EFFECT OF CLIMATIC CONDITIONS ON THE PERFORMANCE OF AN AIR-TO-WATER REVERSIBLE HEAT PUMP USING R290 AS REFRIGERANT: SEASONAL SYSTEM PERFORMANCES EVALUATION BY MEANS OF EXPERIMENTS AND MODELING

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ABSTRACT

Domestic heating and cooling is responsible for a fair percentage of world energy consumption. Heat pumps offer the most energy efficiency way to provide heating and cooling in many applications. They can use alternative refrigerants, such as hydrocarbons, e.g. propane (R290) as one of natural refrigerants is already being used specifically in western and northern Europe, predominantly in small capacity equipment. This paper presents a wide analysis of the influence of Mediterranean climatic conditions (of great importance for Cooling), on the performance of a specifically modified 20kW air-to-water reversible heat pump unit using propane (R290) as working fluid in both modes of operation: heating and cooling. The characterization points were selected according to the EN 255-1 and EN-12055 Standards European Normative. Experimental results are compared with model predictions (ART Model).

Using the ART model, the performance of the heat pump for different geometries of a finned tube heat exchanger (number of circuits and the tube diameter) and air temperate conditions are calculated. The performance analysis of the heat pump system was made taking into account the COP and the Seasonal System Performance Factors (SPF) in both modes of operation: heating and cooling. For this purpose, an extensive study of relevant climatological parameters of Valencia-Spain was made. These results were transformed in bin-hour data which are used for calculating the heating seasonal performance factor (HSPF) and cooling seasonal performance factor CSPF according to the ANSI/ASHRAE 116-1995 and ARI Standard 210/240-2003. As a result, it was concluded that climatic conditions has a strongly effects on the seasonal system performance in both modes of operation. In the heating mode the HSPF varies between 3.1 and 3.3; whereas the CSPF varies between 3.1 and 3.4 for the three years studied (2000, 2001 & 2002).

Keywords*: cooling, heating, heat pump; heat exchangers; propane (R290); seasonal performance factors (SPF).*

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1 INTRODUCTION

Public awareness on the destruction of the stratospheric ozone layer grew in the last decade and the most harmful materials were banned. Under the terms of the Montreal Protocol, the United States agreed to meet certain obligations that have brought challenges to the Heating Ventilation Air Conditioning and

Refrigeration (HVAC&R) industry. Chlorofluorocarbons (CFCs) were used to a large extent as refrigerants but have high Ozone-Depletion Potential (ODP) and they were completely phased out in the USA in 1995. Hydrochlorofluorocarbons refrigerants, e. g. R22, are harmful environmentally and exclusively used as refrigerant in residential heat pumps and air-condition systems.

The Environmental Protection Agency (EPA) has published regulations prohibiting the production of R22 after 2010 except for servicing equipment produced prior to 2010 and it will be completely banned in 2020 (EPA, 2001). In contrast, technological developments in Europe are strongly directed towards substitutes with natural fluids, such as CO_2, hydrocarbons (HC's) and ammonia, as evidenced by the large number of projects organized in the framework of the International Energy Agency Implementing Agreement for R&D on Advance Heat Pumps, (IEA, 1996).

Some authors (Lystad 1995, Halozan 1995, Rice 1997, Purkayastha 1998, Pelletier 1998, Bredesen et al. 1999, Corberán et al. 2000, Chang et al. 2000, Nan 2000, Granryd 2001, Hwang et al. 2002, Urchueguía et al. 2003, Blanco et al. 2005) have investigated the use of hydrocarbons and particularly propane in refrigeration applications. They have pointed out that propane (R290) is a refrigerant, which has not ODP and extremely low (< 20) Global Warming Potential (GWP) compared to many currently used refrigerants. Also thermodynamic and transport properties are the same as or better than the most popular refrigerants used in refrigeration systems. Propane is not corrosive in combination with many materials such as aluminum, bronze, copper, stainless steel, silver, etc. therefore it is fully compatible with existing components such as heat exchangers, expansion valves, compressors, lubricants and copper tubing, which are commonly used in current refrigeration systems.

The main aim of this investigation is to analyze the effects of the climatic conditions and some geometry factors (number of circuits and inner tube diameter) of the finned tube heat exchanger (coil) on the performance of the heat pump, in terms of COP and Seasonal Performance Factor (SPF) using propane as working fluid. For this purpose we have used a simulation model of the heat pump, called ART[1] model. Previously, the simulation model of the heat pump was adjusted and validated using an extensive collection of the experimental results.

2 EXPERIMENTAL SETUP AND TEST PROCEDURE

The heat pump test unit shown in Fig. 1 was designed to perform the characterization of a medium size reversible air-to-water heat pump and it is described more extensively in Refs. (Corberán 2000 and Blanco 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 $50m^3$ 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, an 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).

The heat pump is a modified version of an R22 catalogue model IWA-95´heat pump manufacturer, especially well-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 main components of the analyzed heat pump are: Danfoss Maneurop compressor SM110, Danfoss TDEX 6

[1] Advance Refrigeration Technologies, IMST-Group

thermostatic expansion valve, AlfaLaval CB52X BPHE with 46 plates, Alfa Laval coil with 11 circuits, 4 tube rows, 40 tubes per rows and a 8.9mm tube that acts as condenser/evaporator.

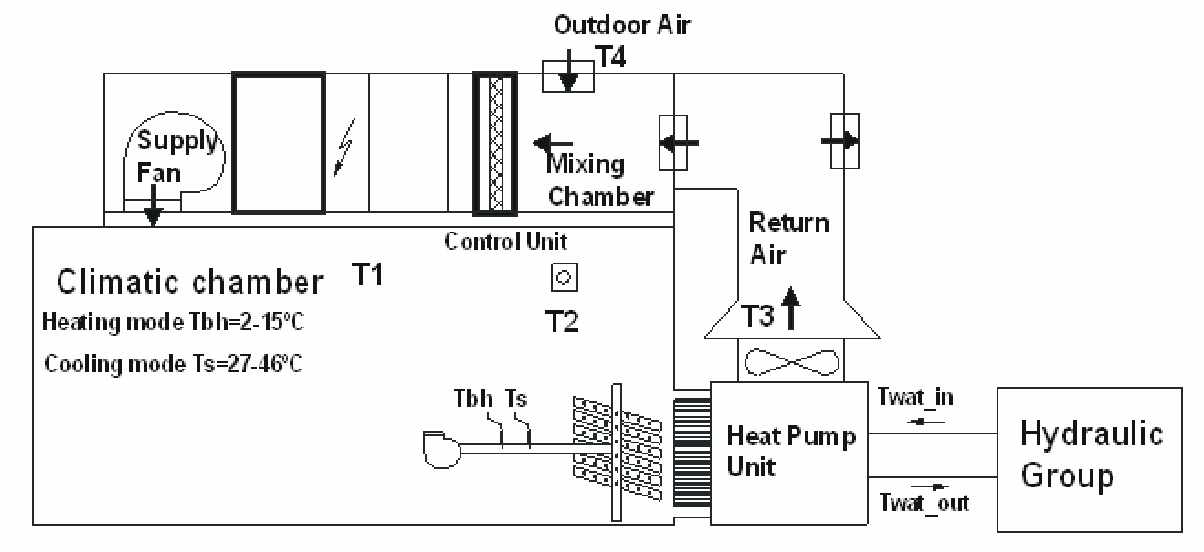

Fig. 1. Scheme of the heat pump test unit

The heat pump 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.

The characterization points of the heat pump were selected according to the EN 255-1 and EN-12055 Standards European Normative. Heating and cooling capacities [$\dot{Q}(T_i)$] were calculated 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. 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.

Figure of Merit

In order to quantitatively evaluate the performance of any air-conditioning system or heat pump system, a figure of merit must be established. For a heat pump system utilizing a vapour compression refrigeration cycle, the efficiency is expressed in terms of the Coefficient of Performance (COP) and the Seasonal Performance Factor (SPF) in both modes of operation.

The COP is a dimensionless quantity. It is the ratio of the rate of cooling or refrigeration capacity (heat absorbed by the evaporator or heat rejected by the condenser), to the electrical or mechanical power used to drive the system (compressor power, condenser fan power, and evaporator fan power). The COP is calculated as:

$$COP = \frac{\dot{Q}(T_i)}{\dot{E}_c(T_i) + \dot{E}_f(T_i)} \quad [1]$$

where $\dot{Q}(T_i)$ is the heating or cooling capacity, as a function of the modes of operation on the heat pump system; $\dot{E}_c(T_i)$, $\dot{E}_f(T_i)$ represent the compressor power input and the fan power, respectively. The fan power is considered negligible.

On the other hand, the Seasonal Performance Factor ($HSPF$ or $CSPF$) takes into account the effect of varying outside temperatures on the performance of the system. It is the ratio of the average cooling load for the system during its normal usage or "cooling load hours" to the average electricity required by the system over all cooling load hours.

Cooling load hours are defined as those in which temperature is above 65ºF(18.3ºC) [2]. Air-conditioning systems typically operate during this period. In warmer climates, there are more cooling load hours, per year than in cooler climates. In Atlanta, for example, the number of cooling load hours is approximately 1300 hours per year, while it is only about 700 hours per year in Cleveland, OH. In these cases, the outside temperature will be between 80 ºF(26.66 ºC) and 84 ºF(28.88 ºC), approximately 16.1% of the cooling season. Table 1 shows the distribution of the cooling load hours.

Table 1. Distribution of cooling load hours

Bin Number	Temperature Range(ºF)	Representative Temperature (ºF)	Fraction of total temperature hours
1	65 – 69	67 (19.44ºC)	0.214
2	70 – 74	72 (22.22ºC)	0.231
3	75 – 79	77 (25ºC)	0.216
4	80 – 84	82 (27.77ºC)	0.161
5	85- 89	87 (30.55ºC)	0.104
6	90 – 94	92 (33.33ºC)	0.052
7	95 – 99	97 (36.11ºC)	0.018
8	100 - 104	102 (38.88ºC)	0.004

The COP and SPF vary with the outside air temperature along the year and according to the ANSI/ASHRAE standard[3], the Cooling Seasonal Performance Factor, $CSPF$ [kW/kW], is calculated as:

$$CSPF = \frac{\sum_{i=1}^{8} \dot{Q}_C(T_i)}{\sum_{i=1}^{8} \dot{E}_{TC}(T_i)} \quad [2]$$

$$\dot{E}_{TC}(T_i) = \dot{E}_{cC}(T_i) + \dot{E}_{fC}(T_i) \quad [3]$$

[2] ARI Standard 210/240-2003, Unitary Air-Conditioning and Air-Source Heat Pump Equipment. Air-Conditioning & refrigeration Institute. Arlington, Virginia, sect. 5.1, p20-27.
[3] ANSI/ASHRAE 116-1995, Methods of Testing for Rating Seasonal Efficiency of Unitary Air Conditioners and Heat Pumps. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc. Atlanta, GA, 1995 p23-35.

where: $\dot{Q}_C(T_i)$ adjusted evaporator capacity at ambient temperature Ti

$\dot{E}_{TC}(T_i)$ adjusted electrical power demand (compressor + fan) at ambient temperature Ti

$\dot{E}_{cC}(T_i)$ adjusted electrical power demand (compressor) at ambient temperature Ti

$\dot{E}_{fC}(T_i)$ adjusted electrical power demand (fan) at ambient temperature Ti

$(\ )_i$ values corresponding to temperature bin i (from Table 1)

Similarly in heating mode, the "heating load hours" are defined as those in which the outdoor temperature is below 65ºF(18.3ºC).

The Heating Seasonal Performance Factor, $HSPF$ [kW/kW], is determined by:

$$HSPF = \frac{\sum_{i=1}^{n} \dot{Q}_H(T_i)}{\sum_{i=1}^{n} \dot{E}_{TH}(T_i) + \sum_{i=1}^{n} RH(T_i)} \quad [4]$$

$$\dot{E}_{TH}(T_i) = \dot{E}_{cH}(T_i) + \dot{E}_{fH}(T_i) \quad [5]$$

where: $\dot{Q}_H(T_i)$ adjusted condenser capacity at ambient temperature Ti

$\dot{E}_{TH}(T_i)$ adjusted electrical power demand (compressor + fan) at ambient temperature Ti

$\dot{E}_{cH}(T_i)$ adjusted electrical power demand (compressor) at ambient temperature Ti

$\dot{E}_{fH}(T_i)$ adjusted electrical power demand (fan) at ambient temperature Ti

$RH(T_i)$ supplementary resistance heat term at temperature Ti (in this case 0)

$(\ )_i$ values corresponding to temperature bin i

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 obtained by Corberán (1998, 2002). The submodel involving the coil was implemented taking into account the real processes of sensible cooling, cooling with 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).

In the present work, only two geometry factor of the coil design was analyzed: the number of circuits NC and inner tube diameter tube ID. The term "number of circuits (NC)" is used to quantify the number of parallel passages in which the refrigerant mass flow is divided. The term "tube ID" define the inner tube diameter used in the coil design. For this investigation, the number of rows, the number of tube per rows and fin spacing were fixed to the values used for the base configuration. The refrigerant flow circuit configurations and inner tube diameter investigated for this study are summarized in Table 2.

Table 2. Geometric parameters analyzed.

	Geometric parameters	
Cases	**Number of circuits**	**Tube ID/OD (mm)**
1	6	6.5/7.0
2	9	7.9/8.5
3	12	8.9/9.5

Figure 2 shows the comparison between theoretical and experimental results in terms of COP. Two different groups of points can be distinguished in the graphs, corresponding to heating or cooling modes. The variations range of the heat pump operation conditions covers: different air temperatures (35ºC and 40ºC in the cooling mode, and 12ºC and 15ºC in the heating mode), air humidity was varied from 60% and 80%, and fan speed from, 35Hz to 50Hz. As can be observed, the agreement between simulation and measurements is good, although a tendency of the model to over predict the measurements, at the higher COPs, can be identified. The model predicts the performance of the heat pump within a 5% of relative error without any specific adjustment.

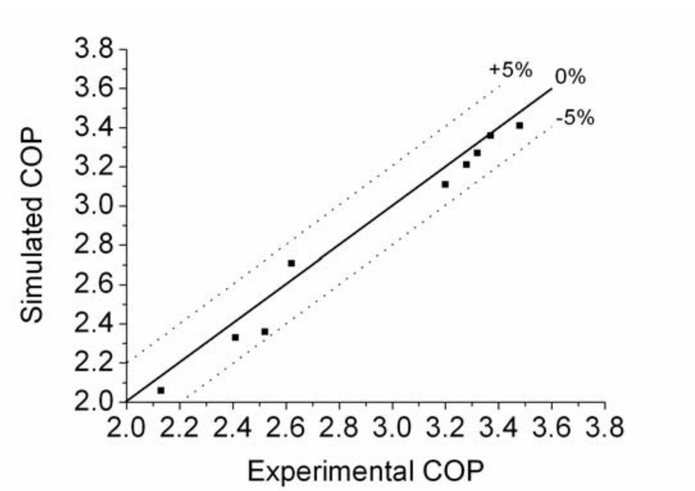

Fig. 2. Comparison of calculated measured and calculated COP in both modes of operation.

For our analysis, design conditions are selected from the EN-255 and EN-12055 standards. The parameters of the heat pump, capacity or COP, compressor and fan power consumption as a function of air temperature (from 1,5ºC to 15,5ºC with air humidity of 80% (in the heating mode) and 21,5ºC to 46ºC (in the cooling mode) has been calculated using the ART model. Input values of subcooling and superheat have been taken from our experimental results (Urchueguía et. al 2003). In this sense, subcooling was fixed in 2K (in cooling mode) and 10K (in heating mode) with fixed superheat of 6K.

4 COP & SPF CALCULATION FOR AN AIR-TO-WATER HEAT PUMP WITH PROPANE (R290): APPLICATIONS TO OF REAL CASE. EXPERIMENTAL RESULTS AND MODELLING

4.1 Evaluation of heating and cooling load profile

The building is a set of spaces in the Applied Thermodynamics Department in UPV with a total surface of about 250 m^2. This area includes a corridor, nine offices, a computer classroom, and a room with copiers and a coffee dispenser. All spaces are equipped with one fan coil except the corridor (not air-conditioned) and the computer class (two fan coils). In Table 3 we show the maximum values of the calculated loads for each of these spaces, except for the corridor. Figure 3 shows the configuration of the air-conditioned spaces.

Table 3. Maximum loads calculated for the building studied

Space	Maximum heating load (kW)	Maximum Cooling load (kW)
Office 1 person	2.05	1. 66
Office 2 persons	1.17	0.92
Office 1 person	1.18	0.99
Office 1 person	1.20	1.02
Office 2 persons	1.20	1.02
Office 1 person	1.36	1.20
Office 1 person	1.37	1.23
Office 1 person	1.09	0.87
Office 1 person	1.78	1.62
Computer Classroom	5.74	2.00
Copiers and Coffee dispenser	3.39	2.00

Additionally, a study of heating and cooling loads for the building was done (Fig. 4). It is important to realize that the load calculated for the month of August is zero corresponding to the vacation period of the university.

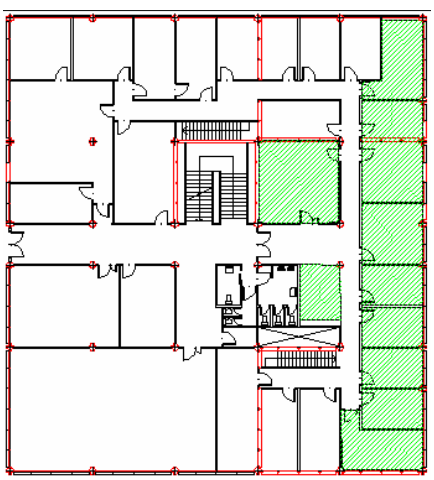

Fig. 3. Air-conditioned spaces

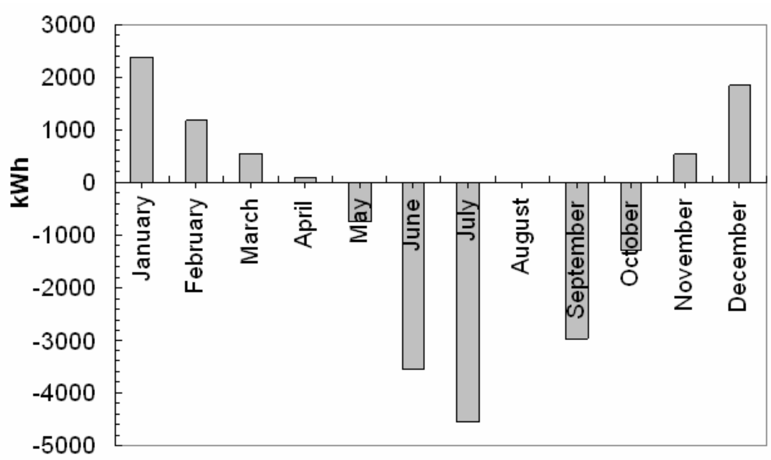

Fig. 4. Load Profile for Building

It can be seen in Fig. 4 that the values of the thermal loads are given in kWh, being positive for heating requirements and negative for cooling requirements. The software used for the evaluation of the heating and cooling load profile is CALENER (2001), a package made to characterize building in Spain.

4.2 Climate data

A method to represent the temperature profile for a certain area in a certain period is the bin-hours method. It consists of adding up the number of hours that the outside temperature lies within a certain range, and repeating this calculation for all temperature ranges that may occur. In this way a temperature histogram is obtained. The information needed to make this histogram is a database of hourly temperature observations in the study area. In the case of Valencia, the Instituto Nacional de Meteorología (Spanish National Meteorological Institute) supplied the raw data for the years 2000, 2001 and 2002.

The data furnished by the INM (1996) were compared with data gathered in the Climatic Atlas of Valencia (Atlas Climático de la Comunidad Valenciana). This book contains tables with absolute minimum temperatures, means of minima, means, absolute maximum temperatures, and means of maximums for each month in the period 1961–1990. A period of 30 years is considered as a statistically representative period, therefore these values can be considered as the expected values for Valencia.

Comparison with data gathered in the Climatic Atlas of Valencia

The "Atlas Climático de la Comunidad Valenciana" unfortunately doesn't give data variance. It does however give information about the mean values, but the mean is defined as follows:

"The annual mean is normally calculated on basis of daily means, which are the average values of the daily maximums and minimums. Other methods to estimate the mean daily temperature exist as well, based on a continuous record of temperatures during the day or regular temperature measurements during the day."

This way of calculating the mean daily temperature, and therefore the mean monthly temperature, does not correspond to the method explained in the previous chapter. Since the histograms are not symmetrical, the average of the minimum and maximum temperatures T_{med} will be higher than the mean temperature based on 24 daily observations. Indeed, for the period 2000-2001-2002 we find important differences; especially in the months of January (see Table 4):

Table 4. Comparison with data from the Climatic Atlas of Valencia

	$\overline{T}$	$\overline{T}_{\min}$	$\overline{T}_{\max}$	T_{med}
January 2000	8.85	4.43	14.94	9.68
July 2000	25.66	21.06	30.61	25.83
January 2001	13.79	9.88	18.31	14.10
July 2001	25.77	21.23	30.50	25.86
January 2002	11.72	7.57	16.68	12.13
July 2002	25.33	20.94	29.51	25.22
Means January	11.45	7.29	16.64	11.97
Means July	25.58	21.08	30.20	25.64

The question is: Do T_{med}, $\overline{T}_{\min}$ and $\overline{T}_{\max}$ for the period 2000-2002 correspond to the equivalent values for the period 1961-1990?

According to the atlas T_{med} for January is 11.5ºC. The mean minimum and maximum temperatures are respectively 7 and 15.9ºC. The average value of the T_{med} for 2000, 2001 and 2002 is the value that comes closer to 11.5ºC than any T_{med} in these three years. Therefore the best way to model the temperatures in January in Valencia is taking the average values for these three years.

According to the atlas T_{med} for July is 24.3ºC. The mean minimum and maximum temperatures are respectively 20.5 and 28.7 ºC. However the months of July during 2000, 2001, and 2002 were all warmer;

the most similar year was 2002. Therefore the best way to model the temperatures in July in Valencia is taking the values of the year 2002.

5 DISCUSSION AND RESULTS

Figs. 5 to 8 show the effect of air temperature variation upon the characteristics of the heat pump system (capacity, COP, compressor consumption and fan power) working with propane (R290) as refrigerant in both modes of operation: heating and cooling.

As the air temperature increases, cooling capacity (Fig. 5) and COP (Fig. 6) drops due to an increase of the condensing temperature and the decrease of the refrigerant mass flow rate through of the evaporator. In addition, the increase of the condensing temperature will deteriorate the compressor efficiency, increasing the compressor consumption (Fig. 5).

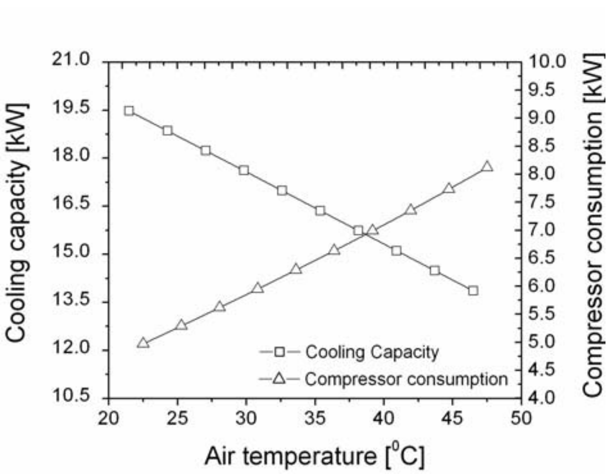

Fig. 5. Simulated Cooling Capacity [$\dot{Q}_C(T_i)$] and Compressor Power Consumption [$\dot{E}_{cC}(T_i)$] as a function of the air temperature

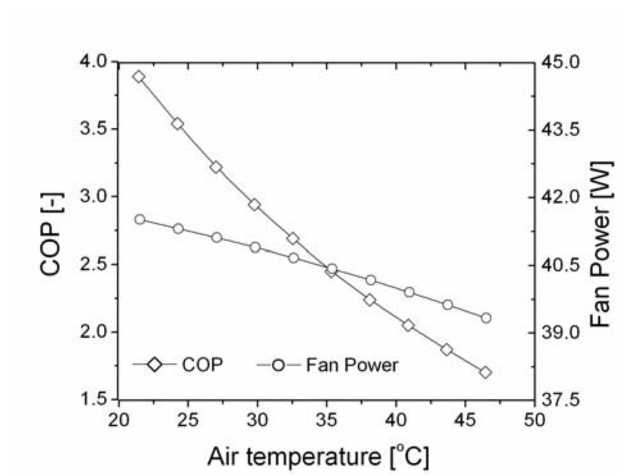

Fig. 6. Simulated COP and fan power as a function of the air temperature in the cooling mode.

In the heating mode (coil as evaporator), the heating capacity (Fig. 7), and COP (Fig. 8) are increases with the air temperature. This behavior can be explained taken in to account the higher evaporator temperature and refrigerant mass flow rate through the evaporator, which allows to a decreasing of the compressor consumption. On the other hand, the variation of the fan power with the air temperature is negligible in both modes of operation, as shown in Fig. 6 (in the cooling mode) and Fig. 8 (in the heating mode).

Finally, after having analyzed all climatic information, calculated the bin-hours for Valencia, modeled the air-to-water heat pump with R290, calculated the heating and cooling load for the building and applying the methodology proposed in the Standard ANSI/ASHRAE 116-1995 and ARI Standard 210/240-2003, the results are presented.

Figures 9 and 10 show the effect of the refrigerant cross-sectional area on the SPF in both modes of operation: heating (HSPF) and cooling (CSPF). The refrigerant cross-sectional area is determined from the ratio between the number of circuits and tube ID in the finned tube heat exchanger. The values of the compressor consumption, fan power and system capacity was calculated using the ART model in all cases.

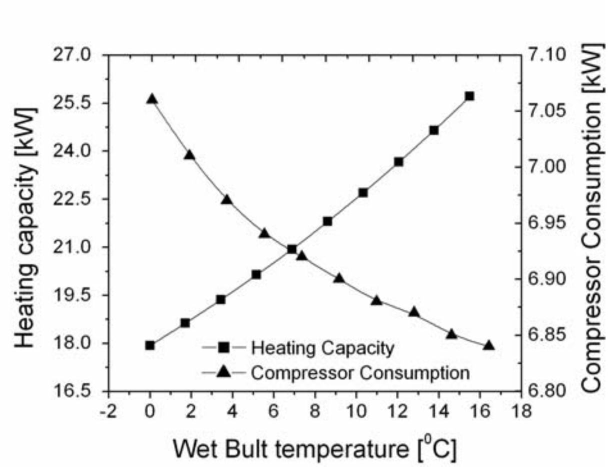

Fig. 7. Simulated Heating Capacity [$\dot{Q}_H(T_i)$] and Compressor Power Consumption [$\dot{E}_{cH}(T_i)$] as a function of the ambient air temperature

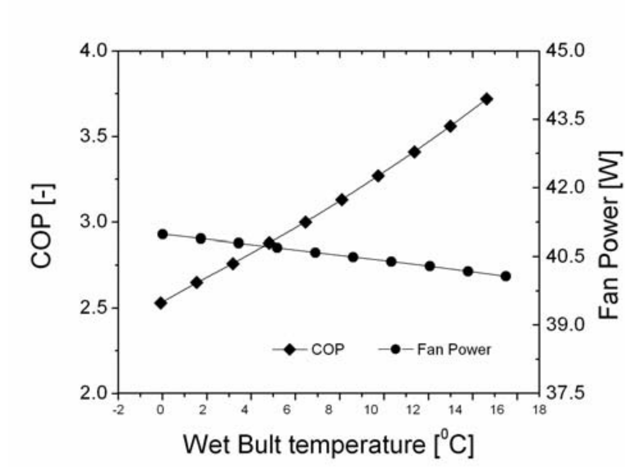

Fig. 8. Simulated COP and fan power as a function of the ambient air temperature in the heating mode

As Figs. 9 and 10 show, the optimum SPF is affected by the sectional-across area. In heating mode (coil as the evaporator) the optimum values are obtained in the range between 0.35-0.5 m^2 (Fig. 9). An increase of the refrigerant cross-sectional area allows to a decrease of the heating seasonal system efficiency (HSPF). This result can be explained from the decreased of the system capacity due to a reduction of the heat exchanger (lower refrigerant velocities and heat transfer coefficient). In addition, Fig. 9 shows that, the optimum HSPF of 3.37 occurs with a tube diameter of 8.9mm for the 2002.

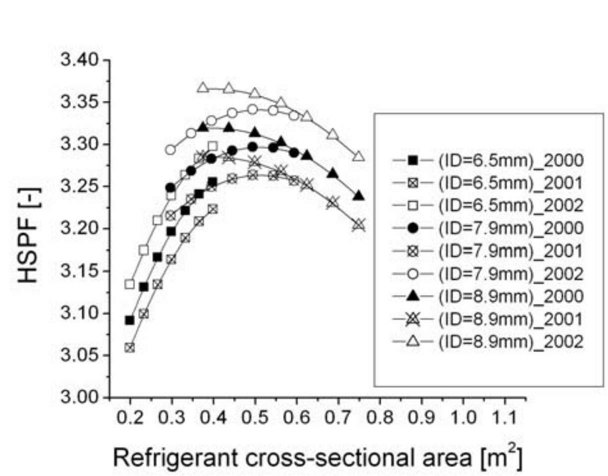

Fig. 9. Heating Seasonal Performance Factor for air-to-water Heat Pump with Propane

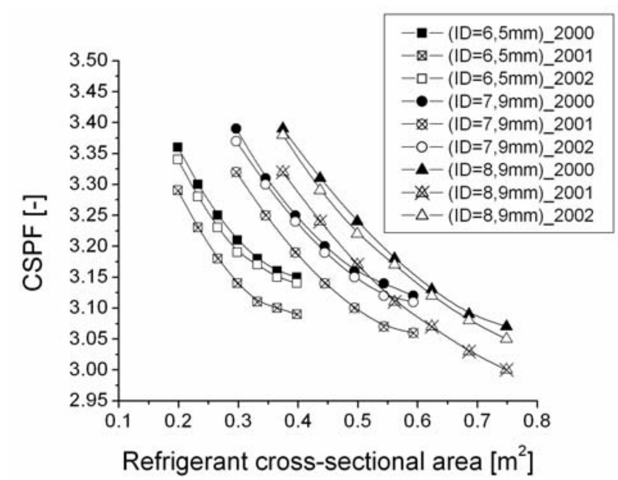

Fig. 10. Cooling Seasonal Performance Factor for air-to-water Heat Pump with Propane

In the cooling mode (coil as a condenser) the figure shows that if the sectional-across area increases, the CSPF drops suddenly. Setting a representative tube diameter of 8.9mm, the CSPF decrease approximately an 10%, but this difference is lower if the comparison is realize using a tube diameter of 6.5mm. This behavior can be explained from the increment of the compressor consumption due to a decrease of the system capacity and the increment of the pressure in the condenser as the number of circuits increase.

CONCLUSIONS

According to the bin-hours method, the best way to model the temperatures in Heating in Valencia is taking the average values for these three years (2000, 2001, & 2002). Also for Cooling, the best way to model the temperatures in Valencia is taking the values of the year 2002.

The effect of the variation of the air temperature upon the characteristics of the heat pump working with propane (R290) as refrigerant in both modes of operation were shown, concluding that in the heating mode (coil as the evaporator) the optimum values for the refrigerant sectional-cross area is attained in the range between 0.35-0.5 m^2, and the optimum HSPF of 3.35 occurs with a tube diameter of 8.9mm. On the other hand, in the cooling mode (coil as a condenser), at a fixed number of circuits, the variation of the inner tube diameter has a little influence on the CSPF.

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