively.

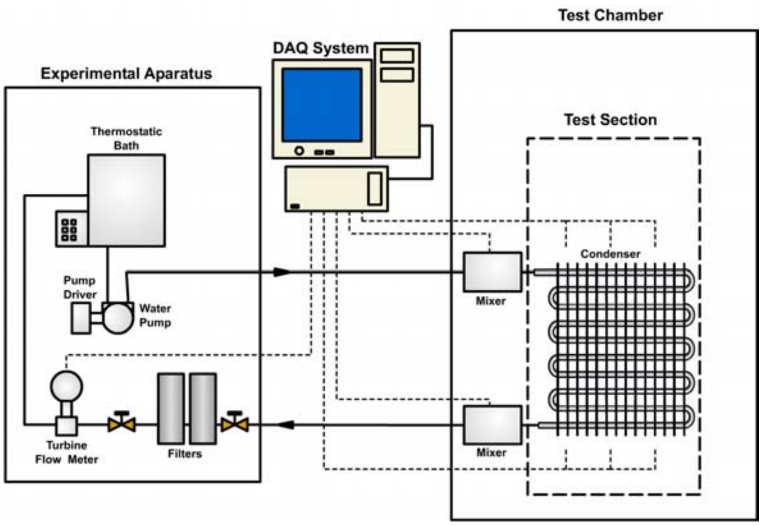

Figure 1: Experimental apparatus

4. TEST SECTION

The test section is shown in detail in Figure 2. This section allows the positions of the condenser, side walls, upper wall and the refrigerator to be varied.

The test section was constructed with wooden plates of 18mm thickness, totally painted with black matt paint. The refrigerator, also made of wood, had dimensions similar to a no-frost, 430 liter refrigerator. The side walls were constructed according to the specifications of the ISO 7371 (1985) standard. An upper wall was additionally used in order to eliminate possible disturbances caused by the supplied air and also to reproduce the conditions in a kitchen.

The effects of positioning were studied using a reference point (0,0) which corresponded to the point where the floor met the rear part of the test section (see Figure 2). The heat exchanger and the refrigerator positions were established by the coordinates (XT, YT) and (XG, YG), respectively. The distance of the refrigerator from the side walls was specified through the ZG variable.

The air temperature at the inlet and outlet of the heat exchanger was evaluated by eight thermocouples embedded in copper cylinders to provide good thermal contact with the surrounding air (ISO 7371, 1985). The air temperature at the inlet of the test section was evaluated by four thermocouples, all embedded in copper cylinders and placed between the side walls and the refrigerator, but at a distance of 1m from the back wall.

The temperature of the back wall of the refrigerator was also measured by three thermocouples, installed in the central plane and at 470, 950 and 1390 mm from the floor. Three thermocouples were also used to measure the temperature of the condenser walls, all fixed in a central plane and equally distributed along the height of the heat exchanger.

5. EXPERIMENTAL RESULTS

The heat exchanger under analysis was comprised of 19 tube passes of 6.5mm external diameter and of 65 pairs of wires of 1.5mm diameter, covering an area of 440 mm width by 1100 mm height. The water flow rate was kept at an almost constant value of 0.49 liters/min. Also, the room temperature was maintained at 32^{o}C. The water temperature at the heat exchanger inlet was maintained at between 43.3 and 44.0 ^{o}C. The aim of this procedure was to minimize the variations of the temperature difference between the heat exchanger and the surrounding air.

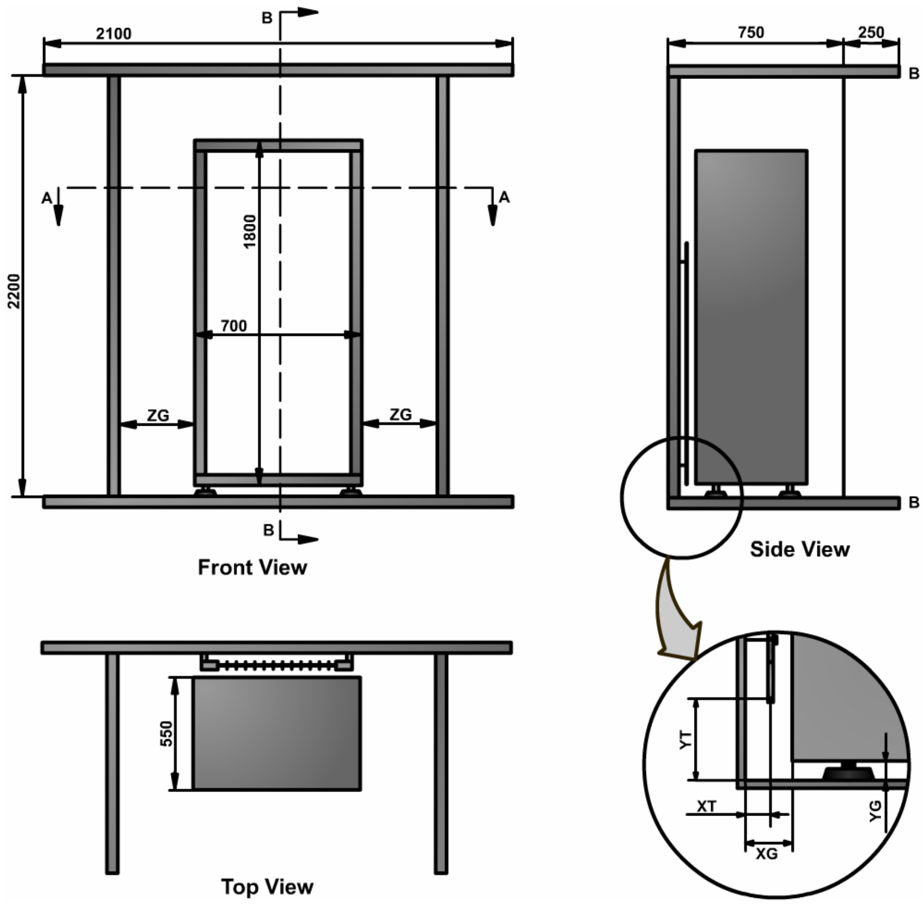

Figure 2: Test section

5.1 Gap between the refrigerator and the rear wall (XG with XG-XT = 30 mm)

The geometric parameters YT = 325 mm, YG = 45 mm and ZG = 300 mm, remained fixed, varying only the parameters XT and XG. The distance of the refrigerator from the rear wall, XG, varied between 30 and 210 mm. As the distance between the heat exchanger and the refrigerator remained fixed at 30 mm, the parameter XT varied between 0 and 180 mm.

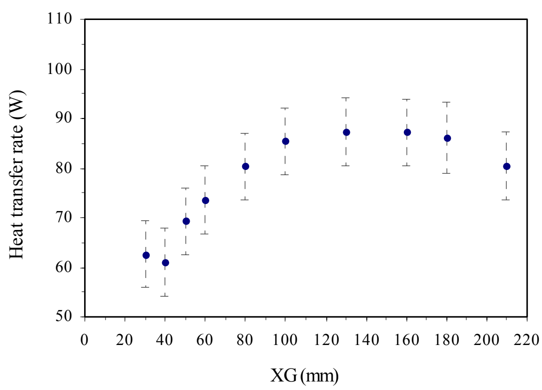

Figure 3: Effect of the gap between the refrigerator and the real wall

Figure 3 illustrates the heat transfer rate variation for the condenser as a function of the parameter XG. As can be seen the heat transfer rate is significantly affected by the gap between the refrigerator and the rear wall of the test section. This result reflects the behavior observed by Cyphers et al. (1959). It can also be noted that as the gap decreases, the heat transfer rate also decreases, mainly due to the reduction in the circulating air flow. With an increase in the gap the heat transfer rate increases, up to a maximum, at around 130 mm.

After this point the heat transfer rate starts to decrease due to a reduction in buoyancy. .It is also evident that a special care must be taken in establishing the parameter XG, during normalized refrigerator tests, since this parameter affects significantly the condenser heat dissipation rate and consequently the refrigerator performance.

5.2 Gap between the heat exchanger and the rear wall (XT)

In this analysis the parameters XG = 100 mm, YG = 45 mm, ZG = 300 mm, YT = 325 mm, remained fixed. The gap between the heat exchanger and the rear wall, characterized by the parameter XT, varied between 0 and 90 mm. Figure 4 shows the condenser heat transfer rate as a function of the parameter XT.

It can be observed that the heat transfer rate is affected significantly by the parameter XT, decreasing with proximity to one of the two adjacent surfaces. The appearance of a maximum point, at XT = 50 mm, when the heat exchanger is positioned at exactly half the distance between the two adjacent surfaces (XG/2) can also be observed. This result reflects the behavior observed by Cyphers et al. (1959).

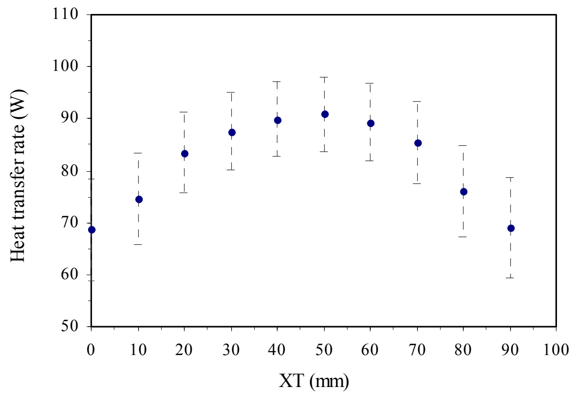

Figure 4: Effect of the gap between the heat exchanger and the real wall

5.3 Gap between the refrigerator and the rear wall (XG with XT = XG/2)

In the above item it was shown that the heat transfer is at a maximum when the heat exchanger is positioned at half the distance between the refrigerator and the rear surface. In this item, the distance between these surfaces, characterized by the parameter XG, varied between 40 and 180 mm, however, keeping the heat exchanger always in the central plane, that is, at XT=XG/2. The other parameters YG = 45 mm, ZG = 300 mm, YT = 325 mm, remained fixed.

As in Figure 3, Figure 5 shows that the heat transfer rate of the condenser increases with an increase in the distance between the adjacent surfaces. In the present case the heat transfer rates are slightly higher than those shown in Figure 3, which is in agreement with the observations of Cyphers et al. (1959).

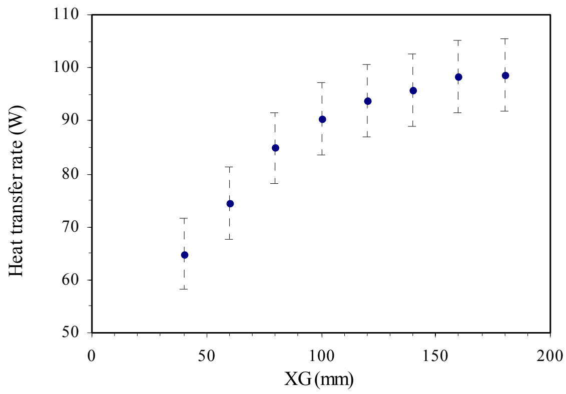

Figure 5: Effect of the gap between the refrigerator and the real wall

It can be seen that the gradient of the heat transfer rate decreases with an increase in the parameter XG. A limit value seems to have been reached at XG = 160 mm. Unfortunately, due to test section limitations, it was not possible to carry out experiments with gaps greater than 180 mm, which would allow the determination of whether or not a maximum point similar to that shown in Figure 3 occurs.

5.4 Gap between the refrigerator and the side walls (ZG)

In this case the only parameter which varied was ZG, representing the gap between the side walls and the refrigerator. The other parameters XG = 100 mm, YG = 45 mm, YT = 325 mm and XT = 50 mm, remained fixed.

Figure 6 shows the effect of the side gap on the condenser heat dissipation rate. It may be seen that the heat transfer rate is not affected by the parameter ZG, for gaps above 80 mm. Between 0 and 80 mm, the heat transfer rate reaches a maximum point, and this is due to the variation of buoyancy resulting from the gap variation.

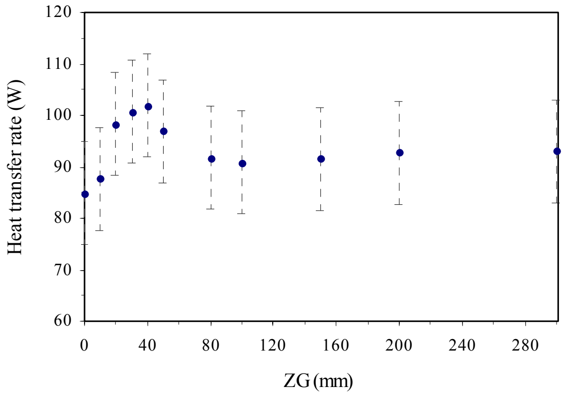

Figure 6: Effect of the gap between the refrigerator and the side walls

It can therefore be concluded, that the establishing of this parameter during normalized refrigerator tests (ZG = 300 mm) does not need to be made with great care, due to its low impact upon the heat dissipation rate of the condenser and consequently upon the refrigerator performance.

5.5 Inclination of the heat exchanger

In this experiment the effect of the inclination of the heat exchanger was studied, since there are indications in the literature that the heat transfer rate may be increased up to 25% when the heat exchanger is moved from the vertical to the horizontal position. All of the geometric parameters, XG =100 mm, YG = 45 mm, XT = 50 mm, YT = 325 mm and ZG = 300 mm, remained fixed. The inclination of the heat exchanger was therefore varied between the vertical (0^o) and the maximum inclination permitted by the gap between the refrigerator and the real wall (6.5^o).

The results are illustrated in Figure 7. It can be seen that there is no effect of the condenser inclination on heat transfer rate, within the limited range of variation considered.

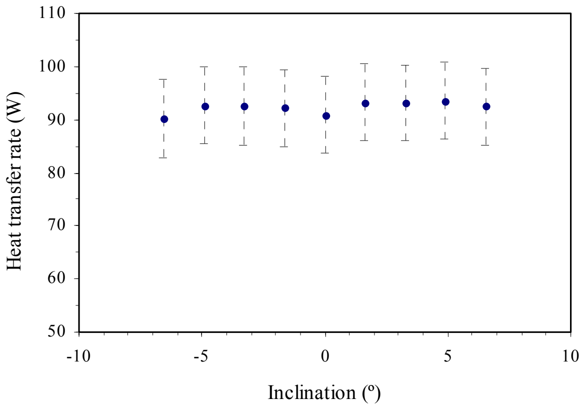

Figure 7: Effect of the inclination of the heat exchanger

6. CONCLUDING REMARKS

A very comprehensive experimental study was carried out, with a focus on the experimental evaluation of wire and tube condenser performance. The experiments were carried out with strict control of the operating conditions which were maintained close to the usual application values.

It was shown that condenser performance may vary greatly, depending on its positioning. Of the parameters analyzed, the gap between the refrigerator and the rear wall of the test section was the most relevant. For a specific gap, it was shown that the heat dissipation rate reaches a maximum value when the heat exchanger is positioned in the mid plane. The other parameters analyzed affected only slightly the heat exchanger performance.

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