# DESIGN ANALYSIS OF A FINNED TUBE HEAT EXCHANGER FOR AN AIR-TO-WATER REVERSIBLE HEAT PUMP WORKING WITH PROPANE (R290) AS REFRIGERANT

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## ABSTRACT

The finned tube heat exchanger "coil" is widely used in refrigeration, air conditioners and heat pump applications. In this study, a statistical method together with a simulation model were employed to analyze the effects of several coil geometry factors on the performance (COP) of an air-to-water reversible heat pump in both modes of operation. The main aim of this investigation is to find a coil design that improves the efficiency of a heat pump working with R290 as refrigerant. As a result, it was found that certain relation exists between the number of circuits and tube diameter for which are possible to obtain the maximum COP in heating or cooling mode. In heating mode (coil as the evaporator), the combination of 9 circuits and tube diameter of 8.9 mm gives a maximum COP of 3.74 along with a pressure drop of 25.2kPa. For the coil acting as a condenser in cooling mode, the optimum COP is obtained with 6 circuits and tube ID of 6.5 mm. This design yields a 5% increase in the COP with a reduction of 49% and 27% in the refrigerant volume and total area with regard to the baseline design configuration, respectively.

**Key Words:** ***heat pump, heat exchanger, propane (R290), factorial design, optimization***

**NOMENCLATURE**

| | |
|---|---|
| A:NC | Number of circuits |
| B:NFT | Number of rows |
| C:NTF | Number of tube per rows |
| D:Tube ID | Inner tube diameter |
| E:FS | Fin spacing |
| AD | Interactions between the factor number of circuits and inner tube diameter |
| AC | Interactions between the factor number of circuits and number of tube per rows |
| BD | Interactions between the factor number of rows and inner tube diameter |
| AB | Interactions between the factor number of circuits and number of rows |
| DD | Inner tube diameter quadratic interactions |
| AA | Number of circuits quadratic interactions |
| HTC | Heat transfer coefficient |
| COP | Coefficient of Performance |
| $\Delta$P | Pressure drop |
| $NT_{active}$ | Total number of active tube |
| $A_p$ | Refrigerant sectional-cross area, (m$^2$) |
| $V_{coil}$ | Refrigerant side volume, (m$^3$) |
| $A_{ref}$ | Inside surface area, (m$^2$) |
| $L_{coil}$ | Length of one circuits, (m$^2$) |

**Subscripts**

| | | | |
|---|---|---|---|
| HM | Heating mode | ref | refrigerant |
| CM | Cooling mode | i | diameter |
| coil | Finned tube heat exchanger | d | diameter |

# 1 INTRODUCTION

The finned tube heat exchangers "coil" is widely used in refrigeration, air conditioners and heat pump applications. The optimization of coil constitutes a very complicated process due to a large number of geometry factors. In the last years, a number of publications analyzing the influence of "coil" geometry factors in several air conditioners and heat pump systems by modelling and experimental test (Douglas et al. 1996, Ragazzi and Perdersen 1996, Rice 1997, Bigot et al. 2000, Sadler 2000). Douglas et al. (1996) looked at cost-optimum smooth tube diameters for R22 and some alternative working fluids, such as: R134a, R290, R407C and R410a. They used water to refrigerant heat exchangers configured to have equivalent air-to-refrigerant resistance ratios of unitary air conditioners. Cooling COP and capacity were fixed and system cost was minimized. The analysis was done assuming one circuit in each heat exchanger. Their results show that the heat exchanger cost was proportional to tube material. For R290, the optimum diameters ratio relative to R22 for evaporator/condensers was 24.8/25.2 mm.

Rice (1997) investigated the relationship between the number of circuits and tube ID on the performance of an air-to-air reversible heat pump working with R22, R134a, R410A and R290. He found that with R410A and R290, the optimal tubes are one tube size smaller than for R22 with a similar numbers of circuits. In addition, R290 requires one extra circuit with respect to R410A for both heat exchangers (indoor and outdoor coils). The reduction in tube size's for R410A and R290 results in 20 to 67% lower charge requirements than for R22. For the outdoor coils with tube ID of 7.9 mm coupled with R410A and R290 delivers slightly lower performance with significantly lower charge while requiring one additional circuit. Sadler (2000) developed a model to design analysis of a finned tube heat exchanger for a residential air-conditioner working as a condenser with R22. The author analyzed the effects of changing the tube diameter, tube circuiting, number of rows, and fin pitch on the performance (COP) and seasonal COP of the system. The outcome of the test showed that tube diameter and tube circuiting could not be considered separately because they both affect the refrigerant side pressure drop.

From the previous work, it can be seen that a variety of optimum values have been obtained depending of the assumptions made, the optimization criteria selected and the refrigerant. In all previous works, the effect of coil geometry factors on the performance systems was carried out using traditional methods (parametric studies), by means of which it, is very difficult to give a general yardstick for the finned heat exchanger design. In this work, the optimized coil design process is conducted in tree stages:

- First, the performance of a heat pump system is measured for two different coil designs at several superheat conditions at the compressor inlet in both modes: heating and cooling.
- Based on these test, the ART[1] model of the heat pump was validated. The model is used to determine COP's value of the heat pump.
- Finally, a statistical method together with a simulation model were employed to analyze the effects of several geometry factors of coil design on the performance (COP) of an air-to-water reversible heat pump in both modes of operation. Keeping fixed the frontal area of the heat exchanger, the studied geometry factors are: number of circuits, number of rows, and number of tube per rows, tube ID and fin spacing. The factor levels used were selected taking into account the availability of standard commercial material. The main aim of these tests was to search for optimized coil design concepts for an air-to-water reversible heat pump working with R290, from the point of view of a better utilization of the heat exchanger area, and minimizing the heat exchanger volume.

[1] Advanced Refrigeration Technologies

## 2 EXPERIMENTAL SETUP AND TEST PROCEDURE

The heat pump test rig, shown in Fig. 1, is described more extensively in (Corberán et al. 2000, Blanco et al. 2004) and was designed to perform the characterization of a medium size reversible air--to--water heat pump. The whole range of climate conditions typical of Southern Europe can be simulated in a 50 m$^3$ climatic chamber in both modes, heating and cooling. The air--side--coil of the heat pump unit is attached to the climatic chamber, ensuring, by means of a variable fan, that the air pressure drop across the unit is null. The air treatment unit is located on the top of the climatic chamber and creates the desired environmental conditions inside the chamber. In the water--side of the heat pump unit, a hydraulic loop simulates the loads of the unit, ensuring that the standard operating conditions are maintained in the refrigerant--to--water brazed plate heat exchanger (BPHE). Within conditions contained in the EN-255 and EN-12055 standards, the unit was operating with an inlet water temperature of 45ºC and outlet water temperature of 50ºC in the heating mode (with outdoor temperature of 12ºC and humidity of around 80% in the climatic chamber) and an inlet water temperature of 12ºC and outlet water temperature of 7ºC in the cooling mode, (with outdoor temperature of 35ºC in the climatic chamber). These conditions are referred in all figures as A12W50 and A35W7, respectively.

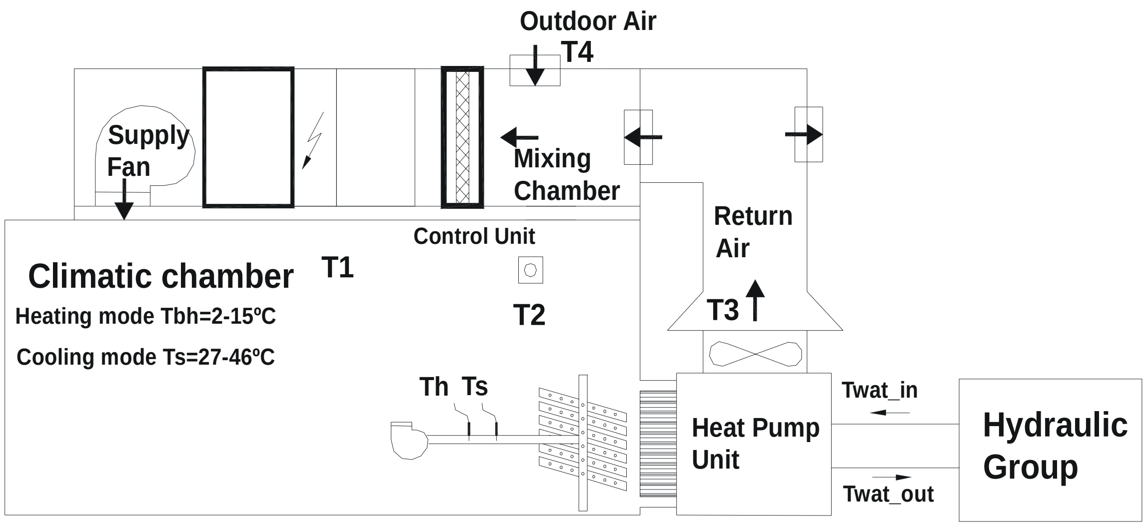


**Fig. 1. Schematic of the heat pump test rig.**

The heat pump used for the test is a modified version of an R22 catalogue model of CIATESA (IWA-95), specifically adapted, from the point of view of safety as well as performance, to be used with propane as refrigerant. All materials and measurement devices utilized were intrinsically safe. The heat pump unit is equipped with a scroll compressor, a thermostatic expansion valve, a BPHE with 46 plates and a fin--and--tube heat exchanger that acts as condenser/evaporator. It is thoroughly instrumented with several resistance temperature detectors, absolute pressure transducers and relative pressure meters. Inside the heat pump, the evaporation and condensation pressure were determined by means of pressure sensors located up-- and downstream of the compressor. A digital energy counter was used to measure compressor power consumption. Several PT-100 thermo-resistance temperature probes were placed to measure the levels of superheat and subcooling. A data acquisition system collected the information related to temperatures, pressures, secondary loop water mass flow rates, air humidity and compressor power input measurements.

Heating and cooling capacities from the secondary loop water mass flow rate and BPHE outlet/inlet water temperature difference. Temperature difference, which is critical for a precise measurement, was determined by means of a pair of carefully calibrated PT-100-type thermo-resistance probes. In all the tests, during the characterization of the different designs, the temperature levels were regulated to get a difference of temperature among the outlet/inlet of the BPHE of 5K. Except for the coil design, units designated on coil-1 and coil-2 are identical. The geometric parameters of the tested coils are listed in Table 1. The values of the inside surface area (Eq: 1), refrigerant side volume (Eq: 2) and the refrigerant cross-sectional area (Eq: 3) are defined as:

$$A_{ref} = \pi d_i^2 L_{coil} NT_{active} \tag{1}$$

$$V_{coil} = \left( NC \frac{\pi d_i^2}{4} \right) \left[ \left( \frac{NFT * NTF}{NC} \right) L_{coil} \right] \tag{2}$$

$$A_p = NC \left( \frac{\pi d_i^2}{4} \right) \tag{3}$$

**Table 1. Design parameters and geometrical data of baseline finned tube heat exchanger**

| Geometric parameters | coil-1 | coil-2 |
|---|---|---|
| Outer tube diameter, mm | 9.52 | |
| Inner tube diameter, mm | 8.92 | |
| Number of rows | 4 | |
| Number of circuits | 15 | 11 |
| Number of tube per rows | 45 | 40 |
| Inner tube surface/Tube material | Smooth/Copper | |
| Inside surface area, $m^2$ | 4.28 | 3.81 |
| Refrigerant side volume, $m^3 x10^{-3}$ | 9.54 | 8.48 |
| Refrigerant cross-sectional area, $m^2$ | 0.94 | 0.69 |
| Fin type/material | Wavy/Aluminium | |
| Fin spacing, mm | 2.5 | 2.1 |

## 3 DESCRIPTION OF THE ART MODEL

For the prediction of the heat pump performance, the ART model was used in both modes of operation: heating and cooling. The model incorporates a number of submodels for the integral components of the heat pump: compressor, brazed plate heat exchanger, finned tube heat exchanger, expansion devices and connecting tubes. The model and submodels have been validated using an extensive collection of experimental results by Corberán et al. (1998, 2000, 2002).

In the present work, the model was calibrated using test results and thereof used to analyze effects of several geometry factors of the air heat exchanger "coil" on the performance (COP) of the heat pump in both modes of operation. The submodel involving the coil was implemented taking into account the real processes of sensible cooling, cooling with the dehumidification, and sensible heating of humid air. The coil submodel enables us to undertake a parametric study of the coil varying the influential geometric parameters (tube material and diameter, number of circuits, fin material, separation between fins, fin type, exchanger width, height and depth). To model the coil, the refrigerant side and air side heat transfer coefficient (HTC) and pressure drop ($\Delta P$) made used of semi--empirical correlations found in the specialized literature. When the coil operates as an evaporator, the VDI correlation (VDI 1990) is used to determine the refrigerant side heat transfer coefficient and the Shah correlation (Shah 1979) is used when it switches to a condenser. In both modes, the refrigerant side pressure drop for two phase flow operation are calculated by means of the Chisholm-Sutherland correlation (Chisholm 1967). The air side heat transfer coefficient and pressure drop are calculated by means of the Chi Chuan Wang et al. (1996, 1997) correlations for both modes of operation.

Figures included in 2 and 3 show a representative comparison of the measured capacity and COP in both modes, heating and cooling under different test conditions with the corresponding predictions. The only free parameter used for the adjustment of the model to test results was the coil inlet air velocity distribution (which is difficult to measure accurately). Nevertheless, the air velocity which was found was very reasonable taking into account our previous estimations and the available fan data. The air velocity required for the adjustment of the experimental data in cooling mode is 1.6 $ms^{-1}$ and 1.5 $ms^{-1}$ in heating mode, respectively.

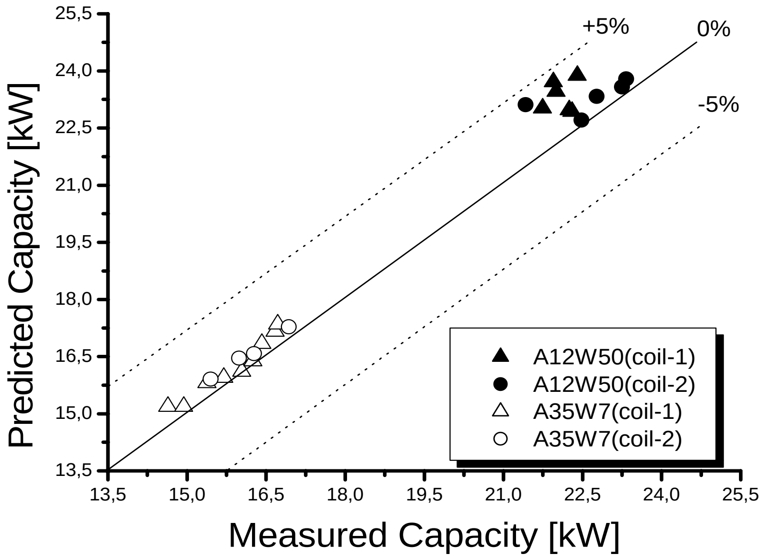


**Fig. 2. Comparison of the capacity predicted by the model with the experimental results in the heat pump systems using both coil designs and modes of operation with R290 as working fluid.**

The tandem of Figs. 2 and 3 indicate that the model was able to predict capacity (Fig. 2) and COP (Fig. 3) within less 5% relative error for all tested conditions. At this point, it should be taken into consideration that for low superheat values, the model prediction could be affected by the possible presence of two-phase flow at the exit of the evaporator in the real experiment. For our analysis, design conditions are selected from the EN-255 and EN-12055 standards, and the subcooling and superheat values require as input data in the model are determine to from the experimental results (Urchueguía et. al 2003). In these sense, the subcooling was fixed in 2K (in the cooling mode) and 10K (in the heating mode) at a fixed superheat of 6K.

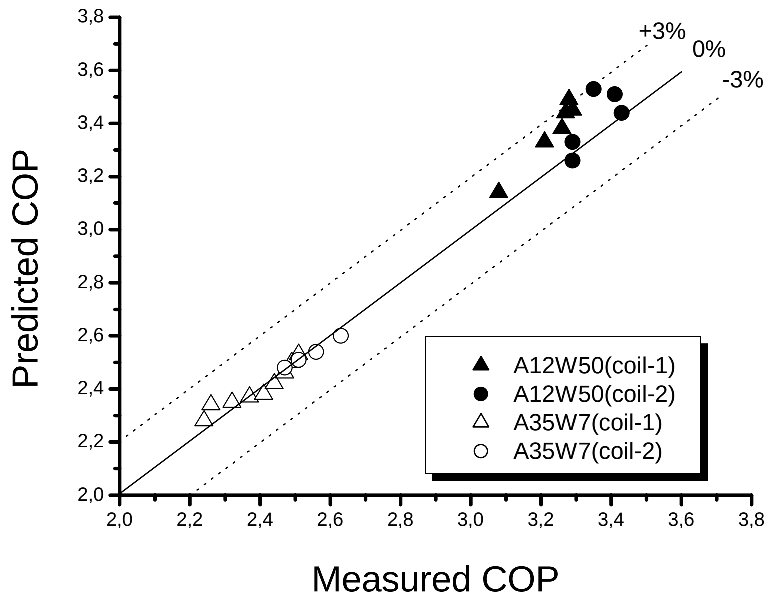


**Fig. 3. Comparison of the COP predicted by the model with the experimental results.**

## 4 STATISTICAL METHODS FOR EXPERIMENT DESIGNS

We have used factorial design in order to study the effects of different coil geometry factors on the performance (COP) of the heat pump in both modes of operation. Experiment designs are an efficient tool that allows detecting interactions between the factors. These designs employ the analysis of variance (ANOVA) as statistical support (Montgomery et al. 1991).

With the aim of finding optimized coil design concepts for the heat pump several geometry factors were analyzed. In all tests, the height and width of the heat exchangers will remain fixed taking into account the characteristics of the heat pump design, but the depth is free to change. Keeping fixed the frontal area of the heat exchanger, the studied geometry factors are: *number of circuits* (NC), *number of tube rows* (NFT), *number of tube per rows* (NTF), *inner tube diameter* (tube ID) and *fin spacing* (FS). The term NC is used to determine the number of parallel passages the refrigerant mass flow rate. The term NFT refers to the number of rows in the directional normal to air flow. The term NTF is used to determine the number of tubes in each row. The term tube ID defines the inner tube diameter used in the coil design. The relation between the number of circuits and tube ID are used to determine the sectional-across area in passing of the refrigerant. The term FS refers to the number of fins per unit length along the axial direction of the tubes. Figure 4 gives an idea of the geometry factor of the tested finned tube heat exchanger.

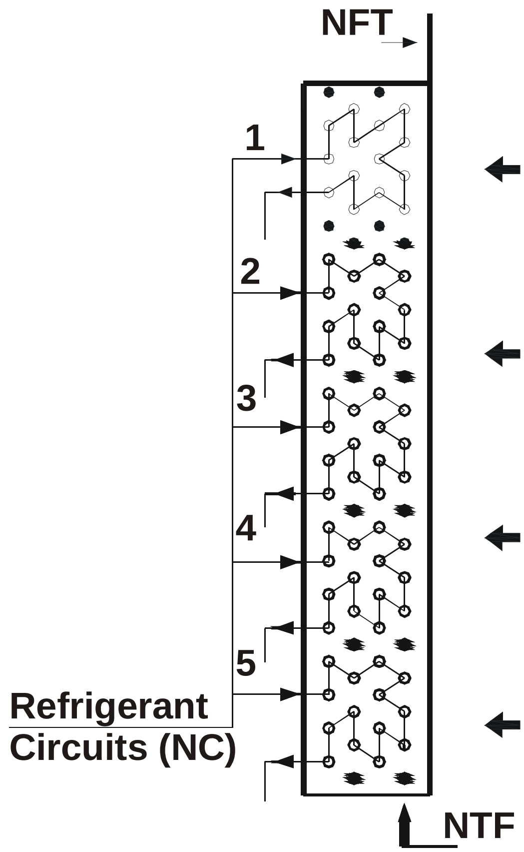


**Fig. 4. Schematic representation of the geometry factors of a finned tube heat exchanger.**

In this study, as we had five factors (five geometry factors) and four levels for each variable, there are $4^5$=1024 possible combinations in the complete factorial design. The response variables were the COP of the heat pump in both modes of operation, and the pressure drop through the heat exchanger in the heating mode (coil as the evaporator). The correspondence between different geometry factors at their different levels for the complete factorial design is shown in Table 2, once the $4^5$ possibilities were calculated using the heat pump simulation model. The obtained values for the response variables (COP and pressure drop) were subjected to a variance analysis. The analysis of the results rendered for each one of the possible combinations has been adjusted through a methodology of response surfaces using the statistical package, Statgraphics 5.1 Plus 2000.

**Table 2. Correspondence between geometry factors at their different levels for the complete factorial design**

| Real | | | | | Coded |
|---|---|---|---|---|---|
| A:NC | B:NFT | C:NTF | D:ID | E:FS | |
| 6 | 2 | 35 | 6.5 | 1.9 | -1 |
| 9 | 3 | 40 | 7.9 | 2.1 | -0.3333 |
| 12 | 4 | 45 | 8.9 | 2.3 | +0.3333 |
| 15 | 5 | 50 | 11.6 | 2.5 | +1 |

## 5 DISCUSSION OF RESULTS

Figure 5 shows the normal probability plot of the estimated effect on the COP of the heat pump in heating. The normal probability plot consists of a horizontal axis scaled for the data and a vertical axis scaled so the cumulative distribution function of a normal distribution plots as a straight line. The closer the data are to the reference line, the more likely they are to follow a normal distribution. This analysis is commonly used to analyze data from unreplicated factorial designs, see Montgomery et al. 1991.

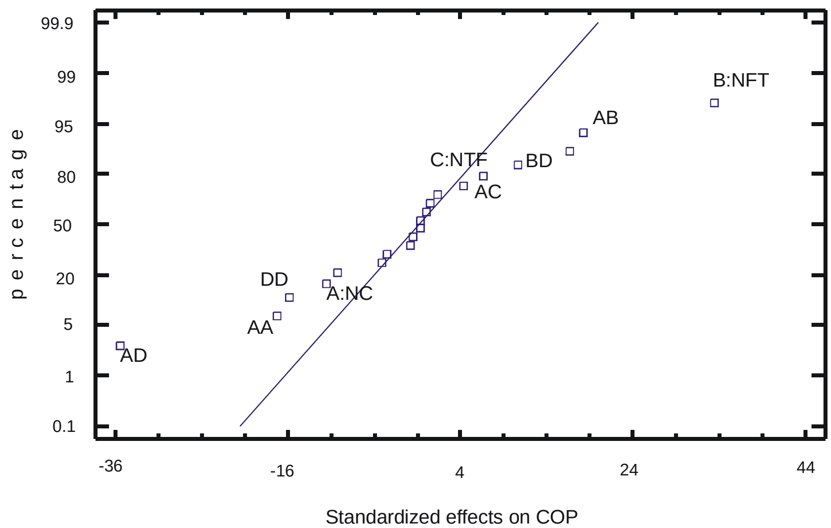


**Fig. 5. Normal probability plot for COP in the heating mode (coil as evaporator). The labels**

Based on these plots, in the heating mode (Fig. 4) all the main effects of the number of rows (NFT), the number of tube per rows (NTF), the number of circuits (NC) and inner tube diameter (tube ID) along with the interactions between NFT-ID(BD), NC-NTF(AC), NC-NFT(AB) and NC-ID(AD) require interpretation. As can be observed, the interactions between the number of circuits and tube ID (AD) posse's an important influence on the COP of the heat pump. Also, the main effect of the fin spacing (FS) could be considered negligible. This behaviour is similar in the cooling mode (coil as the condenser), see Blanco 2004.

Figure 6 shows a two-dimensional plot for the relationship between the COP and the total refrigerant area for two of the five considered factors: the number of rows and number of tube per rows. The total refrigerant area was define as the sum of products of the number of rows, number of tube per rows, tube ID and the width of the finned tube heat exchanger. The maximum COP corresponds to both factors is higher levels in both modes of operation. This would be the obvious solution for improving the COP, but choosing the high level of each factor would also result in an increase of the total refrigerant area. The area added by increasing of the number of rows or the number of tube per rows performs inefficiently.

Figure 7 shows a three-dimensional plot of the COP at the design conditions versus the number of circuits and tube ID in the heating mode. As can be observed in figure (Fig. 7), an inverse relation exists between both geometry factors in heating mode (coil as the evaporator). Setting a representative number of circuits of 6, the COP is increased of approximately 13% with the tube ID. This difference falls among tubes with highest diameter. As the number of circuits increases, the highest COP is obtained in tubes with the smallest diameter. Obviously, smaller tubes diameter results in higher heat transfer coefficients (lower surface area and higher velocities), but require the longer heat exchangers with greater pressure drop. Exploring the response surfaces finds that, the maximum COP of 3.73 is obtained in combination with a pressure drop of 25.2kPa at the design condition.

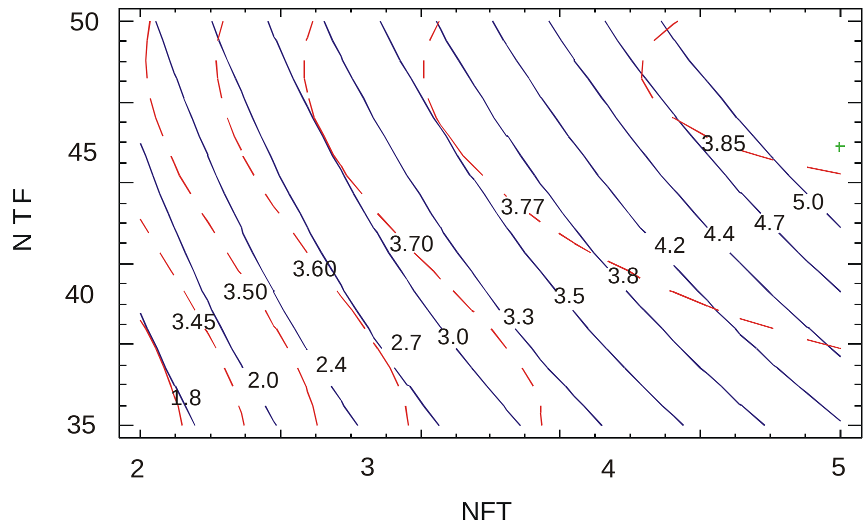


**Fig. 6. Heating COP and refrigerant total area levels as a function of the number of rows (NFT) and number of tubes per rows (NTF). The COP level is represented by discontinuous lines and total area with continuous lines.**

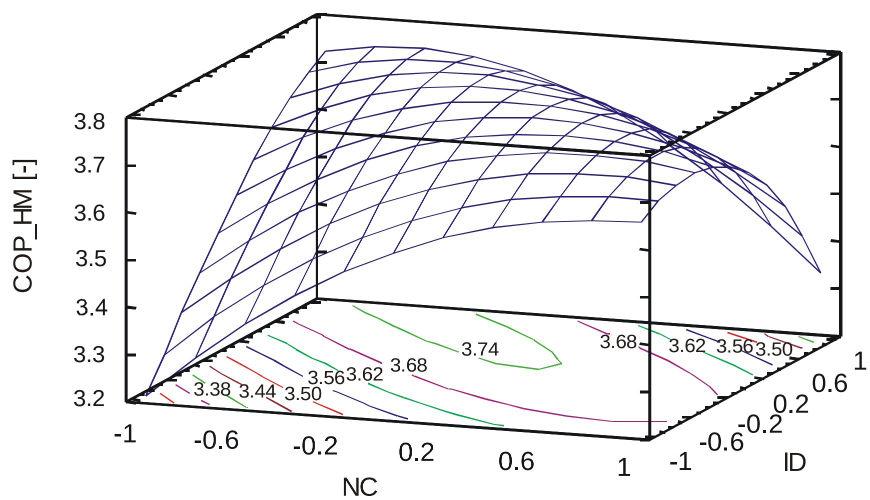


**Fig. 7. Heating COP estimated response surface as a function of number of circuits (NC) and tube ID.**

## 6 OUTCOME OF THE OPTIMIZATION PROCESS

For a reversible system, the best design in cooling mode is conditioned by the pressure drop through the heat exchanger in the heating mode (coil as the evaporator). Table 3 shows the optimal number of circuits and tube ID required maximizing the COP in the cooling mode (coil as condenser). Except for the case II, which corresponds to the optimal design in heating mode, four (cases) significant alternatives of a finned tube heat exchanger were extracted from the analysis of the response surface in cooling mode. The COP, refrigerant volume and total area values are presented for each design. These values are obtained by using the third-order models resulting from variance analysis as results of the treatments that created the response surfaces obtained. These models explained the variability observed with 99.9% (refrigerant total area) and 99.8% (refrigerant volume), respectively. The maximum COP is obtained using two different designs, as shown in Table 3. Case V with the smallest tube diameter and number of circuits is the most economical design. This design yields an 46% and 27% reduction in the refrigerant volume and total area with respect to the baseline design, respectively.

**Table 3. Geometric parameters of baseline and optimized designs. The values of the geometry factors NFT, NTF and FS are fixed in correspondence with the baseline design.**

| | **Baseline design** | **Case I** | **Case II** | **Case III** | **Case IV** | **Case V** |
|---|---|---|---|---|---|---|
| **Number of circuits, NC** | 11 | 11 | 9 | 8 | 6 | 6 |
| **Tube ID, mm** | 8.9 | 6.5 | 8.9 | 7.9 | 7.9 | 6.5 |
| **COP** | 2.69 | 2.78 | 2.75 | 2.8 | 2.82 | 2.82 |
| **Refrigerant total area, $m^2$** | 3.81 | 2.78 | 3.81 | 3.37 | 3.37 | 2.78 |
| **Refrigerant volume, $m^3$** | 8.48 | 4.54 | 8.52 | 6.65 | 6.65 | 4.54 |

## 7 CONCLUSIONS

In this study, a statistical method (complete factorial design) and a simulation model were used in order to analyze the effects of several geometry factors of a finned tube heat exchanger on the performance (COP) of an air-to-water reversible heat pump in both modes of operation: heating and cooling. The studied geometry factors are: number of circuits, number of rows, number of tube per rows, tube ID and fin spacing. The factor levels for the complete factorial design were selected taking into account the availability of material. The response variables (COP and pressure drop) values for each one of the possible combinations ($4^5$) were calculated using the ART model and the analysis of the results rendered has been adjusted by means of response surfaces using the statistical package. Previously, the model has been adjusted and validated using an extensive collection of the experimental results. The comparison between experimental and calculated results show that the model was able to predict capacity and COP within less than 5% and 3% relative error for all tested conditions, respectively.

As a result of tests, it was found that a certain relation exists between the number of circuits and tube ID for which is possible to obtain the maximum benefits (COP) in the heating or cooling mode. In the heating mode (coil as the evaporator), the combination of 9 circuits and tube ID of 8.9 mm gives a maximum COP of 3.74 along with a pressure drop of 25.2 kPa. In the cooling mode, when the coil acting as a condenser an improvement of the COP around 5% is obtained with 6 circuits and tube ID of 6.5mm. This alternative is interesting from the point of view of reducing the costs of the heat exchanger and the refrigerant charge. On the other hand, as the number of rows and number of tube per rows increases, the COP increased due to an increment of the total area. Also, the effects of fin spacing on the COP of the heat pump could be considered negligible in both modes of operation.

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