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SYNTHESYS

RESEARCH ON THE IMPLEMENTATION OF UNCONVENTIONAL SYSTEMS FOR THE USE OF RENEWABLE RESOURCES FOR CONSUMERS IN THE CATEGORY OF RESIDENTIAL AND NON-RESIDENTIAL BUILDINGS.

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PREFACE

Of major interest and current international, community and national, the subject of this study, so debated, of global warming, is the first on the table in all technical fields, the forerunner of development and strategic point of new technologies. The consideration of global warming also generates solutions that will lead to a reduction in the future of the human footprint on the environment, thus eliminating the least desired forecasts of the continued increase in annual average temperature.

In this context, in 1992 the foundations of the United Nations Framework Convention on Climate Change (UNFCCC) were laid, today with 197 members, including the European Union, being the main pawn governing good organization and prevention of human interference with global warming (1). To this approach was added in 2015 a written agreement, signed by all members, through the 21st Congress of the 21st Congress of Parties (COP21) which took place, through such a historic event, in Paris, in November 30 - December 11, 2015 (2). The purpose of this agreement is also a common goal of UNFCCC members by limiting the increase in average annual global temperature below 2 ° C, with the aim of keeping it below 1.5 ° C.

In the European Community, known internationally as a pioneer of involvement in reducing the effects of human actions on global warming, and also the main economic direction of development in the medium and long term, a major economy and a neutral society has been proposed as a major objective. in view of global warming in 2050. This goal remains the prelude to the future development of the European Union, following the adoption of the "European Climate Law" in 2020, in Brussels, through the "European Green Pact" (3).

The content of the paper is organized in six chapters, where are presented a series of data necessary for mathematical modeling, physical models of unconventional systems in a mathematical approach to heat and energy transfer processes; data processed following parametric determination experiments, as well as mathematical simulations to highlight the benefits of using renewable resources.

The first chapter situates Romania's position in the European Union regarding renewable resources, emphasizing mainly the motivation of this study in the context of investments and the implementation of technical solutions for the use of renewable resources.

In the second chapter, we study Romania's problem in terms of energy needs to ensure interior comfort in residential and non-residential buildings and highlight the distinct climatic zones of the national territory.

The third chapter describes in detail the mathematical methods developed in this research for modeling solar panel systems and heat pump models. Also, the analyzed and developed models are compared with models already developed at European and international level, but which offers a number of advantages such as simplicity of application and calibration of models on the national calculation methodology in the field of thermal consumers, thus being useful for engineering. Romanian already familiar with the chosen methodology.

The fourth chapter presents an experimental study to determine the specific parameters of an air-to-water heat pump, highlighting the real values of the efficiencies of such a system but also the calibration of simulation methods, based on the values obtained in experimental measurements.

The fifth chapter, the prelude to the chapter of conclusions, is the simulation of complex unconventional, hybrid, serial and parallel models, by conducting economic studies on the implementation of such systems for well-defined consumers in different geographical areas of the country. Relevant for future studies is the sixth chapter which sums up a series of conclusions regarding the research study of the paper, the personal contributions of the author but also the highlighting of possible lines of study for further research.

Last but not least, I would like to express my gratitude to the entire team that was part of my training at this important stage in my life. I would like to thank the coordinating professor Prof. Florin IORDACHE, Ph.D., Dr. Prof. professional and family life. I would like to personally thank the professor from the guidance committee, Prof. Dr. Anica ILIE, Dr. to the measurements and approach of the experimental plan in the second research report. At the same time, I express my sincere thanks to the chairman of the doctoral committee and to the distinguished officials. I would also like to thank the members of the Department of Thermotechnics and Thermal Equipment within the Faculty of Installations for their advice and support during their doctoral studies.

Obtaining experimental data would not have been possible without the help of Mr. Teohari Tudor, who made available to me, without exception, the heating system with air-water heat pump that he owns in the commune of Ciocănești in Dâmbovița County. I would like to thank you for all the support and patience I had from my family but especially from my wife, for the encouragement and encouragement I received during my doctoral studies, without which I would not have been able to complete this study and this extremely stage. important in my pre-professional career.

CHAPTER 1. INTRODUCTORY ASPECTS

1.1 Introductory notions on renewable energy resources.

Unconventional renewable energy use systems, defined in this research study, refer to the use of thermal equipment that does not convert energy by burning fossil fuels, but delivers thermal energy, for heating and cooling, only by converting solar energy or electricity. Heat pumps are thermal equipment that converts electricity into thermal energy, but can also be considered as indirect consumers of polluting energy from the burning of fossil fuels. This paper does not aim to study the impact of the implementation of heat pumps in the combustion of fossil fuels, but mainly opts for the use of these systems that have the ability to deliver and compensate for any gaps that solar thermal systems may have during cloudy days and in night interval. Having the ability to use electricity, which can come from renewable resources, solar, hydro, wind, etc., the heat pump is therefore an equipment that has the quality to transform this clean energy into thermal energy, so necessary for human demand. thermal comfort.

Solar panels that deliver heat are also part of a series of "green technologies" that can support this paper, to meet the chosen title and field of research. This technology is a pioneer in the use of direct solar energy that is able to deliver heat immediately to the final consumer, thus significantly reducing system losses through conversion and transportation. In the residential and non-residential field, solar thermal energy for heating and cooling totaled in 2019 a total of 389 TWh or 1402 petajoules (PJ) out of a total installed and operational of 479GWth, unpolished and polished solar panels (tubes or plates) according to the same report REN21 from 2020. Also in this report we can note for a period of 5 years, the world has managed to double the capacity of installed and operational solar panels, through policies to encourage their installation but also the proven energy independence capacity for preparation hot water consumption (DHW), between 2009 and 2014, reaching in another 5 years to add 479GWth, thermal energy produced by operational facilities.

1.2 National programs. Implementing solutions that use renewable resources.

In order to meet the objectives of the European Union, in 2010 the National Action Plan in the Field of Renewable Energy was finalized, a plan that prevents the post-accession development direction, and which will be the backbone of a national environmental policy, by encouraging and imposing investments and implementations of technical solutions that will meet this goal. This plan was the result of a sustained effort on an inter-ministerial line of drafting and drafting coordinated by a working group, in the study and information gathering phase already following the European model established by Commission Decision 2009/548 / EC and relating to when adopting a common model of action in the field of renewable resources (4). In a difficult context in which our country was in the transition period between the period of centralized economy and the market economy in which it is now, but also of the difficult post-transition period, being also the first country in Annex 1 of the UNFCCC (5) of the Kyoto Protocol (6) which it ratified (7) and by which it was obliged to reduce greenhouse gas emissions by 8% compared to the reference year 1989.

The program for the installation of photovoltaic systems in 2018, which supports investments with the implementation and modernization of photovoltaic installations in proportion of 90%, extended for the year 2021 with the related completions (8); this program adds competitiveness to heat pumps that can use electricity produced indirectly from solar radiation to ensure the required thermal level for heating and cooling.

1.3 Motivation, purpose and objectives of the paper.

The doctoral thesis entitled "Research on the implementation of unconventional systems for the use of renewable resources in the field of consumers in the category of residential and non-residential buildings" responds to this global approach, community and national, finding its motivation in contributing to current technical solutions and equipment available on the market and in support of the achievement of the common goals of the United Nations on climate change. In the field of Romanian university research, the chosen field of study is a priority and seen as a medium and long term development framework. In Romania it has been legislated (9) the allocation and development of national education programs being an important component in the national strategy, to help economic agents that implement, operate and maintain new types of equipment, providing them with qualified personnel in the post-secondary and university field.

In order to meet the purpose of the doctoral thesis, a series of objectives have been defined that will ensure the simplified models prior to the final evaluation model of unconventional systems. These objectives were part of the research study carried out during the doctoral period, objectives that ensured a step-by-step development of the mathematical model that evaluates the entire unconventional system proposed to be established by the doctoral thesis. The project of the research program carried out at the beginning of the doctoral studies also outlined the main objectives on which the development of the entire project was based as follows:

1. Documentary research on the current state of energy assessment methods of buildings, which are consumers of thermal energy delivered by the unconventional system. This goal was achieved through the study in Chapter 2.
2. Documentary research on the implementation of solutions and types of equipment with solar panels and heat pumps, objective conducted in the study in Chapter 3, paragraphs 3.1 and 3.2.
3. Documentary research of current methods of evaluating unconventional systems and presentation of their advantages and disadvantages. Objective achieved in Chapter 3, paragraph 3.3 letter a and 3.3 letter b.
4. Creating a mathematical model for both solar systems and systems that use heat pumps. Objective achieved in Chapter 3, paragraph 3.3, letter c and 3.4.
5. Achieving a quick simulation tool of the previously established evaluation methods, with the help of Matlab software. Objective achieved in Chapter 3, paragraph 3.5.

6. Experimental determination of the parameters of heat pump systems whose values are uncertain. In the 2nd research report, and in the 4th chapter of the doctoral thesis, the experimental determinations led to the achievement of this objective.
7. Study of the energy market, its prices and the equipment necessary for the realization of an unconventional system of series type and of a parallel hybrid system, objective achieved and presented in detail in chapter 5 of the paper.
8. Carrying out simulations, highlighting the economic and energy aspects of the implementation of unconventional systems for the use of RES. This objective was achieved and presented in Chapter 5, paragraph 5.4, which also summarizes the doctoral thesis through the final model of energy and economic evaluation of some systems in the chosen field of research.

CHAPTER 2. THERMO-ENERGY ASPECTS OF THE CLIMATE IN ROMANIA

2.1 Evaluation methodology MC001

In order to establish the thermal demand of the building, the thermal loss coefficient of the building, defined in MC001, part II, point II.1.5.6.3 given by equation 1, composed of two transfer elements, namely the heat transfer coefficient by transmission and heat transfer coefficient by ventilation, coefficients that can be calculated according to MC001 part I-III points 2.4.7 and 2.4.8.

$$H = H_T + H_V \tag{1}$$

This coefficient is calculated for a uniform indoor temperature and for an average outdoor temperature corresponding to an energy assessment period or subperiods, taking into account constant.

It is also noted that this coefficient of heat loss is calculated only on the basis of the building elements of the building and natural ventilation through them, possibly artificial where appropriate, which is a quick process to assess the thermal energy released by the building to the outside environment, QL , for each period or calculation, τ, according to equation 2.

$$Q_L = H * (\theta_i - \theta_e) * \tau \tag{2}$$

The national methodology also presents the procedure for the economic analysis of the solutions for the rehabilitation and energy modernization of existing buildings (10). This method is necessary for the economic valuation in terms of the cost of the investment and its recovery in case of application of a period of payment of the investment, possibly a crediting of it. Equation 3 shows the relationship to determine the net present value of the "NAV" related to an investment due to the application of an energy rehabilitation or modernization project and energy saving resulting from its application.

$$VNA = C_0 + \sum_{k=1}^{3} C_{E_k} \cdot \sum_{t=1}^{N} \left(\frac{1 + f_k}{1 + i}\right)^t + C_M \sum_{t=1}^{N} \left(\frac{1}{1 + i}\right)^t \tag{3}$$

Between 1998 and 2013, ANRE reports the price of thermal energy delivered to the population, a price that has tended to increase from year to year on a relatively constant upward slope of 5% and shown graphically in Figure 1 (11).

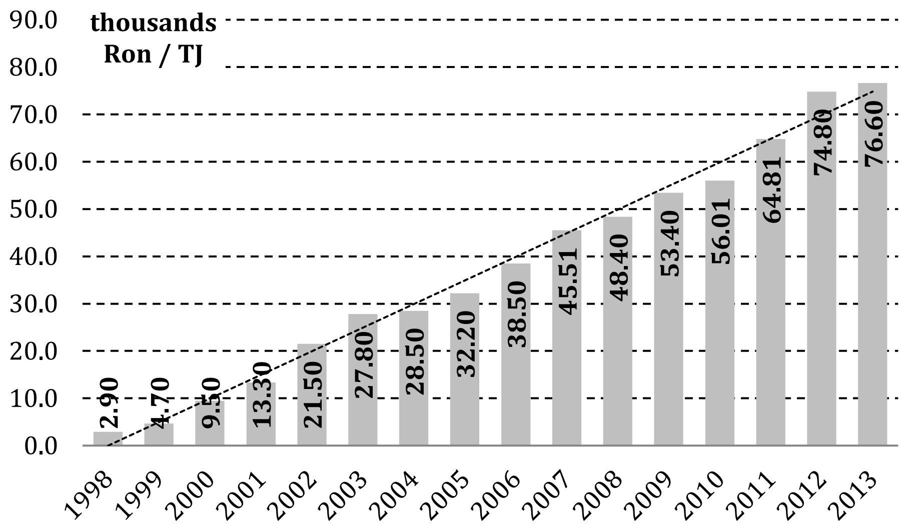

Figure 1 Romania Energy price 1998 – 2013

Data on the price of thermal energy in the country are available for the period represented graphically, obtained the information provided by ANRE, an authority that since 2016 has not regulated the price of this type of energy, the price liberalizing and becoming a tracked indicator, recorded and declared by the National Institute of Statistics "INS" as it appears from a letter issued for this purpose by ANRE, and annexed to this paper.

2.2 Romania climate.

Climatic zoning according to the conventional outdoor temperature calculation is an extremely important feature in terms of the purpose of this work, the sizing of thermal equipment for preparing the heating agent required for heating is based on outdoor temperature and thermal properties of construction elements. The thermal power of the equipment must be sufficient to generate the thermal energy required for heating, as shown in Equation 2, the computational thermal power being directly influenced by the outside temperature, the indoor temperature and the heat loss coefficient of the building. Equation 4 isolates the two components constant in the case of the indoor calculation temperature, θ_0, respectively variable depending on the climatic zone, the outdoor temperature.

$$\Phi_{th0} = H \cdot (\theta_{i_0} - \theta_{e_0}) \quad (4)$$

The coefficient of heat loss of the building, according to MC001, part I, is a parameter dependent on the constructive elements of the building, on the air exchanges between the indoor and outdoor environment, the coupling with the ground and the unheated spaces. Thus, the heat loss coefficient of the building can be written:

$$H = H_T + H_v \quad (5)$$

Equation 7 separates the coefficient of heat loss into two components of coupling with the external environment, one of contact by transmission of the building tire, denoted HT, which shows qualitatively the degree of thermal insulation of the tire and the second component representing the degree of air permeability through the building envelope, denoted HV. Therefore, as in equation 4, the temperature difference is constant regardless of the type of building evaluated, of course for the same outside temperature, the heat flow yielded to the outside environment, under limit conditions, is dependent only on the building elements of the building. The degree of thermal insulation of the tire and the control of its air permeability are two qualitative elements that can quickly assess the thermal power required to maintain the temperature of indoor comfort, depending on the climatic zone of Romania, where the analyzed building is built.

The coefficient of heat loss through the building envelope, HT, is in turn divided into 3 distinct elements from a constructive point of view. Its expression, synthesized, is given by equation 6.

$$H_T = L + L_S + H_u \quad (6)$$

- The thermal coupling with the external environment, L, which is made by the constructive elements being calculated as the sum of the products between their surfaces and the corresponding transmittance.

- Thermal coupling to unheated spaces, HU, its evaluation being recommended to be performed with SR EN 13789.
- Thermal coupling to the ground, LS, which is allowed to be calculated in steady state, in accordance with SR EN 13789 and SR EN 13370.

The coefficient of thermal losses by ventilation, HV, is calculated on the basis of the ventilation rate, na, applied to the volume of indoor air, V, summing up the volume of all rooms of the analyzed building, characterized by the same comfort temperature and taking into account the thermal thermal capacity. air, ρa · ca. Equation 7, expresses the coefficient of thermal losses due to the air permeability of the building, this being an important part of the coupling of the building with the external environment, as will be shown below.

$$H_V = \rho_a \cdot c_a \cdot n_a \cdot V \tag{7}$$

Hourly air exchanges, recommended by the national methodology, has the value of 0.5, this representing the minimum at which an air exchange is performed necessary to ensure indoor air quality.

The calculation equation corresponding to the corrected thermal resistances, correlated with the minimum necessary to obtain the normative values given in Table 1 of the mentioned annex, together with the relation of the thermal coupling coefficient, L, and the volume of indoor air led to equation 8, of the global insulation coefficient thermal, GN, through a simplified procedure.

$$GN = \frac{\sum(L_j \cdot \tau_j)}{V} + 0{,}34 \cdot n_a \tag{8}$$

As can be seen, in the calculation are synthesized the two components of the calculation of energy requirements, by conduction of the construction elements and by ventilation, related to their permeability to air exchange with the external environment, of them. Therefore, the first element of the equation is nothing more than the ratio between the heat loss coefficient of the building envelope elements, HT, and the volume of indoor air, V, of the building. This coefficient expresses the amount of energy lost and related to the volume of indoor air as well as the surface of the tire elements. Thus, it is apparent from this assessment that the heat loss of the tire elements depends on the one hand on the corrected thermal resistance and on the other hand by the A / V ratio between the surface of the tire elements and the volume of indoor air.

Building levels	A/V	GN
	[m^2/m^3]	[W/m^3K]
1	0.80	0.55
	0.85	0.58
	0.90	0.61
	0.95	0.63
	1.00	0.66
	1.05	0.67
	≥ 1.10	0.68
2	0.45	0.41
	0.50	0.44
	0.55	0.48
	0.60	0.5
	0.65	0.52
	0.70	0.53
	≥ 0.75	0.54

Tabel 1 GN values based on national normatives

The component that expresses the air permeability of the building elements, so the free exchange of air between inside and outside, and which does not depend on the thermal resistance of the building elements but simply on the hourly air volume, which is transferred between indoor environments and externally, it is summarized by the second term in Equation 8, where 0.34 is the volume thermal capacity of air, expressed in Wh / m3K, given by the product of air density, ρa, and its thermal capacity, as its. The sum of the two components expresses the overall coefficient of thermal insulation of buildings, a coefficient that is subject to obtaining minimum values provided in the Ministerial Order, and which obviously supports building designers, through a simple indicator, which imposes a minimum energy efficiency of global residential buildings.

A useful table for the simplification of the calculation method is Table 1, "Standard values of the global thermal insulation coefficient - GN" (12). Table 1 shows the values of the overall thermal insulation coefficient, based on the ratio between the surface area of the building envelope elements and the volume of heated / cooled air, for one and two level buildings. The values of the global thermal insulation coefficient have been insulated for this paper, for buildings with a maximum of two levels, because they are the subject of the analysis that will follow in subchapter 5.4.

CHAPTER 3. THERMAL EQUIPPMENTS

3.1 Introduction

Reducing the dependence on primary energy used in heating or cooling spaces in order to maintain an indoor comfort temperature is the main motivation of this chapter. Establishing working methods to analyze the efficiency of installations, establishing the state parameters of the various components of the systems and comparative study between currently available methods in research and international methods for assessing energy efficiency for installations, are the subject of a study theoretical and numerically complex methodological.

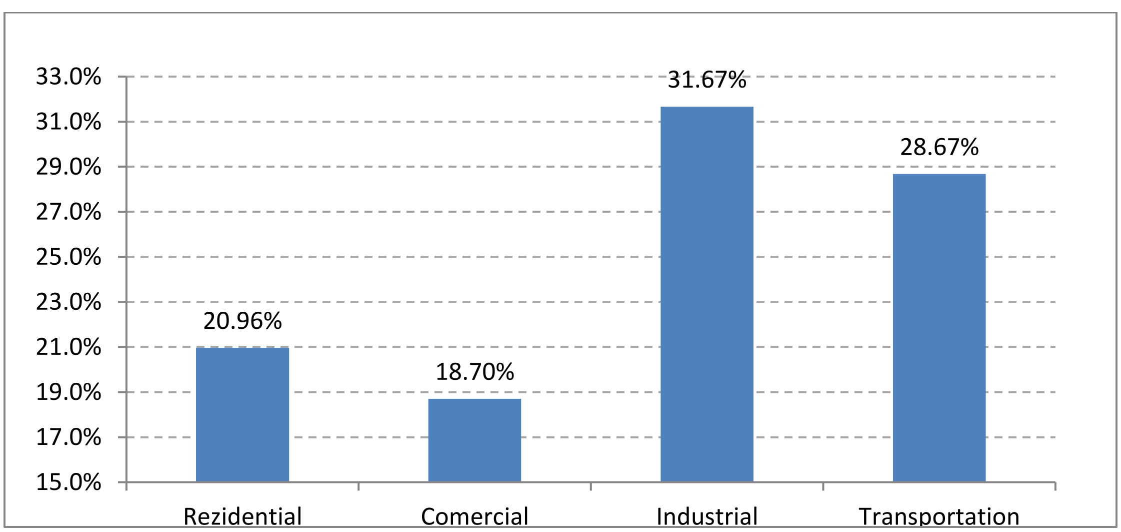

Figure 2 Romania 2016 energy demand distribution

In order to reduce the dependence on primary energy sources, the possibility of installing thermal systems that use solar energy or alternative energy sources, is an additional reason why this analysis is also of real interest.

Hybrid systems can use the energy captured by the solar system throughout the day by raising the thermal potential or using the accumulated energy when the consumer's demand for thermal agent imposes certain temperature or flow parameters, which without an additional source of the solar system cannot be delivered.

3.2 Mathematical evaluation methodes of renewable energy usage systems. Performances

For the residential field, the implementation of solutions for preparing the heating agent for heating, generally follows a low cost, the implementation is often done with pre-configured systems, the actual design no longer being the object of interest of companies in domain. For this reason, a number of quick methods have been adopted today. In this report, a number of mathematical models are listed, presented and compared to highlight the advantages and disadvantages that any type of model inevitably has.

a. F-Chart.

The f-chart method aims to evaluate the solar fraction of the total energy delivered by the solar system, for a consumer and a given system. The method is also a correlation of the results of numerous simulations of solar systems implemented around the world, and which could be recovered by the analysis teams (13).

The f-chart method has been developed for 3 types of system configurations, space heating with heat or air and domestic hot water preparation and for domestic hot water only systems.

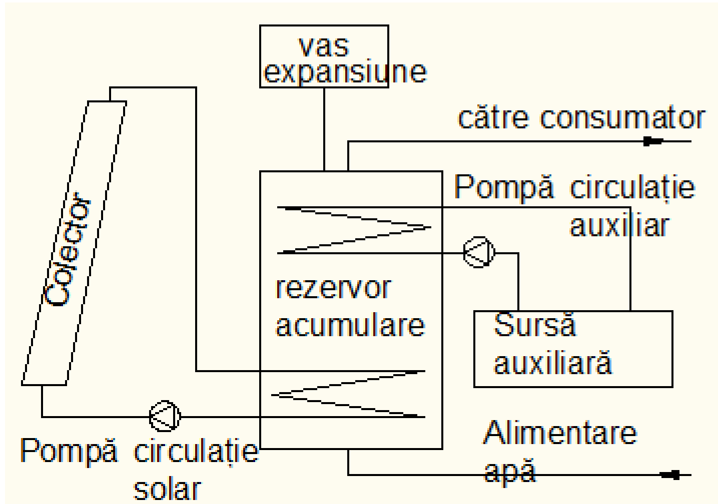

Figure 3 Sollar panels heating system

Figure 3 shows a standard hot water heating scheme for use in the preparation of heat or hot water. The collector heats a primary thermal agent which generally has specific thermal properties such as frost protection, which is recirculated through the collector and the lower coil of the tank by a circulation pump. Frost protection generally decreases the performance of the entire system due to decreased specific heat compared to water resulting in a lower amount of energy transferred per unit time. At the same time, it is worth mentioning the advantage of a lower thermal capacity, a feature that reduces the thermal inertia of the agent, in fact achieving a faster energy recovery. In general, the decrease in the thermal capacity of the thermal agent is compensated by increasing its flow through the sensor, by several methods.

The two parameters, X and Y of the method synthesize as contributions in the Y parameter as well as losses in the X parameter, the constructive elements through the descriptive parameters of the elements, transparency, absorbability, thermal losses but also the specific physical parameters of the thermal agent and the calculation period. An important role is also played by the consumer's task by which the solar fraction is directly influenced and an inadequate sizing of the solar installation leading to low performance.

$$X = \frac{A_c \cdot F'_R \cdot k_c \cdot (t_{ref} - \overline{t_e}) \cdot \Delta\tau}{Q_{cons}}$$
$$Y = \frac{A_c \cdot F'_R \cdot (\alpha\tau) \cdot I \cdot N_{zl}}{Q_{cons}} \tag{9}$$

Parameters X and Y lead to monthly fractions of heat produced in the solar installation for the consumer's needs. In total, the total annual fraction is calculated using the ratio 10 in which the total heat generated in the solar installation is related in this case to the annual needs of the consumer.

$$F = \frac{\sum f_i \cdot Q_{cons,i}}{\sum Q_{cons,i}} \tag{10}$$

The lunar solar fraction is calculated according to the system constants by which the function f depends on X and Y is built. These parameters are different from the system to the system, generally being obtained from simulations. In relation 11, the form of the function f is written in relation to X and Y, the index i representing the evaluation of the fraction in sub-periods when solar radiation and outside temperature are quasi-constant.

$$f_i = a_1 \cdot Y + a_2 \cdot X + b_1 \cdot Y^2 + b_2 \cdot X^2 + c_1 \cdot Y^3 \tag{11}$$

	Liquid thermal agent	Air thermal agent
a_1	1.029	1.040
a_2	-0.065	-0.065
b_1	-0.245	-0.159
b_2	0.0018	0.00187
c_1	0.0215	-0.095

Tabel 2 f_i constant values by heating agent type

For systems with liquid or air heat, the values of the coefficients of the function f_i,are found in Table 2.

Correction of the factor Y to the reference factor considered when establishing the f-chart equation, due to different values of the dimensionless coefficient $\varepsilon_L C_{min}/(k \cdot A)h$, where the ε_L is the efficiency of the heat exchanger,

C_{min} represents the heat transfer capacity of the liquid in the heating circuit in W/°C, and the group (k· A)h is the overall coefficient of thermal losses of the consumer, on the surface of the tyre, calculated according to the method of heating the spaces.

The f-chart method described can also be used in the calculation of the solar fraction in the case of the hot water consumption water preparation service, by correcting factor X with the relationship 12.

$$\frac{Y_c}{Y} = 0.39 + 0.65 \cdot \exp\left(-0.139 \frac{(k \cdot A)_h}{\varepsilon_L \cdot C_{min}}\right)$$
$$0.5 \le \frac{\varepsilon_L \cdot C_{min}}{(kA)_h} \le 50 \qquad (12)$$

$$\frac{X_c}{X} = \frac{11.6 + 1.18 \cdot t_{min} + 3.86 \cdot t_r - 2.32 \cdot \overline{t_e}}{100 - \overline{t_e}} \qquad (13)$$

The f-chart method can generate not very accurate results if the consumer's load varies significantly from a set value and the specific, reference accumulation volume for which the method was developed being 75 liters per square meter of capture area, here mentioning the first disadvantage of the method.

For easy writing of equations for X and Y, the relationship 14 is used.

$$X = F_R k_c \cdot \frac{F'_R}{F_R} \cdot (t_{ref} - \overline{t_e}) \cdot \Delta\tau \cdot \frac{A_c}{Q_{cons}}$$
$$Y = F_R(\alpha\tau)_n \cdot \frac{F'_R}{F_R} \cdot \frac{(\overline{\alpha\tau})}{(\alpha\tau)_n} \cdot I \cdot N_{zl} \cdot \frac{A_c}{Q_{cons}} \qquad (14)$$

a. European method.

The solar radiation in the case of this method is calculated taking into account the solar radiation on the inclined plane of the capturer but also the shading that may occur on the capture surface. The European method does not provide an annex in which the correction factor is found according to the inclination and the azimuthal angle. This decision is taken on account of the differences that would arise depending on latitude and longitude, this factor being left to the member countries. A series of methodologies for calculating the correction factor can be applied, using as input data the meteonorm records for different cities, and by correlation an annex can be obtained by which the user of the method can calculate the value of the corrected radiation on the inclined plane.

The transformation of the reference table, B.7, was performed for a temperature difference of 50°C between the hot water delivery temperature and the cold water inlet temperature in the solar system.

	S	M	L	XL	XXL	
Needs	37	102	203	337	427	l/day

Tabel 3 Daily water consumption references

The method is based on the determination of the fraction of the consumption hot water from the total energy for space heating and domestic hot water. Relationship 15 establishes this factor by reporting the energy needs for hot water preparation to the total necessary for heating.

$$f_{w,use,m} = \frac{Q_{w,sol,us,m}}{Q_{W,sol,us,m} + Q_{h,sol,us,m}} \quad (15)$$

Together with the f-chart method, the reference accumulation volume used in the development of the methodology is 75 liters per square meter of capture, the correction of the solar fraction according to it is made using the relation 16.

$$f_{sto,m} = \left(\frac{75 \cdot A_{w,sol,m}}{V_{sto,sol}}\right)^{0.25} \quad (16)$$

In the European method, solar collector loop losses take into account both coefficients of thermal loss with the environment, generally available in the catalog sheet of solar collectors on the market. The total coefficient of losses of the buffer loop, the pipes and the solar energy accumulation tank is also a parameter that is calculated separately or can be extracted from the data of the system installed with the consumer or even from an annex to this method. Together with the factor of thermal losses and the factors a1 and a2 respectively in the catalogue sheet, one can calculate the total coefficient of losses of the solar system using the relation 20. The solar surface is also used this time to obtain a real value, applied to the solar loop containing both specific armatures and solar capturers. Coefficients a1 and a2 are system parameters that show the size of thermal losses compared to the ambient environment, a1 but also due to heating of the liquid in the catcher a2, value that indicates that the losses of the solar capturer are higher in case of an increased output temperature.

$$H_{loop} = a_1 + a_2 \cdot 40 + \frac{H_{loop,p}}{A_{w,sol,m}} \tag{17}$$

One of the differences in writing the European methodology compared to the f-chart is the calculation of a reference temperature for the preparation of hot water consumption, which in the case of the f-chart method is encountered as a correction to the loss factor, according to the relationship 16. Thus, the reference temperature of the European method, used later in the calculation of factor X, is calculated with the relation 18.

$$\theta_{ref,m} = 11.6 + 1.18 \cdot \theta_{w,srv,m} + 3.86 \cdot \theta_{w,cw,m} - 1.32 \cdot \theta_{e,m} \tag{18}$$

The solar part is also applied to the energy lost by heat transfer of the insulation, calculated with the relation 19.

$$Q_{w,sol,sto,ls} = H_{sto,ls} \cdot \frac{V_{sol}}{V_{tot}} \cdot \left(\theta_{low} + \left(\theta_{high} - \theta_{low}\right) \cdot f_{tmp} - \theta_{amb}\right) \cdot f_{tmp,m} \cdot \frac{t_{ci}}{1000} \tag{19}$$

For the calculation of the solar fraction, the whole relationship 13 with the same coefficients is used, but the factors X and Y respectively, are calculated this time with the relation 20 in which the contributions and losses are the parameters of the system as follows:

$$\begin{gathered} X = \frac{A_{sol} \cdot H_{loop} \cdot \eta_{loop}\left(\theta_{ref,m} - \theta_{e,m}\right) \cdot f_{sto,m} \cdot t_{ci,m}}{Q_{sol,ls,us} \cdot 1000} \\ Y = \frac{A_{sol} \cdot K_{hem}(50^{\circ}) \cdot \eta_0 \cdot \eta_{loop} \cdot I_{sol,m} \cdot t_{ci,m}}{Q_{sol,ls,us} \cdot 1000} \end{gathered} \tag{20}$$

The losses related to the auxiliary source section, relationship 21, the contribution of the auxiliary source, relationship 22, the thermal losses related to the loop of the auxiliary source, the relationship 23, are set out in the European methodology for all 3 types of system, hot water preparation, heat preparation or mixed heat and hot water preparation systems.

$$Q_{w,bu,sto,ls,m} = H_{sto,ls} \cdot \frac{V_{sto,tot} - V_{sto,sol}}{V_{sto,tot}} \cdot \left(\theta_{w,bu,set} - \theta_{amb}\right) \cdot \frac{t_{ci}}{1000} \tag{21}$$

$$Q_{w,bu,out} = Q_{w,sol,us} - f_{dis} \cdot \left(Q_{w,sol,us,an} - Q_{w,bu,out,an}\right) \tag{22}$$

$$Q_{w,bu,dis,ls} = f_{bu,ins} \cdot Q_{w,bu,out} \tag{23}$$

In relationship 21, the heat loss is to the environment and storage, through the auxiliary storage temperature at the temperature set to the backup source. The heat delivered by the auxiliary source is established on the basis of the distribution factor and the records of the consumer's data on the annual auxiliary consumption and that of the solar source.

The distribution factor is calculated simply as the share for each month of the year, calculated with the relationship 24.

$$f_{dis} = \frac{I_{sol} \cdot t_{ci}}{I_{sol,s45,an} \cdot t_{ci,an}} \tag{24}$$

A number of additional results can be generated, knowing the electrical powers of the circulation pumps related to the solar loop and the auxiliary source. The total energy consumed including the total electric consumption, and by reporting the captured energy to it, month by month, the solar fraction can be recalculated, offering the possibility to track the distribution associated with a real installation, in which thermal losses play an important role. Finally, the actual solar fraction is calculated with the relation 25.

$$f_{sol} = \frac{f_{tmp} \cdot Q_{sol,ls,us} - Q_{w,sol,sto,ls,us}}{Q_{sol,ls,us}} \tag{25}$$

b. Solar loop efficiency method.

The method itself involves a list of initial system calculations. If in the f-chart and the European methods are calculated the two factors X and Y necessary for the implementation of methods, in the methodology of the solar loop yield are generated the monthly variables and system constants necessary for the calculation of the final solar fraction.

Relationship 26 establishes the efficiency of the solar loop for each month, the equation underlying the solar yield method.

$$\eta_{CC} = F_R^B \cdot \frac{F_C}{C_{CC}+1} - F_R^B \cdot \frac{1}{C_{CC}+1} \cdot \beta_{CC} \tag{26}$$

$$F_R^B = a \cdot \rho \cdot c \cdot (1 - E_{CS}) \tag{27}$$

$$F_C = \frac{\alpha \cdot \tau}{k_C} \tag{28}$$

With the relationship 29 it is possible to calculate the value of the coefficient of the consumer - collector.

$$C_{CC} = 0.43 \cdot \frac{S_C \cdot F_R^B}{G_{cons}} \tag{29}$$

$$\beta_{CC} = \frac{t_r - t_e}{I} \tag{30}$$

As in the case of the f-chart method, a factor of use of the energy captured by the collector surface is used by correlation between the accumulation volume and the capture surface, having as a reference a specific volume of 75 liters per square meter of capture. This factor is given by the relationship 31

$$f_u = 0.35 + 0.71836 \cdot e^{-0.1/(V_a/75/S_C)} \tag{31}$$

$$E_{CS} = \frac{E_C \cdot (1 - E_S) + E_S \cdot (1 - E_C)}{1 - E_C \cdot E_S} \tag{32}$$

$$\begin{gathered} E_C = e^{-\frac{F' \cdot k_C}{a \cdot \rho \cdot c}} \\ E_S = e^{-\frac{k_S}{a \cdot \rho \cdot c} \cdot \frac{1}{S_C/S_S}} \end{gathered} \tag{33}$$

Using the correction factor, f_{cap}, and the radiation in the horizontal plane, the corrected value of the solar radiation in the inclined plane is obtained with relation 34.

$$I = f_{cap} \cdot I_O \tag{34}$$

3.3 Methods of mathematical evaluation and modeling of the heat pump systems performance.

Multiple studies, both theoretical and experimental, have led to a detailed understanding of the dynamic behavior, efficiency, thermal capacity or consumer's needs, in terms of the thermal machine with vapor compression.

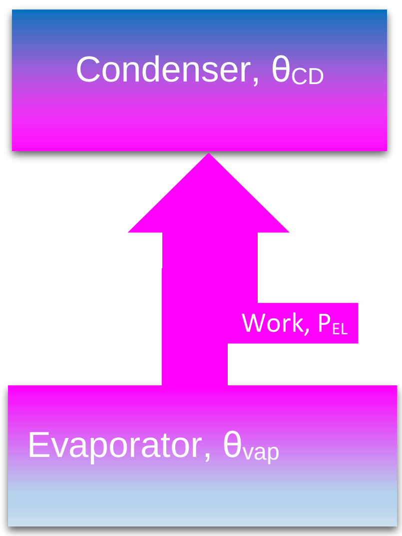

Figure 4 Carnot model

The Carnot cycle, with its simplified and ideal model, exposed by Figure 4, has been widely used as a good starting point, in defining the physical and mathematical model of vapor compression equipment, which can convert mechanical work into heat (14)

$$P_{CD} = P_{VP} + \eta_{iz} \cdot f_{CD} \cdot P_{EL} \tag{35}$$

The inverted Carnot cycle is also described by the coefficient of performance defined as the ratio of the powers from condensation and vaporization, described by equation 36. This term is used in exchange for efficiency, subunit, expressing the multiple thermal power from the evaporator found at condensation. This coefficient, due to the properties of the Carnot cycle, can be described by equation 36, relative to the temperatures related to the cold and hot environments, between which the equipment works (15)(16)(17).

$$\mathrm{COP_{CD}} = \frac{\mathrm{P_{CD}}}{\mathrm{P_{EL}}} \tag{36}$$

The mathematical representation of this physical model is synthetically exposed in equation 37, in which the absolute temperature was also transcribed in order to facilitate the calculations.

$$\mathrm{COP_{CD}} = \eta_{iz} \cdot f_{CD} \cdot \frac{\mathrm{T_{CD}}}{\mathrm{T_{CD}} - \mathrm{T_{VP}}} = \eta_{iz} \cdot f_{CD} \cdot \frac{\theta_{\mathrm{CD}} + \Delta_{\mathrm{CD}} + 273.15}{\theta_{\mathrm{CD}} - \theta_{\mathrm{VP}} + \Delta_{\mathrm{CD}} + \Delta_{\mathrm{VP}}} \tag{37}$$

$$\mathrm{P_{CD}} = \eta_{iz} \cdot f_{CD} \cdot \mathrm{P_{EL}} \cdot \frac{\theta_{\mathrm{CD}} + \Delta_{\mathrm{CD}} + 273.15}{\theta_{\mathrm{CD}} - \theta_{\mathrm{VP}} + \Delta_{\mathrm{CD}} + \Delta_{\mathrm{VP}}} \tag{38}$$

All parameters provide clear dynamic behavior or, if the constant functionality has static values, being able to create a correlation between the states of the refrigerant cycle of the heat pump and the need for thermal energy of the consumer. Constant value 0.016 coming from parameter correlation.

$$P_{CD} = 0.016 \cdot Q_{CD} \cdot \rho_w \cdot c_W \cdot (t_C - t_R) \cdot 10^{-3} \tag{39}$$

By equalization, equations 38 and 39 together become the expression of the thermal balance given by equation 40.

$$0.016 \cdot Q_{CD} \cdot \rho_w \cdot c_W \cdot (t_C - t_R) \cdot 10^{-3} = \eta_{iz} \cdot f_{CD} \cdot P_{EL} \cdot \frac{\theta_{CD} + \Delta_{CD} + 273.15}{\theta_{CD} - \theta_{VP} + \Delta_{CD} + \Delta_{VP}} \tag{41}$$

If we rewrite equation 38, in which the parameters cth and ctl are described by equation 43, defined as the temperature coefficients of the refrigerated cycle (CTCF), we are able to estimate the electrical power, at the input of the compressor's electric motor as shown in equation 42.

$$P_{EL} = \frac{1}{\eta_{iz} \cdot f_{CD}} \cdot P_{CD} \cdot (c_{th} - c_{tl}) \tag{42}$$

$$c_{tl} = \frac{\theta_{VP} - \Delta_{VP}}{\theta_{CD} + \Delta_{CD} + 273.15}$$
$$c_{th} = \frac{\theta_{CD} + \Delta_{CD}}{\theta_{CD} + \Delta_{CD} + 273.15} \tag{43}$$

The power of the evaporator is generally written by the difference from the power of the capacitor and part of the powerof the electric.

$$P_{vp} = P_{CD} - \eta_{iz} \cdot f_{CD} \cdot P_{EL} \tag{44}$$

CHAPTER 4. EXPERIMENTAL MEASUREMENTS

4.1 Numerical simulation program

The numerical simulation program is based on data obtained with the help of COOLpack and DUPREXsoftware.

In DUPREX, the refrigerant characteristics are evaluated according to REFPROP, an acronym for PROPerties fluid REFeference and developed by NIST (National Institute of Standards and Technology) USA.

Simulation in duprex environment is a stationary state, the calculation being made taking into account a constant behavior of parameters inside the system. Also, the refrigerant driven by the compressor has standard lubricating oil used in R410A systems and taken into account by DUPREX.

The COP for the evaporator and capacitor are also outputs from the DUPREX software environment. For the input data, the algorithm is able to reveal the stationary behavior of the equipment when constant parameters are involved. To compare the results, a MATLAB script was written based on CTCF equations:

```
% refrigerant cycle parameters------------------------------------

teta_cd = 50;% [Celsius]
teta_vp = xlsread('matlab_simulation.xlsx','Date_50','B1:AF1');
delta_cd = 5;% [Celsius]
delta_vp = 5;% [Celsius]
%-----------------------------------------------------------------

%sistem parameters-----------------------------------------------
eta_c = 0.61;% compressor efficiency [-]
P_CD = 10;% condenser power [kW]
eta_vp = 0.785;
%-----------------------------------------------------------------
```

```
%RCTC--------------------------------------------------------------
ctl = (teta_vp-delta_vp)/(teta_cd+delta_cd+273.15);% [-]
    xlswrite('matlab_simulation.xlsx',ctl,'Date_50','B2:AF2')   ;
cth = (teta_cd+delta_cd)/(teta_cd+delta_cd+273.15);% [-]
    xlswrite('matlab_simulation.xlsx',cth,'Date_50','B3:AF3')   ;
%------------------------------------------------------------------

%results-----------------------------------------------------------

%compressor power--------------------------------------------------
P_EL = eta_c^(-1)*P_CD*(cth-ctl);% [kW]
    xlswrite('matlab_simulation.xlsx',P_EL,'Date_50','B4:AF4')   ;

%heating COP-------------------------------------------------------
COP_CD = P_CD./P_EL;% [-]
    xlswrite('matlab_simulation.xlsx',COP_CD,'Date_50','B6:AF6')   ;
%------------------------------------------------------------------

%cooling capacity--------------------------------------------------
P_VP = eta_vp*(P_CD-(eta_c*P_EL));
    xlswrite('matlab_simulation.xlsx',P_VP,'Date_50','B8:AF8');
%end---------------------------------------------------------------
```

The curves, comparing both DUPREX results and the mathematical model from MATLAB are graphically represented in figure41. The COP in both simulations have similar results with a small gap between values, we note here an error below 5.00% of the COP differences recorded.

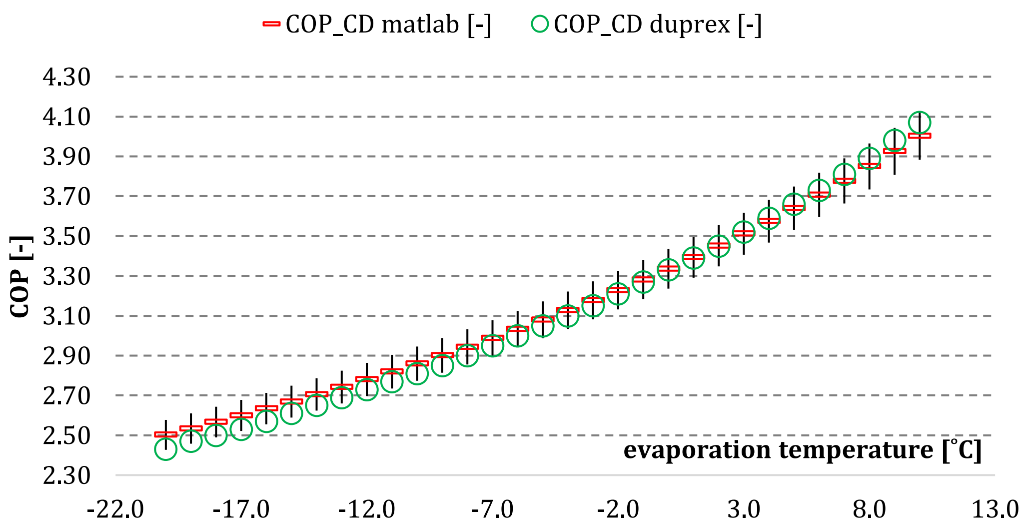

Figure 5 COP heating

4.2 Incertenty parameters identification.

The identification of specific parameters is an important step in defining the experimental plan. It is an important objective to justify an experimental study and the need to identify certain parameters. As a priority of the experimental plan, a SW-based simulation is required to establish what is known in the functionality of the equipment, what can be measured directly on the test bench and what data cannot be measured directly and which requires a mathematical calculation.

4.3 Measured data.

The electric power can be recorded using a data acquisition station, a power meter or an energy recording unit. The electric power for three-phase AC is a product between current, voltage and power factor and is given by equation 45. The parameters measured during the experiment can be tracked in tables 4 and 5.

$$P_{EL\_sym} = 3^{0.5} \cdot U \cdot I \cdot \cos(\Phi) \cdot 10^{-3} \tag{45}$$

Primary parameters		
I_1	Current phase 1	A
I_2	Current phase 2	A
I_3	Current phase 3	A
t_{CD}	Condensing temperature	°C
t_{VP}	Evaporation temperature	°C
t_C	Heat exchanger secondary exit	°C
t_R	Heat exchanger secondary input	°C
Q_{CD}	Liquid flow	$l \cdot min^{-1}$
U_1	Voltage phase 1	V
U_2	Voltage phase 2	V
U_3	Voltage phase 3	V
p_{VP}	Evaporation pressure	bar
p_{CD}	Condensing pressure	bar
τ	Sampling rate	hour

Tabel 4 measured parameters

The results of the calibration of the thermocouples shall be applied to all temperatures considered relevant or subject to a new measurement. If throughout the temperature range the values are constant, a single table may be sufficient to be used to calibrate their measurementsand future calculations.

The experimental study, the main objective of this chapter, to be used in model calibration, requires a deeper analysis of the equipment studied, the stations for the acquisition of their measurementsand the need for calibration for the sensors used.

Secondary parameters		
t_{11}	Heat exchanger input	°C
t_{12}	Heat exchanger output	°C
t_{OH}	Overheaing temperature	°C
t_{SC}	Subcooling temperature	°C

Tabel 5 refrigerant based secondary measurements

The differences in the medium temperatures were evaluated with equation 46 and equation 47 for which the logarithmic average of the temperature from the condesation is given by equation 48. Tcondensation emperature is considered to be a value of the temperature in the range of the temperature difference of the secondary circuit, in which the secondary, liquid heat agent is recirculated and the value of the condensation temperature is considered to be the logarithmicmean between theinput and output temperatures of the secondary circuit, respectively.

$$\Delta_{VP} = t_{ext} - t_{VP\_man} \tag{46}$$

$$\Delta_{CD} = t_{ml\_cond} - t_{CD\_man} \tag{47}$$

$$t_{ml\_cond} = \frac{t_{12} - t_{11}}{ln\frac{t_{12}}{t_{11}}} \tag{48}$$

Measurements of the temperature differences with the collector show for the temperature of the evaporator medium a constant value, the compressor driver being able to maintain a constant pressure inside the evaporator for better control of the heat transfer rate inside the equipment . The pressure in the condenser corresponds to a condensing temperature setting that has been varied during measurements between 35°C and 45°C with temperature steps of 5°C. For the 3 condenser temperature configurations, the heat pump sets only the condensing temperature, varying the compressor's working frequency or changing the stopping threshold in accordance with the pressure value corresponding to the set temperature.

4.4 Data processing

The data recorded are the result of the experimental study over the course of 4 days from 21.02.2018 to 24.02.2018, the time interval was more than 25000 samples for each of the 13 measurement parameters. A large database for the system has been made to be used in the processing that follows.

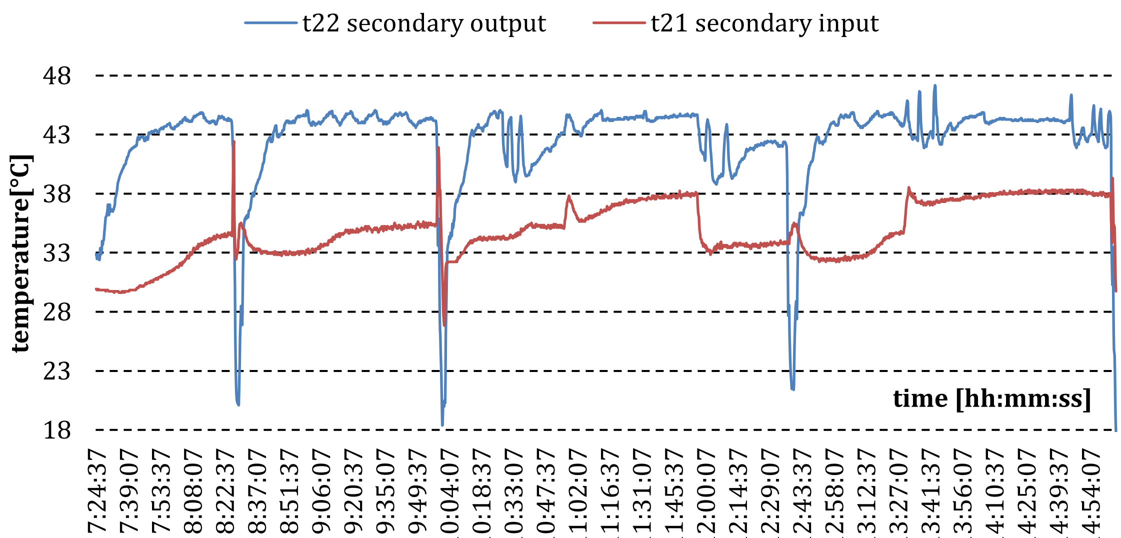

Figure 6 Experimental measurements input and output heat exchanger temperatures

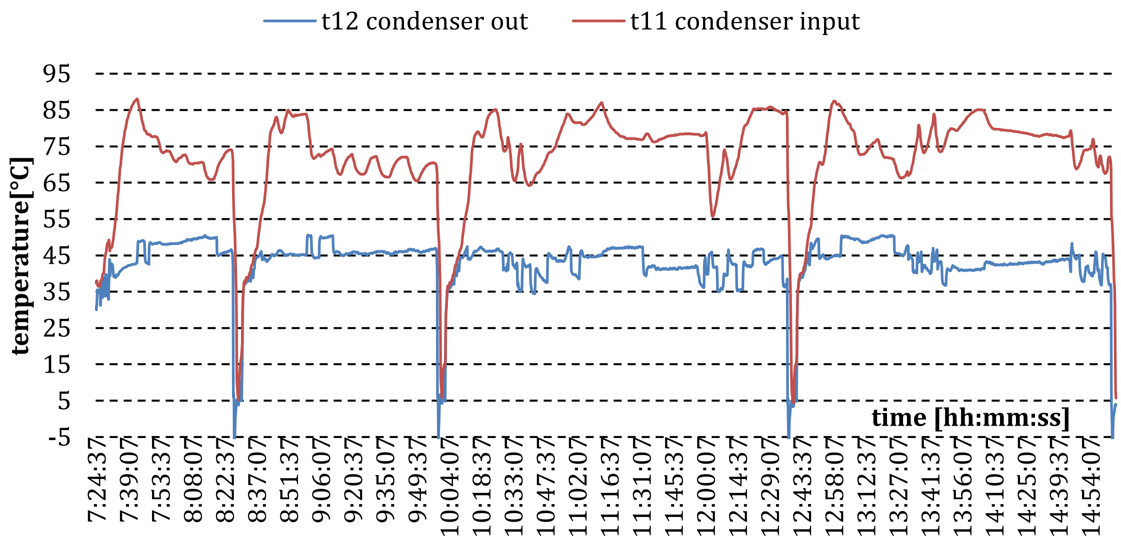

Figure 7 21.02.2018 measurements of refrigerant heat exchanger primary input and output temperature values

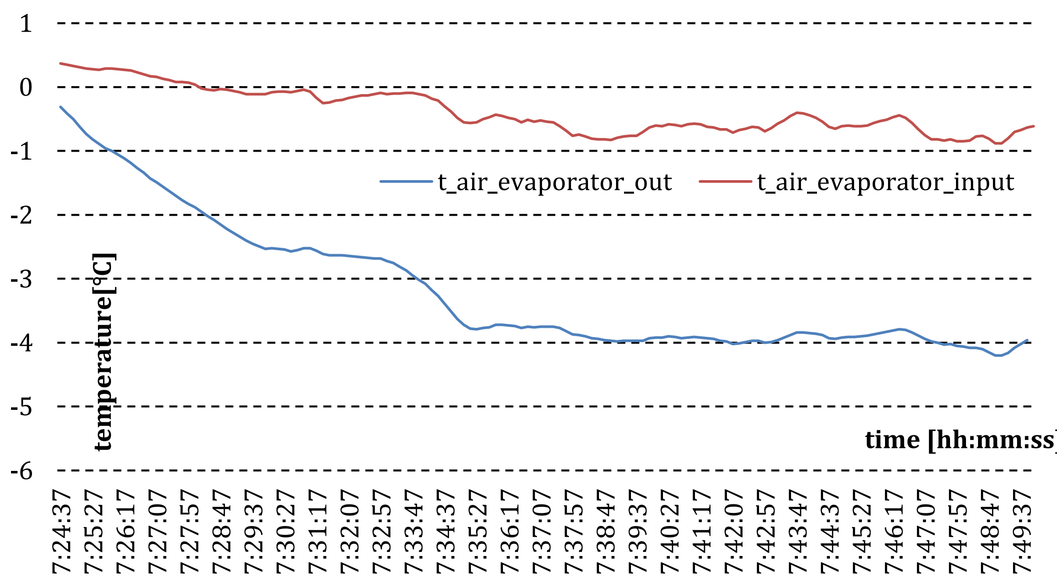

Figure 8 21.02.2018 external unit air input and output temperatures

The average value of the temperature drop, shown graphically in Figure 8, of the air in the evaporator, has a value of 1.19°C. Since measurements of the inlet temperature of the evaporator were stopped during the experimental study of 21.02.2018, the decrease in the outside temperature of the evaporator air was performed again on 28.03.2018. The average value obtained was used to establish the correct external temperature of the medium from which the heat is extracted using the refrigerant.

$$\eta_{iz} \cdot f_{CD} = \left(\frac{t_{ml\_cond} + 293.32}{t_{ml\_cond} - \theta_{VP} + 27.97}\right)^{-1} \cdot \frac{P_{CD}}{P_{EL}} \tag{49}$$

The electrical power is obtained after summing up the power on all three lines, using a coefficient of the power factor equal to 0,9.

Figure 9 shows the condensation and vaporization temperatures of the mathematical model. That temperature is the actual condensation temperatures and there vaporize for which the temperature difference for the condenser or evaporator is added or subtracted.

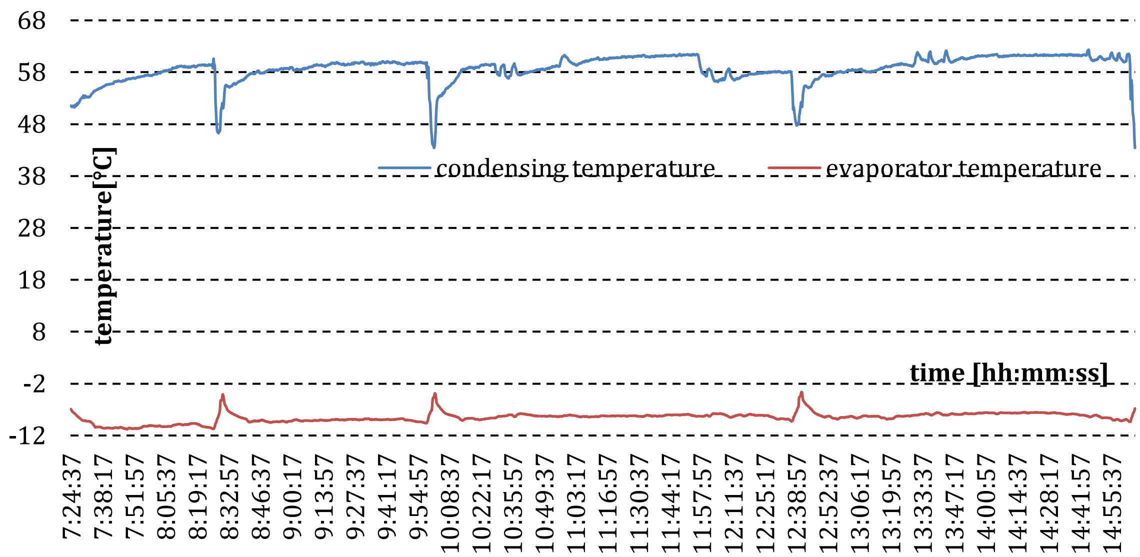

Figure 9 21.02.2018 measurements of evaporation and condensing temperatures of refrigerant

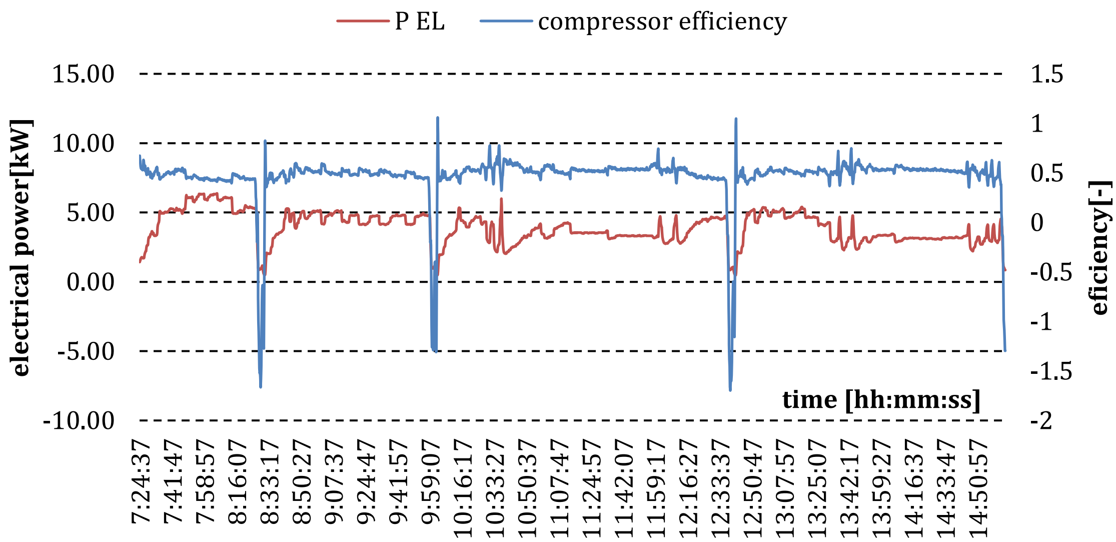

Figure 10 electrical power and compressor efficiency for measured data from 21.02.2018

4.5 Mathematical model calibration

Calibration of the simulation program is performed to configure the simulation program to assess the equivalence between the mathematical model and the experimental recording. For comparison, the electrical power measured directly on the heat pump inlet jacks and the calculated electrical power.

CHAPTER 5. ENERGY AND ECONOMIC ASPECTS REGARDING THE IMPLEMENTATION OF UNCONVENTIONAL SYSTEMS FOR THE USE OF RENEWABLE RESOURCES

5.1 Introduction. Energy and equippments prices.

The European Union provides through Eurostat, the general statistical references to the data and information taking place on its territory. Among other things, Eurostat, in the field of energy, reports annually or biannually the price per kilowatt hour for several types of energy used. The report NRG_PC_202 reports the price of energy from natural gas, on several categories of consumers and on several types of price. Of these, the price with all taxes and services included, in the Romanian national currency, for the period 2013, the second semester, 2018, was retained in the first semester, with an average value of 0.1434 RON / kWh, because it wears out the price of the energy converted from the cubic meter of gas into kilowatt hour.

In this paper, the reference price used in investment or depreciation calculations, will be the average of the years in the statistics presented in this chapter. Therefore, for electricity, the price for kilowatt-hour of energy will be 0.5739 RON, respectively 0.1434 RON for natural gas, as shown in Table 10.

Tabel 10 Average energy prices for 2013 S2 – 2018 S1

Type	Electrical	Natural gases
Abreviation in paper	p^{el}	p^{gn}
Price [RON/kWh]	0.5739	0.1434

Having the prices related to electricity, respectively from natural gas and the prices of the equipment for the production of thermal energy necessary for consumers, one can easily proceedto imagining a strategy to satisfy the needs ofmoney,of a specific consortium. To support this work, theimagined equipment is a hybrid type, which uses renewable energy, the environment, prin the heatpump. Also byusing the heat pump with mechanical vapor compression, two operating regimes of the equipments can be defined, respecitv for heating inthe cold season orrations during the warm season. This aspect brings a major advantage in the thermodynamic scheme used to satisfy its heat needs. Thus, a single piece of equipment can support a pre-existing installation, in which the heating and preparation of domestic hot water is carried out with a classical source, using fossil fuel or various hydrocarbon mixtures.

5.2 Heat pump mathematical model

In general, the relationship between the power delivered and the electric power consumed is defined by the real efficiency ofthe refrigerated midwife in the composition of a heat tree, and written in the form given by the Equation 50.

$$COP_{cd} = \frac{P_{cd}}{P_{EL}} = \varepsilon_{cd}^{C} \cdot \eta_{iz} \cdot f_{CD}$$
$$COP_{vp} = \frac{P_{vp}}{P_{el}} = \varepsilon_{vp}^{C} \cdot \eta_{iz} \cdot f_{vp} \tag{50}$$

The coefficients of fcd, respectively fvp,are definedas factors of connection (18) between the efficiency of Carnot and the coefficient of performance consistentto each typeof configurationor indicator of interest, respectively tocondensation or vaporization. Thesefactors obviously depend on the operating conditionsof the heat pump in which the heat pump is located, respectively the tempers of the refrigerant, as evidenced by the Equation 51 and written in multiple published articles.

$$f_{cd} = \frac{1}{\eta_{iz}} \cdot \frac{P_{CD}}{P_{EL}} \cdot \frac{t_{cd} - t_{vp} + \Delta_{cd} + \Delta_{vp}}{t_{cd} + \Delta_{cd} + 273.15}$$
$$f_{vp} = \frac{1}{\eta_{iz}} \cdot \frac{P_{vp}}{P_{el}} \cdot \frac{t_{cd} - t_{vp} + \Delta_{cd} + \Delta_{vp}}{t_{vp} - \Delta_{vp} + 273.15} \tag{51}$$

The two coefficients are permanently variable depending onthe refrigerant temperatures, and thelectric power used by the electric compressor to satisfy the power at the condenser, respectively the evaporator, being dependent on the freon type, make the two parameters the quality indicatorsorcorrection factors,of thefreon type, for the condensing temperatures, respectively the vaporization usedis by the system. From the paper cited for this chapter in which the two parameters for the freon R134a were evaluated, the Coolpack calculation software, (19) it emerged that the distribution of the correction factors represents a family of curves, dependent on the freon type on the one hand respectively the temperatures of the refrigerants on the other hand, so that a polynomial equation representative of the obtained results can be written inute, the form of Equation 52 and 53. Regressioncoefficients, with the calculation of the calculation factorsgiven by the equations 54 - 57, are also used to calculate at each configurationof temperaturi, the values of the correction factors. The mention to be made for these results is that the isentropic efficiency used when entering data inCoolpack, is 0.7, the values obtained for the correction factors being directly related to this parameter.

$$f_{CD} = (C_{cd} \cdot M_{cd})' \cdot M_{vp} \tag{52}$$

$$f_{VP} = \left(C_{vp} \cdot M_{cd}\right)' \cdot M_{vp} \tag{53}$$

$$C_{cd} = \begin{bmatrix} b_1 & m_1 \\ b_2 & m_2 \\ b_3 & m_3 \end{bmatrix} \tag{54}$$

$$C_{vp} = \begin{bmatrix} c_1 & n_1 \\ c_2 & n_2 \\ c_3 & n_3 \end{bmatrix} \tag{55}$$

$$M_{cd} = \begin{bmatrix} 1 \\ T_{cd} \end{bmatrix} \tag{56}$$

$$M_{vp} = \begin{bmatrix} 1 \\ T_{vp} \\ T_{vp}^2 \end{bmatrix} \tag{57}$$

5.3 Hybrid systems mathematical models

For the study that will follow, it was tried to include in the heatingsystem several elements studied on the par the course of the research. I would therefore, solar instalation, with the help of which the solar energy incident on the surface is used, in the form of radiation and the possibility offered by the hard pumps that use electricity. The advantage of using electricity is that this type of energy can be generated in photovoltaic systems or cogeneration, and, therefore, ensuring the achievement of the purpose ofthe work.

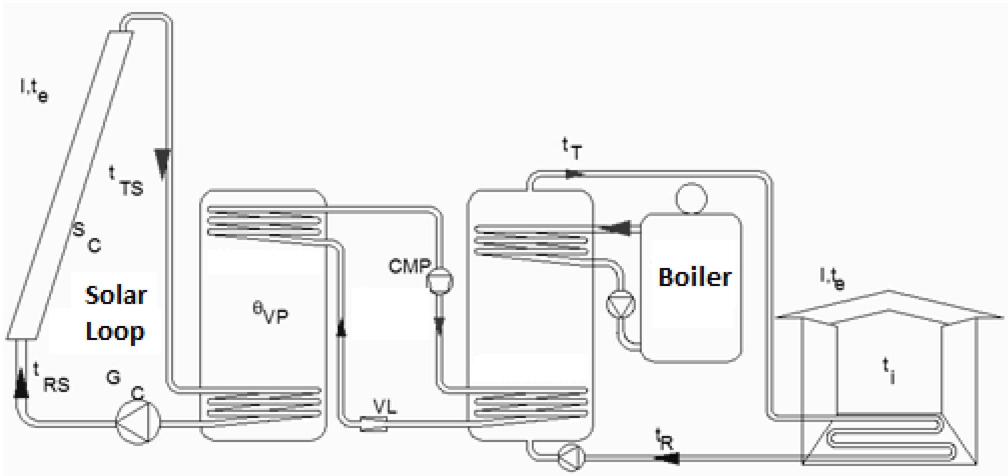

Figure 11 Series hybrid sistem

One of the mentioned is one of the hybrid systems being the solar one with the series configuration pump, shown in Figure 11. The name of the series configuration comes from the constructive type of equipment, through which initially the heat agent in the solar loop tank is heated. Thus, in the second part, the heat pump raises the thermal potential of this agent, adding thermal energy and providing heating to the consumer's loop tank. This is how step heating is achieved, through the model of a system in which the equipment is connected in series, from the solar loop to the consumer.

$$P_{CS} = S_c \cdot I \cdot F_R^B \cdot [\alpha \cdot \tau - k_c \cdot \beta_B] \tag{58}$$

Where:

$$\beta_B = \frac{\theta_{VP} - t_e}{I} \tag{59}$$

$$F_R^B = \frac{a \cdot \rho \cdot c}{k_C} \cdot (1 - E_{CS}) \tag{60}$$

$$E_{CS} = \frac{E_C \cdot (1 - E_S) + E_S \cdot (1 - E_C)}{1 - E_C \cdot E_S} \tag{61}$$

$$E_S = \exp\left(-\frac{k_S}{a \cdot \rho \cdot c} \cdot \frac{S_S}{S_C}\right) \tag{62}$$

$$E_C = \exp\left(-NTU_C\right) \tag{63}$$

Following the same strategy, the average temperature of the thermal agent on the secondary of the solar heat exchanger was expressed according to the fear of the heat agent at the exit of the heating installation of the building. Of course, to achieve this, it was passed through the heat pump that contributed so much to the increase of the thermal power transferred from its evaporator immersed in the tank de accumulation 1 to the thermal power to the capacitor that is immersed in the storage tank 2. The relationshipis related to the average temperatures of the thermal agents on the secondary circuits of the evaporator and the condenser of the heat pump are written in the works (20), (21).

$$\theta_{vp} = A \cdot \theta_{cd} + B \tag{64}$$

$$A = \left(1 - \frac{0.7}{COP_{CD}}\right) \tag{65}$$

$$B = \left[\, 2 \cdot \Delta t - (273.15 + \Delta t) \cdot \frac{0.7}{COP_{CD}}\right] \tag{66}$$

By introducing the relations describing the thermal procedure of the heat pump, the relationships already presented have the same form only as the descriptive parameters are rewritten as follows:

$$P_{CS} = S_c \cdot I \cdot F_R^{BC} \cdot [\alpha \cdot \tau - k_c \cdot \beta_{BC}] \tag{67}$$

Where:

$$\beta_{BC} = \frac{t_R^* - t_e}{I} \tag{68}$$

$$F_R^{BC} = \left(\frac{1}{F_R^B} + \frac{1}{F_R^C}\right)^{-1} \tag{69}$$

$$t_R^* = A \cdot t_R + B \tag{70}$$

$$F_R^C = 2 \cdot \frac{H}{S_C \cdot k_C} \cdot \frac{(t_{i0} - t_{e0})}{(t_{T0} - t_{R0})} \tag{71}$$

Thermal power required by buiding:

$$P_{INC} = H \cdot (t_{i0} - t_e) \tag{72}$$

In order to achieve the same goal, namely the implementation of hybrid solutions using renewable energy resources, the configuration of the solar system in parallel pumping, schematically shown in Figure 12, can also be used. One of the advantages of this type of system, comparing it with the one presented above, would bethe solar loop that can be stopped under certain weather conditions, when the solar component is disadvantageous.

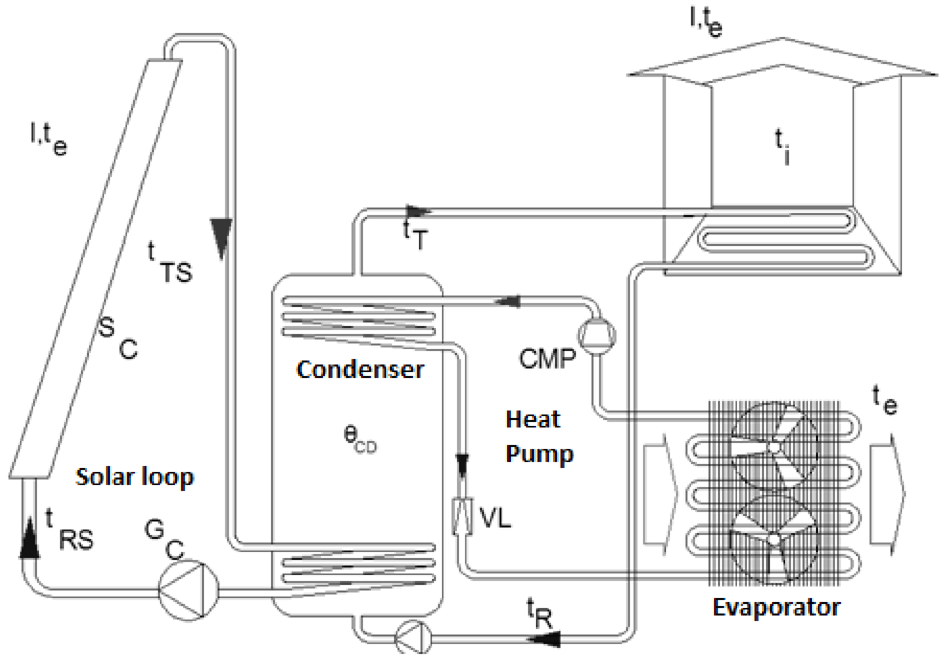

Figure 12 Paralel hybrid system

The system in Figure 46 also hasthe advantage of reducing equipment costs, with a single storage tank being necessary for the storage of thermal energy produced by the solar installation and the heating pump. The operationof this type of configurationis imposed as well as in the first type, by the fact tha tthe central locking is ensured according to the qualitative adjustment curves by which the respective turn temperature of the return of the installation of the the crates are determined on the basis of the exterior design temperature, the calculation outside temperature, the interior design temperature and the sizing temperatures of the installation. The sizing temperatures of the installation are recommended to be of low values, such as those for installation of large areas, e.g. installation of underfloor heating or by ventilation. The working temperatures corresponding to the outside temperature have the expressions given by the Equation 73 and the Equation 74 for the tour and return temperatures respectively.

$$t_T = \frac{t_{T0} - t_{e0}}{t_{i0} - t_{e0}} \cdot t_{i0} - \frac{t_{T0} - t_{i0}}{t_{i0} - t_{e0}} \cdot t_e \tag{73}$$

$$t_R = \frac{t_{R0} - t_{e0}}{t_{i0} - t_{e0}} \cdot t_{i0} - \frac{t_{R0} - t_{i0}}{t_{i0} - t_{e0}} \cdot t_e \tag{74}$$

The thermal power of the solar installation is written according to Equation 75, the solar installation operating between the outdoor temperature and the inlet temperature of the heating medium, respectively t_{RS}.

$$P_{CS} = S_c \cdot I \cdot F_R \cdot [\alpha \cdot \tau - k_c \cdot \beta_0] \tag{75}$$

The synthesis of the effects of temperature and solar radiation is performed by the ratio of thermal parameters related to the capture surface, respectively β0, with the expression given by Equation (78) which shows that the thermal power of the solar surface decreases with decreasing outdoor temperature and increases by incident radiation is higher, as highlighted by Equation 76

$$\beta_0 = \frac{t_{RS} - t_e}{I} \tag{76}$$

The correction factor of the captured heat flux, FR, correlated with β0, has this time the expression given by Equation 77, in which the thermal module EC, related to the solar installation is evaluated by the exponential number of thermal capture units and the parameters of the heat agent, respectively the global heat transfer coefficient of the solar surface.

$$F_R = \frac{a \cdot \rho \cdot c}{k_C} \cdot (1 - E_C) \tag{77}$$

And in this configuration, the thermal power of the solar surface is rewritten, for which the ratio of thermal parameters related to the capture loop, which involves, this time, the average temperature of the thermal agent in the secondary heat exchanger related to the solar loop, given by Equation 78.

$$\beta_B = \frac{t_S - t_e}{I} \tag{78}$$

Similarly, it will be followed the dependence of the thermal capture power as a function of the return temperature of the heating installation and further its dependence on the sizing temperature ti0 for the interior of the building, through Equation 79.

$$P_{CS} = S_c \cdot I \cdot \eta_{BC} = S_c \cdot I \cdot F_R^{BC} \cdot [\alpha \cdot \tau - k_c \cdot F_{INC} \cdot \beta_{i0}] \tag{79}$$

The correction factor for the heating system, Finc, has the expression Equation 80, β_{i0}, the value given by Equation 81.

$$F_{INC} = \frac{t_{R0} - t_{e0}}{t_{i0} - t_{e0}} \tag{80}$$

$$\beta_{i0} = \frac{t_{i0} - t_e}{I} \tag{81}$$

The condensing power of the heat pump is written between the condensing temperature, the temperature in the storage tank plus the temperature difference between it and the freon operating temperature and the vaporization temperature which is the outside temperature minus the freon operating temperature difference. Therefore, the expression of the condensing power is given by the relation 82.

$$P_{CD} = \eta_{iz} \cdot f_{CD} \cdot \frac{\theta_{CD} + (273.15 + \Delta t)}{\theta_{CD} - t_e + 2 \cdot \Delta t} \cdot P_{EL} \tag{82}$$

By correlating the heat pump with the solar system and the indoor heating system, by means of the temperature in the storage tank, the condensing performance coefficient becomes:

$$\mathrm{COP_{CD}} = \eta_{iz} \cdot f_{CD} \cdot \frac{\mathrm{t_T - C \cdot P_{CD} + (273.15 + \Delta t)}}{\mathrm{t_T - C \cdot P_{CD} - t_e + 2 \cdot \Delta t}} \cdot \mathrm{P_{EL}} \tag{83}$$

In this equation, the factor C has the expression 84 depending on the area and the overall heat transfer coefficient of the solar collector and the correction factor for the building.

$$\mathrm{C} = \frac{1}{\mathrm{k_C \cdot S_C \cdot F_R^C}} \tag{84}$$

Steps are:

Determine the power required by the heating system for the outdoor temperature corresponding to the calculation step. Calculate the correction temperature, t_E, according to Equation 85:

$$\mathrm{t_E} = \frac{\alpha \cdot \tau}{\mathrm{k_C}} \cdot \mathrm{I} + \mathrm{t_e} \tag{85}$$

5.4 Simulation and conclusions

The previous chapter detailed the procedures for energy evaluation of the different types of hybrid systems for the use of renewable resources. Thus, two constructive types were proposed, namely series and parallel. At the same time, a number of advantages and disadvantages were listed, both constructive and functional.

In order to study the feasibility of implementing such solutions, a numerical simulation is required, based on the models presented, for the same consumer, being interesting also its geographical location. For this it is also necessary to choose the consumer and the type of heating used to meet the needs of thermal comfort.

5.5 Energetical study

This chapter aims to highlight the economic benefits of a possible investment in the heating system for the cold season. Therefore, the distribution of power presented in the previous chapter led to the calculation of the energy delivered by each piece of equipment compared to the thermal energy required by the consumer. The results presented together with their description highlighted a series of constructive and quantitative advantages / disadvantages from a thermal point of view. This chapter will present the level of investment, the energy recovered based on the degree of energy coverage from renewable resources and the recovery period of the investment. All this will be done based on the prices of the equipment and the energy already presented at the beginning of the study.

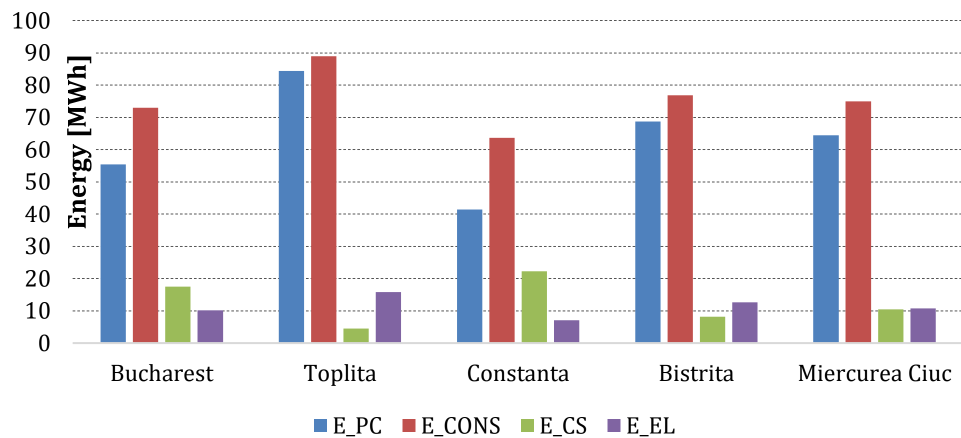

Figure 13 Energies for paralel system

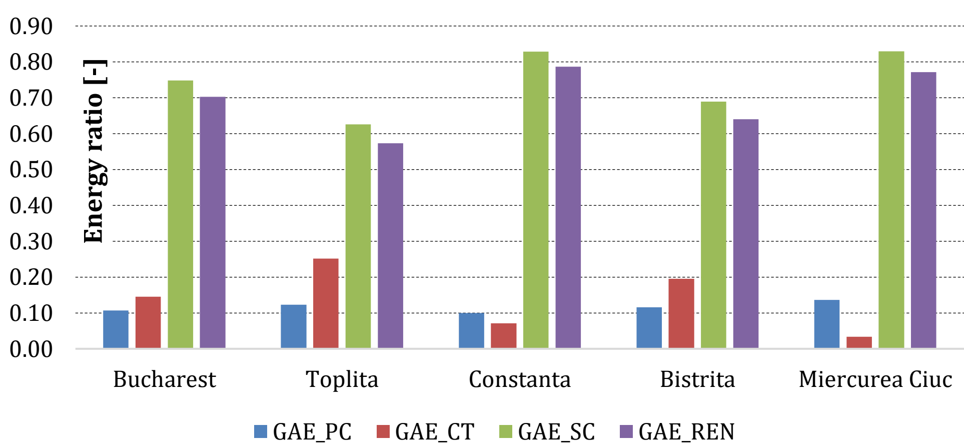

Figure 14 Energie ratios for series system

The energy delivered by the various construction elements and the energy required by the consumer, elements contained in the series hybrid system, represented graphically in Figure 13 for the solar surface, heat pump and boiler together with the total energy required by the consumer. It can be easily deduced that cities located in more unfavorable climatic zones require more energy from the total of the 5 selected cities. This graph also shows that compared to the heat pump, the solar surface provides a significant amount of energy, as an input of the total required.

CHAPTER 6. PERSPECTIVES AND PERSONAL CONTRIBUTIONS.

6.1 Personal contributions.

Scilab / Matlab software development and Scilab / Matlab simulations,

Mathematical research and comparison of different mathematical models used and proposed in the present study to highlight the system parameters necessary to increase energy efficiency and solar fraction in terms of the system of use of solar energy,

Documentary research and bibliographic study in 73 specialized works, Participation in conferences (listing them and published articles), 22 articles,

Realization of experimental plan, realization of experimental bench, calibration of thermocouples in UTCB / thermo's own laboratory,

Development of a matlab program for the model using neural networks, Research and highlighting the parameters of the refrigeration machine, as being described by a second degree polynomial,

The matrix writing of the mathematical description of the refrigerating machine that facilitates the implementation in the Matlab calculation program,

Research of the solar correction coefficient of the solar radiation according to the geographical position, using the Meteonorm database, Simulating and highlighting the advantages of different types of freon, using the useful coolpack,

Various simulations for obtaining the correction factor of the solar radiation, using the useful TRNsys,

Realization of fcap correction factor, used in the MC001 methodology for calculating inclined plane solar radiation.

Realization of thermo-hydraulic schemes for the studied systems and the proposal of improvements that should be brought in order to increase the energy efficiency.

6.2 Future research perspectives.

Continued research of different hybrid thermal schemes,

Research on the possibility of integrating the studied thermal solutions in the hot season,

Research and proposal of experimental programs for determining the specific parameters of a heat pump system for the cold season,

Further research and development of mathematical models using neural networks.

Carrying out a predictive analysis plan on the experimentally studied system for identifying possible equipment with malfunction and proposing improvements.

Propose hybrid systems focused on the contributions and disadvantages of climate zones and conduct an economic study in this regard.

Creating a dynamic model for the operation of a PC.

Nomenclature

COP_{CD} – Coefficient of Performance in heating, -,
COP_{VP} – Coefficient of Performance in cooling, -,

η_c – compressor efficiency, -
η_{vp} – evaporator efficiency, -
P_{CD} – condenser Power, kW,
P_{VP} – evaporator Power, kW,
P_{EL} – electrical power consumption, kW,
P_{EL_sym} – electrical power for symmetrical loads, kW,
Q_{CD} – liquid flow at condenser, l/min,
Δ_{CD} – temperature difference between condenser and condenser environment, K
Δ_{VP} – temperature difference between evaporator and evaporator environment, K
t_{ml_cond} – secondary line condenser medium logarithmic temperature, K
Δ_{VP_drop} – evaporator temperature drop between inlet and outlet of air heat exchanger, K
c_{tl}, c_{th} – low, respectively high refrigerant temperatures coefficient, -
θ_{CD} – condenser temperature, ˚C
θ_{VP} – evaporator temperature, ˚C
t_C – hot water temperature, ˚C,
t_R – cold water temperature, ˚C,
t_{LL} – liquid line temperature, ˚C,
t_{GL} – gas line temperature, ˚C,
t_{SC} – sub cooling temperature, ˚C,
t_C – overheating temperature, ˚C,
t_{11} - heat exchanger primary input temperature, ˚C,
t_{12} - heat exchanger primary output temperature, ˚C,
t_{evap_man} – evaporator temperature on manifold measurement, ˚C,
t_{cond_man} – condenser temperature on manifold measurement, ˚C,
t_{out_vp} – evaporator air output temperature, ˚C,
ρ_w – water density, kg/m^3,
c_w – water specific heat, kJ/(kg·K),
U_1 – voltage phase 1, V,
U_2 – voltage phase 2, V,
U_3 – voltage phase 3, V,
I_1 – phase 1 current, A,

I_2 – phase 2 current, A,
I_3 – phase 3 current, A,
cos(Φ) – electrical power factor, -,
E_{el} – electrical energy for nonsymmetrical loads, kW·h,
E_{sim} – electrical energy for symmetrical loads, kWh,
τ – sampling time, h,

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