

EXPERIMENTAL INVESTIGATION AND SYSTEM DESIGN OF DISHWASHER
HEAT PUMP SYSTEM WITH R600A/R290 MIXTURE AND SYSTEM DESIGN
MODELING

by

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B.S., Mechanical Engineering, Kocaeli University, 2021

Submitted to the Institute for Graduate Studies in
Science and Engineering in partial fulfilment of
the requirements for the degree of
Master of Science

Graduate Program in Mechanical Engineering
Boğaziçi University
2023

ACKNOWLEDGEMENTS

I am immensely honored and proud to have completed my master's thesis under the guidance and supervision of Prof. Hasan Bedir. I express my deepest gratitude for his endless support and guidance and his unwavering endeavor for my academic and professional development. His experience, time, and valuable input have been instrumental in completing this study, and I am deeply grateful for the opportunity to work with him.

I would also like to sincerely thank Arçelik A.Ş. for providing me with the necessary laboratory infrastructure and technical support throughout my thesis. In particular, I am grateful to the Research and Development Center and my managers, Ph.D. Emre Oğuz and Ph.D. Serdar Kocatürk, for their considerable support and encouragement throughout this journey.

I want to acknowledge and express my heartfelt thanks to Mutlu İpek and Okan Aydın for their unwavering support throughout my professional and academic journey. Their generous sharing of knowledge and experience, and unwavering contributions to my success, have been truly invaluable. Additionally, I wish to thank İlhan Balıkçı for his invaluable contributions to the prototyping process and for his dedication to solving technical challenges.

I would also like to extend my sincere gratitude to my esteemed colleagues Atacan Oral and Aykut Bacak and all valuable members of the Arçelik family who contributed to my thesis. Their support and encouragement have been remarkable, and I am deeply grateful for their contributions. Finally, I would also like to thank Onur Poyraz for his valuable support at the outset of my professional journey.

Finally, I wish to express my profound gratitude to every member of my family whose unwavering support and encouragement have been the cornerstone of my success. Their moral support has been a source of immense strength and inspiration, and I am truly grateful for their love, reliance, and support throughout this journey.

ABSTRACT

EXPERIMENTAL INVESTIGATION AND SYSTEM DESIGN OF DISHWASHER HEAT PUMP SYSTEM WITH R600A/R290 MIXTURE AND SYSTEM DESIGN MODELING

In this thesis study, the effect of R600a/R290 zeotropic mixture on system performance was investigated in detail on a heat pump system adapted to the dishwasher and using pure R600a as the refrigerant. In this context, five different compositions of the mixtures were investigated experimentally, with the R290 composition in the refrigerant mixture ranging from 10% to 50% by mass. In experimental studies, energy consumption, power consumption, cycle time, heating capacity, and heating efficiency were mainly compared with the base system using pure R600a. Fundamentally, a gradual decrease in the dishwasher cycle time was noted, with the percentage of R290 increasing and the cycle time in the experiment using 50% R290 in the mixture decreased by 54 minutes (28.4%) compared to the base experiment with a cycle time of 190 minutes. Although increasing R290 ratios in the mixture offered a higher power consumption value, it increased the heating capacity, so increased power consumption did not always bring high energy consumption with it. In the experiment where a 60% R600a and 40% R290 mixture was applied, a cycle time of 160 minutes was obtained, and it was observed that the increase in heating capacity provided optimum heating efficiency by surpassing power consumption. In parallel, the energy consumption decreased by 10 Wh compared to the base experiment with an energy consumption value of 470 Wh. Also, this study created a sub-model for each component of the heat pump dishwasher, and then all models were combined to form a heat pump dishwasher cycle model. The dynamic behavior of the dishwasher has been validated for both pure R600a and the R600a/R290 mixtures. While the maximum deviation in the cabinet temperature of the dishwasher was 1.5 °C, the cycle time variance was 2 minutes throughout the cycle. In addition, although the maximum deviation in power consumption was 5.2%, a maximum deviation of 2.9% was observed in the energy consumption value. Thus, the outputs of the model and the experiment were relatively comparable.

ÖZET

ISI POMPALI BULAŞIK MAKİNESİNDE R600A/R290 KARIŞIMININ DENEYSEL İNCELEMESİ VE SİSTEM TASARIMININ MODELLEMESİ

Bu tez çalışmasında, bulaşık makinasına adapte edilmiş ve soğutucu akışkan olarak saf R600a kullanılan bir ısı pompası üzerinde, R600a/R290 zeotropik karışımının sistem performansı üzerindeki etkisi detaylı bir biçimde araştırılmıştır. Bu kapsamda, soğutucu akışkan karışımındaki R290 bileşimi kütle ağırlığına göre %10 ile %50 arasında değişecek şekilde karışımların 5 farklı bileşimi deneysel olarak incelenmiştir. Deneysel çalışmalarda: enerji tüketimi, güç tüketimi, çevrim süresi, ısıtma kapasitesi ve ısıtma verimliliği saf R600a kullanılan sistem referans alınarak karşılaştırılmıştır. Temel olarak soğutucu akışkan miktarında yüzdece artan R290 oranıyla bulaşık makinası çevrim süresinde kademeli bir düşüş gözlenmiştir. Çevrim süresi, karışımda %50 R290 kullanılan deneyde, 190 dakika çevrim süresine sahip baz deneye göre 54 dakika (%28,4) azalmıştır. Karışımda artan R290 oranları daha yüksek güç tüketim değeri sunmasına rağmen ısıtma kapasitesini de arttırdığından dolayı artan güç tüketimi her zaman beraberinde yüksek enerji tüketimi getirmemiştir. %60 R600a ve %40 R290 karışımının uygulandığı 160 dakika çevrim süresi sunan deneyde, ısıtma kapasitesindeki artış güç tüketimini yenerek optimum ısıtma verimi sağlamıştır. Buna paralel olarak, enerji tüketimi, 470 Wh enerji tüketim değerine sahip baz deneye göre 10 Wh azalmıştır. Ayrıca bu çalışmada, ısı pompalı bulaşık makinasının her bir bileşeni için bir alt model oluşturmuş ve ardından tüm modeller bir araya getirilerek ısı pompalı bir bulaşık makinesi çevrim modeli oluşturmuştur. Bulaşık makinasının döngüsel davranışı soğutucu akışkan olarak hem saf R600a hem de R600a/R290 karışımı için doğrulanmıştır. Bulaşık makinası kabin içi sıcaklığındaki maksimum sapma 1,5 °C iken çevrim süresinde bu değer 2 dakikadır. Ayrıca, güç tüketimindeki maksimum sapma değeri %5,2 olmasına karşın enerji tüketim değerinde maksimum %2,9'luk bir sapma gözlemlenmiştir. Sonuç olarak, model ile deney oldukça benzer çıktılar göstermiştir.

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LIST OF SYMBOLS

A	Area (m^2)
C_p	Specific Heat Capacity (J/kgK)
d	Displacement (m^3)
D	Diameter (m)
E	Energy (Wh)
g	Acceleration due to gravity (m/s^2)
G	Mass flux (kg/m^2s)
h	Enthalpy (J/kg) or Convective heat transfer coefficient (W/m^2K)
k	Thermal conductivity (W/mK)
L	Length (m)
m	Mass flow rate (kg/s)
n	Rotational speed (Hz)
Nu	Nusselt number
p	Pressure (Pa)
P	Power (W)
Pr	Prandtl number
Q	Heat transfer rate (W)
R	Heat resistance (K/W)
Re	Reynolds number
t	Time (s)
T	Temperature ($^{\circ}C$ or K)
U	Overall heat transfer coefficient (W/m^2K)
V	Volume (m^3)
w	Uncertainty
W	Compressor Power (W)

α	Heat transfer coefficient in Dymola (W/m^2K)
γ	Specific heat ratio
ε	Emissivity or tube roughness (m)
λ	Thickness of tube (m)
ν	Kinematic Viscosity (m^2/s)
ρ	Density (kg/m^3)
τ	Shear Stress (Pa)

LIST OF ACRONYMS/ABBREVIATIONS

1-D	One Dimensional
CFC	Chlorofluorocarbons
COP	Coefficient of Performance
GWP	Global Warming Potential
HC	Hydrocarbon
HCFC	Hydrochlorofluorocarbons
HEX	Heat Exchanger
HFC	Hydrofluorocarbons
LMTD	Logarithmic Mean Temperature Difference
ODP	Ozone Depletion Potential
SGD	Sustainable Developments Goals
TEWI	Total Equivalent Warming Impact

1. INTRODUCTION

The destructive effect of global warming has been one of the most discussed worldwide in the last decades. Depending on global warming, climate change emerges that weather patterns are shifting, sea levels are increasing, and extreme weather events are intensifying. On the other hand, travel restrictions and economic recession growing out of the COVID-19 pandemic are estimated to reduce greenhouse gas emissions by roughly 6% in 2020, although this reduction is ephemeral. In other words, once the world economy starts to get over the epidemic, emissions are anticipated to rise again. For these reasons, United Nations has set objectives called Sustainable Developments Goals (SGD) to draw attention to global challenges like climate change and environmental degradation [1]. Besides, the European Union (EU) has set an ambitious domestic greenhouse gas reduction target of at least 40 percent by 2030. The aim is based on global predictions consistent with the Paris Agreement's medium-term objective. Furthermore, the agreement adopted in 2015 by 195 countries has aimed to diminish the threats of global warming by trying to keep global temperature rise below 2 °C [2].

In parallel with electricity usage, household appliances such as washing machines, dishwashers, and refrigerators contribute the most to the overall global warming effects since they have a high impact per unit and have occupied a vast place in our lives beginning of the 21st century [3]. A prime example is that, according to estimates, 44% of EU houses are equipped with dishwashers, meaning that around 96 million machines are used actively. As a result, energy consumption will reach 40 TWh/year in 2030, equivalent to 15 million tons of carbon emission [4]. However, the energy consumption value can be remarkably reduced by utilizing a heat pump integrated with conventional dishwashers or household water heater applications instead of using an electric heater.

Flück et al. [5], for example, evaluated the energy consumption of conventional and heat pump-integrated dishwashers. The results pointed out that a heat pump system can reduce electricity consumption by up to 50% compared to a conventional electric heater.

Another study utilizing a heat pump instead of a conventional electric water heater was conducted by Atasoy et al. [6]. They worked on a heat pump system that heats 4 L of water to 50°C to integrate it into electrical home appliances like dishwashers. The study optimized the system by choosing compressor speed and airflow rate passing on the evaporator as variables. The optimized system was compared with a conventional electric water heater system, and the results indicated a 17% decrease in energy consumption. In contrast, the system's operating time was almost 80 minutes longer.

Moreover, Dong et al. [7] presented a multi-functional dishwasher heat pump system through a thermodynamic analysis. The heat pump system could be worked as a single-stage or two-stage. Using a single-stage mode, R22 was used as working fluid, and hot water and hot air cycle were used for cleaning and drying. On the other hand, through the two-stage heat pump cycle run with R134a, water was heated to high temperatures, almost 85°C, to achieve further hygiene. Besides, in this study, the advantages of the heat pump system were evaluated for the carbon dioxide emission and energy consumption compared to the dishwasher operated with a conventional electric heater.

Using a heat pump in home appliances is insufficient to restrict greenhouse gas emissions; therefore, the refrigerant used in the heat pumps should also be considered. Refrigerants used widely in heat pump applications include chlorofluorocarbons (CFC), hydrochlorofluorocarbons (HCFC), and hydrofluorocarbons (HFC). After long research, it was determined that CFC and HCFC refrigerants have a pretty high ozone depletion potential (ODP) and global warming potential (GWP) since the halogen substance they contain reacts with ozone and thins the ozone layer [8]. Thus, their usage was extinguished via Montreal Protocol, contributing to the SGD [9]. Even though HFC refrigerants do not harm the ozone layer, they are classified as greenhouse gas that causes climate change since they contain fluoride. Hence, taking radical actions concerning these issues had become indispensable, and the emission of some greenhouse gases was limited by the Kyoto Protocol [10]. Besides, HFC refrigerants like R134a have low ODP values, whereas their GPW values are not negligible because of their fluoride-containing nature. Another example is the F-gas regulation, and according to the regulation published in 2014, the use of more than 150 refrigerants was limited to reduce fluorinated greenhouse gases [11].

However, due to the abolition of the use of gases specified in the F-gas regulation, it is inevitable to investigate alternative refrigerants, and studies have mainly been carried out on hydrocarbon (HC) refrigerants since hydrocarbons, such as R600a, R290, R1270, and also their mixtures, are natural refrigerants with zero ODP and low GWP value. For instance, Bodio et al. [12] proposed an isobutane/propane mixture as an alternative for certain refrigerants whose ODP is relatively high, like R12.

For zeotropic mixtures like R600a and R290 mixture, temperature glide issue takes place in both the condenser and evaporator. The temperature glide can be defined as the absolute temperature difference between the point where the phase change starts and ends at constant pressure. The difference stems from the composition change of refrigerants that boiling point changes as the composition changes. Radermacher et al. [13] investigated the refrigeration system performance using an R600a/R290 mixture as the refrigerant. They indicated that the large temperature glide sparked better heat transfer in both heat exchangers: condenser, and evaporator. Thus, through the temperature glide, the performance of the cycle could be enhanced.

Sanchez et al. [14] experimented with R134a alternatives with a low GWP value. R152a, R1234yf, R1234ze(E), R290, and R600a are evaluated with variable operating circumstances without making a specific change in the system equipped with a hermetic compressor for R134a. In R600a systems, the condensation and evaporation temperatures are lower compared to R134a systems operating under the same conditions. Due to its low specific volume, R600a significantly decreases the cooling capacity and energy consumption. R600a compressors require more displacement to attain the same cooling capacity as R134a compressors. R290, on the other hand, provided the best results for cooling capacity and COP. Nevertheless, the system operating with R290 required smaller displacement or a larger electric motor. Eventually, R290 cannot be an alternative for direct implementation to the system with R134a.

In this thesis study, the effects of various outputs, such as heating efficiency and power consumption, based on a refrigerant mixture of R600a and R290 are examined to reduce the energy consumption and cycle time of a household dishwasher system with a heat pump in which R600a is used as a working fluid.

Also, the dishwasher with a heat pump is modeled dynamically, the system design stage is conducted experimentally, and the model is validated for pure refrigerant and mixture.

The heat pump system used in a dishwasher comprises a compressor, condenser, capillary tube, and an evaporator. The compressor is a reciprocating compressor, and the condenser is a water source coil condenser located in the washing waterway. In other words, the washing water is heated by passing over the coil condenser. The refrigerant is expanded through the capillary tube rather than the expansion valve in the heat pump system. As for the evaporator, it is an air source fin and tube crossflow heat exchanger. Besides employing a fan, the ambient air is blown on the evaporator, and the evaporator can draw heat on itself from the ambient.

The study presents a detailed literature review in Chapter 2 regarding refrigerant evaluation, system design, and optimization studies on integrated heat pump dishwashers. Then, the objective is emphasized in Chapter 3 by pointing out the gap in the literature. Next, the system design and specifications of the dishwasher with heat pump are stated in Chapter 4. The following section introduces a detailed modeling methodology, and the experimental test setup and method are clarified. Finally, experimental results, including model validation and discussion, are submitted in Chapter 7. The conclusion and references sections form the last two parts of the thesis.

2. LITERATURE REVIEW

Since the regulations and restrictions regarding energy efficiency and usage of certain refrigerants have been pushing the sector, the studies on heat pump technology and alternative refrigerants have remarkably increased. Under these circumstances, this trend is reflected in the literature at the same rate. The studies on environmentally friendly refrigerants as alternative halogenated gases date back to the beginning of the 90s. Didion and Bivens [15] suggested eco-friendly refrigerant mixtures as CFC substitutes. They classified them into three categories (azeotropes, near azeotropes, and zeotropes) in 1990. Also, they revealed the potential and characteristics of these mixtures.

Moreover, a comprehensive review study by Mohanraj et al. [16] evaluated the studies carried out from those years to the beginning of the 2010s on different systems with numerous refrigerants and their mixtures. First, the study presented characteristics of different refrigerants such as hydrocarbon, hydrofluorocarbon, and their mixtures as an ecology-friendly alternative to certain refrigerants based on ozone layer depletion (ODP) and global warming potential (GWP). Then the experimental and theoretical studies performed on domestic and commercial refrigeration systems, air conditioners, heat pumps, and chillers were reviewed, analyzing their working fluids. Finally, hydrocarbon refrigerants and their mixtures were recommended to be utilized as promising long-term alternatives.

In this chapter, a detailed literature survey on various refrigerant evaluations in heat pump systems and dishwasher system design and optimization has been done, paying sufficient attention to the top-rated papers. Section 2.1 gives papers in which refrigerants used in heat pump systems or their mixtures are applied in refrigerator and water heating systems under two distinct headings. The following section (Section 2.2) presents papers covering the design and optimization of heat pump systems adapted to dishwashers.

2.1. The Refrigerant Evaluation in Heat Pump Systems

Many studies have been done on refrigerants to limit the usage of refrigerants with toxic effects by looking for alternatives and enhancing the system's efficiency, especially in water heaters and refrigeration systems.

Nawaz et al. [17] built a heat pump model to evaluate the performance of R600a and R290, which has a low global warming potential value and are considered alternative for R134a. Heat pumps performance ratings, such as the amount of refrigerant and cycle time, are evaluated and compared with a system with R134a. As for the results, the heat pump with R600a has the longest cycle time of refrigerants since it has the lowest volumetric heat of vaporization among the examined refrigerants, R290, R600a, and R134a. On the other hand, as R290 has the highest volumetric heat of vaporization, it has the lowest cycle time. Consequently, the required refrigerant in the heat pump system generated with R290 and R600a is almost 50% less than R134a. Moreover, when the different performance factors were considered, R290 showed more excellent prospects as a drop-in replacement for R134a. However, R600a needed a considerably larger compressor size to reach the baseline heating performance.

Bai et al. [18] conducted a simulation study on the performance of a dual-source hybrid heat pump with alternative refrigerants R32, R290, and R600a for R134a. This dual-source hybrid heat pump can be used as heat pipe and vapor compression heating modes. As a result, regarding the coefficient of performance, R600a offered better performance than other refrigerants in vapor compression mode. Besides, in heat pipe mode, the R32 provided higher heating capacity and evaporator pressure, while R290 and R600a had a lower heating capacity and pressure than R134a. Moreover, refrigerant mixtures were analyzed with different mass fractions by utilizing the characteristic properties of the refrigerant in the mixture.

Chang et al. [19] investigated heat pump system performance by considering propane (R290), n-butane (R600), isobutane (R600a), and propylene (R1270) as an alternative for R22 experimentally. In addition to these refrigerants, propane/isobutane and propane/butane blends were also examined. At various compressor speeds, the coefficient of performance (COP) and cooling and heating capacity of heat pump systems with different refrigerants were presented. The condenser and evaporator used in this study were copper tubes with similar dimensions. Besides, ethyl alcohol was used as a secondary fluid for each heat exchanger to restrain freezing and refrigerant flow in a counter-flow arrangement. As a result, R290 offered lower cooling and heating capacity than R22, while its COP was higher than R22.

On the other hand, the capacities and COP value of R1270 were somewhat higher than R22. As for the refrigerant mixtures' performance, as the R290 proportion in the mixture increased, the cooling and heating capacity also increased linearly. Another consequence of this study was refrigerants' average heat transfer coefficients; hydrocarbon refrigerants showed better performance, whereas zeotropic mixtures reduced the average heat transfer coefficients owing to compound distinctness between vapor and liquid phases.

2.1.1. Refrigeration Systems

A study by Richardson et al. [20] pointed out that a propane/isobutane refrigerant mixture can be directly substituted for R12 in vapor compression refrigeration systems. The results indicated that the propane/isobutane mixture by 56/44% provided higher COP than R12 for all temperatures examined. In contrast, 43/57% was the only mixture that achieved higher COP at temperatures above -10°C . Moreover, the system operated with the mentioned mixture in half shares showed similar saturation characteristics with R12. As another result, when the propane mass fraction was increased, the COP value was improved.

In the literature, refrigerant mixtures, mainly propane, and isobutane, are compared to the hydrofluorocarbons like R134a. For example, Lee et al. [21] investigated refrigerators' energy consumption, optimizing refrigerant charge amount and capillary tube length by using an isobutane and propane mixture (45/55) as the refrigerant and compared the results

to the case in which R134a was used as the refrigerant. As a result, it was reported that the optimized system offered much better performance than R134a.

The specific volume of propane and isobutane was quite more than R134a, and the amount of refrigerant was reduced by half in optimized conditions for both refrigerants. At the same time, the capillary length was made longer by 0.5 meters. Therefore, compared to the optimized systems, the system's energy consumption used R134a was almost %12 more than a system with a propane and isobutane mixture.

A similar study was carried out by Jwo et al. [22]. They experimentally investigated using R290/R600a refrigerant mixture (50/50%) as an alternative for R134a used in a 400L household refrigerator. At the end of the study, the energy consumption and the amount of refrigerant charge were evaluated. A performance comparison was made among three refrigerants: R12, R134a, and R290/R600a. The COP values of the R290/R600a mixture and R134a revealed that using 90 grams of R290/R600a showed the best cooling behavior for the refrigerator, and the charge was reduced by 40% compared to the reference 150 grams of R134a. The refrigerator achieved the best cooling effect, in which the R290/R600a mixture was used as the refrigerant, accelerated the temperature changes, and shortened the operating time. Through the reduced working time, the refrigerator's energy consumption using R290/R600a mixture decreased by 4.4% compared to R134a.

Another study, which showed that R600a/R290 mixture could be considered an alternative to R134a, was done by Chandrasekar et al. [23]. In this study, the propane and isobutane mixture ratios were 54.8% and 45.2%, respectively. The experiments were conducted in a 200L single evaporator domestic refrigerator under various ambient conditions such as 24, 32, and 43 °C. Again, the results revealed were similar to the studies in literature; the hydrocarbon mixture offered lower energy consumption than R134a and a roughly 3.5% increase in the COP. In addition, the refrigerator, in which hydrocarbon mixture was used as the refrigerant, provides almost 10 K lower discharge temperature than systems using R134a.

A study was explicitly done for refrigerator-freezer applications by Yan et al. [24] in which a modified vapor compression cycle was developed. Furthermore, a composition

separator was utilized differently from the conventional refrigeration cycle to improve further. This modified cycle was compared to a nonmodified, traditional refrigeration cycle for the COP, volumetric refrigeration capacity, and exergy efficiency. In a conventional system, the performance analysis used R600a and R600a/R290 refrigerant blend, whereas the modified system only uses R600a/R290 mixture as the refrigerant.

In addition, a mathematical model was introduced to examine the modified cycle from the point of thermodynamic performance. The simulation model results showed that the modified refrigeration cycle performs much better under all situations in the study owing to the temperature glide and composition shift of the zeotropic mixture.

2.1.2. Water Heater Applications

In addition to experimental and theoretical refrigerant mixture studies conducted on a refrigerator, some studies evaluate using refrigerant mixtures for water heater applications to utilize refrigerants' distinctive chemical and thermodynamical properties. For example, Tian et al. [25] made a thermodynamic analysis for the performance of a small capacity system, a kind of air source water heater providing high temperatures up to 90°C. The system comprises a compressor, gas cooler, internal heat exchanger, expansion device, and evaporator. An R32/R290 azeotropic mixture with a mass fraction of 70/30% is used as the refrigerant. It is worth mentioning that high heating COP values and volumetric heating capacity were acquired through the refrigerant mixture.

CO₂ mixtures in heat pump systems are also considered due to their low GWP value. Nevertheless, certain characteristic features of carbon dioxide limit its use as a refrigerant. For example, it is not widely used since it operates above critical pressure, which is remarkably higher than the operating pressures of heat pump systems using conventional refrigerants. Moreover, this characteristic creates huge pressure differences between evaporation and condensation temperatures, and depending on it, high throttling loss and irreversibility throughout the expansion decrease system performance.

Baomin et al. [26] examined the effects of pure CO₂ and CO₂ mixtures with R41, R32, R1270, and R161 on discharge pressure, hot water outlet temperature, and thermal

performance of the heat pump system. The binary mixtures enhanced the energy performance by reducing the working pressure of heat pump systems. Besides, the CO₂/R41 blend heat removal process occurred at a supercritical state and provided higher COP than pure CO₂. It was also presented that the temperature glide issue of zeotropic mixtures enhanced COP and declined irreversibility. Following this, it was introduced in this study that the CO₂ mixtures with R32, R1270, and R161 offered higher temperature glide.

Wang et al. [27] evaluated a CO₂/R41 refrigerant mixture as an alternative to pure CO₂. Unlike other papers, this study assessed the mixture for usage in a refrigerated cabinet, air-source heat pump water heater, and water-source pump water heater systems based on different R41 mass fractions. The thermal performance of systems, including the ideal high pressure, maximum COP, compression ratio, discharge temperature, cooling/heating capacities per unit, irreversible loss of each component, and exergy efficiency, was examined. The principal findings indicated that the mixture could be considered an alternative for pure CO₂ for each system. The half mix of mentioned refrigerants declined the optimal high pressure by roughly 30%, with a 20% increase in COP.

Moreover, a remarkable decrease was achieved in the compressor ratio and discharge temperature. It was also deduced that air-source heat pump water heaters were best suited for pure CO₂ refrigerant under the same operational circumstances, followed by water-source heat pump water heaters and refrigerated cabinets. The CO₂/R41 combination may significantly decrease the irreversible loss of each component and improve the exergy efficiency of pure CO₂ systems.

Wang et al. [28] investigated a CO₂/R170 mixture to consider an alternative for R134a in heat pump applications. This study used the azeotropic CO₂/R170 mixture with a mass fraction of 78/22% instead of R134a to evaluate system performance and exergy efficiency. They reported a 31.3% and 30.6% increase in COP and exergy efficiency when the mixture was used as the refrigerant. In addition, the compressor discharge temperature was much lower than the system with R134a. Also, this mixture's compression ratio was decreased by almost four times, and the compressor's lifetime was enhanced. Therefore, CO₂/R170 can be regarded as a mixture that might be a favorable candidate for heat pump systems.

Xiao et al. [29] introduced a two-stage air source compression heat pump model based on MATLAB and REFPROB. The refrigerant mixture consisted of three refrigerants: R290, R600a, and R13I1 (15/20/65%), which was suggested considering the feasibility of substituting R134a. In conclusion, the condensing pressure of R134a was almost 25 bar at 80°C, while the proposed mixture operated at lower pressure, 90 °C. Although considering COP was 4.0 at 70 °C, even if the temperature reached 90 °C, it exceeded 3.8.

In parallel with the results, the mixture refrigerant offered better performance than the R134a, considering lower condenser pressure, higher heating capacity, and coefficient of performance (COP). In other words, the COP value improved by 6.2%. At the same time, due to refrigerants' low GPW, Total Equivalent Warming Impact TEWI can be declined by approximately 19%. Accordingly, it was clearly understood that replacing the refrigerant mixture with R134a could be reasonable.

Another study about heat pump water heaters consisting of a water source evaporator and condenser was performed by Ju et al. [30]. This study investigated the heat pump performance using refrigerant mixture R744 and R290 by comparing R22 experimentally and on simulation. The mixture of R744/R290 with an optimum mass fraction of 12/88% was proposed substituting for R22. It can be seen from the study that the system with a refrigerant mixture offered a much lower discharge temperature while there was a slight rise in the discharge pressure. Therefore, the refrigerant mixture increased heating capacity and COP by 11% and 17.5%, successively, compared to the nominal condition outputs of the system with R22.

Roskosch et al. [31] conducted an experimental investigation on heat pumps for underfloor heating systems. This study asserted that substantial theoretical advantages of zeotropic mixtures have never been seen in experiments. In other words, the studies in the literature have a typical result: the COPs of both pure refrigerants are different, and the maximum COP is usually observed for a specific composition. However, the COP of the best blend composition was slightly higher than the better-performing pure refrigerant. In theoretical studies, fluid-dependent compressor efficiency was generally neglected. This study first described the differences between simulations and experiments. Second, it was demonstrated how zeotropic mixtures might boost system efficiency.

Experimental data from a heat pump with three zeotropic mixtures and theoretical investigations, which include a physical compressor model, form the basis of the analysis. Finally, the compressor's performance was demonstrated to be significantly composition dependent.

Consequently, compressor efficiency is the determining factor for the considerable advantages of zeotropic mixes beyond temperature matching. Searching a fluid database for fluids with well-matching temperature glides and high compressor efficiency resulted in R152a/R32 zeotropic combination for a selected compressor. According to the relevant calculations, zeotropic mixtures can substantially improve heat pump efficiency if the pure components have comparable and high compressor efficiencies.

2.2. System Design and Optimization of Dishwasher with Heat Pump

It is possible to enhance the energy efficiency of dishwashers in various ways, such as modifying the algorithm, using additional water tanks, and specific washing systems. Nevertheless, utilizing a heat pump system to heat water in a dishwasher can be regarded as a ground-breaking implementation for energy efficiency. Also, in the literature, there is a dearth of research on dishwashers that employ heat pump systems.

Bengtsson et al. [32] presented a study in which the heat pump system was used in a household dishwasher instead of an electric heater. Both experimental setup and transient simulation model were introduced to investigate the effect of the cylinder volume of the compressor on energy consumption. Replacing a conventional heater with a heat pump system in a dishwasher provided considerable energy efficiency, up to 25%, even in the same operating time. Furthermore, the results from this study indicated that in conditions where the compressor operates for less than 50 minutes, energy consumption was also reduced depending on the compressor operating time. In other words, the longer the compressor runs, the lower the energy consumption. However, in conditions where the compressor's operating time exceeds 60 minutes, the change in compressor size hardly changes the energy consumption.

The refrigerant evaluation without mixtures was also investigated in the heat pump system integrated with the dishwasher. Bengtsson et al. [33] studied the dishwasher with heat pump simulation by considering R134a, R600a, and R290 as refrigerants to diminish global warming effects. The compressor cylinder volume was optimized to achieve the lowest energy consumption for each refrigerant. The total electricity usage and the CO₂ emission equivalent for three types of refrigerants were calculated according to three different regions: Sweden, European countries, and non-European countries. As a result, using R600a, the heating capacity was increased by 9% compared to R134a. On the other hand, using R290, it decreased by 8%.

Moreover, it was determined that the lowest Total Equivalent Warming Impact (TEWI) value was found in hydrocarbon (R290 and R600a) refrigerants. The TEWI value of R290 and R600a was 74.5% less than R134a and 25% less than the electrical element. Therefore, it was concluded that heat pump dishwashers using hydrocarbon refrigerants with very low GWP could replace dishwashers with heaters or heat pumps with high GWP.

In addition to these studies, Bengtsson et al. [34] optimized a capillary tube on a dishwasher heat pump system with a water source evaporator and R600a as the refrigerant to obtain the lowest energy consumption. They compared the results to the system with an expansion device. Four refrigerant amounts were examined based on the lengths of five capillary tubes. In addition, multiple approaches were utilized to determine the length of the capillary tubes. Consequently, the energy consumption of a single capillary tube during the heating phase was comparable to that of the expansion device.

3. OBJECTIVE

In the literature, certain refrigerant mixtures are examined in heat pump systems. For example, R600a and R290 have been analyzed separately in a heat pump system integrated dishwasher. However, no study can be found that tackles R600a and R290 refrigerant mixture in heat pump systems for water heating. For this reason, this thesis study presents a novel method for dishwasher heat pump systems using R600a and R290 refrigerant mixture and also provide heat pump integrated dishwasher model for both pure and mixture refrigerant cases.

This thesis study investigates the use of R600a/R290 zeotropic mixture on the system performance of a heat pump, which has been adapted to a dishwasher and utilizes pure R600a as its refrigerant. The experimental analysis examines five distinct compositions of refrigerant mixtures, with R290 composition varying between 10% and 50% based on mass. In addition, the experimental study compares the energy consumption, power consumption, cycle time, heating capacity, heating efficiency, high-low pressure differential, compressor outlet temperature, and evaporator behavior with that of a reference system that uses pure R600a. The primary objective is to improve the system performance by leveraging the higher volumetric capacity and gas density of R290 compared to R600a and exploiting the unique temperature glide characteristic of the zeotropic mixture created.

In addition, this study includes the creation of a sub-model for each constituent of the heat pump dishwasher and the subsequent combination of all models to yield a comprehensive dynamic cycle model for the heat pump dishwasher. The cyclic behavior of the dishwasher is empirically substantiated for both pure R600a and R600a/R290 mixture as the refrigerant. Consequently, a heat pump dishwasher is modeled for pure and mixture systems, providing the potential for system-level analysis in a parametric manner, obviating the necessity for experimental testing.

4. SYSTEM DESIGN AND SPECIFICATIONS

4.1. Heat Pump Cycle

A heat pump is a system transferring heat from low-temperature ambient to a high-temperature one. A heat pump aims to absorb heat from a low-temperature source to the region to be heated. Figure 4.1 shows the schematic diagram of the heat pump cycle.

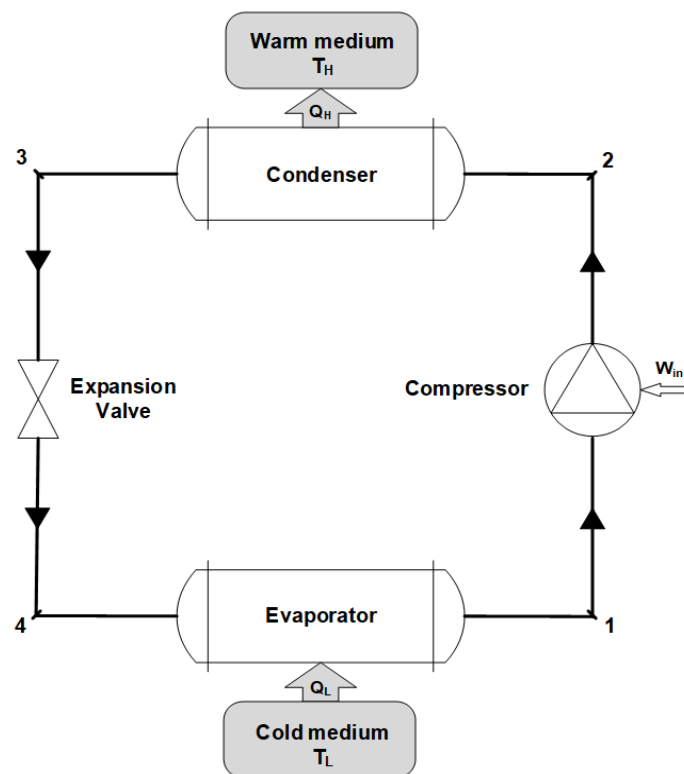


Figure 4.1. Schematic heat pump cycle.

The heat pump system has four main components: compressor, condenser, expansion valve, and evaporator. The ideal heat pump cycle consists of four processes:

- 1-2: isentropic compression is achieved through a compressor,
- 2-3: heat rejection at constant pressure occurs in a condenser,
- 3-4: irreversible throttling occurs in an expansion device,
- 4-1: heat absorption at constant pressure happens in an evaporator.

The ideal p-h diagram is illustrated in Figure 4.2, and according to the first law of thermodynamics, the following equation can be written as

$$Q_H - Q_L = W_{in}, \quad (4.1)$$

here, Q_L represents the heat absorbed from the cooling ambient at T_L temperature, whereas Q_H represents the heat removed to the ambient at T_H temperature. In addition, W_{in} is the magnitude of the work input to the compressor.

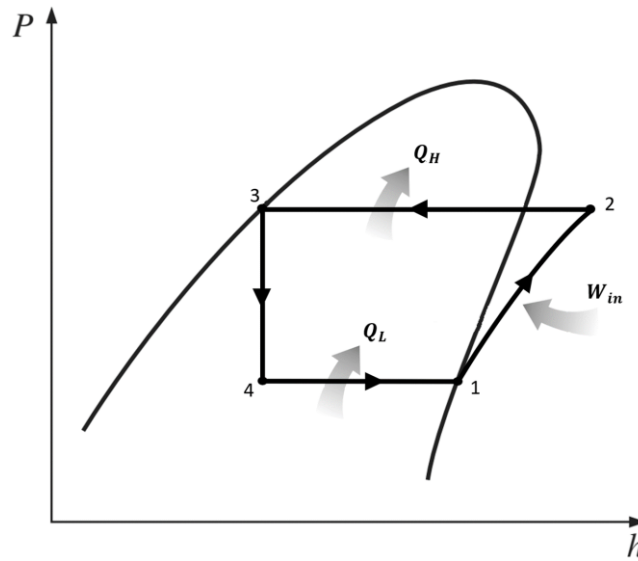


Figure 4.2. Ideal heat pump cycle p-h diagram.

The coefficient of performance (COP) of a heat pump system is defined as

$$COP = \frac{\text{heating effect}}{\text{Work input}} = \frac{Q_H}{W_{in}}. \quad (4.2)$$

In the ideal heat pump cycle, pressure drops due to fluid friction and irreversibility associated with heat transfer processes are not considered. Moreover, even though compression is considered isentropic in the ideal cycle, non-isotropic compression occurs in the compressor.

Actual heat pump cycles differ from the ideal heat pump cycle. Figure 4.3 shows the four processes comprising the actual heat pump cycle in a p-h diagram. First, in the process 1-2, low-pressure, low-temperature superheated vapor refrigerant is compressed and discharged out of the compressor as high-pressure, high-temperature superheated vapor.

Next, this high-pressure superheated refrigerant flows into the condenser via a discharge line. In the process 2-3 occurring in the condenser, heat is removed from the refrigerant. There are three regions in the condenser in terms of the refrigerant phase. The first one is the de-superheating region, where refrigerant becomes saturated vapor. Then, in the condensing region, the refrigerant condenses at constant pressure. Finally, saturated liquid refrigerant is cooled to a lower temperature in the subcooling region. The high-pressure subcooled liquid refrigerant flows to the expansion device through the liquid line from the condenser. In the process 3-4, the refrigerant pressure drops in the expansion device as it flows through a narrow cross-section flow area. At the exit of the expansion device, the refrigerant is a low-temperature, low-pressure mixture of saturated liquid and saturated vapor. This two-phase refrigerant enters the evaporator and absorbs heat from the ambient in process 4-1. In the evaporator, there are two regions. In the first region, named evaporating, refrigerant evaporates and becomes saturated vapor. The second region is the superheating region, where the refrigerant temperature increases due to further heat absorption. Finally, the high-temperature, low-pressure refrigerant vapor leaves the evaporator through the suction line and returns to the compressor. Therefore, the cycle starts over.

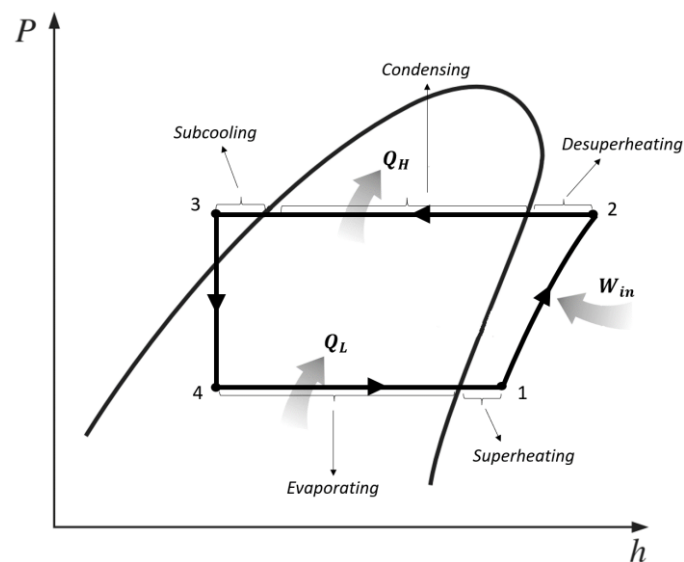


Figure 4.3. Actual heat pump cycle p-h diagram.

4.2. Heat Pump Integrated Dishwasher

The present study conducts experiments on a dishwasher with a heat pump prototype. Heat pump integrated dishwasher includes a compressor, condenser, capillary tube with a dryer as an expansion device, evaporator, and fans. In addition, the heat pump system is assembled with copper pipes through welding processes. The heat pump system is located on the bottom chassis of the dishwasher. The washing water is heated effectively utilizing the heat pump. The schematic diagram of the dishwasher with a heat pump is shown in Figure 4.4.

It can be seen in Figure 4.4 that there are two cycles in a dishwasher integrated with a heat pump: the water cycle and the heat pump cycle. In such a system, water used for washing is heated through a condenser instead of an electric heater. A pump pressurizes the water and continuously circulates it through the condenser until it reaches the desired temperature, and then sent into the dishwasher through spray arms. Details of the components that make up the heat pump are explained individually in the following sections.

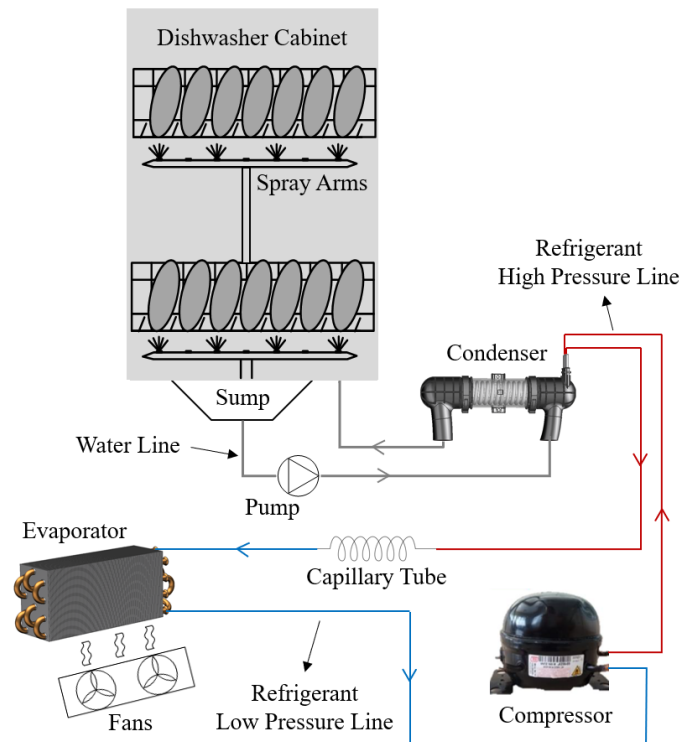


Figure 4.4. Heat pump integrated dishwasher flow diagram.

4.3. Heat Pump Components Used in Dishwasher

4.3.1. Compressor

A reciprocating type hermetic compressor is used in the present study, and an electronic card arranges the compressor speed during the experiment. In addition, the compressor speed is a fixed parameter of 2500 rpm. The compressor capacity can be calculated as

$$\dot{W}_{compressor} = \dot{m}_{ref}(h_2 - h_1). \quad (4.3)$$

Table 4.1 [35] lists the compressor's characteristics according to the ASHRAE standards (at -23.3°C evaporation and 54.4°C condensation).

Table 4.1. The performance values of the compressor [35].

Displacement (cc)	Output Frequency (Hz)	Speed (rpm)	Capacity (kcal/h)	COP (W/W)
8.4	43.3	1300	55	1,77
	70	2100	92	1,80
	100	3000	130	1,76
	136.6	4100	169	1,66

4.3.2. Condenser

In this thesis study, a helical tube stainless steel condenser is used. The condenser is located in an EPDM shell and plastic cover. The washing water is heated by passing it through the condenser. Therefore, heat transfer occurs between the tube condenser and washing water. The water and refrigerant flow in a counter flow arrangement to enhance the heat transfer rate. The condenser capacity is

$$\dot{Q}_{condenser} = \dot{m}_{ref}(h_3 - h_2). \quad (4.4)$$

Furthermore, the heating capacity can be calculated from the water side as

$$\dot{Q}_{water} = \dot{m}_{water} C_{p,water} (T_{max} - T_{in}). \quad (4.5)$$

The equation calculates the heat required for a fixed mass of water used for washing in a dishwasher to reach the specified maximum temperature. Condenser parameters such as diameter, wall thickness, and length are fixed. A schematic illustration of the condenser is shown in Figure 4.5.

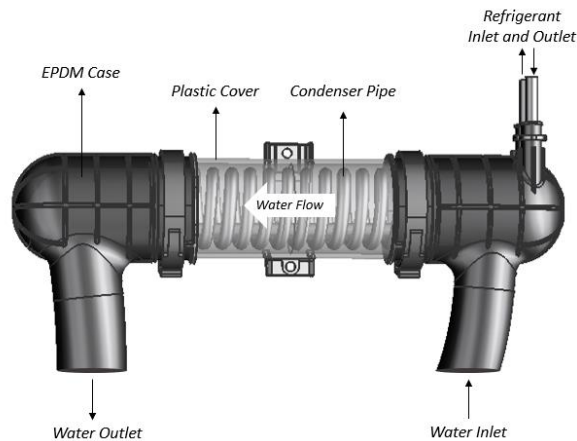


Figure 4.5. Condenser illustration.

4.3.3. Capillary Tube

In this heat pump integrated dishwasher, a capillary tube provides the necessary pressure drop as an expansion device. The capillary tube is a kind of pipe whose cross section is very narrow and is one of the critical elements determining the evaporation and condensation temperature. Besides, the liquid refrigerant enters the filter dryer before the capillary tube at the condenser outlet. The filter dryer removes dirt, foreign particles, and moisture from the refrigerant. The capillary tube with filter dryer is shown in Figure 4. 6.

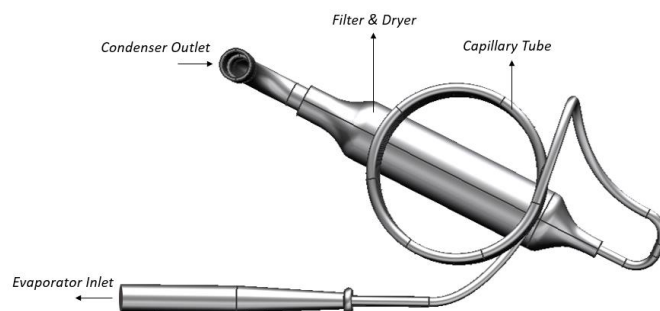


Figure 4.6. Capillary tube and dryer.

4.3.4. Evaporator

A fin and tube evaporator absorbs heat from the ambient in the dishwasher with a heat pump. While the material of the evaporator pipe is copper, the fin material is aluminum. Forced heat transfer occurs between the evaporator and the ambient due to the axial DC fans. In addition, there is space at the bottom of the evaporator to accumulate condensed water on the evaporator and a mini pump to discharge the accumulated water from this space. Figure 4. 7 shows the drawing of the fin and tube evaporator. The parameters of the fin and tube evaporator are considered fixed parameters.

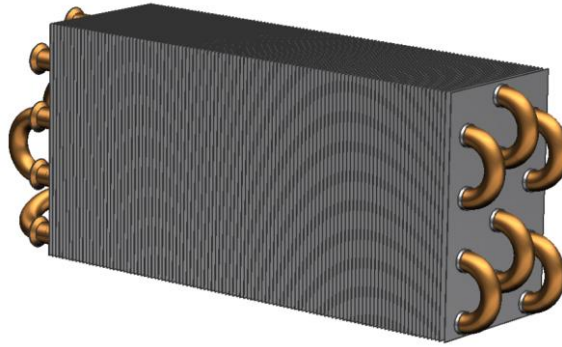


Figure 4.7. Fin and tube evaporator view.

The refrigerant side capacity from the evaporator is

$$\dot{Q}_{evaporator} = \dot{m}_{ref}(h_1 - h_4). \quad (4.6)$$

4.4. Refrigerant

Even though using a heat pump-integrated dishwasher for household applications offers energy efficiency, it may not be sufficient to restrict greenhouse gas emissions. Therefore, the type of refrigerant should also be considered. Since global warming has reached critical levels, restrictions have been applied on using certain fluoride-containing refrigerants by the EU via F-Gas regulations [11]. On the other hand, Hydrofluorocarbon refrigerants (HFCs) like R134a are widely used in heat pump systems. Therefore, evaluating hydrocarbon (HC) refrigerants complying with F-gas regulations is necessary.

A prime example of this is that R600a (Iso-Butane) and R290 (Propane) are hydrocarbons that are becoming increasingly popular due to their lower Global Warming Potential (GWP). The GWP values of refrigerants widely used in heat pump systems are listed in Table 4.2 [36].

Table 4.2. GWP values of widely used refrigerants [36].

Refrigerant	Type	GWP
R12	Chlorofluorocarbon	10910
R32	Hydrofluorocarbon	675
R134a	Hydrofluorocarbon	1430
R404A	Hydrofluorocarbon Blend	3920
R410A	Hydrofluorocarbon Blend	2088
R290	Hydrocarbon	3
R600a	Hydrocarbon	3

On the other hand, a refrigerant mixture can be divided into two types azeotropic and zeotropic. The azeotropic mixtures such as R502, R404A, and R407C behave as a pure refrigerants, whereas zeotropic mixtures like R600a and R290 mixture have a trademark that non-isothermal phase change occurs, and temperature glide effect is observed at the evaporation and condensation temperatures. In other words, during the evaporation process, the refrigerant having a higher volatility than the other refrigerant boils first, and the least volatile refrigerant boils last. However, the opposite of this case takes place in the course of condensing.

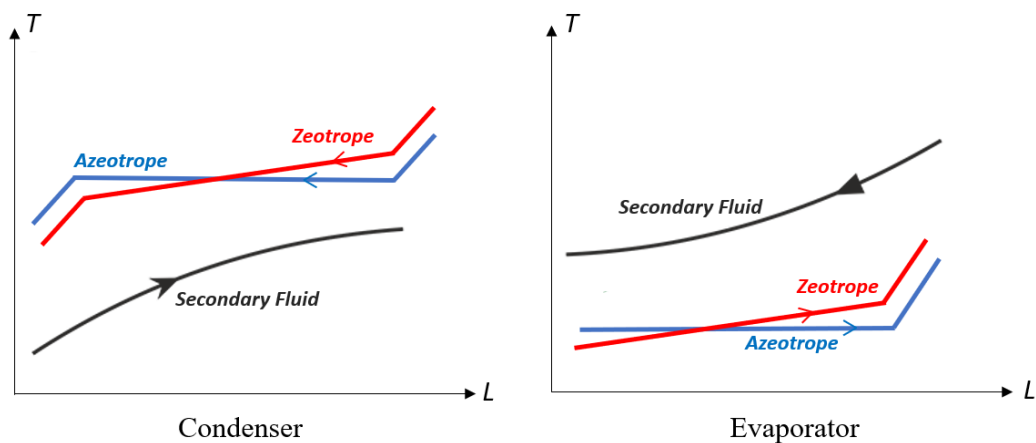


Figure 4.8. Temperature glide demonstration for counter-current flow HEXs.

Thus, the saturation temperature is lower at the evaporator's inlet than at the outlet. On the contrary, the inlet saturation temperature is higher than the outlet in the condenser. The temperature change brings about the gliding effect through the heat transfer surface. As a result, the temperature of the refrigerant in the evaporator changes continuously during the phase change process, allowing for a more efficient heat transfer. Figure 4.8. illustrate the temperature glide for condensing and evaporating.

Moreover, the temperature glide issue reduces the mean temperature differential between the refrigerant and the secondary fluid in the heat exchanger. A decreased difference between the thermodynamic mean temperatures of the heat source and refrigerant led to enhanced overall process performance in energy and exergetic efficiency [37].

The comparison of R600a and R290 properties at the average evaporation and condensation temperatures in the dishwasher heat pump system can be shown in Table 4.3.

Table 4.3. Specific properties R600a and R290.

Refrigerant	Temperature (°C)	Pressure (bar)	Liquid Density (kg/m ³)	Vapour Density (kg/m ³)	Liquid Entalphy (kJ/kg)	Vapour Entalphy (kJ/kg)	Volumetric Capacity (kJ/m ³)
R600a	0	1,57	580,58	4,26	200,0	554,3	1508,4
R290	0	4,74	528,59	10,35	200,0	574,9	3880,3
R600a	50	6,85	517,37	17,60	322,1	620,8	-
R290	50	17,13	448,87	38,63	336,8	621,7	-

Regarding the refrigerants' specifications in dishwasher heat pump systems, R600a is a hydrocarbon widely used in heat pump systems integrated household applications like refrigerators. The low gas density of the R600a and its operation at low-pressure differences provide advantages for the compressor's power consumption for the compression work and the total energy consumption of the dishwasher.

Condenser and evaporator capacity should be increased to reduce energy consumption and cycle time; the refrigerant flow rate should not be low to achieve this. However, the features of the R600a limit this situation at one point.

Although another solution is considered to increase the compressor speed, operating at high speeds may pose a problem regarding compressor life.

On the other hand, R290 offers higher vapor density and the opportunity to work at higher pressures than R600a. In other words, by using R290 in the system, the refrigerant mass flow rate can be enhanced. Accordingly, the evaporator and condenser capacity can be raised.

Furthermore, due to the significantly higher volumetric capacity of R290 compared to R600a, an increase in the mass fraction of R290 is expected to increase the volumetric capacity in mixture states. The refrigerant mixture is applied to the system without any parametric changes to the system components in a drop-in manner, allowing for greater capacity than fixed heat exchanger heat transfer areas and providing high capacity at a constant displacement value on the compressor side.

Thus, since compressor work is calculated as

$$\dot{W}_{compressor} = - \int_1^2 v \, dP. \quad (4.7)$$

Thus, when the pressure and mass flow rate increase, an increase in compressor power is inevitable. At first sight, this can be considered a drawback for energy consumption. However, the increased condenser capacity allows faster heating and rapid heating can precede high compressor power.

This thesis evaluates the case where pure R600a is the refrigerant as the baseline experiment. The system optimization is performed according to this refrigerant. Subsequently, the R600a/R290 zeotropic mixture is experimentally investigated on the optimized system by increasing the R290 mass ratio in the mixture up to 50% in terms of performance outputs.

4.5. Algorithm

Since dishwashers' energy consumption and program duration are declared according to the eco-program, its algorithm is a crucial element affecting energy consumption, washing-drying performance, noise level, and cycle time. In this study, all experiments are carried out based on the eco-program. The algorithm originated from the main, cold, and hot wash. Specific steps in the algorithm depend on washing water temperature, while some are time oriented. First, the algorithm activates discharge pumps if water remains in sump from the previous cycle. Then, 4 L of water is taken for main washing. Finally, the water taken is circulated in the dishwasher tub for a particular time without heating.

Following this, the compressor is activated, and the heating starts. The heating process and compressor work continue until water reaches the determined main wash's maximum temperature of 48 °C. At the end of the heating process, when water comes to the maximum temperature, the compressor is de-activated, and circulation continues for 20 minutes. Subsequently, the dirty main wash water is discharged, and 2.6 L of fresh water is drawn from the water supply network for cold washing. The water is not heated for cold washing and only circulates for 10 minutes. At the end of this process, cold-wash water is also discharged. The next step is hot washing, and for this period, 3.3 L of water is taken into the dishwasher sump. In this process, the compressor is activated, and water is heated up to 45 °C together with circulation. Afterward, the hot wash water is discharged, and the machine door is opened for the drying step 10 minutes after the hot wash water is drained. At this moment, the cycle is completed. Figure 4.9 shows the basket temperature in the dishwasher cabinet throughout the cycle. In this thesis study, the drying step is not examined, whereas the cycle duration is predicated on the compressor's stopping point in which the aimed temperature is reached for the hot rinsing step.

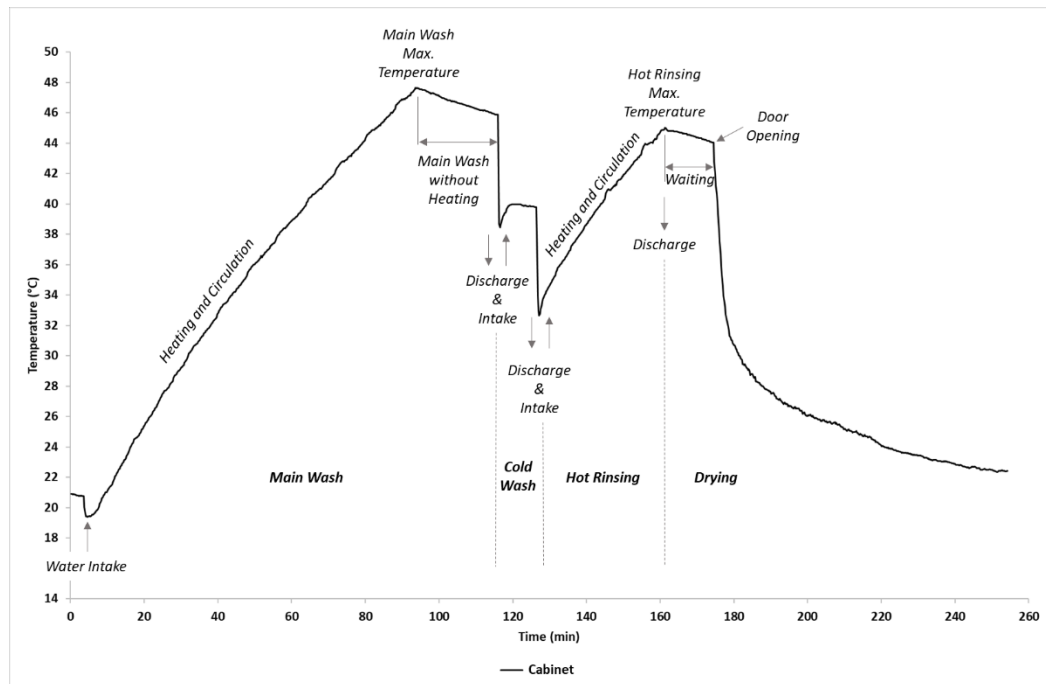


Figure 4.9. Cabinet temperature changes along the cycle.

5. MODELING METHODOLOGY

Each component must be modeled individually to develop a dynamic dishwasher with a heat pump model. The Dymola program, which uses the Modelica language, is used to perform the modeling studies. While the object-oriented Modelica language is used to model cabinets and algorithms, TIL library developed by TLK-Thermo GmbH is utilized for the heat pump system modeling by making detailed modifications to models presented in the library. However, Modelica models the components and situations not available in the library. Having created each model making up the dishwasher and heat pump system, the dynamic model of the dishwasher with heat pump is composed. In this chapter, the component models and the creation of the heat pump cycle model are explained in detail.

5.1. Dishwasher Cabinet Model

The cabinet is one of the essential parts, considering the heat losses from the cabinet to the ambient. Heat losses are the most influential parameter on the energy consumption of dishwashers. For this reason, the model's primary goal to be developed should be to determine the heat loss to the environment where the dishwasher is located through the cabinet walls, as well as the time-dependent variation of the cabinet air temperature.

Each wall of the cabinet does not consist of a single layer. Instead, there is a thin sheet in the innermost part of the cabinet; there are different successive layers, which comprise insulation materials such as felt, bitumen, and foam, on top of this thin sheet in each wall structure. In addition, the dimensions of each layer can differ from each other on each wall. For this reason, each wall's thermal resistance structures are defined to calculate the heat transfer between layers and the environment. Besides, as the length and width of each layer are much larger than their thicknesses, heat transfers in the length and width directions can be neglected for the heat transfer calculation. In other words, the value of the temperature gradient in the thickness direction is significantly larger than in the width and length direction. Thus, a 1-D approximation is evaluated when calculating the heat transfer rates in the model. The layers on each wall are divided into as neat pieces as possible to enhance modeling accuracy. This operation can be called meshing.

The meshing operation of the dishwasher's right wall is shown in Figure 5.1. to illustrate.

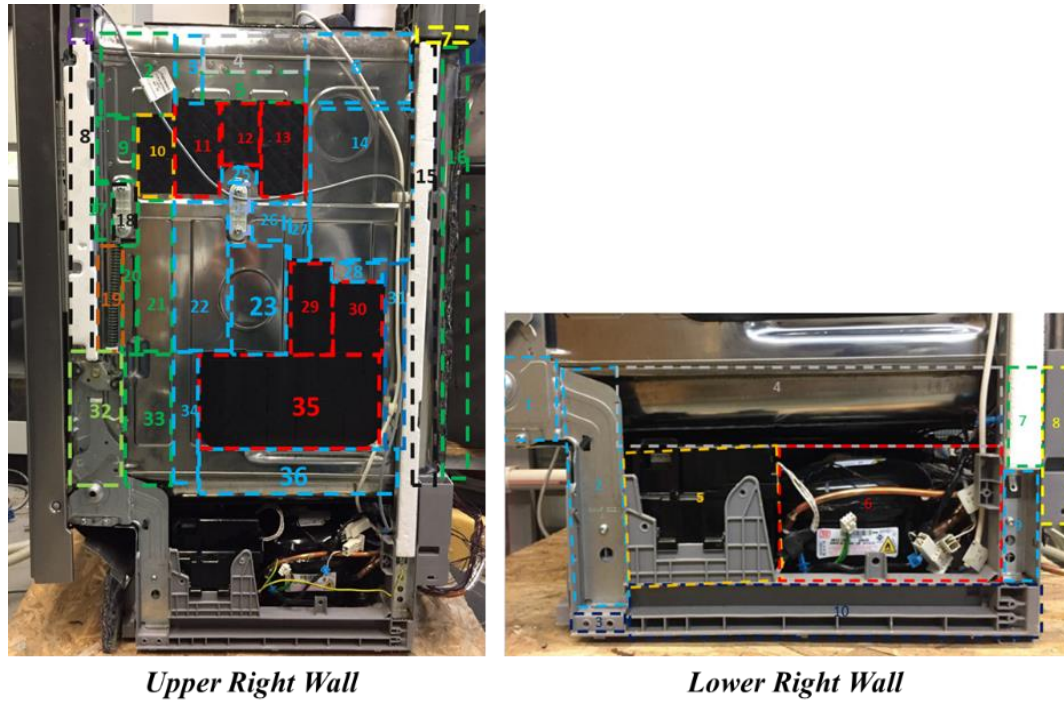


Figure 5.1. Discretization operation of the right wall of the dishwasher.

Table 5.1. Thermophysical properties of dishwasher wall layers.

Materials	Thermal Conductivity (W/mK)	Density (kg/m³)	Specific Heat Capacity (J/kgK)
Inner Metal Sheet	20	7900	480
Cabinet Wood	0,12	650	1760
Upper Plastic Sheet	0,22	900	1800
Air	0,026	1,276	1,006
Plastic	0,177	1000	1400
Bitumen	0,085	1776	1200
Thinsulate	0,047	16	1350
Foam	0,035	27	1100
Metal Sheet	26	7800	460
Felt	0,037	43,6	1380
Compressed Felt	0,037	182,9	1380
Outer Metal Sheet	26	7800	460
Bitumen on Felt	0,0519	2190	905

Although the created imaginary areas are unequal, they are easily calculated since they are divided into regular geometric segments. Then, the layers in each area were determined, and their thermophysical properties were identified. The properties are shown in Table 5.1.

Afterward, the thermal resistance structure of the dishwasher's right wall is made up depending on the meshing operation, as shown in Figure 5.2.

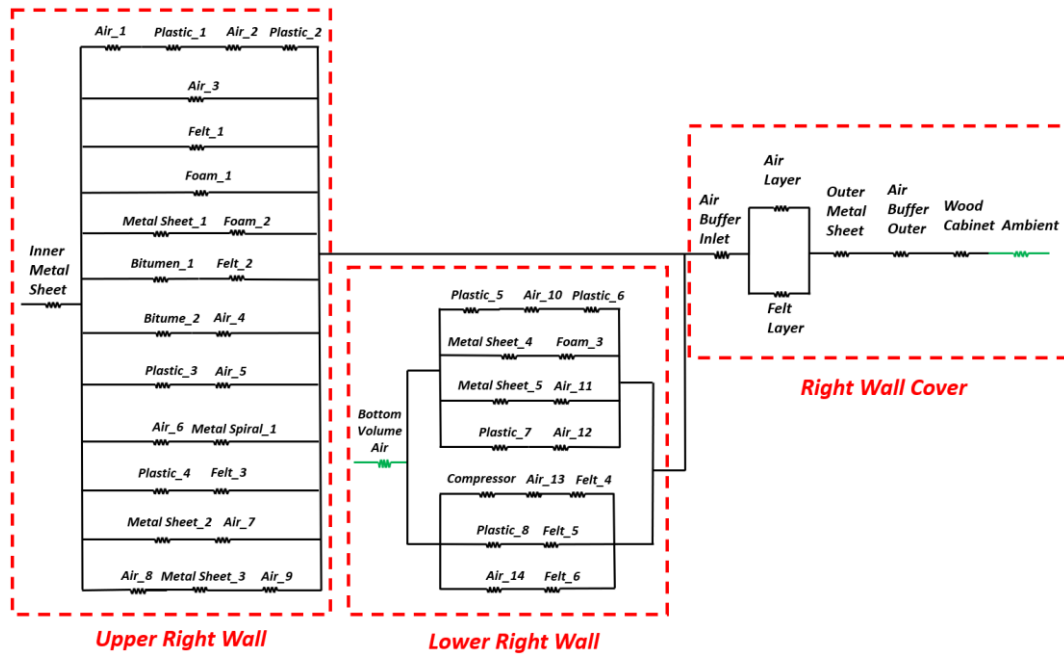


Figure 5.2. Thermal resistance structure of the right wall of dishwasher.

Each layer on an area can be called a wall cell; in Figure 5.2, each resistance represents a wall cell. Also, as seen from the Figure, the resistance structure is created separately for the upper, lower, and wall cover and then put together. Besides, in the Figure, the heat transfer mechanism via conduction is pointed out as black resistances, while the heat transfer by convection is indicated in green.

The primary goal of this model is to get the time-dependent behavior of the cabinet air temperature. In this case, the heat input to the cabinet must be calculated depending on time. The one-dimensional time-dependent heat transfer equation along the wall is supposed to be solved to obtain the desired condition.

The equation can be described as

$$\frac{\partial^2 T}{\partial x^2} = \frac{\rho c_p}{k} \frac{\partial T}{\partial t}. \quad (5.1)$$

Instead of solving this equation directly, each layer is discretized into two parts. For a better understanding of this complex structure in Figure 5.2, the partial schematic presentation of wall cells is illustrated, and the heat flow direction is shown in Figure 5.3.

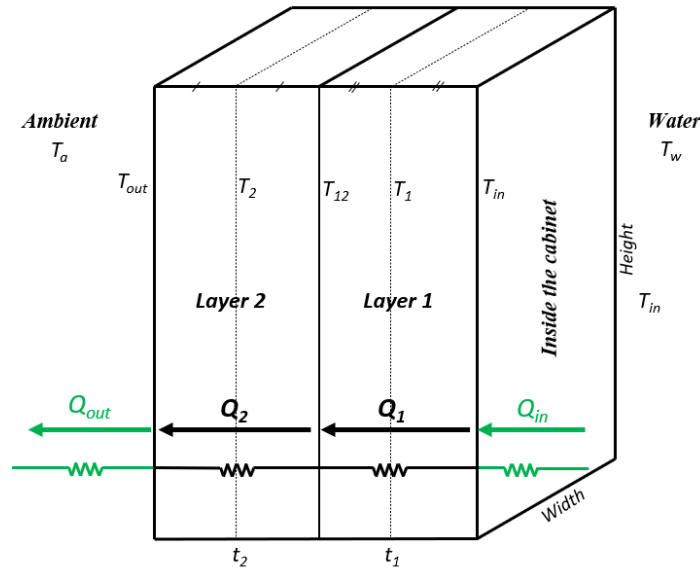


Figure 5.3. Heat transfer mechanism of layered wall structure.

Temperature variables are defined for all points from the inlet of the dishwasher cabinet to the ambient. Since each layer has a specific heat capacity according to mass and material type, they have a thermal capacity depending on these two components. For this reason, the heat entering each layer cannot be equal to the heat leaving. Besides, each layer is regarded as a lumped mass, and therefore the 1st Law gives the temperature for lumped masses,

$$m_{1,2} c_{p,1,2} \frac{dT_{1,2}}{dt} = Q_{1,2}. \quad (5.2)$$

The following equations calculate the heat transfer mechanism, intermediate temperatures, and transient heat loss from the cabinet through the wall layers.

Fourier's Law is used for the net heat transfer through conduction through each wall element as

$$Q_1 = -k_1 A_1 \frac{T_{12} - T_1}{t_1/2} + k_1 A_1 \frac{T_1 - T_{in}}{t_1/2}, \quad (5.3)$$

$$Q_2 = -k_2 A_2 \frac{T_{out} - T_2}{t_2/2} + k_2 A_2 \frac{T_2 - T_{12}}{t_2/2}. \quad (5.4)$$

Heat transfer occurs through convection from the inner and outer surfaces wall structure to the air. For this, Newton's cooling Law is used. For the inner side, it is calculated as

$$Q_{in} = h_{in} A_{in} (T_w - T_{in}), \quad (5.5)$$

while for the outer side, it is described as

$$Q_{out} = h_{out} A_{out} (T_{outer} - T_{amb}). \quad (5.6)$$

In the cabinet's interior, the hot washing water chaotically circulates through the spray arms and hits the cabinet's inner walls. Therefore, the convection coefficient in the interior can be taken as infinitely large. In other words, the water temperature inside and the interior surface of the cabinet can be considered at the same level.

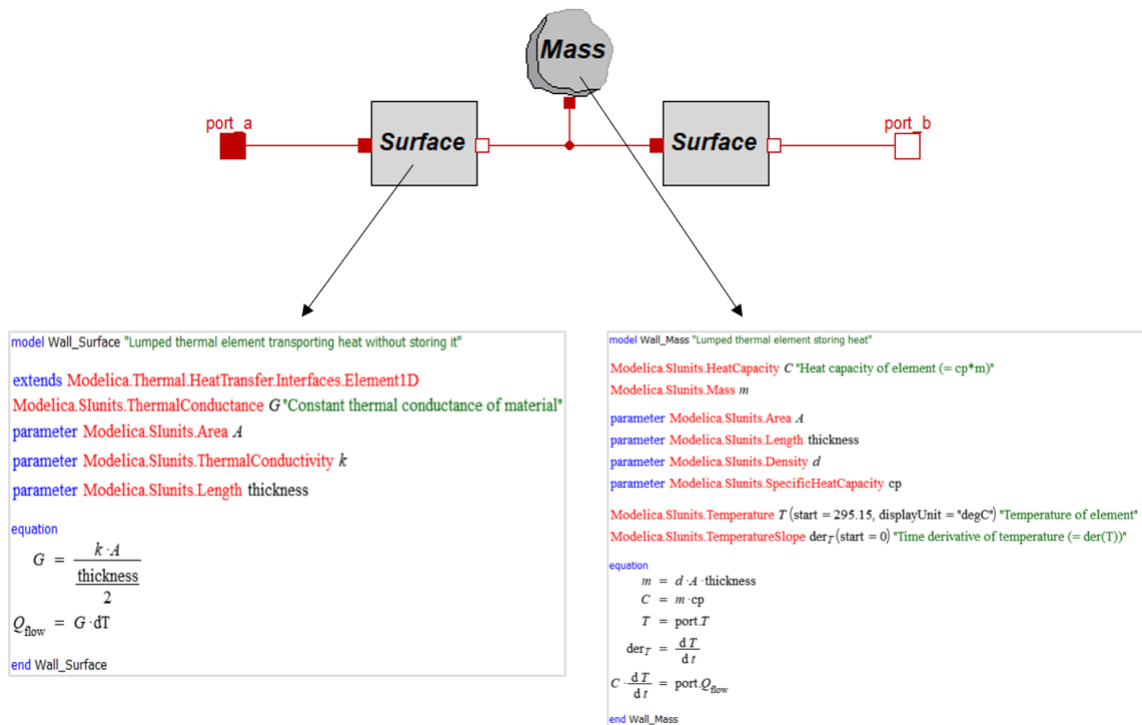


Figure 5.4. The wall cell and its code structure in Dymola.

An object-based wall cell was created to include the equations mentioned above. The wall cell and its code structure in the Dymola program are illustrated in Figure 5.4.

After each wall cell is formed, they are thermally interconnected within the model, as shown in the thermal resistance structure. Figure 5.5 shows the object-based equivalent of the whole thermal resistance structure of the right wall in the Dymola program.

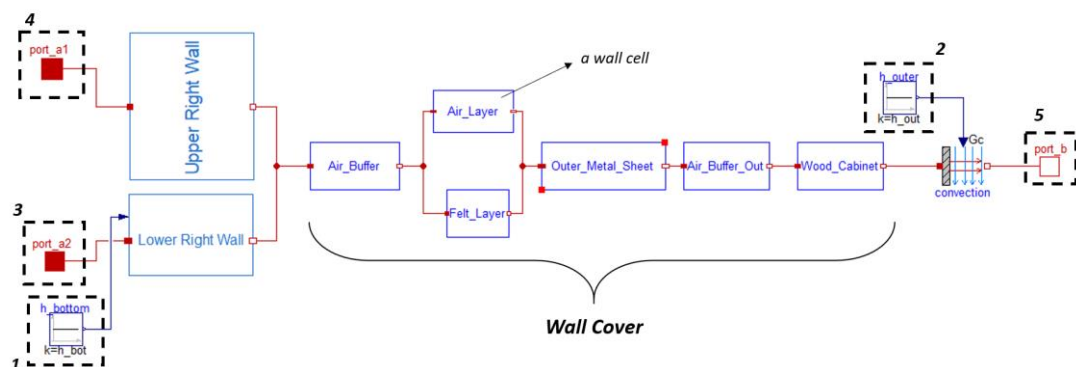


Figure 5.5. The right wall thermal resistance structure in Dymola.

The upper and lower right wall models in Figure 5.5 include their resistance structure comprising wall cells depending on their thermal resistance structures demonstrated in Figure 5.2. Each resistance's area, thickness, mass, and specific heat capacity values are assigned to each model as a parameter. Thus, the right-wall model of the dishwasher is created. Besides, using the heat port structure shown with number 3, the lower wall volumes are thermally connected to the bottom wall model shown in Figure 5.6, and also the heat transfer coefficient in the lower wall volume is assigned to the model thanks to structure number 1. In the same way, the heat transfer coefficient between cabinet and ambient, which is $9.26 \text{ W/m}^2\text{K}$ (ASHRAE handbook [38]), is assigned to the convection model through structure number 2, and the connection with the outer temperature is established with the heat port structure number 5. Similarly, all wall models are thermally connected via the heat port structure number 4, as in Figure 5.6. All other wall model structures were created following the procedures performed while creating the right-wall structure.

In addition, the structure of the cabinet's walls is not the only parameter affecting the thermal behavior of a dishwasher. Inevitably, other components in the cabinet would also affect this thermal behavior to a great extent. The components inside the cabinet can be examined in two parts. The first one is place settings. The number of standard loads in the machine also varies, depending on how many people a dinner set includes. To illustrate, while there are seven dinner plates in a dishwasher with a capacity of 14 people, this number is 8 in a dishwasher with 16 people capacity. Therefore, in the model created as place settings, all components that make up the dinner set, such as plates, cutlery, and pots, are defined as lumped mass depending on the dishwasher's capacity.

On the other hand, the other group of components affecting the thermal behavior of the dishwasher consists of the dishwasher's core parts which are generally plastic and metal. Baskets, spray arms, and filters are examples of these materials. As in the place settings structure, these are defined to the model as lumped mass. Besides, the thermophysical properties of place settings and core components of the dishwasher referred to as goods, are assigned to the lumped mass because heat transfer between these materials and cabinet air occurs by convection.

The heat transfer mechanism between cabinet air and materials is defined in equations below. As mentioned above, the heat transfer coefficient is identified as incredibly high since the hot water is hit all materials and cabinet walls by significantly circulating. The heat transfer equations can be written as

$$\dot{Q}_{Good} = h_{inner} A_{Good} (T_{Cabinet} - T_{Good}), \quad (5.7)$$

$$\frac{dT_{Good}}{dt} = \frac{\dot{Q}_{Good}}{m_{Good}c_{p,Good}}. \quad (5.8)$$

After this stage, there is no obstacle to creating the cabinet model holistically. The created cabinet model is shown in Figure 5.6. The heat port shown as number 1 on this model is associated with the algorithm and is discussed in detail under the heading 5.6 with the cabinet model. Similarly, the bottom chassis air is defined as the lumped mass in the bottom wall model in the output structure mentioned with number 5, and the air temperature is associated with the heat pump components system because the heat pump system is located in the bottom volume of the dishwasher. Through the heat port shown with number 2, the thermal relationship of the cabinet with the ambient temperature has been established. In addition, the component models in the cabinet are shown in boxes 3 and 4, and they are thermally associated with the wall structures.

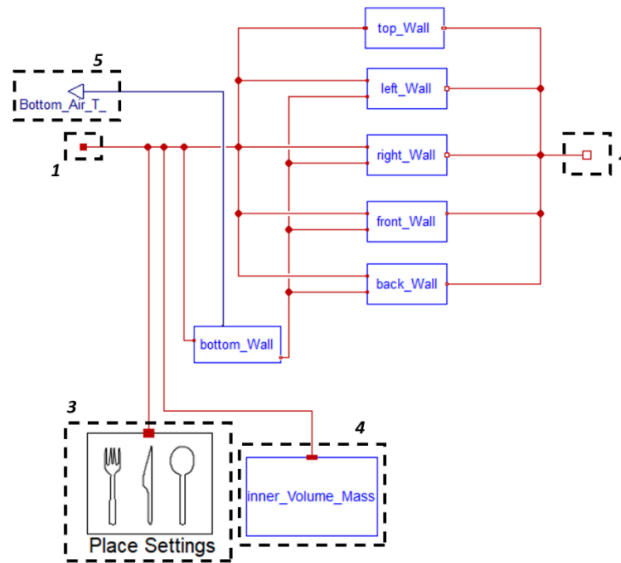


Figure 5.6. The whole cabinet model in Dymola.

5.1.1. Cabinet Model Validation

A heating test was performed on the dishwasher to validate the cabinet model individually. In this test, the dishwasher is heated to a specific temperature. Then, the heating process of the dishwasher cabinet is terminated, and the dishwasher is left to cool down for a specific time. Finally, the same operation is carried out in the model environment. The model created is shown in Figure 5.7.

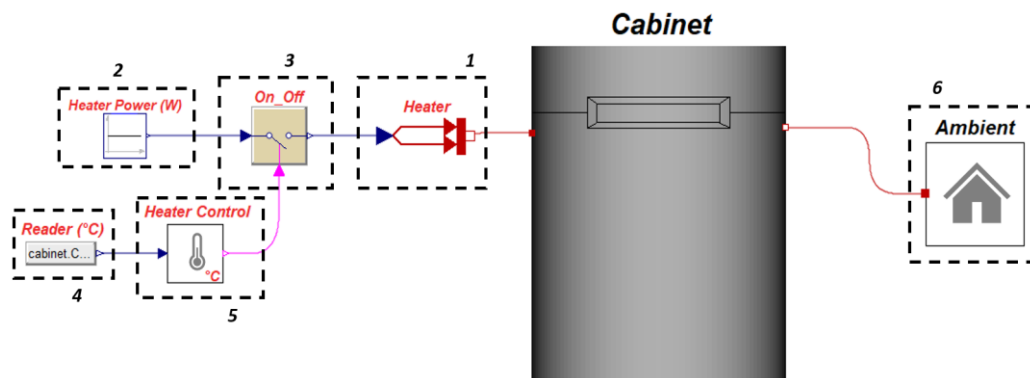


Figure 5.7. The cabinet heating test model in Dymola.

The model created for cabinet validation is stirred through specific components. The heater model available in Dymola thermal library, shown as number 1, is used to heat the cabinet, and heater power is defined in component number 2. The heater is activated or deactivated by utilizing the trigger model (number 3), depending on the cabinet temperature read by the reader component shown as number 4. As the cabinet temperature reaches the set temperature, the heater control model consisting of the algorithm structure, number 5, transmits the false information to the trigger model, and the heater power is turned off. In addition, the ambient temperature and outer heat transfer coefficient are also provided to the model in virtue of the ambient model shown in number 6. Through this validation model, test results and model outputs are compared. The cabinet inner temperature results are shown in Figure 5.8, whereas the air temperature comparison of the bottom chassis is illustrated in Figure 5.9. Besides, the air temperature at the bottom chassis is measured from two points, right and left, in the experiments. As regards the model output, the temperature of the bottom chassis air was obtained as the average of the two measured values. The maximum deviation is less than ± 1 °C in the cabinet inner temperature comparison.

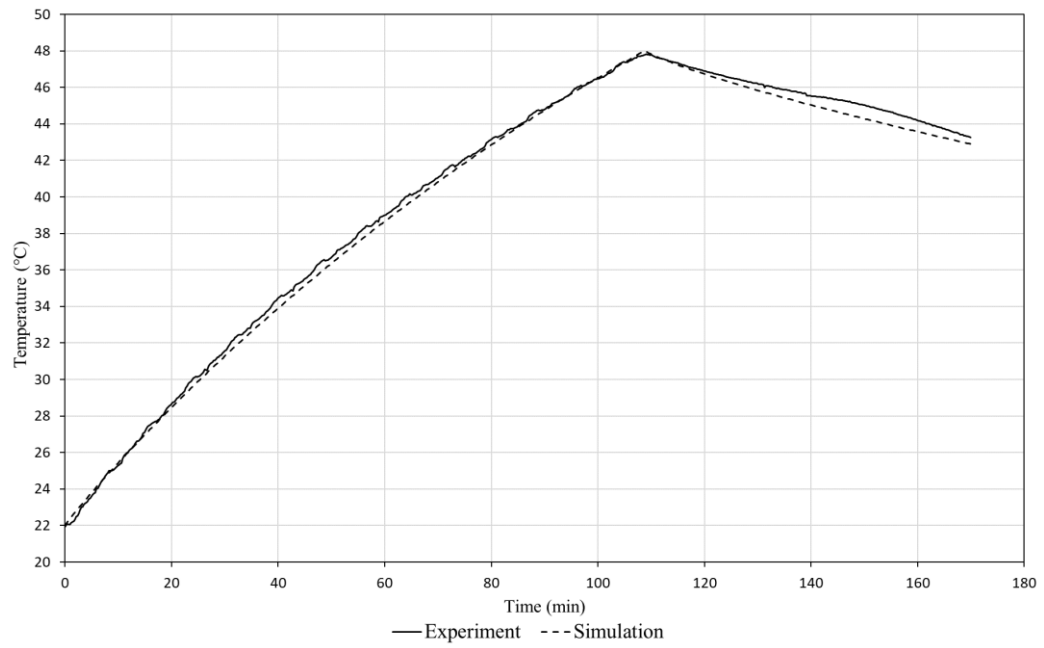


Figure 5.8. The inner cabinet temperature – Exp. vs. Sim.

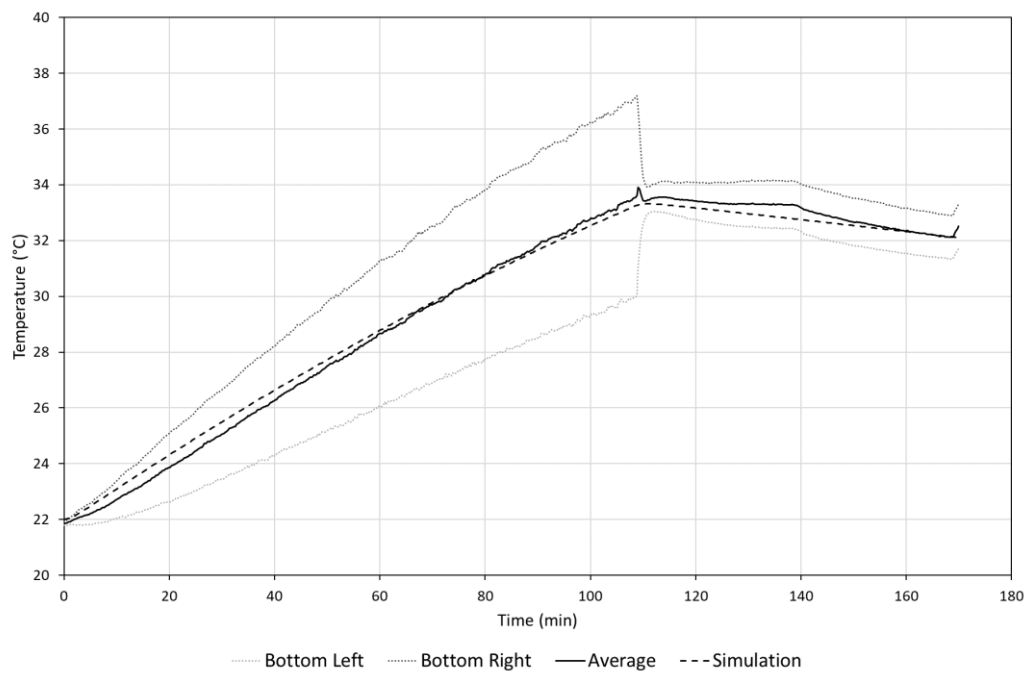


Figure 5.9. The bottom air temperature – Exp. vs. Sim.

5.2. Compressor Model

A compressor can be regarded as the heart of the heat pump system because it provides the pressure difference between the condenser and the evaporator, ensuring that the refrigerant continues to flow. Besides, the power consumed by the compressor is considered a performance parameter, and the compressor's power consumption contributes significantly to the power consumption of the dishwasher system. For this reason, it is essential to model the compressor accurately and holistically to obtain results compatible with experiments. Even though there are several compressor types like scroll and rotary, the heat pump system integrated dishwasher has a reciprocating type compressor, as mentioned in Chapter 4. This chapter explains the compressor model's structure, operating logic, and work applied to the heat pump system model.

It is generally considered that a compressor carries out compression operation as isentropic. In this case, the power can be calculated in simple terms by the first law relation as

$$P_{isentropic} = \dot{m}(h_{2s} - h_1). \quad (5.9)$$

However, in reality, the compression process does not occur isentropically since a few losses bring about irreversibility and make the compression process non-isentropic. These losses, together with their brief explanations, are listed as follows:

- Pressure drops at the suction line: The pressure drop occurs to some extent while refrigerant goes to the compressor through the inlet valve. Thus, the inlet pressure reduces, and the compressor's compression ratio increases.
- Pressure drops at the discharge line: When the refrigerant at high pressure starts to move from the discharge valve to the condenser, a pressure drop occurs in this part.
- Internal leakage: Refrigerant leakage is inevitable along the cylinder walls and the piston towards the compressor body.
- Mechanic friction: Due to the movement of the piston, the friction causing losses occurs between the cylinder and piston.

- Heat transfer at suction and discharge line: the compressor losses heat to the ambient from both suction and discharge lines.
- Re-expansion of the refrigerant: This case arises from the clearance volume at the end of the compressor stroke volume. The clearance volume is created to allow for thermal expansion and manufacturing tolerances. Because of the clearance volume, some refrigerant remains in the cylinder after compression. Because there is high-pressure refrigerant inside the cylinder, the suction valve cannot be opened directly. Instead, the volume inside must grow, and the refrigerant pressure inside must fall for the low-pressure refrigerant from the evaporator to enter the compressor. As a result, there is some re-expansion of the refrigerant left inside after compression. A particular volume is sacrificed to allow for re-expansion. Eventually, the entire compressor volume cannot be used to pressurize the refrigerant, and the mass flow rate would be lower than the ideal case in which re-expansion is neglected.

All losses mentioned above are demonstrated in Figure 5.10.

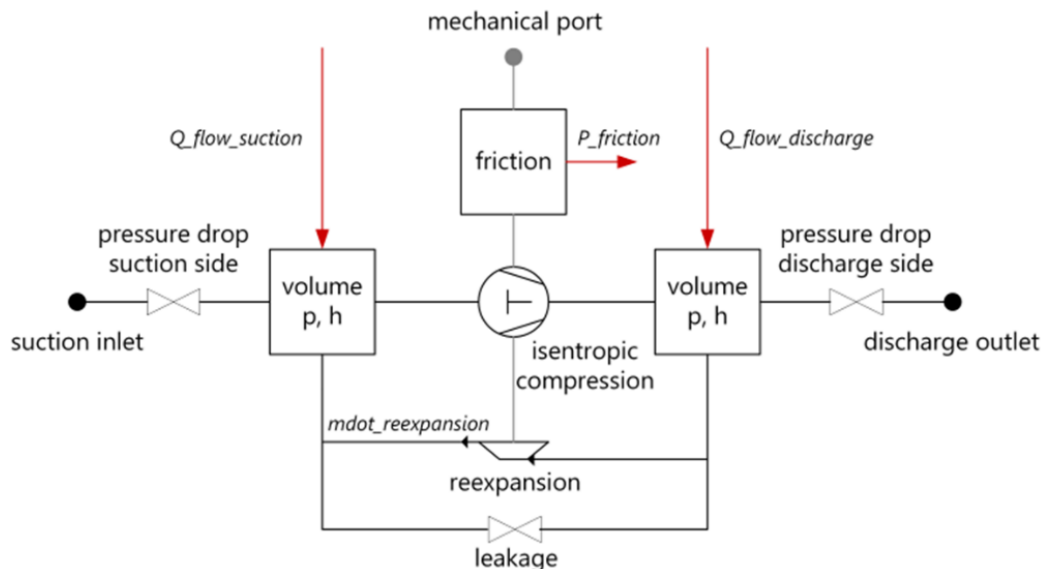


Figure 5.10. The losses in a compressor.

5.2.1. Compressor Model Structure and Equations

There is a reciprocating compressor model in the TIL library. This model considers all the losses mentioned above except for the suction side heat loss since the refrigerant temperature at the suction side is not entirely different from the ambient. These losses are combined with the transient form of the mass and energy balances to derive the actual mass flow rate and power values. Following that, the equations needed to determine each loss are thoroughly discussed.

Considering the pressure drops and internal leakage, the Saint-Venant-Wantzel [39] equation describes the pressure drop and internal leakage effects stemming from the pressure difference between the discharge and suction line. The equation considers an effective flow area while relating the mass flow rate to the pressure ratio. The flow area can be configured as a model parameter by the user. The pressure drop along the narrow section is shown schematically in Figure 5.11.

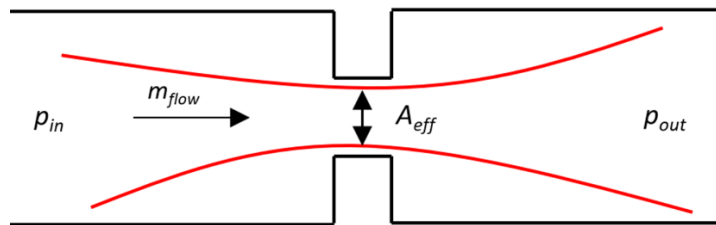


Figure 5.11. Pressure loss and flow behavior in constriction.

The equation used is shown as

$$\dot{m}_{flow} = \varphi A_{eff} \sqrt{2p_{in}\rho_{in}} \quad (5.10)$$

where, $\dot{m}_{flow}$ is the mass flow rate due to internal leakages and is shown in Figure 5.10. A_{eff} is one of the nine parameters the compressor model requires, and how to obtain these parameters is explained in detail in the next sections. Also, p_{in} is the inlet pressure, whereas ρ_{in} is the density at the inlet.

In addition, φ is a factor calculated by comparing the pressure ratio with the critical value. The pressure ratio (Π) is defined as

$$\Pi = \frac{p_{out}}{p_{in}}, \quad (5.11)$$

and the critical pressure ratio (Π_{crit}) is given as

$$\Pi_{crit} = \left[\frac{2}{\gamma + 1} \right]^{\frac{\gamma}{\gamma - 1}}. \quad (5.12)$$

In the case of $\Pi > \Pi_{crit}$, the equation transforms that

$$\varphi = \sqrt{\frac{\gamma}{\gamma - 1} \left[\left(\frac{p_{out}}{p_{in}} \right)^{\frac{2}{\gamma}} - \left(\frac{p_{out}}{p_{in}} \right)^{\frac{\gamma + 1}{\gamma}} \right]}, \quad (5.13)$$

while if $\Pi < \Pi_{crit}$, it can be written that

$$\varphi = \left(\frac{2}{\gamma + 1} \right)^{\frac{1}{\gamma - 1}} \sqrt{\frac{\gamma}{\gamma + 1}}, \quad (5.14)$$

where γ is the specific heat ratio of the refrigerant.

As for the friction losses, the equation that expresses the friction loss is shown as

$$P_{friction} = fn + n^2 f_{piston} d, \quad (5.15)$$

where $P_{friction}$ indicates the power loss owing to friction, f is the friction factor, n represents the rotational speed of the compressor, f_{piston} is another friction factor related to the piston, and d presents the compressor's stroke volume called displacement.

In regard to the re-expansion of the compressed refrigerant, the mass flow rate equation depending on the re-expansion of the refrigerant, is shown below. The space above the piston in the compression process is named dead space, S_{dead} . Also, t_{delay} represents the time it takes to close the valve at high speeds. ρ is the density of refrigerant in the vapor phase. Then, n represents the rotational speed of the compressor, and d is the displacement. The equation is stated as

$$m_{re-expansion} = (S_{dead} + t_{delay}) n p d. \quad (5.16)$$

Another loss is the heat loss and the net heat transfer is indicated as Q_{flow} . As mentioned above, $P_{friction}$ is the power loss owing to friction, considered as a heat-generation parameter. α is the heat transfer coefficient, and A is the compressor's outer shell area. Lastly, ΔT is the temperature difference between ambient and compressor discharge temperature. The heat transfer is expressed as

$$\dot{Q}_{flow} = P_{frict} + \alpha A \Delta T. \quad (5.17)$$

5.2.2. Determination of Model Parameters

As the equations in the previous section are examined, it can be seen that the best parts of terms are unknown physical parameters of the compressor. These parameters below cannot be calculated theoretically, yet they must be defined in the compressor model. To that purpose, a model fitting technique is used to acquire the requisite nine parameters semi-empirically:

- Flow area, A_{eff} (suction, discharge, and leakage)
- Displacement, d
- Friction coefficient, f
- Friction coefficient piston, f_{piston}
- Relative dead space, S_{dead}
- Discharge valve delay, t_{delay}
- Heat transfer coefficient, αA

Picking up these parameters consists of several steps. First, the compressor map data must be obtained. The data is achieved by a calorimeter experiment in which the compressor is attached to a heat pump system. The system keeps evaporator and condenser temperatures constant using external heaters and a controlled expansion device. Moreover, the subcool and superheat temperatures can be arranged desired value and kept fixed. Various input parameters, which are temperature values of condensation, evaporation, subcooling superheating, and compressor speed, are defined through the calorimeter system.

In addition, the system provides discharge temperature, capacity, power consumption, and mass flow rate as outputs as a means of its measurement system. Finally, the compressor map is obtained; in other words, the outputs corresponding to the inputs reveal the behavior of the compressor under different evaporation, condensation, and speed conditions broadly.

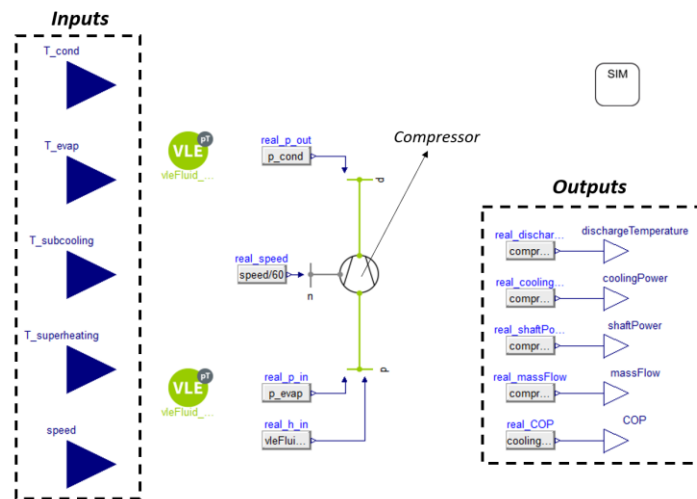


Figure 5.12. Compressor testing model.

In parallel with the calorimeter experiment, a compressor test model is created to simulate the calorimeter experiment. First, however, the nine physical parameters must be defined in the compressor model. The test model is shown in Figure 5.12.

The next step is to create a functional mock-up unit (fmu) from the compressor test model. The fmu file is a part of the functional mock-up interface [40]. This cross-platform interface created a standard in which other modeling and simulation environments can produce and use the C code of a system model of a modeling environment. Then, the fmu procured from the test model is read on an Excel-based program (called "Modelfitter") solved with a statistical approach and created by TLK-GmbH. Besides, the input data of the calorimeter system are defined to the Modelfitter as dependent variables, while output data are defined as independent variables. After these processes, the Modelfitter is run, and model fitting is carried out. Thus, all unknown physical parameters are achieved.

The Modelfitter program includes an embedded model-fitting algorithm in a macro excel file to explain the model-fitting operation in more detail. The model uses the initial values for the parameters mentioned above in this method to generate an initial output set. Then, the initial outputs are compared to the calorimeter outputs' independent variables. Next, the residuals are compared to a user-specified minimum desirable value, and the difference is used by an algorithm to estimate new parameter values. This procedure is continued until the minimal residual level is achieved. The Levenberg-Marquardt Algorithm [41] was used in the study. This approach combines the well-known Gradient Descent and Gauss-Newton methods. Although the results are far from the optimum, gradient descent can be evaluated as more effective, while Gauss-Newton is more feasible when the results are close to the optimum. Levenberg-Marquardt uses these advantages by creating a parameter whose value is modified depending on the distance from the optimal.

Consequently, Modelfitter detects the optimal parameter values to guarantee that the models are best fitted to fixed measurement data. After achieving fitted parameters, simulation and experiment outputs are compared according to the calorimeter results. The comparison for the discharge temperature is shown in Figure 5.13, and it can be seen that the deviation is less than 1°C. Similar comparisons are made for capacity, input power, and mass flow rates. These are seen in Figures 5.14-5.16, and the model results are compatible with the experimental data within $\pm 5\%$ error range.

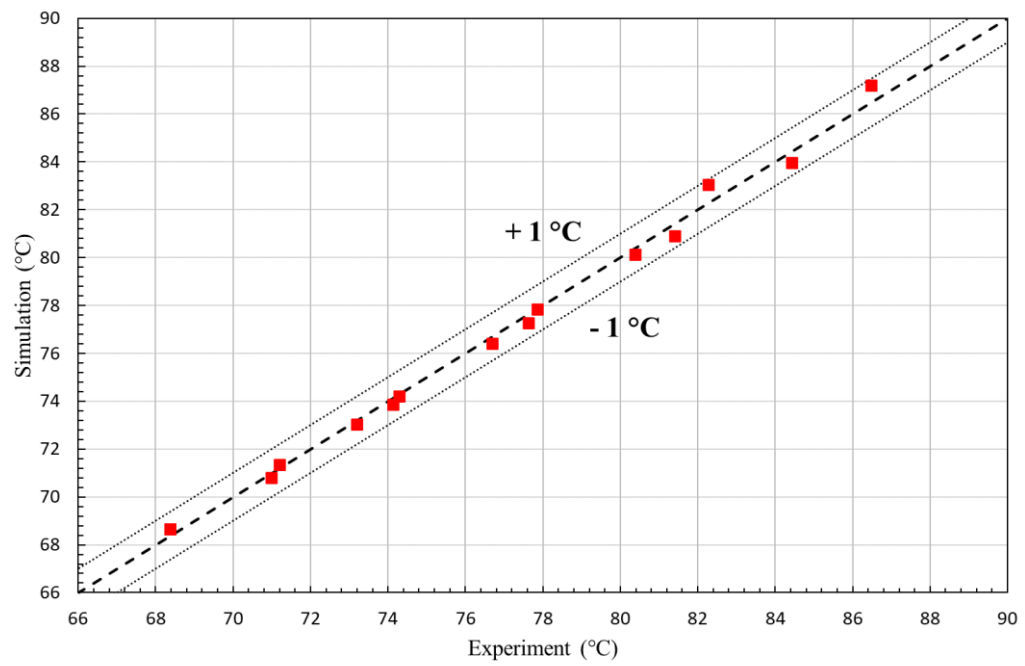


Figure 5.13. Comparison of simulation and experiment results for discharge.

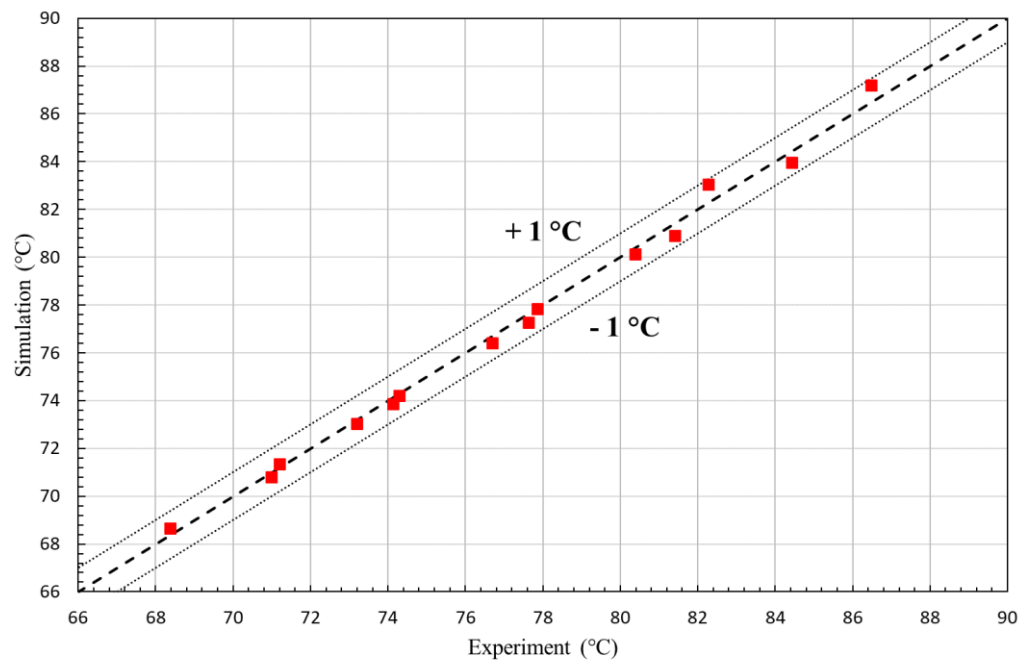


Figure 5.14. Comparison of simulation and experiment results for capacity.

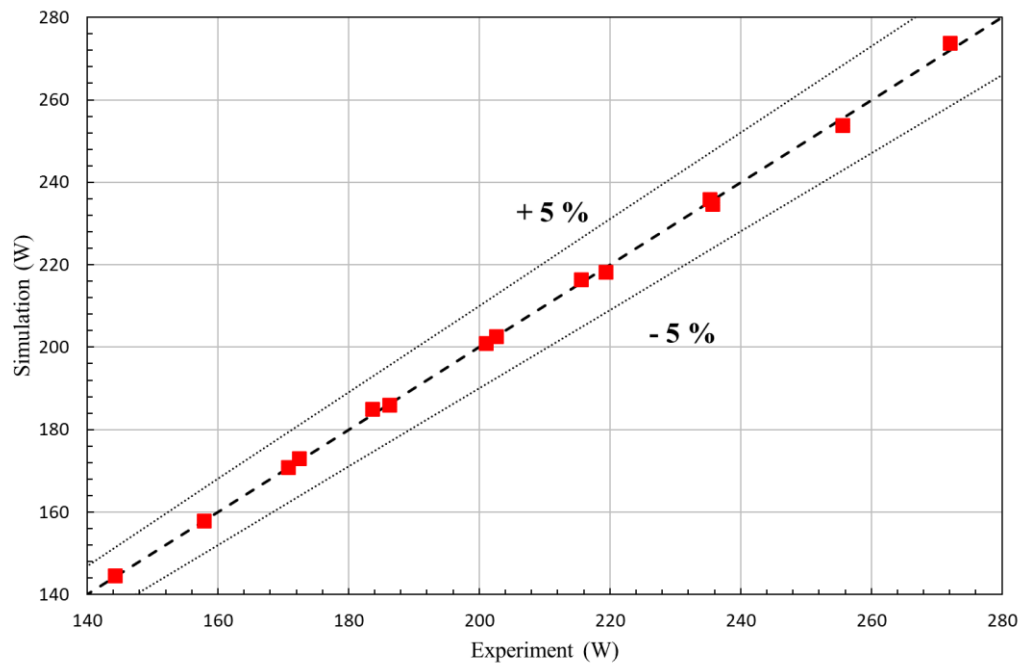


Figure 5.15. Comparison of simulation and experiment results for power.

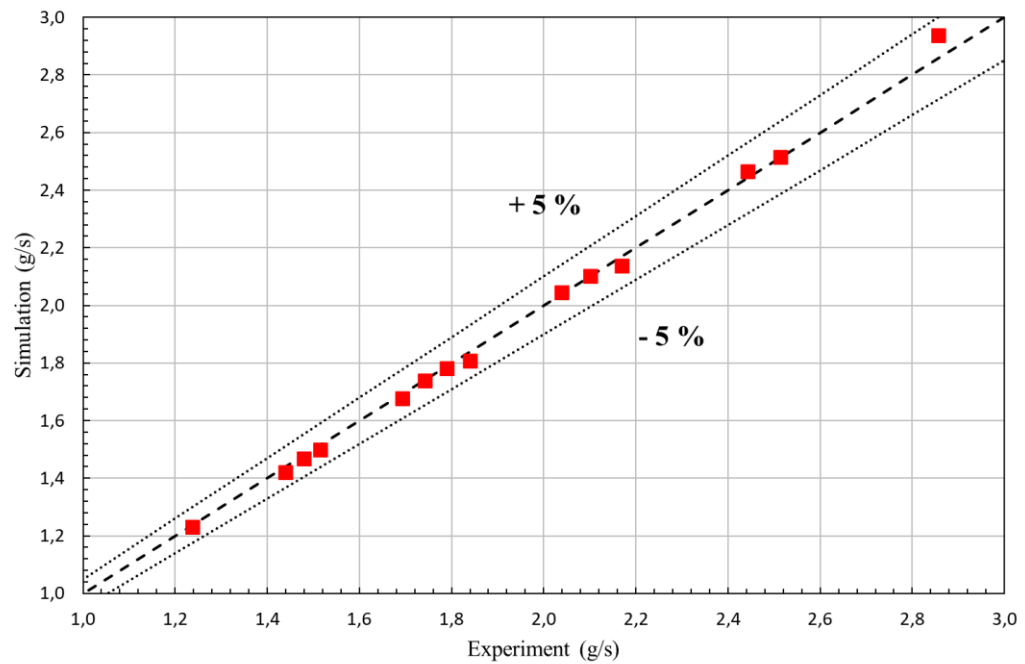


Figure 5.16. Comparison of simulation and experiment results for mass flow rate.

5.3. Heat Exchanger Model

The heat pump system consists of two heat exchangers, a condenser, and an evaporator. Next, their types and modeling methodology are explained.

5.3.1. Condenser Model

The water is heated in the heat pump dishwasher by passing through the helical condenser mentioned in Chapter 4. The condenser is modeled using the TIL library's "Tube" component. This model expects to define the geometric parameters required for the solution: total pipe length and diameter. Moreover, the library offers heat transfer coefficient and pressure drop correlations for the user to choose the appropriate one in the model. However, the pressure drops can be neglected since the dishwasher heat pump system is compact. The tube model is illustrated in Figure 5.17.

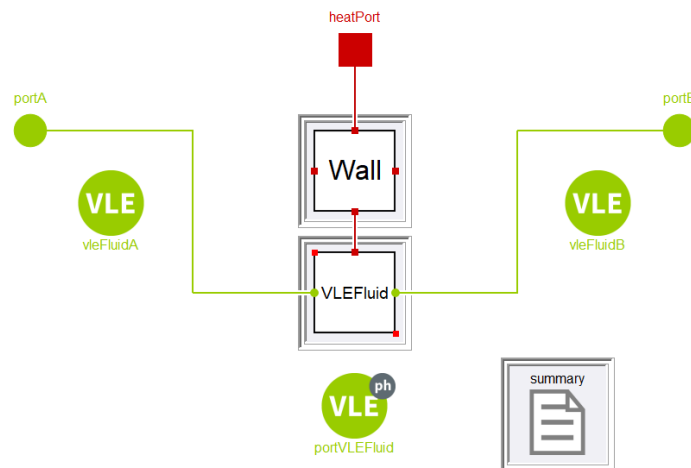


Figure 5.17. Tube model demonstration in TIL library.

Considering the tube model in TIL library shown in Figure 5.17, green lines represent the refrigerant side called VLEFluid (Vapor Liquid Equilibrium Fluids). Besides, "portA" indicates the tube's inlet, whereas "portB" relates to the tube's outlet. Also, the thermo-physical properties of refrigerant, enthalpy, and pressure are represented through sub-models

around the "Wall" and the "VLEFluid" blocks. These blocks are comprised of mentioned discretized cells. The discretized cells are illustrated in Figure 5.18.

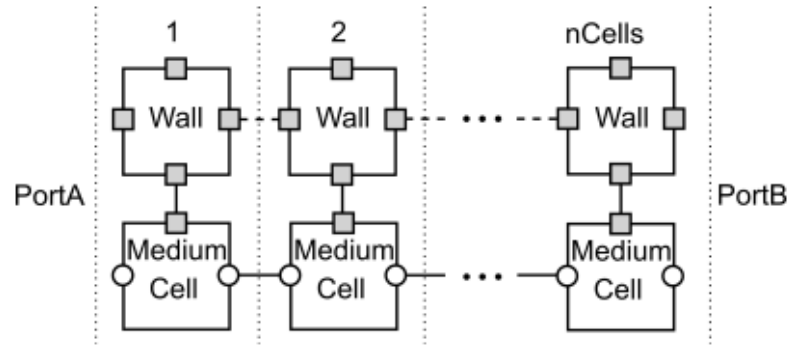


Figure 5.18. The discretized cells compose the tube model.

The discretization operation is taken place by aligning cells in the flow direction. The cells are connected by considering the finite element approach, and the user states the number of cells. Besides, each cell's mass, momentum, and energy balance equations are solved individually as time-dependent. While the medium cells correspond to working fluid and involve its properties, the tube's wall is discretized by solid masses, represented by wall cell models. The user can specify the wall material; accordingly, the density, heat capacity, and thermal conductivity are assigned. Moreover, the transient equations consider the medium's inertia obtained from the tube geometry.

The wall heat conduction model represents the heat transfer via the tube's wall. In other words, the heat transfer in the tube component is not only on the refrigerant side, but the tube wall is also considered in the heat transfer calculations. Besides, the component shown in Figure 5.17 allows an external heat flow rate to be considered. The "heatPort" element defines the thermal resistance value and temperature of the medium in which the tube is located. The heat transfer mechanism is shown in Figure 5.19.

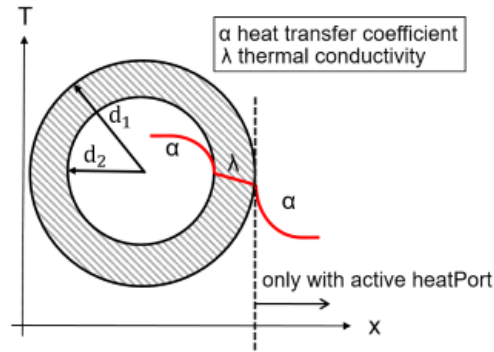


Figure 5.19. The heat transfer mechanism of the tube model.

On the other hand, changeable heat transfer models from the literature are included in the tube model, which estimates heat transfer from the medium to the wall. Also, the user can define the constant heat transfer coefficient instead of using a correlation. The correlations used in the single-phase situation vary with the Reynolds number value.

- In $Re < 2300$, the flow is considered laminar, and the following Nusselt number is used and indicated as

$$Nu = 3,6568. \quad (5.18)$$

- In $2300 < Re < 100000$, the flow is considered in the transition region. The following correlations obtained by Gnielinski [42] are used for the Nusselt number. The Gnielinski correlation is

$$Nu = \frac{\left(\frac{\zeta}{8}\right) (Re - 1000) Pr}{1.07 + 12,7 \sqrt{\frac{\zeta}{8}} (Pr^{2/3} - 1)}, \quad (5.19)$$

and

$$\zeta = (0,79 \ln Re - 1,64)^{-2}, \quad (5.20)$$

where ζ represents the friction factor.

- In $Re > 100000$, the flow is turbulent, and Dittus-Boelter [43] correlation is used as

$$Nu = 0,023Re^{4/5}Pr^{1/3}. \quad (5.21)$$

However, since no correlation was defined in the dual-phase state and the helical condenser model, the dual-phase transport coefficient was calculated experimentally and added to the model. While making this calculation, first, the condenser capacity by the refrigerant is calculated by multiplying the inlet and outlet enthalpy difference with the refrigerant flow rate. Afterward, the capacity obtained in the single-phase part is subtracted from the total capacity, and the capacity taken by the two-phase is attained.

Since the water inlet and outlet temperature and water flow rate are known, the convection coefficient on the waterside h_{water} , is calculated using the Gnielinski correlation. However, the friction factor (ζ) is calculated according to the Moody diagram as

$$\zeta = 0.184Re^{-0.2}. \quad (5.22)$$

Then, the UA value was obtained using the $LMTD$ method, and the fluid side two-phase heat transfer coefficient was achieved from here. These calculations were made by establishing a thermal resistance network, as shown in Figure 5.20.

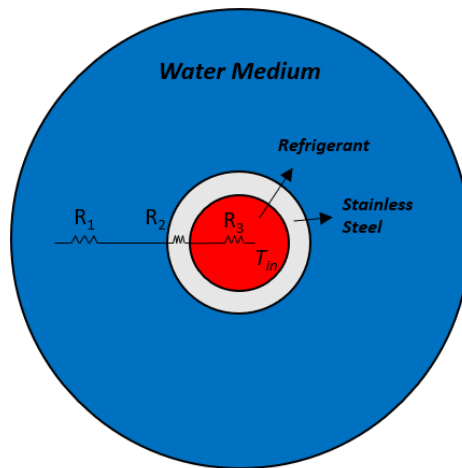


Figure 5.20. The thermal resistance structure of the condenser.

The resistance structures in the thermal resistance network are formulated as follows:

$$Q = UA \Delta T_{lm}, \quad (5.23)$$

$$UA = \frac{1}{R_{eq}}, \quad (5.24)$$

$$R_{eq} = R_1 + R_2 + R_3. \quad (5.25)$$

Each resistance can be calculated as follows:

$$R_1 = \frac{1}{h_{water} A_{water}}, \quad (5.26)$$

$$R_2 = \frac{\ln \left(\frac{D_o}{D_i} \right)}{2\pi L k}, \quad (5.27)$$

$$R_3 = \frac{1}{h_i A_i}. \quad (5.28)$$

Thus, looking at the equations, it is seen that the only unknown value is h_i , and in Equation 5.28, h_i is obtained by substituting all the calculated values.

The condenser model using the Tube model is illustrated in Figure 5.21.

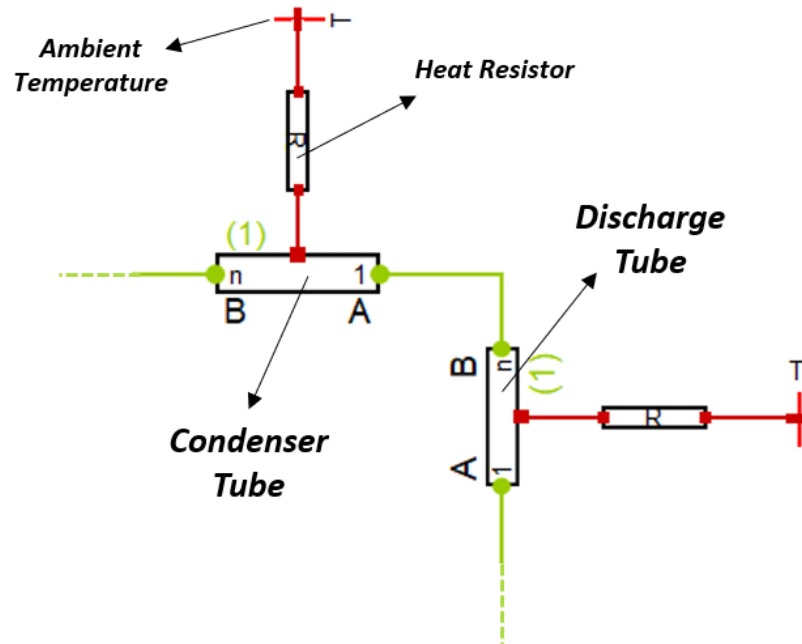


Figure 5.21. The tube condenser model.

In the condenser model created, a heating resistor is defined by activating the heat port to simulate the heat loss by the condenser to the water, and this resistor is associated with the ambient temperature in the bottom chassis volume of the cabinet.

In addition, a discharge tube is modeled with a tube model because the refrigerant coming out of the compressor passes through a copper pipe. Before entering the condenser, this part has heat transfer to the environment. In this section, since the refrigerant is in a single phase, single-phase correlations are considered in the discharge tube model. Again, a heating resistor is used to simulate the heat transferred by this tube to the water.

In addition, the heat loss from the water to the environment is calculated from the experimental results. The difference between waterside and fluid side enthalpy changes is considered the loss while modeling.

5.3.2. Evaporator Model

The other heat exchanger in the dishwasher heat pump system is the fin and tube evaporator. The two-phase refrigerant from the capillary is evaporated by drawing heat from the ambient and sent to the compressor as superheated vapor. To model the evaporator, fin and tube heat exchanger structure presented in TIL library is utilized. The fin and tube geometry is illustrated in Figure 5.22. The user defines all the geometric dimensions.

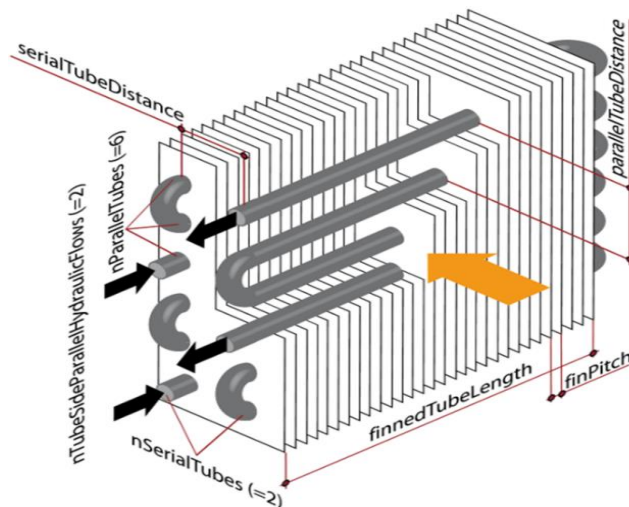


Figure 5.22. Fin and tube geometry in TIL library.

The TIL library has two types of fin and tube heat exchanger models: parallel and crossflow. In the heat pump system discussed in this study, an evaporator with a crossflow arrangement is used, as detailed in Chapter 4.

The structure of the model contains a tube model, and the solution mechanism is not far from that described in the condenser. However, the discretized cell structure is slightly different as it contains an additional secondary fluid. This discretized cell structure is shown in Figure 5.22.

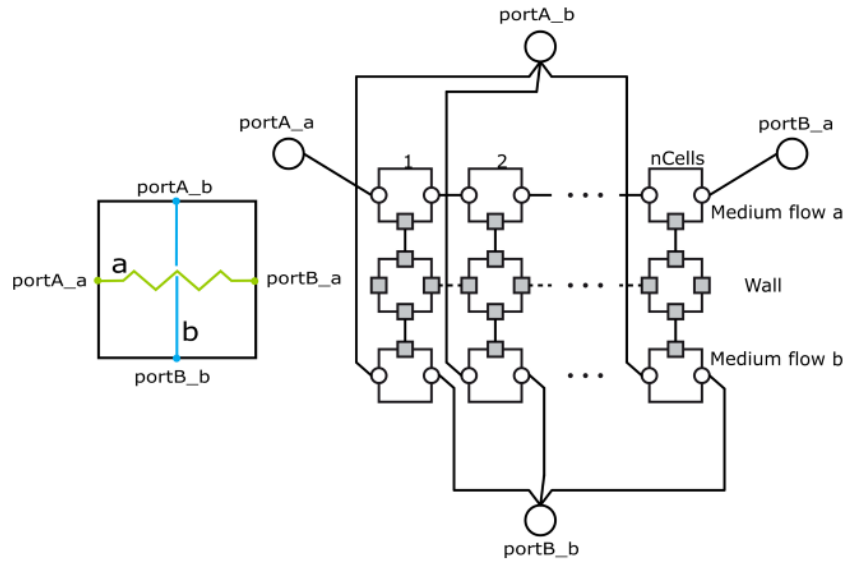


Figure 5.23. The cell structure of the crossflow fin and tube evaporator.

In Figure 5.23, "Medium flow a" and related ports indicates the refrigerant flow, whereas "Medium flow b" and "b" ports represent the secondary fluid flow. Moreover, ports a and b are associated throughout the wall cells. The blocks are discretized along the flow direction, and the mass-momentum and energy equations are solved for each cell according to the replaceable heat transfer correlations presented in the model. While the correlations in the condenser in the single-phase region were used for the refrigerant side, the Shah [44] correlation was used for the two-phase region as

$$\alpha = \alpha_{liq} \left[(1 - x)^{0.8} + \frac{3.8x^{0.76}(1 - x)^{0.04}}{p_r^{0.38}} \right]. \quad (5.29)$$

On the other hand, while simulating the forced convection on the air side, the library's air side correlations are unsuitable for the structure used in the existing system. In other words, no correlation was found considering the wavy fin structure. For this reason, an experimentally developed correlation for a wavy or corrugated fin structure by Chi-Chuan Wang et al. [45] fits Dymola by defining it in Modelica language. After testing the correlation, it determines the heat transfer coefficient on the air side. The evaporator model created is shown in Figure 5.24.

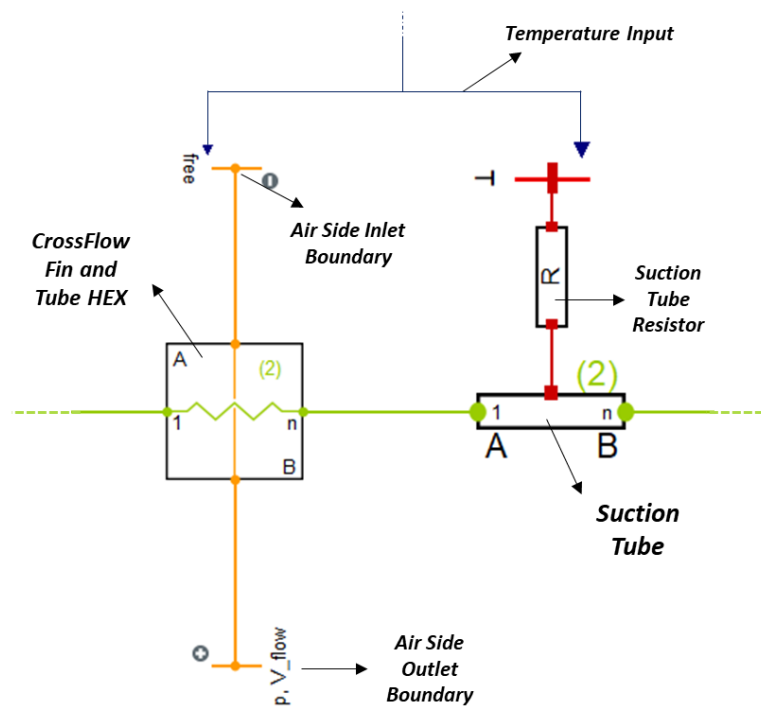


Figure 5.24. The evaporator model representation in Dymola.

The suction tube is also modeled like the discharge tube in the condenser model since the refrigerant draws heat from the ambient as it flows through the suction tube. The resistance structure here has been experimentally fine-tuned. In addition, looking at the HEX model, the line shown in orange represents the air side by selecting moist air as secondary fluid, and the boundary conditions of the air are determined as inlet and outlet boundaries. Parameters such as air flow rate and relative humidity are determined from the outlet boundary, representing the fans' blowing part.

On the other hand, considering the suction part of the fans, the bottom volume temperature modeled and mentioned in Chapter 5.1. is used as input to the boundary. Thus, the evaporator's suction air temperature is matched to the bottom volume temperature, and the location of the evaporator is simulated.

Modeling this bottom volume is highly vital at this point. Because in the actual situation, the temperature is constantly increasing, and the evaporator constantly changes the suction air temperature, adding a dynamic state to the heat pump.

5.4. Capillary Tube Model

The capillary tube is one of the most crucial components of the heat pump cycle. The capillary tube, a long pipe with quite a narrow cross-section, acts as an expansion device and ensures the continuity of the cycle by providing the necessary pressure drop. In this way, while the liquid phase refrigerant coming out of the condenser enters the capillary tube, its temperature drops significantly along with its pressure expanding. As a result, the refrigerant at low pressure and low temperature enters the evaporator as a two-phase, liquid-vapor mixture at the capillary tube outlet.

The pressure drop in the capillary tube is linear up to a certain point. In this part, the fluid is single-phase and liquid. After a certain pressure, the first vapor bubble starts to originate, and the flow now takes place as a two-phase. In the two-phase region, the pressure drop is nonlinear, and the pressure drop per unit length increases toward the end of the capillary tube. This phenomenon mentioned above can be explained as follows; while only friction affects the pressure drop in the flow in the single-phase region, the momentum effect increases as the vapor ratio increases in the two-phase region, and total density decreases depending on the matter of fact. Therefore, the total density decreases gradually, creating a steep acceleration in the flow, which brings about a rapid pressure drop [46]. This case is illustrated in Figure 5.25.

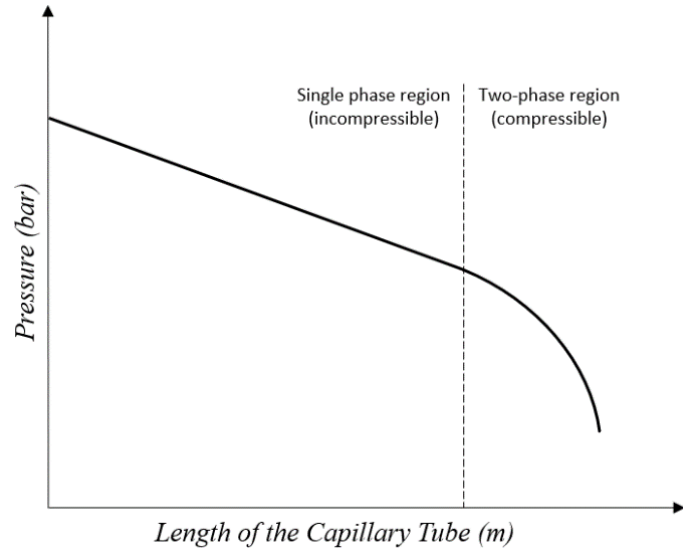


Figure 5.25. Theoretical pressure drop characteristic of the capillary tube.

While modeling the capillary tube, the main aim is to achieve the pressure drop characteristic shown in Figure 5.25. Thus, the flow regime could be examined as single or two-phase. The flow is supposed to be fully developed in the single-phase region, and the one-dimensional Navier-Stokes equation is reduced to

$$\frac{dp}{dx} - \mu \frac{d^2 u}{dy^2} = 0. \quad (5.30)$$

By neglecting body force terms and disappearing inertia terms, the Navier-Stokes equation is integrated throughout the tube area, and the equation is transformed to

$$p \cdot \pi \cdot \frac{D^2}{4} - (p + \Delta p) \cdot \pi \cdot \frac{D^2}{4} - \tau_{wall} \cdot \pi \cdot D \cdot \Delta L = 0. \quad (5.31)$$

in which p is pressure, D is tube diameter, and ΔL is the unit length of the tube.

In Equation 5.31, the shear stress, τ_{wall} , remains, and the pressure drop is only associated with the viscous shear stress term. In single-phase flow, the friction gives rise to the pressure drop, and is shown as

$$\Delta p = \left(f \frac{G^2}{2\rho_{liq} D} \right) \Delta L, \quad (5.32)$$

where f represents the Darcy friction factor, whereas G is the mass flux.

The Colebrook correlation [57] can be utilized to figure out the friction,

$$\frac{1}{f^{0.5}} = -2 \log \left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{Re f^{0.5}} \right), \quad (5.33)$$

However, the correlation offers an implicit solution. Therefore, instead of using this correlation Swamee-Jain's [48] correlation putting explicit solution forward is evaluated. The Swamee-Jain correlation is given as

$$f = \left[-2 \log \left(\frac{\varepsilon/D}{3.7} + \frac{5.74}{Re^{0.9}} \right) \right]^{-2}, \quad (5.34)$$

where ε represents the tube roughness.

The pressure drop in the two-phase region is considered more sophisticated than single phase case because of the change in the density of the refrigerant. In this case, the inertia term is added to the reduced and integrated version of the Navier-Stokes equation. The formula is shown as

$$p \cdot \pi \cdot \frac{D^2}{4} - (p + \Delta p) \cdot \pi \cdot \frac{D^2}{4} - \tau_{wall} \cdot \pi \cdot D \cdot \Delta L = m \cdot \Delta V, \quad (5.35)$$

where ΔV represents the velocity change.

Unlike the single-phase case, the pressure drop in two-phase flow consists of friction and momentum effects. In the literature, numerous correlations are proposed to analyze pressure drop growing friction and momentum effect. However, here again, the Swamee-Jain correlation, which is formed to cover both effects, is used.

As a result, the equation below is solved for each cell in the capillary tube model that makes the solution by dividing it into cells, and the total pressure loss is calculated considering the effect of the two-phase situation as

$$\Delta p = \left[\left(\frac{f}{2D} \right) \Delta L + \frac{\rho_{in} - \rho_{out}}{\rho} \right] \frac{G^2}{\rho}. \quad (5.36)$$

5.5. Algorithm

The working logic of the algorithm is explained in detail in Chapter 4. Two basic moves trigger the algorithm. While one of these basic movements is the set temperature value, the other is the time step. In addition, each algorithm step needs to be modeled separately since the algorithm structure throughout a cycle does not consist of loops that start and end with regular steps. In other words, each step's time or set temperature value differs.

In the model, the algorithm structure was established according to a logical chain by using the modeling elements in the Modelica logical library and revising the code structure. In addition, the structures not in the library were created as a new model using the Modelica modeling language, which is generally composed of "if" and "when" structures. Besides, since the structures used in the algorithm model receive the inputs and values to be controlled from the heat pump system and the cabin model, it is crucial to establish a logical connection between them. Therefore, while creating this model, mostly "timer" and "trigger" structures were used to simulate the steps containing periods and the set temperature values. These structures are illustrated in Figure 5.26.

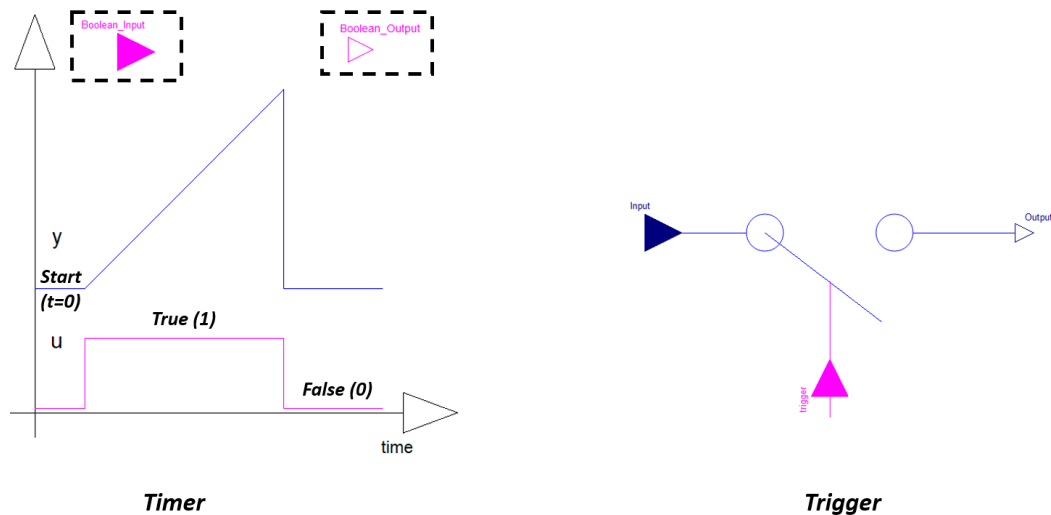


Figure 5.26. Timer and Trigger demonstration.

First, looking at the timer model, it works with a "Boolean" input that contains only true/false information. According to the revision, a timer runs in the code when the input is true. It outputs false when the time specified for the timer is over.

Alternatively, with a different code revision, it takes the true input and runs the timer until the input returns false and dynamically outputs the result of the timer. In this way, the desired period is obtained independently of the simulation time and is controlled by the outputs.

Concerning the Trigger structure, it takes a "Boolean" input, and the given input to the model is out of the model as output only if the "Boolean" input is true. Otherwise, the predetermined start value is output. Namely, when the "Boolean" returns false, the predetermined initial value is output out of the model. The algorithm model is created by associating these and similar structures.

On the other hand, the water to be heated is modeled as a lumped mass in the algorithm structure. In each washing step (main wash, cold wash, hot rinsing), a different water model is used, and thanks to the structures mentioned above, the heating systems of the waters are switched on and off and associated with the cabinet structure. The heater model in the Modelica thermal library was used to heat the water structure. In the dishwasher model with a heater, the heater capacity is written to the heater system, while in the heat pump model, the condenser capacity is written to simulate the heating of the water with the heat pump system. Therefore, the algorithm model must be linked to the cabin and the heat pump model.

Besides, since the power consumed by the elements such as the circulation pump, compressor, and discharge pump are known during the working period and working time, the dynamic power consumption and the total energy consumption amount are calculated using the modeling components mentioned again.

In addition, the time, temperature, and power values defined in the system to model the algorithm have been moved out of the algorithm model to create a user-oriented algorithm. In other words, a user who wants to change these values can easily change these values parametrically without interfering with the code structure.

5.6. Model of Dishwasher with Heat Pump

The heat pump model was created by combining the components mentioned in the modeling steps in the previous sections in the flow direction, and each component is numbered and shown in Figure 5.27. Here, the algorithm activates the compressor (1) by its mechanical port in which the compressor speed is defined. Next, the refrigerant leaving the compressor loses heat as it passes through the discharge tube (2) and enters the condenser (3). Then, the refrigerant passing through the capillary tube (4) returns to the compressor by passing through the evaporator (5) and the suction tube (6). Finally, the bottom volume modeled temperature is provided as input to the discharge tube, condenser tube, evaporator air suction line, and suction tube cabinet model via the resistance element to simulate the environment of the heat pump system on the model. In addition, with the system information manager (8) component called "SIM," the fluid types like moist air and the refrigerant used in the system are selected, and the relevant fluid is automatically defined for all components. In addition, sensors (7) are positioned after each component to check the system's temperature, enthalpy, and pressure values.

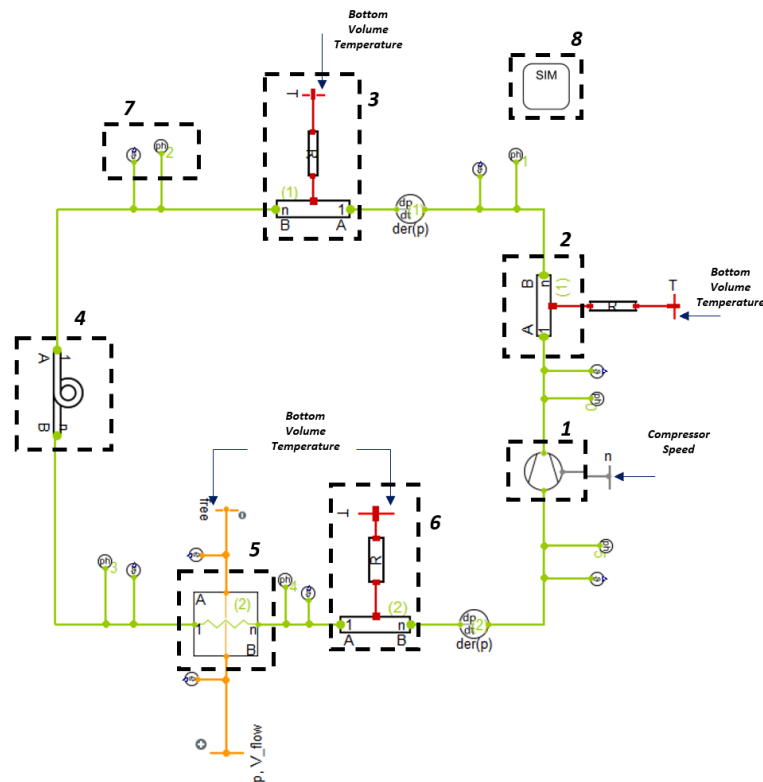


Figure 5.27. The dishwasher heat pump model.

Therefore, the whole model can be presented in Figure 5.28. Furthermore, it can be seen in the whole model that the information-bearing parts are combined with the relevant places in the relevant models. In other words, each model is related to the other.

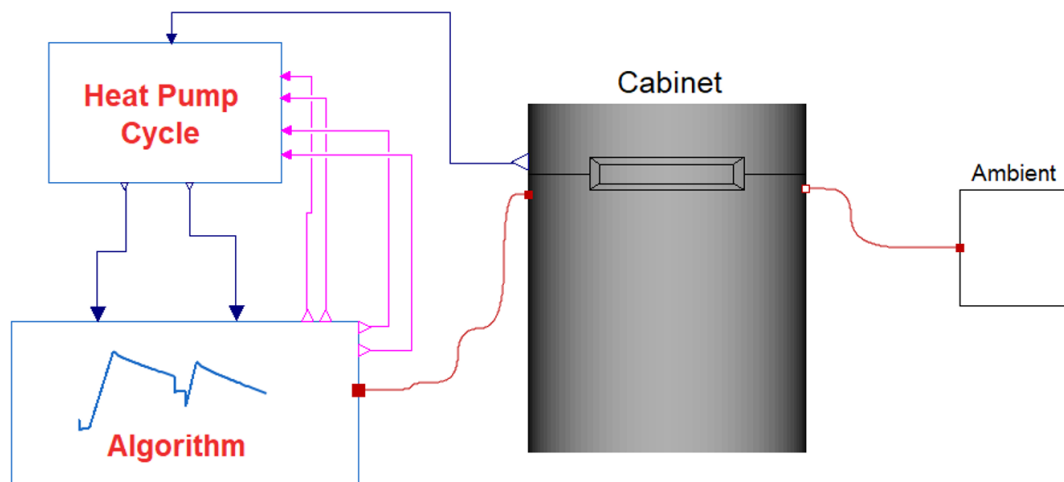


Figure 5.28. The whole dishwasher model in Dymola.

While the heat pump system works with the inputs it receives from the cabin, it transmits its outputs to the algorithm model. With heating the water masses in the algorithm model, the cabin also gets warm thanks to the heat ports connected to the cabin. The outputs are transferred to the heat pump model, allowing the system to stop at any algorithm step. In addition, with the ambient model, which includes the ambient temperature and the external convection coefficient, the heat transfer between the cabinet and the outside environment is simulated.

6. EXPERIMENTAL PROCEDURE

6.1. General Assessment

This study performs experiments on a heat pump integrated dishwasher by examining refrigerant mixture rate by increasing R290 amount in total mass fraction. The experiments are carried out in Arçelik Central R&D Laboratory. The laboratory conditions are prepared according to EN 60436 [49], the European dishwasher standard. The ambient conditions in the standard are listed in Table 6.1.

Table 6.1. Experiment test conditions [49].

Voltage	$230 \pm 1\% \text{ V}$
Frequency	$50 \pm 1\% \text{ Hz}$
Ambient Temperature	$23 \pm 2 \text{ }^{\circ}\text{C}$
Relative Humidity	$55 \pm 5\% \text{ RH}$
Water Feed Temperature	$15 \pm 2 \text{ }^{\circ}\text{C}$
Water Pressure	$240 \pm 20 \text{ kPa}$
Water Hardness	$2.5 \pm 0.5 \text{ mmol/l}$

Besides, since the heat pump system has an evaporator in contact with the ambient, the heat pump system is quite sensitive to environmental conditions. For this reason, the environmental conditions of the experiment are kept under control throughout the experiment to minimize oscillations, especially in the ambient temperature and humidity.

Besides, a standard wood cabinet is worn on the dishwasher to simulate kitchen conditions. A rectangular opening at the back of the cabinet where the air passes over the evaporator is evacuated, and the system water intake and discharge pipes are output.

6.2. Experimental Setup

The heat pump dishwasher is prepared by placing thermocouples in the heat pump and at specific points of the dishwasher to understand the effects of parametric changes on system behavior. In addition to thermocouples, two pressure transducers are attached to the high and low-pressure lines to measure condenser and evaporator pressure, respectively.

The list of thermocouples is shown below:

- Compressor outlet
- Condenser inlet
- Condenser outlet
- Evaporator inlet
- Evaporator 1st pas
- Evaporator 2nd pas
- Evaporator 3rd pas
- Evaporator 4th pas
- Evaporator 5th pas
- Evaporator 6th pas
- Evaporator outlet
- Compressor inlet

Thermocouples position on dishwasher system is listed below:

- Sump
- Basket
- Condenser water inlet
- Condenser water outlet
- Evaporator air suction
- Evaporator air blowing

The thermocouples and transducers' locations are illustrated schematically in Figure 6.1.

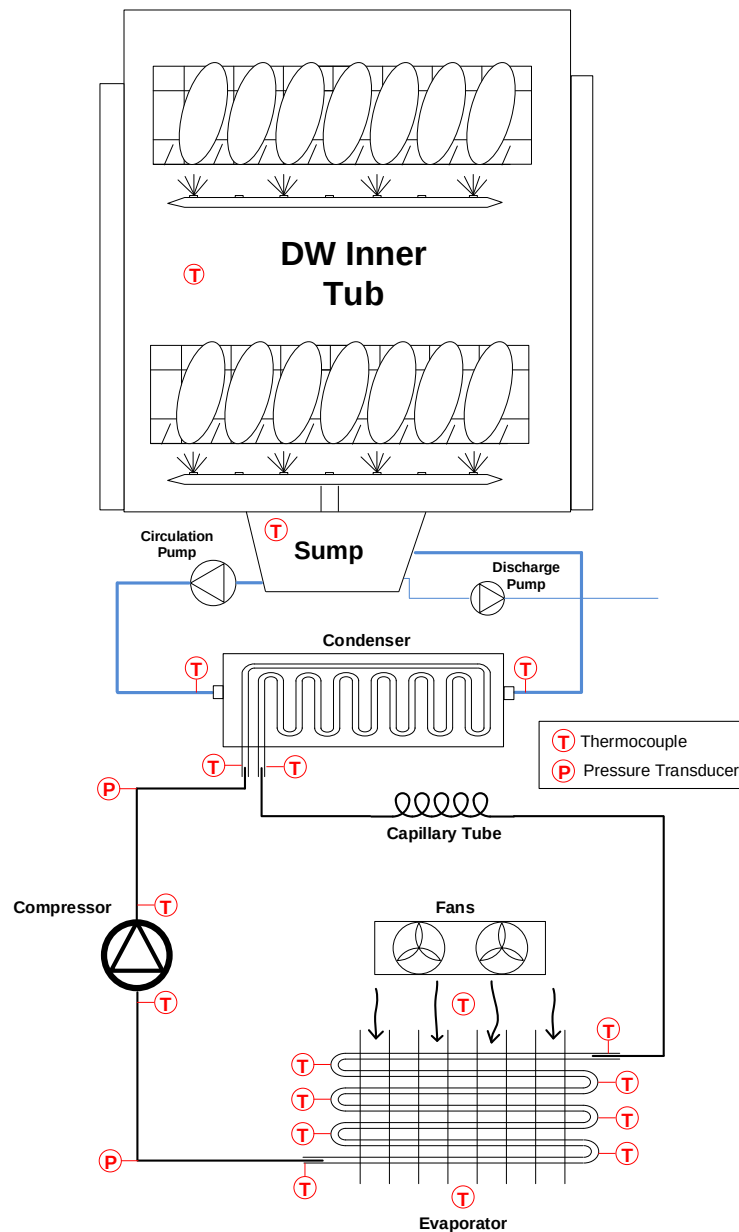


Figure 6.1. The experimental setup demonstration dishwasher with heat pump.

Many thermocouples are used in the test machine to monitor the system and ensure that the system works under optimal conditions. A prime example is that the refrigerant phase must be superheated at the evaporator outlet to prevent liquid from overflowing into the compressor. If an overflow occurs, the compressor might be damaged, and its lifetime can be reduced remarkably.

The refrigerant phase can be inferred from the thermocouple data, and overflowing can be avoided by taking necessary actions, such as amending the refrigerant charge amount.

Agilent 34970A [50] data acquisition system is used for gathering temperature and pressure data. Besides, through the multi-function metering system, Power Logic ION6200 [51], voltage (V), current (A), power (W), and energy (Wh) are measured. All the data collected during the experiment from the data acquisition system and multi-meter are saved in a computer.

6.3. Uncertainty Analysis

Experimental results' accuracy is affected by measurement devices and experimental setup errors. Several methods have been proposed to determine the error values of the parameters calculated using data from the experiments. One of these methods is uncertainty analysis, developed by Kline and McClintock [52]. In this method, the value to be calculated is R , and n independent variables corresponding to this value are $x_1, x_2, x_3 \dots x_n$.

The error rates of the independent variables are denoted as $w_1, w_2, w_3 \dots w_n$, and the total error rate of the measurand in the experimental study is W_R . In the uncertainty analysis, the total error of the measured quantity is

$$W_R = \sqrt{\left(\frac{\partial R}{\partial x_1} w_1\right)^2 + \left(\frac{\partial R}{\partial x_2} w_2\right)^2 + \dots + \left(\frac{\partial R}{\partial x_n} w_n\right)^2}. \quad (6.1)$$

The uncertainty values of sensors are listed in Table 6.2.

Table 6.2. Measurement sensors uncertainty values.

Sensor	Uncertainty
Temperature	$\pm 0.5 \text{ } ^\circ\text{C}$
Voltage	$\pm 0.25\%$
Current	$\pm 0.25\%$
Power	$\pm 0.5\%$

Depending on the algorithm, several components are activated, deactivated, or worked synchronously in the dishwasher cycle. For this reason, the power value changes

continuously in a cycle. Therefore, the elapsed time, depending on data retrieval frequency, is multiplied by the power value to calculate the dishwasher's energy consumption. The energy value calculated is summed up cumulatively. It can be written as

$$E = (Time)(P), \quad (6.2)$$

where E is the energy in Wh and P is the power in W.

As for the calculation of uncertainty in the energy value, it can be expressed as

$$W_E = (Time)(w_1). \quad (6.3)$$

Looking at the equation, W_E and w_1 are the uncertainty in energy consumption and power, respectively. The percentage uncertainty value of power is listed in Table 6.2. The following relation shows that the value of uncertainty for power is equal to the value of uncertainty of energy which is 0.75%,

$$\frac{E}{W_E} = \frac{P}{w_1}. \quad (6.4)$$

6.4. Experimental Method

Thermocouples and pressure transducers are attached to the dishwasher and heat pump system. The dishwasher inside the cabinet is placed in a climate chamber where the temperature and relative humidity are controlled strictly. The data logger is turned on to monitor the temperature and pressure inside the dishwasher. In the climate chamber, the dishwasher waits until all thermocouples on the systems reach the same value as the tested chamber's arranged temperature value. When the dishwasher reaches equilibrium, the test is started. At the end of the test, the data logger is turned off, and data are processed in Excel software. Figure 6.3 shows the dishwasher placed on the platform inside the climate chamber.



Figure 6.2. The dishwasher in the climate chamber.

On the other hand, the primary purpose of this thesis study is to reduce energy consumption and cycle time by optimizing heat pump systems with different propane and isobutane mixture compositions used as the refrigerant. Firstly, a heat pump dishwasher was tested using only isobutane as the refrigerant. Following this, the R290 amount in the mass fraction is increased by %10 in each test up to 50%. Then, all mixing ratios are compared by evaluating system performance outputs such as energy consumption, cycle time, and heating capacity. Namely, the system behavior is examined each time the refrigerant mixing ratio is changed. The experimental phases are listed in Table 6.3.

Table 6.3. Experimental phases.

Experimental Study	
100% R600a (Base Exp.)	Energy Consumption & Cycle Time Comparison
90% R600a / 10% R290	
80% R600a / 20% R290	
70% R600a / 30% R290	
60% R600a / 40% R290	
50% R600a / 50% R290	

Furthermore, the details of the dishwasher with heat pump on which base experiments are conducted are shown in Table 6.4.

Table 6.4. Details of the dishwasher with heat pump.

Dishwasher with Heat Pump	
Dimensions (H x W x D)	818 x 598 x 550 mm (standard)
Place Settings	16 pcs
Water Consumption	9.9 L
Energy Consumption	470 Wh
Program Duration	190 min.
Compressor	Reciprocating
Condenser	Stainless Steel Helical Tube Condenser
Evaporator	Fin and Tube Evaporator
Refrigerant	R600a

7. RESULTS AND DISCUSSION

Experiments were performed on a heat pump system integrated into a dishwasher, whose specifications are presented in Chapter 4, according to EN 60436 [49], the European dishwasher standard. The following processes were completed using the experimental methodology and modeling approach mentioned in the previous chapters:

- i) Refrigerant charge amount optimization with fixed heat pump components (base experiment with 100% R600a)
- ii) R600a and R290 refrigerant mixture experiments with increasing ratios of R290 in mass:
 - 90% R600a 10% R290 mixture
 - 80% R600a 20% R290 mixture
 - 70% R600a 30% R290 mixture
 - 60% R600a 40% R290 mixture
 - 50% R600a 50% R290 mixture
- iii) Model validation with the experiment:
 - Model validation of optimized system
 - Model validation of the system with a refrigerant mixture

7.1. System Refrigerant Charge Amount Optimization

Before proceeding to the refrigerant mixture tests, it should be ensured that the system works at the optimum point. Thus, the amount of refrigerant in the heat pump system was optimized. Furthermore, the amount of refrigerant was optimized by considering the refrigerant phase at the inlet of the compressor. Coming of a two-phase refrigerant mixture into the cylinder, reed valve, piston connecting rods, and compressor crankshaft can result in a critical defect. It may even cause a complete failure of the compressor, or there may be a significant reduction in its expected life.

In detail, whenever the amount of refrigerant is increased in a heat pump system where a constant fan speed is used and the secondary fluid (air) parameters are stable, the refrigerant mass flow rate in the system also increases. Thus, the amount of superheat may decrease. In other words, the desired heat transfer between the secondary fluid and the refrigerant cannot be realized. Therefore, complete phase change does not occur in the evaporator, and a two-phase flow is encountered at the compressor inlet.

On the other hand, if the amount of charge in the system is reduced, the mass flow rate of refrigerant declines. Accordingly, the expected heat transfer occurs at a point far behind the outlet of the evaporator. In such a system, the evaporation temperature tends to decrease, and the refrigerant quickly turns into gas. Therefore, the benefits of the heat of transformation cannot be fully realized, which brings about inefficiency in system performance. Also, it is known that when the quantity of superheat grows, the density of the gas phase refrigerant lowers, and the mass flow rate in the system correspondingly falls depending on the low-density value of the refrigerant vapor phase. Thus, the evaporator does not provide sufficient capacity and the system's COP deteriorates. At this point, the optimization process is in question.

Temperature measurements were made with thermocouples from each tube (pass) of the fin and tube evaporator used in the system. In addition, high- and low-pressure values on the system were determined using pressure transducers. Finally, the evaporation temperature was resolved with the pressure values taken from the evaporator, low-pressure line. Afterward, the relationship between the temperatures measured on the evaporator and the evaporation temperature was examined to verify that the phase change of the refrigerant was complete. The evaporator behavior of the optimized system is illustrated in Figure 7.1.

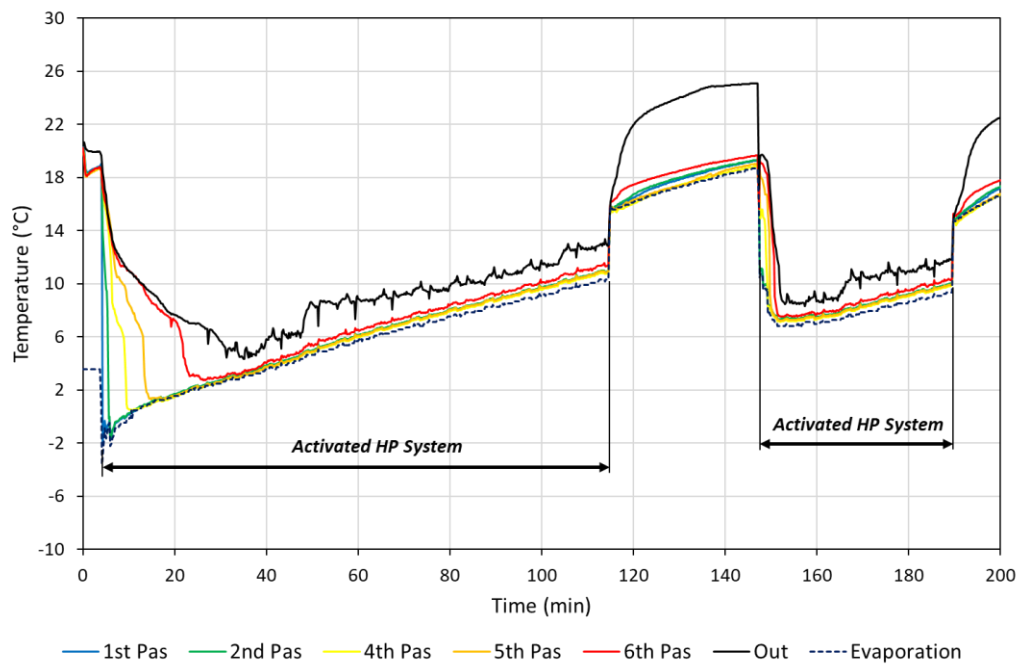


Figure 7.1. Evaporator behavior of the optimized system with 100% R600a.

In Figure 7.1, the number of tubes for temperature measurement increased from the evaporator's inlet (1st pass) to the outlet (out). It can be seen from the curves that the refrigerant temperature in the first six tubes did not differ from the evaporation temperature. Up to the sixth tube, the refrigerant was in a two-phase state. However, the outlet temperature value, shown with the black line, was higher than the evaporation temperature under all conditions in which the heat pump system operates. As a result, the refrigerant completed its phase change at the evaporator outlet, leaving it as superheat. Thus, superheat amount at the inlet of the compressor was kept at a relatively minimum value, and a complete phase change was ensured. Namely, the superheat phase was assured at the evaporator outlet, the suction temperature was maintained at a minimum, and the refrigerant flow rate reduction was prevented.

7.2. Refrigerant Mixture Experiments

This thesis studied the impacts of various parameters based on a blend of R600a and R290 refrigerants to lower the energy consumption and cycle time of a household dishwasher system with an R600a-based heat pump. In the optimized system, the experiments depended on a single variable parameter: the mass fraction of R290 in the mixture up to 50 percent.

Table 7.1. Energy consumption and cycle time depend on the increased mixing ratio.

Refrigerant Mixture	Energy (Wh)	Cycle Time (min.)
100% R600a	470	190
90% 600a / 10% R290	485	170
80% R600a / 20% R290	475	162
70% R600a / 30% R290	470	150
60% R600a / 40% R290	460	140
50% R600a / 50% R290	472	136

Table 7.1 presents the energy consumption and cycle time values depending on the increasing mixture rate. The cycle times examined in the experiments were evaluated by considering the time at the end of the second heating step. Because after this point, there is no change depending on the heat pump's operating performance. In parallel with this, since there is no energy-consuming component after the second heating step.

The table shows the lowest cycle time was reached, with a cycle time of 136 minutes at a 50% mixing ratio compared to the base experiment in which the cycle time was 190 minutes. In addition, despite attaining a significant reduction in cycle time at this mixing ratio, the energy consumption value, which is 472Wh, was nearly identical to that of the base experiment (470Wh). On the other hand, the lowest energy consumption value was obtained as 460Wh at the mixing ratio of 60% to 40% of R600a and R290, respectively, in which the cycle time was 140 minutes.

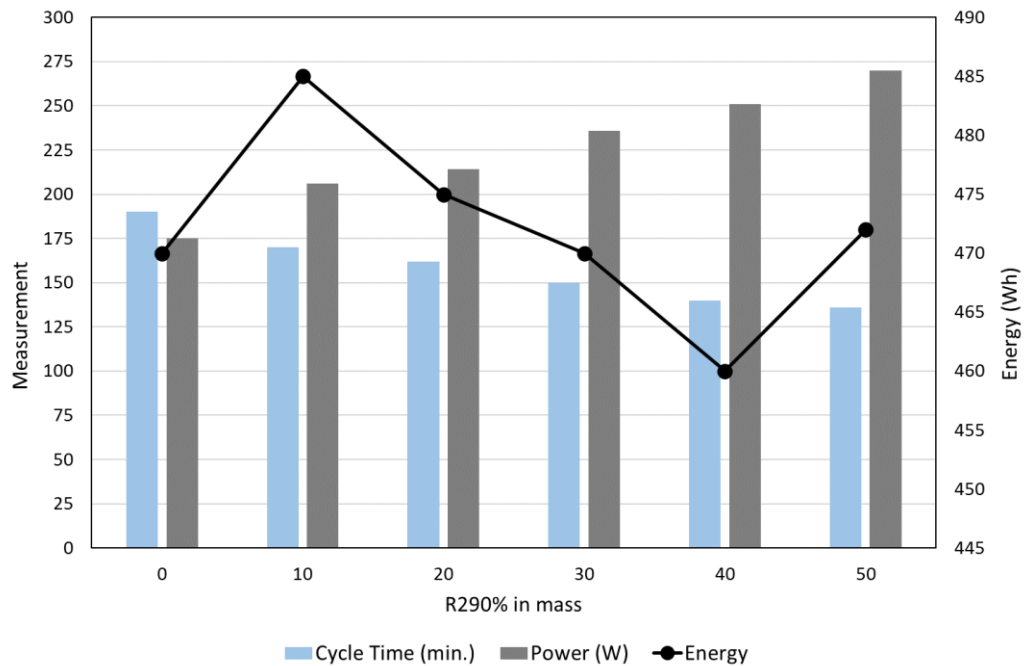


Figure 7.2. Evaporator behavior of the optimized system with 100% R600a.

Figure 7.2 illustrates data on the experiments in which the R290 fraction in total mass was gradually increased concerning the cycle time, power, and energy consumption. Overall, the power consumption governed by the compressor power had a general upward trend because of the higher vapor density of R290 compared to the R600a. In contrast, all cycle time figures saw a decline at a higher mass fraction of R290, depending on the increase in power consumption values. Besides, the rate of cycle time reduction was mainly ruled by heating capacity.

It can be regarded as while the energy consumption value increased in the test containing 10% R290 compared to the base experiment comprising 100% R600: it steadily declined until the test with a 40% R290 mixture. In other words, the minimum energy consumption value, 460Wh, was achieved in the experiment where the R290 ratio is 40% in total mass; even relative to the base experiment, this mixing ratio produced a 2% reduction in energy consumption. Moreover, at this point, the increased heating capacity depending on the rising mass flow rate due to higher R290 vapor density, allows faster and rapid heating to prevail over high compressor power. Nevertheless, the energy consumption increased again in the test in which R290 makes up half of the mixture, yet this mixing ratio offered the shortest cycle time with 136 minutes.

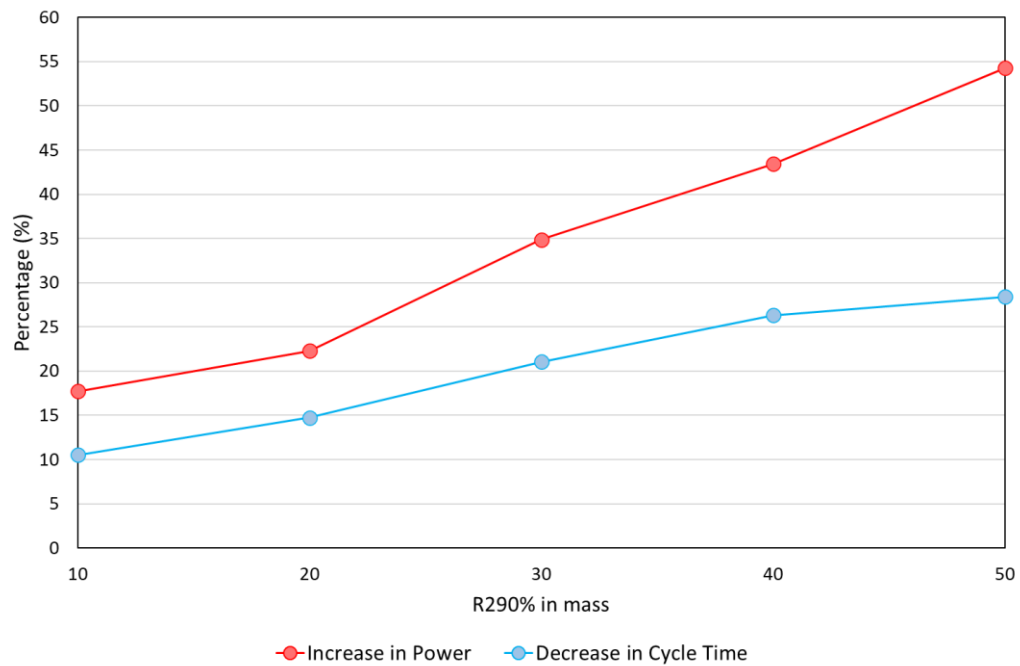


Figure 7.3. Comparison of the decrease in cycle time and the increase in compressor power.

As can be seen from Figure 7.3, with a 50/50 mixing ratio of R290 and R600a, the highest cycle time reduction of 28% was attained compared to the base experiment, and the power consumption value rose by 54.3% in the same case. As for the primary outcome, the rate of rise in compressor power had consistently outpaced the rate of decrease in operating time rate. In this regard, higher energy consumption is expected as the R290 mass ratio in the mixture increases in such a circumstance. However, it would be incorrect to directly correlate the rise in power with energy output, particularly with heat pump systems. To put a finer point on it, if the heating capacity can compensate for the increase in power consumption or if the system's coefficient of performance (COP) rises, energy consumption may be reduced.

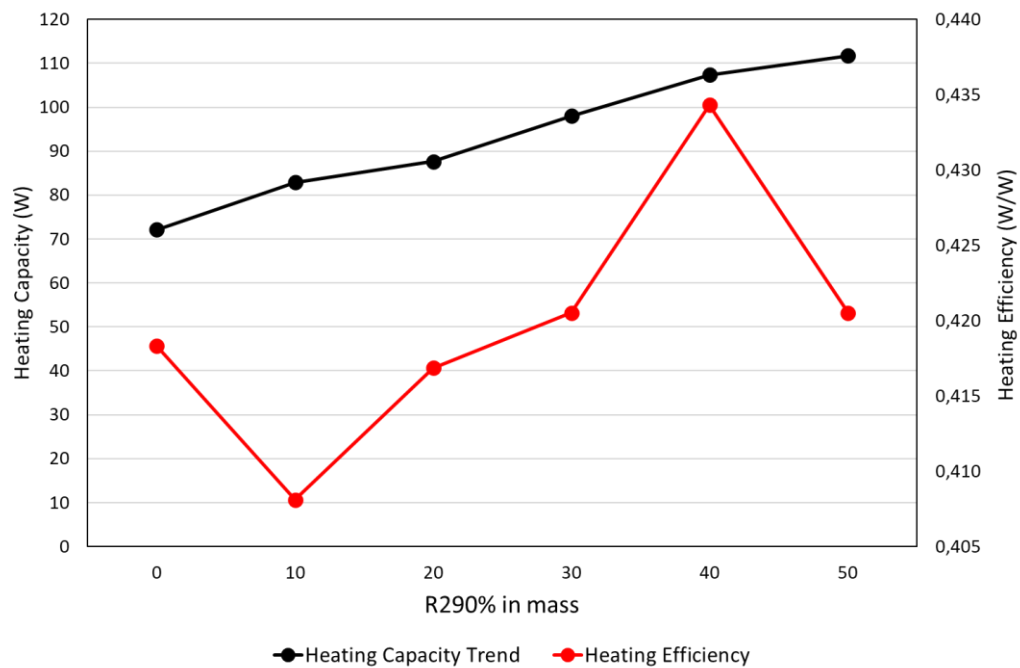


Figure 7.4. Heating capacity and efficiency comparison along the main wash step.

The line graph in Figure 7.4 demonstrates the heating capacity and corresponding heating efficiency in the main wash step of the whole cycle. The heating capacity figure was determined by calculating the heating energy required to heat 4 liters of water until the completion of the main washing step for each mixing ratio. In addition, the heating efficiency was calculated by dividing the average power consumed during the mean wash heating period by the heating capacity. It may be regarded as heating capacity tends to rise, while the heating efficiency had an extraordinarily similar but opposite trend to energy consumption as expected. Put differently, the decrease in heating efficiency affected the value of energy consumption adversely. Besides, increasing heating capacity may not necessarily improve efficiency due to the predominance of the variable compressor power. To illustrate, the heating efficiency in the 10% R290 mixture decreased significantly compared to the baseline test conducted with 100% R600a since the proportion of consumed power was considerably more than the increase in heating capacity. By contrast, the heating capacity grew sharply up to 40% R290 by mass and reached its maximum at 40%. It is reasonably pointed out that at this mixing ratio, the increase in heating capacity surpassed the rise in power consumption, resulting in a 10Wh reduction in energy usage compared to the base experiment. A decline in heating capacity was noted when R290 comprised half of the total mass, approximately identical to 30% of R290 in mass.

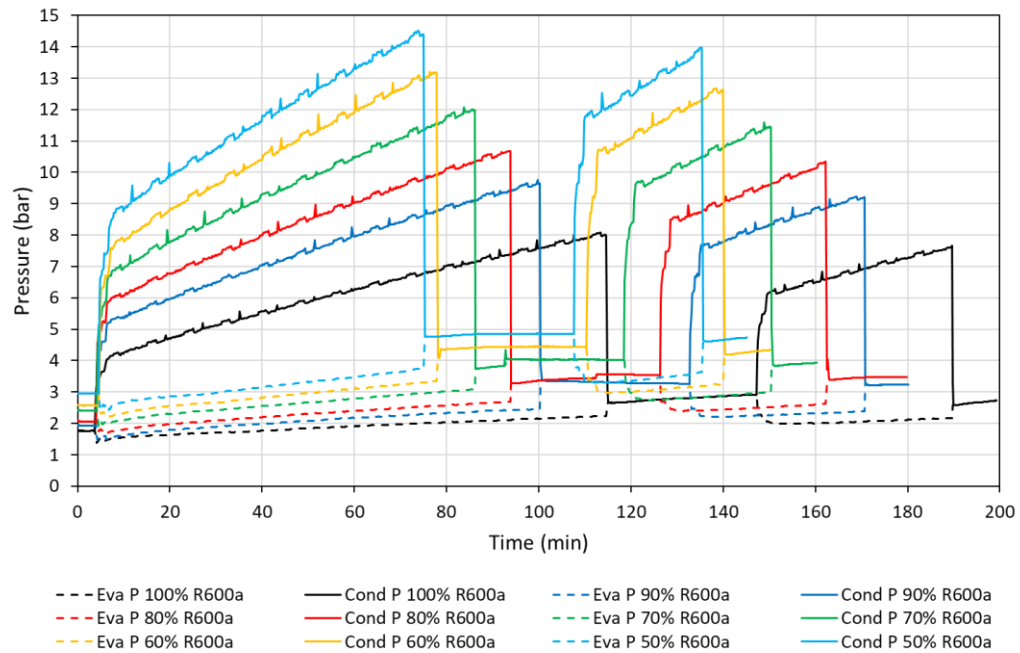


Figure 7.5. The variation of the high- and low-pressure levels.

Figure 7.5 demonstrates the shift in high- and low-pressure levels (condenser and evaporator pressure) during the cycle. In general, both pressure values tend to increase as the temperatures of the secondary fluids on both the condenser and the evaporator increase continuously during the heating steps. High- and low-pressure levels observed at every mixing ratio looked higher than for pure R600a. Besides, as the mass flow rate in the system increased with the percentage rise of R290, the pressure values in both pressure lines increased proportionally. In the base experiment, the average high-pressure level of R600a was around 6.2 bar, yet the average high-pressure level of the system containing 50% R290 by mass was roughly 11.5 bar. Likewise, at low-pressure levels, in the case of pure R600a, the average low-pressure level was measured to be 1.9 bar, but the low-pressure levels in the half-mixed condition were recorded to be 3.2 bar.

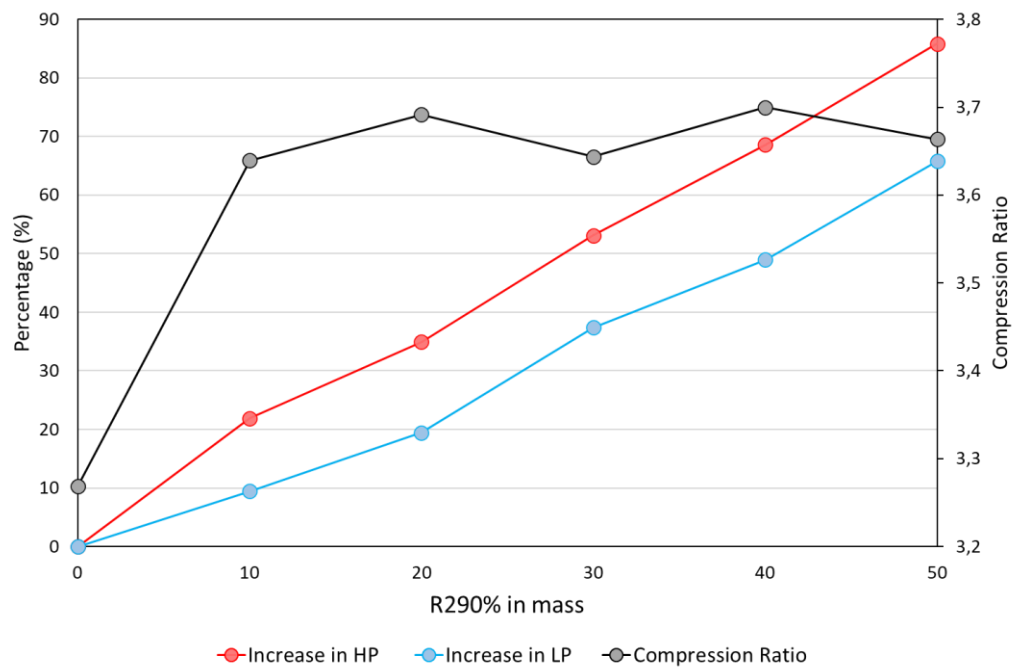


Figure 7.6. Percentage change in the low-high pressures and compression ratio.

Figure 7.6 illustrates the relationship between the growing pressure levels and the compressor's compression ratio. The line graph was created based on the average pressure values in the main washing step. It is evident from the graph that the rise in condenser pressure was always more than the increase in evaporation pressure for each mixing ratio. In these mixing configurations, the compression ratio remained nearly stable. In such a circumstance, when just the compression ratio is examined, an increase in power consumption may not be anticipated.

Nevertheless, as stated previously, since the rise in condenser pressure was considerably more than the increase in evaporation pressure, the pressure differential grew dramatically, resulting in a considerable increase in power consumption. Noticed power consumption can be interpreted differently; the rising pressure values at the compressor's inlet (evaporator pressure) may also have contributed to the power consumption because the typical return pressure is the primary dominant factor in the power value.

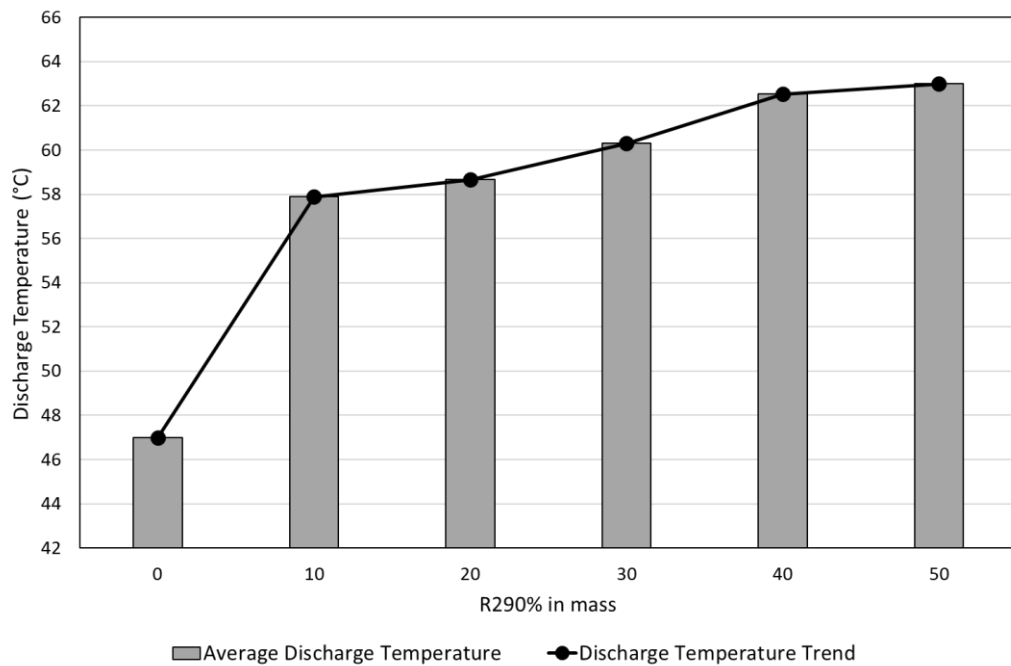


Figure 7.7. Average compressor discharge temperature.

The compressor's discharge temperature is crucial for stabilizing lubricants, compressor components, and heat pumps. If the compressor's discharge temperature is consistently too high, the viscosity of the lubricating oil drops and the compressor's performance declines, giving rise to considerable damage. The increasing compression ratio brings about an impairment in the compressor's isentropic efficiency. In parallel, it increases compressor discharge temperature and power consumption, as seen in this study. Figure 7.7 shows the compressor average discharge temperature shifting according to the increasing proportions of R290 in the mixture. As can be seen, following the compression ratio (see Figure 7.6), the discharge temperature went up dramatically in all experiments utilizing a refrigerant mixture compared to the base experiment using pure R600a. Besides, when the experiments conducted with the mixture are analyzed independently, an increase in the mass ratio of 290 results in a modest rise in discharge temperature. When the base experiment and the experiment employing 10% R290 were compared, the discharge temperature difference was 11 K, whereas this variation between the 10% and 50% R290 mixing states was 5 K on average.

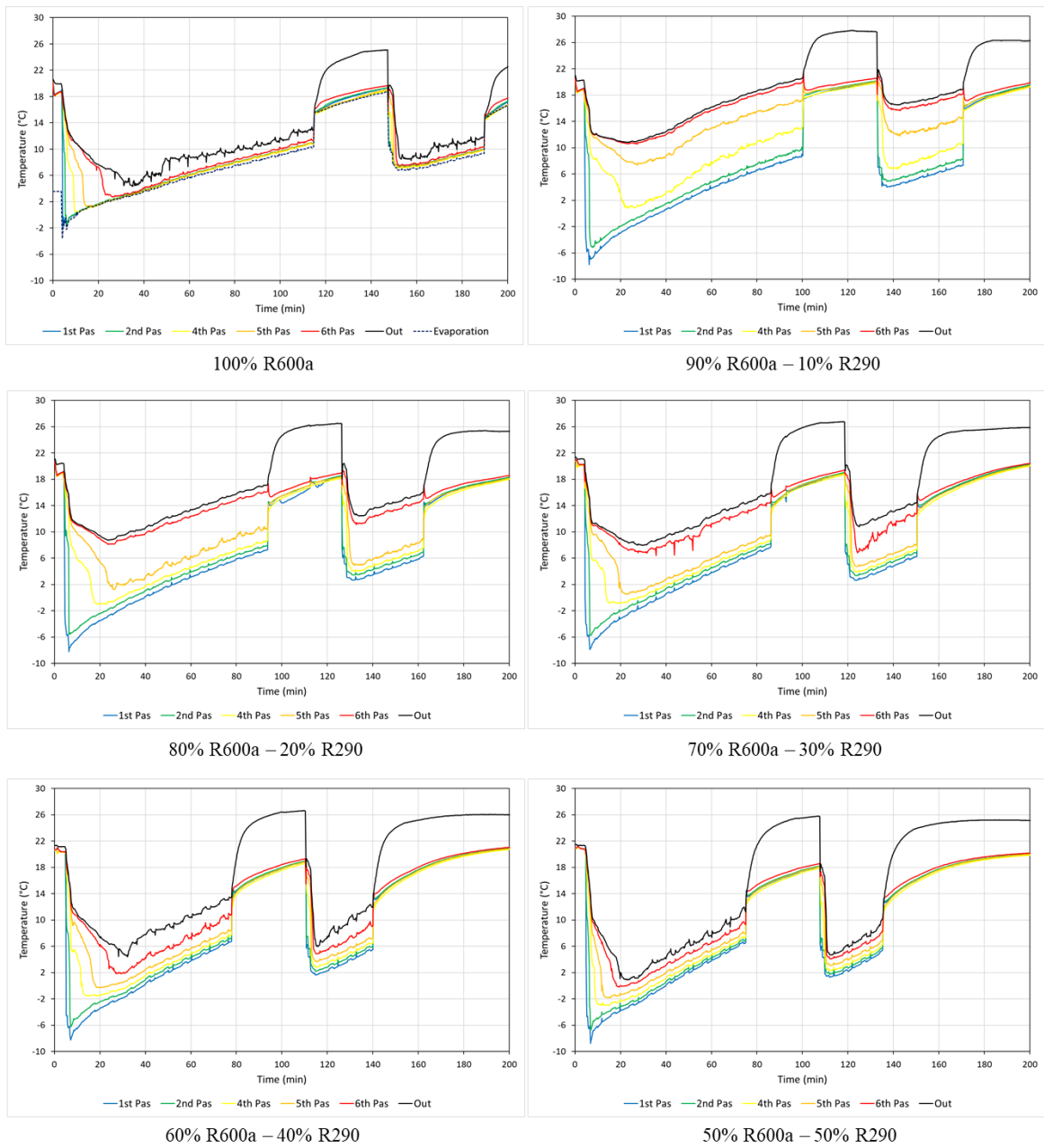


Figure 7.8. The evaporator temperature behavior of all experiments.

The fin and tube evaporator employed in the system has a 4x3 design, 12 pipes, and 6 pipe return terminals outside the fin structure. Temperature measurements were taken from each pipe return (pas) on the evaporator-like discretization. The measurements reveal the evaporator's behavior straightforwardly. The temperatures were measured from each pass of the base experiment (a), and each mixing ratio (b-f) is depicted in Figure 7.8.

The graphs show that the evaporator temperature trends were lower in the mixing experiments than in the base experiment. However, there was no discernible change in evaporator inlet temperatures as the mass fraction of R290 rose in the mixing cases. Therefore, it can be explained in a way that upon the introduction of R290 into the system (10% R290 in mass), a subsequent increase in the mass flow rate was observed due to the higher vapor density of R290, leading to an elevation in evaporating pressure (see Figure 7.5). However, given that R290 exhibits a considerably lower saturation temperature than R600a at an equivalent pressure, the evaporation temperature decreased under mixed conditions compared to that of pure R600a. Consequently, the temperature differential between the ambient and evaporation temperatures rose, resulting in a significant surge in the amount of superheat within the evaporator. Furthermore, in the mixed experiments, the temperature glide is visible; at these places, the temperatures of the two phases do not lie on a single line, unlike the base pure refrigerant experiment. As a zeotropic blend of R600a and R290, distinct saturation temperatures resulted in a wider range of evaporation process (temperature glide) which may have caused continuous variation in the refrigerant temperature in the evaporator and lead to more effective evaporation. This, in turn, caused a phase change (evaporation) to occur proximal to the evaporator inlet passes, after which the refrigerant continued to extract heat from the ambient, thus increasing its temperature gradually. Namely, the mean temperature difference between the refrigerant and the ambient diminished. As a result, the system efficiency is negatively affected due to the insufficient capacity obtained from the evaporator. Thus, despite the increase in the heating capacity associated with the rise in mass flow rate, the decrease in heating efficiency due to the inability to overcome the increase in power consumption resulted in a noticeable increase in energy consumption value. Nonetheless, this scenario produced a latent potential within the evaporator.

With the progressive amplification of the R290 ratio in the system, the mass flow rate and evaporator pressure experienced a gradual increment. While such a circumstance would typically result in an increment of the evaporation temperature, the lower saturation temperature of R290 compared to R600a ensured that the inlet temperatures of the evaporator remained consistent. Consequently, the increased R290 ratio facilitated more efficient utilization of the evaporator by reducing the amount of superheat rather than modifying the evaporator inlet temperature.

In other words, as the mass flow rate in the system increased, the superheat level in the evaporator reduced. In the case of decreasing superheat, the fluid in the evaporator changes phase towards the outlet passes of the evaporator, and greater utilization can be achieved from the heat of transformation. In other words, the mean temperature difference between refrigerant and ambient did not reduce rapidly as in the mixture of 90% R600a / 10% R290.

The increasing heating capacity dominates against the increasing power consumption from 20% to 40% R290 mixing ratio. Therefore, the heating efficiency goes up, and the energy consumption decreases accordingly at this point. Considering the system outputs, the most efficient evaporator was produced with a mixture of 40% R290 and 60% R600a, where the least amount of superheat was obtained according to higher LMTD, and in this case, the enhanced heating capacity preceded the power consumption. Thus, this mixing ratio provided the lowest energy consumption. In this so-called optimum case, since the volumetric capacity of R290 is significantly greater than R600a, an evaporator with the same surface area may produce a higher capacity.

In the case of half-mixing, the flow rate continued to increase with enhancing R290 ratio. Due to the evaporator dimensions and the second refrigerant flow in a constant trend, 100% phase change could not be achieved, and the refrigerant in the evaporator could not be separated from the two-phase region. In such a case, the refrigerant enters the compressor as a high-density two-phase, increasing the power consumption. As the increased power consumption dominates the highest achieved heating capacity, the energy consumption figure climbed again.

In addition, Figures 7.9 and 7.10 display the behavior of the system in the p-h diagram for experiments using pure R600a and 60% R600a and 40% R290 mixture offering optimum outputs. The reference point for the cycle plots is the 60th minute of the cycle.

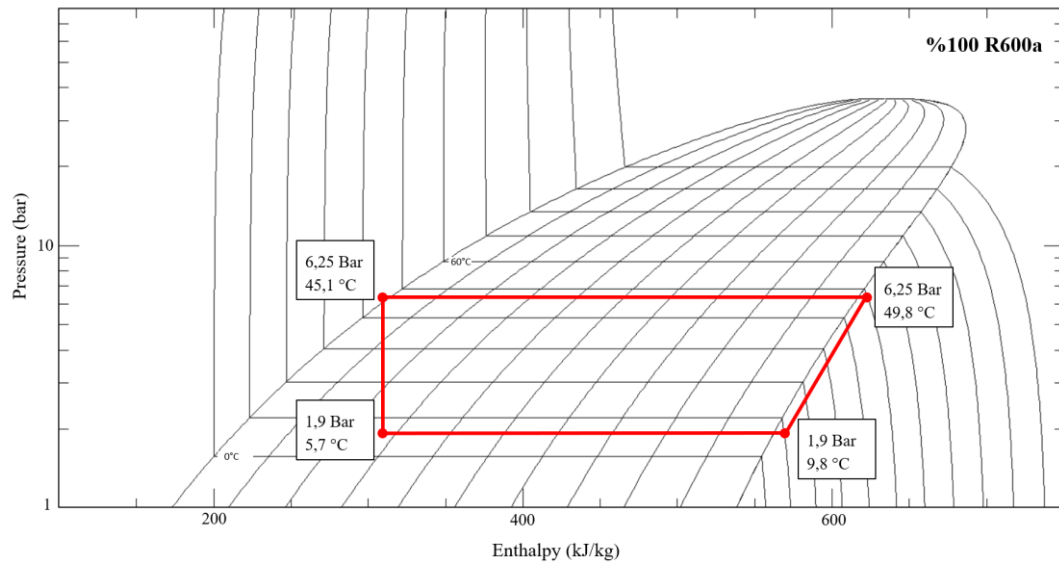


Figure 7.9. P-h diagram of 100% R600a experiment.

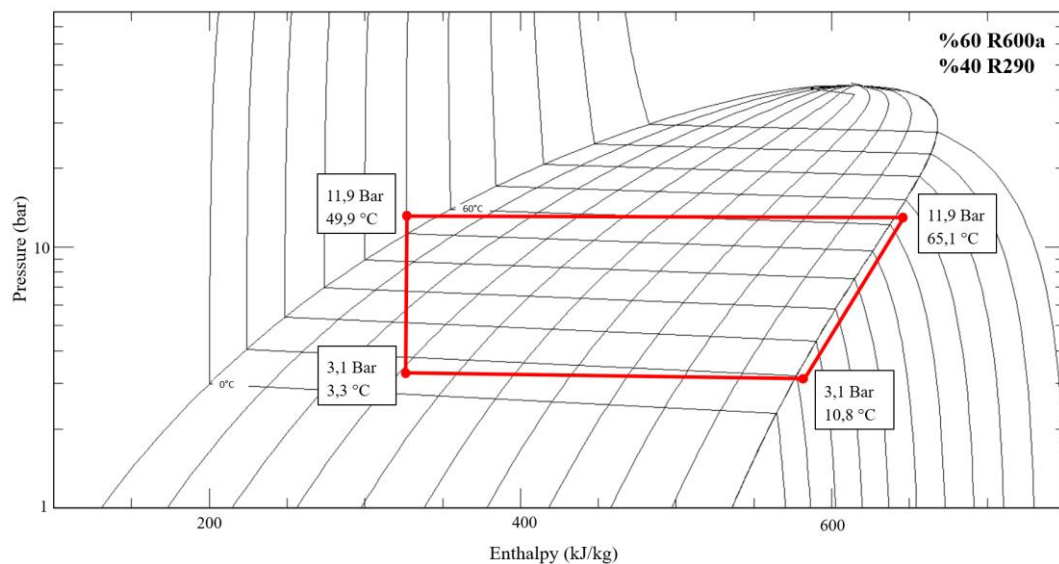


Figure 7.10. P-h diagram of 60% R600a / 40% R290 mixture experiment.

7.3. Modeling Validation of Optimized System

In the refrigerant-amount-optimized system, 100% R600a was employed, and the test findings were regarded as the baseline outputs. The thermal loss parameter to simulate heat losses in the course of water heated on the condenser surface to the cabinet and the "tube roughness" in the capillary tube to fit the system pressures were utilized as fine-tuning parameters. The fine-tuning processes of the model were mainly carried out by considering the cabinet temperature and heating time. Then, the base experiment's model and test results were compared considering the dishwashing cabinet's temperature, power consumption value, energy consumption, and low-high pressure values.

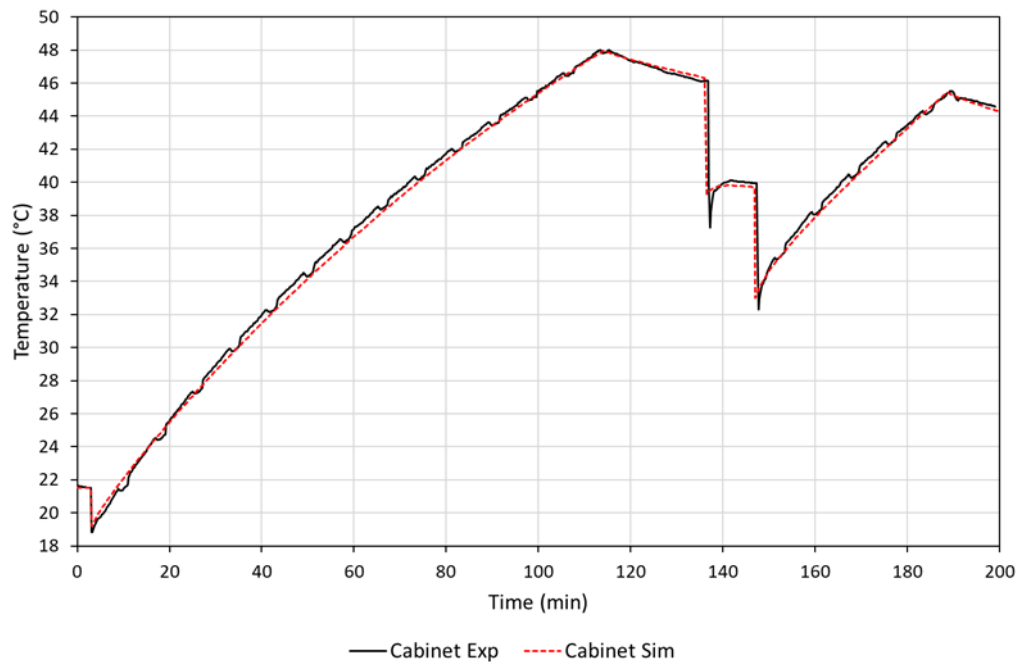


Figure 7.11. Cabinet temperature during the cycle – baseline test.

Figure 7.11 depicts the behavior of experimental and modeled cabinet temperatures. The maximum deviation between the experimental and simulation results is ± 0.5 °C. Furthermore, the difference in cycle time between the model and experiment is 0,5 minutes. The model's behavior in the baseline case aligns with the experiment's outcomes.

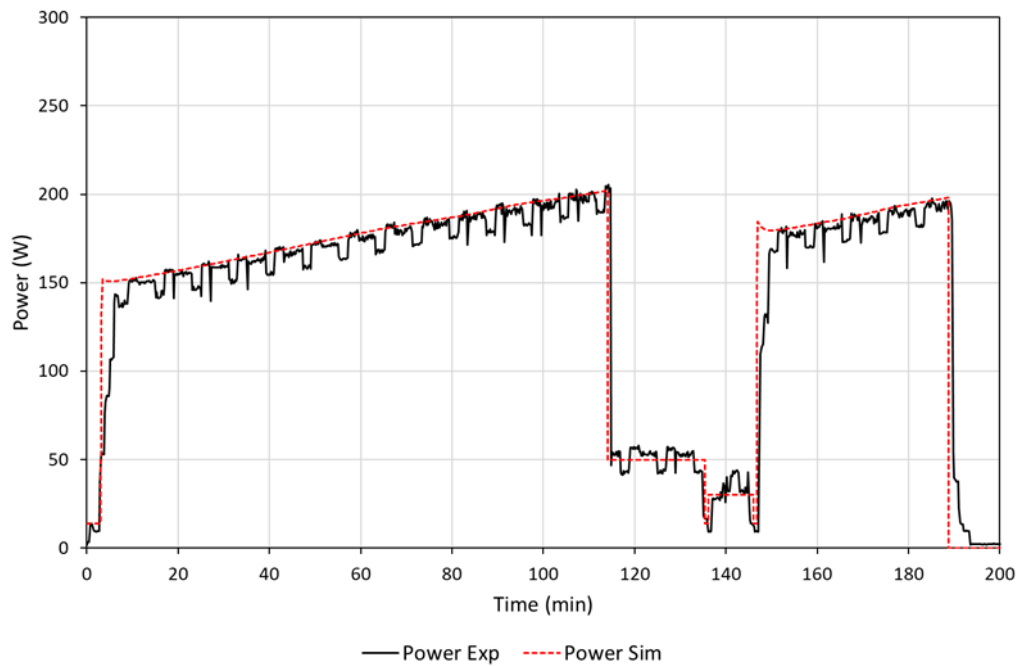


Figure 7.12. Power consumption along the cycle – baseline test.

In Figure 7.12, the time-dependent power consumption is illustrated during the cycle. Considering the power consumption value of the dishwasher, components such as the compressor, circulation pump, and drain pump contribute to the total power consumption. Regardless of the compressor's operational stages, the circulation pump operates for almost the whole cycle. However, the dominant component in power consumption is the compressor. The circulation pump is responsible for sucking and pressurizing the water and sending it back to the cabinet. However, it can operate at various speeds and in different spray arm positions (lower-middle-upper) depending on the algorithm to achieve the desired washing performance. Changing spray arm positions and driving the pump at different speeds frequently change during the cycle. Therefore, it consumes different power at every moment as the pressure in the water line is different for each variable situation. The oscillation in the power consumption value, which is seen in the graph, can be explained in this way. Since the pump power consumption is defined as the average in the model, this oscillation is not encountered.

For the above reasons, the deviation for the average power consumption during the cycle time is 2,4%. However, when the figure is examined, it is seen that the model consumes more power than the experiment when the compressor starts to work.

The discrepancy is because, in the actual case, it takes some time for the refrigerant to reach the evaporation temperature with each pass of the evaporator (see Figure 7.1), while in the model, this evaporation temperature is reached at each point simultaneously. In other words, it takes some time for the refrigerant to fill the evaporator fully, although, in the model, this filling occurs much more rapidly. Another reason is that the compressor does not directly reach the determined operating speed when it starts to work due to system requirements. Instead, it gradually increases to the determined speed value to guarantee the lubrication of the components inside the compressor. Since these chaotic lubrication cycles are not reflected in the model, it is customary to see a noticeable difference in power consumption values at the point where the compressor starts to work. The same trend can be visible in pressure lines (see Figure 7.14). For these reasons, the compressor power in the experiment is greater during the initial operation and gradually lowers as the experiment progresses. Despite these differences, the overall trend is consistent with the model and the experiment. This deviation in the power consumption value is reflected as a 2,9 % difference in the overall energy consumption value.

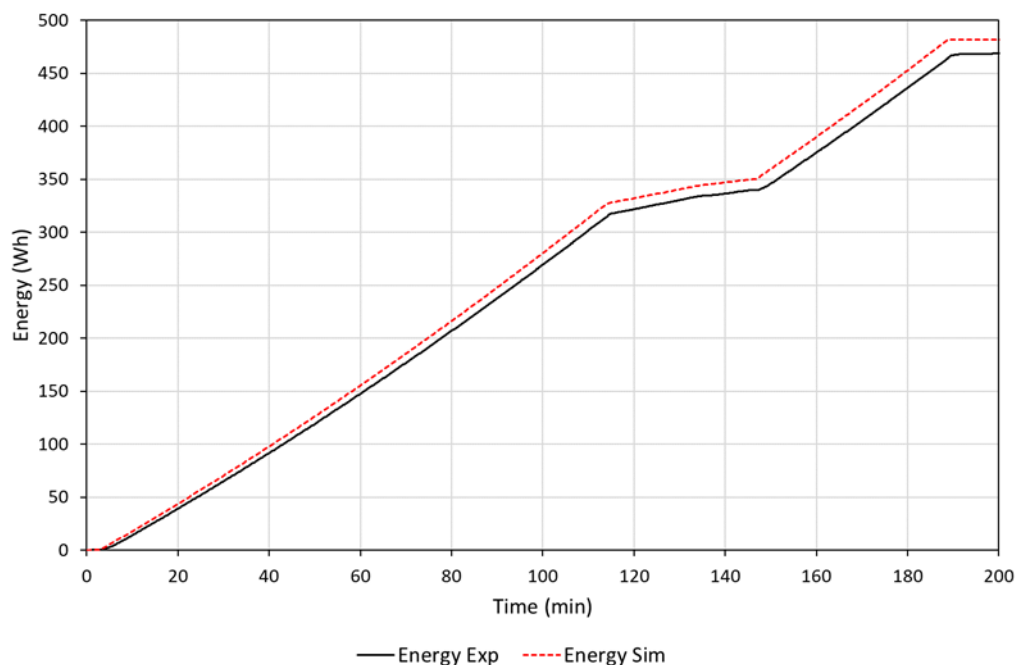


Figure 7.13. Energy consumption along the cycle – baseline test.

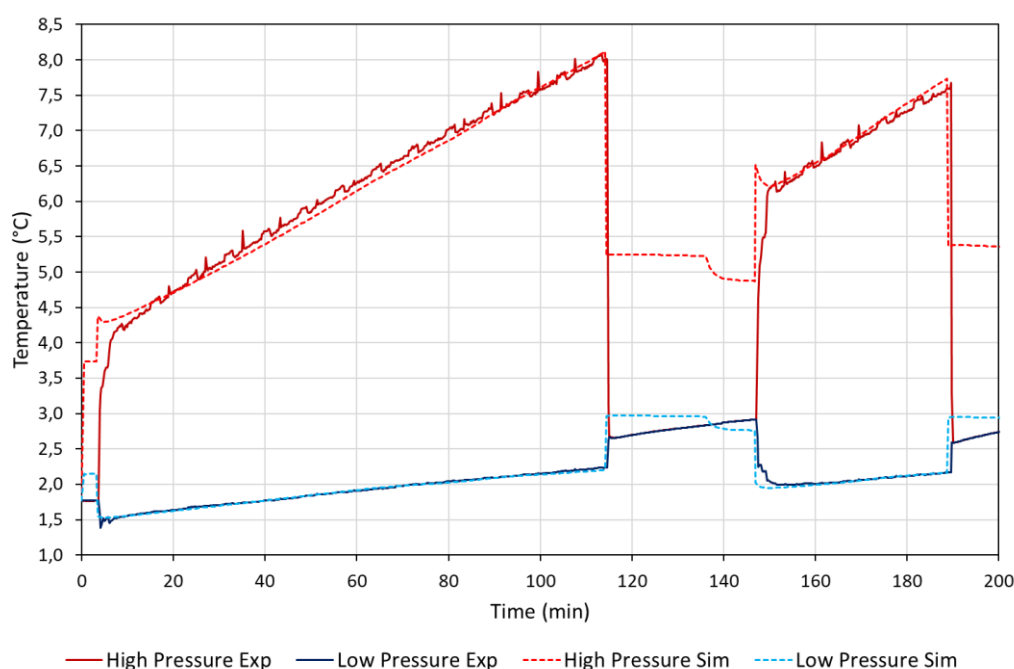


Figure 7.14. High and low-pressure change during the cycle – baseline test.

Figure 7.14 demonstrates high and low-pressure values over a cycle. The proximity of the actual and model pressures during compressor operation demonstrates that the model can accurately predict the pressure drop along the capillary tube and compressor performance. The maximum inconsistency is observed in the condenser (high) pressure line, which is almost 0,23 bar corresponding to nearly 1°C for condensation temperature in the case of R600a. The deviation on the low-pressure side is 0,1 bar. It can be explained that, as described in Chapter 5, the physical parameters of the compressor model are derived from calorimetric results using statistical analysis to create the compressor model. Owing to the limits of the calorimeter equipment, unlike the actual operating circumstances, the superheat value is maintained at 25 °C, and evaporation condensation values are entered into the system as input to account for the working conditions. However, condensation, evaporation, and superheat levels vary according to operating circumstances. Although the evaporation pressure is fine-tuned with the "tube roughness" parameter in the capillary tube model, some discrepancies in the condensation pressure can be detected with the experiment due to the factors above because the compression ratio of the compressor changes. Therefore, it is only logical to notice corresponding variations in power consumption. Despite this unpredictability, the model fits the data well and allows parametric analysis.

Also note that in circumstances where the compressor is not running, numerical errors can explain why the high-pressure result deviates significantly from the experiment as the compressor switches from on to off.

7.4. Model Validation of System with Refrigerant Mixture

After the validation of the base experiment, the experiment and model validation of the 60% R600a 40% R290 combination, which emerged as the optimal mixing ratio based on the experimental outputs, were carried out. The primary goal of picking this experiment is that it gives a considerable cycle time reduction, 50 minutes, compared to the base experiment, and offers the lowest energy consumption among all experiments. In addition, although the cycle time decline (54min) is more significant in the experiment in which R290 is used to generate half of the mixed mass, the energy consumption value does not far from the base experiment. Additionally, the improvement in cycle time gained by mixing 60% R600a with 40% R290 is not notable for such long cycle times. Figures 7.15-7.18 show the cabinet temperature, power consumption, energy consumption, and the heat pump system pressures successively.

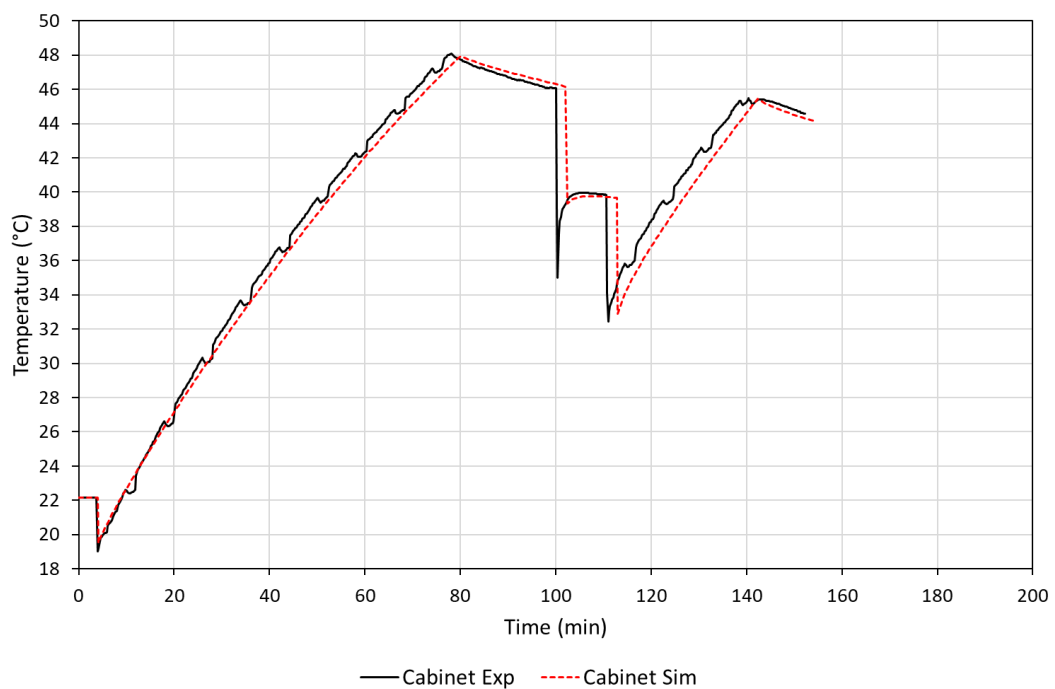


Figure 7.15. Cabinet temperature during the cycle – 60%R600a / 40%R290 mixture.

The cabinet temperature trend is quite close to the proposed mixture experiment with a maximum deviation of 1,5 °C, yet it may be observed that the deviation is more than the validation of the base experiment. The deviation arises because the masses in the model are considered a "lumped" mass and may not have fully reacted to the fine-tuned thermal loss parameter in the base experiment. Moreover, the variance in cycle time is around 2 minutes. Therefore, a heating capacity consistent with the experiment could not be attained to the same extent as in the validation of the base experiment. Nonetheless, even in these instances, the temperature trend in the cabin is pretty consistent with the experiment.

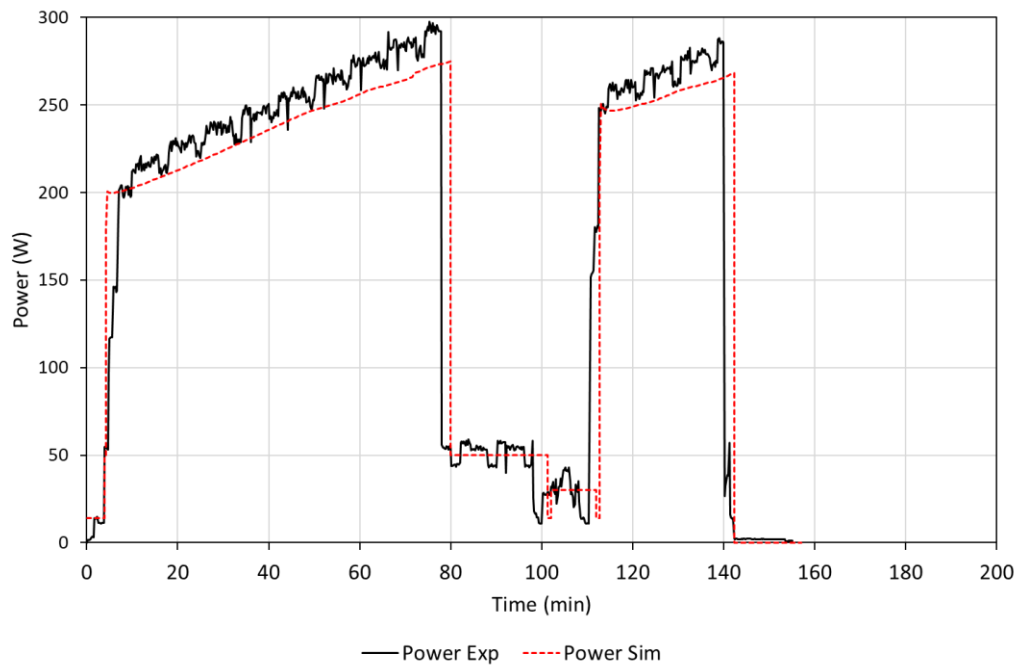


Figure 7.16. Power consumption along the cycle – 60% R600a / 40% R290 mixture.

The compressor power consumption is seen in Figure 7.16. The model somewhat overestimates the average value by 5,2%. As previously indicated, the difference in the starting point, in which the power consumption increases rapidly, is related to the lubricating speeds and the constraint of the refrigerant to fill the evaporator at the beginning of the cycle completely. This circumstance is repeated in pressure graphs (see Figure 7.16).

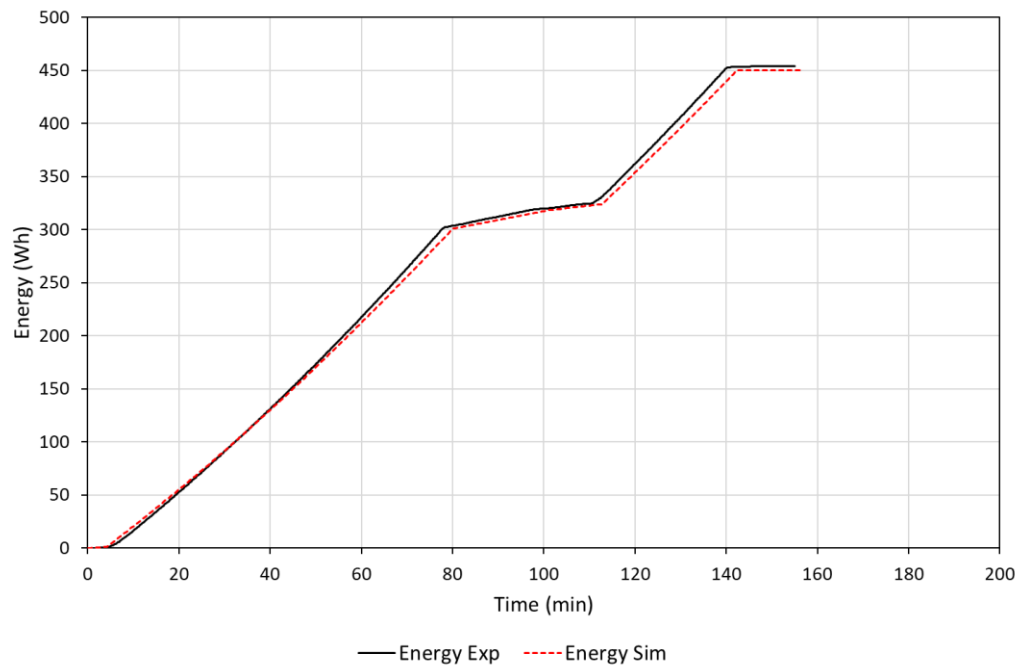


Figure 7.17. Energy consumption along the cycle – 60% R600a / 40% R290 mixture.

The reflection of the trend in the power curve to the energy consumption can be seen in Figure 7.17. Although the average value of power consumption (2.4%) in the validation of the base experiment was less than that of the mixed state, the mixing experiment demonstrated a more consistent result in terms of energy consumption value than the validation of the base experiment, with a deviation of approximately 2%. The confusing oscillation can explain this disparity in power consumption from the circulation pump and the dispersion in energy power consumption.

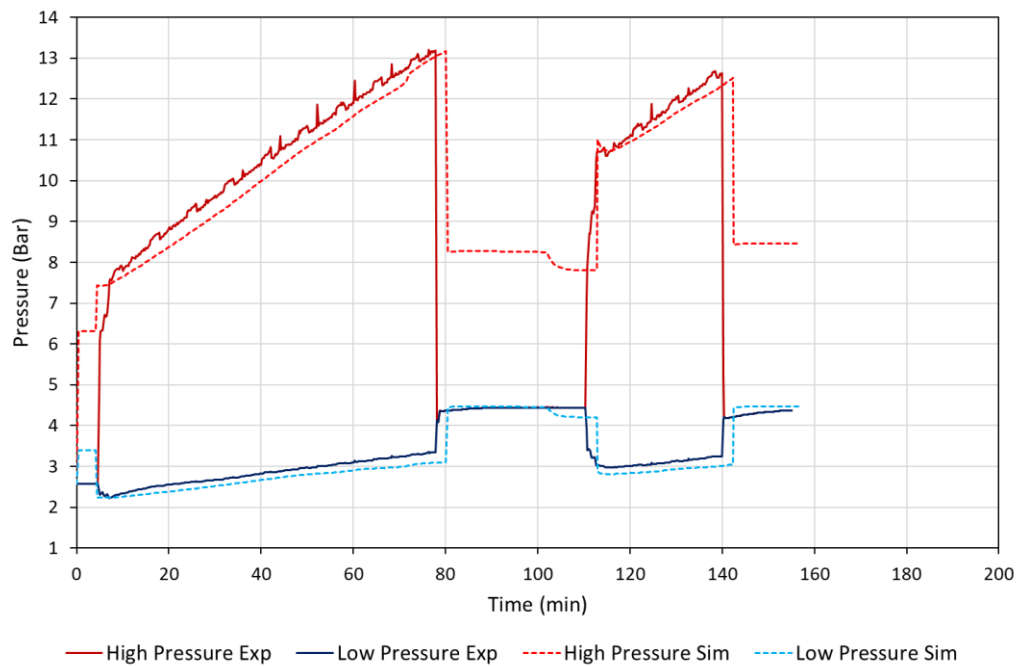


Figure 7.18. High and low-pressure change – 60% R600a / 40% R290 mixture.

Figure 7.18 illustrates the model's validation with the system experiment containing 40% R290. The maximum deviation in the condenser (high) pressure was almost 0,5 bar corresponding to a 2,3°C difference in condensation temperature, while it is 2,5°C evaporation temperature with a maximum 0,25 bar deviation in evaporator (low) pressure considering the mixture rate of 60% / 40% R600a and R290 respectively. These deviations in condensation and evaporation pressure clarify the power inconsistency because compressor power depends on evaporation and condensation pressures.

On the other hand, it would be significantly more accurate to describe the difference between the experiment and the model on the compressor model. Since the calorimetric measurements were utilized in the compressor model generation, steps described in Chapter 5 were conducted with R600a at a concentration of 100% percent. As anticipated, the reported physical compressor characteristics demonstrated more consistency at 100% R600a circumstances. However, notable variances were seen in mixed states.

In other words, the flow rate and power consumption values produced by the compressor model in response to a certain evaporation and condensation pressure are considerably more in line with the experiment in the base experiment model using 100% R600a compared to the system employing fluid mixture. Contrary to expectation, the model demonstrated a method for mixing refrigerant conditions with responses not significantly far from the experiment.

Table 7.2 shows the test and model verification outputs of the systems in which the refrigerant is pure R600a, and the R600a / R290 mixture is evaluated over the aforementioned different parameters and shown holistically.

Table 7.2. Deviation amounts of modeling results.

Variable	100% R600a	60% R600a / 40% R290
Cabinet Temperature	$\pm 0.5\text{ }^{\circ}\text{C}$	$\pm 1.5\text{ }^{\circ}\text{C}$
Cycle Time	0,5 min	2 min
Power	2,4%	5,2%
Energy	2,9%	2%
High Pressure	0,23 bar	0,5 bar
Low Pressure	0,1 bar	0,25 bar

This section used model validations for the pure R600a and R600a, and R290 in the specified mixture (60%/40%, respectively) experiments. In addition, potential causes for the inconsistencies between the experiment and the model are widely discussed. Finally, looking at Table 7.2, which presents the results from all modeling validations, the experiments and the model converged remarkably, but the model offered more accurate results when pure refrigerant was used in the system. Therefore, using the proposed model, the effect of parametric analyses such as heat exchanger sizing to be performed in heat pump systems integrated with a dishwasher using pure R600a or a mixture of R600a and R290 as refrigerant may be evaluated with minimal system-wide error rates.

8. CONCLUSION

In this thesis, the R600a/R290 refrigerant mixture was experimentally applied to a heat pump dishwasher, which was adapted to the dishwasher and used pure R600a. In experimental studies, energy consumption, cycle time, power consumption, heating capacity, heating efficiency, high- and low-pressure values, compressor discharge temperature, and evaporator behavior referencing the optimized base system were examined and discussed in detail depending on the increasing R290 ratio in the mass amount of refrigerant.

Moreover, this work modeled a dishwasher operated with a heat pump by constituting a sub-model for each component and then piecing together all the models to create a dishwasher with a heat pump. The modeling was performed utilizing the TIL library in Dymola, based on the computer language Modelica. Using the generated model, both the pure R600a base test and the 60% R600a / 40% R290 refrigerant mixture experiment, which yields the optimum results in terms of system performance characteristics, were independently validated based on the experimental findings. During the verification procedure, the following characteristics are evaluated for consistency: dishwasher interior temperature, power consumption, energy consumption, and low- and high-pressure values. For both validations, it can be said absolutely that the outcomes were quite similar.

The following are some significant insights from this study, which are discussed in Chapter 7 in detail:

- In general, using an R600a/R290 mixture instead of just pure R600a in such a system has considerably reduced the cycle time and energy consumption to some extent.
- When the mass of R290 in the refrigerant rose, the system's cycle time reduced, although the power consumption climbed steadily. The 50% / 50% mixing condition reduces cycle time by 54 minutes (28.4% compared to the baseline trial, whose cycle time is 190 minutes).

- A gradual increase in power consumption could not be observed in energy consumption, which differed for each refrigerant mixture because the heating capacity increased in response to the increased power consumption in the system.
- In the experiment, which lasted only 4 minutes longer than the experiment with the lowest cycle time, in which a mixture of 60% R600a and 40% R290 was applied, the increase in heating capacity defeated the power consumption and ensured optimum heating efficiency. In parallel with this, energy consumption decreased by 10 Wh compared to the base experiment.
- Considering the system pressures in the case of a refrigerant mixture, both high- and low-pressure values increased with increasing R290 ratio. In other words, the most significant difference in mean pressure rise was obtained between the base experiment and 50% R600a / 50% R290 mixture experiments. The average high-pressure value increased from 6.2 to 11.5 bar, while the low pressure was from 1.9 bar to 3.9 bar in these experiments.
- In the case of the first applied mixture ratio (90%R600a/10%R600a), the discharge temperature rises by as much as 11 K compared to the base test. Nevertheless, when R290 ratios increased, this abrupt upward trend became less pronounced, rising by 16 K in the 50% mixture condition relative to the base test. In addition, the discharge temperature changes are directly related to the compression ratio.
- Comparing the evaporator temperatures under all mixing settings, including the base test, reveals that 60% R600a/40% R290 produces the optimal evaporator behavior.
- Validation has been undertaken for the pure R600a state of the fine-tuned model. While the variance in temperature within the cabinet is a maximum of 0.5 °C, the deviation in cycle time is 1 minute. While the temperature difference grew to a maximum of 1.5 °C in the case of a refrigerant mixture, the cycle duration diversion reached 2 minutes.
- In the case of pure refrigerant, the power consumption differential rose from 2.4% to 5.2%. In contrast, energy consumption declined from 2.9% to 2% due to power consumption fluctuations.

- Considering the system's operating pressures in pure refrigerant and mixing conditions, the maximum deviation between the experiment and the model increased from 0.23 bar to 0.5 bar on the high-pressure side, while it increased from 0.1 bar to 0.25 bar on the low-pressure side, respectively.

This thesis reveals the impact of mixing states on the parameters that govern the system's performance to a limited extent. In this research, where the mixing ratio is the only variable, the system may be optimized for each mixing ratio. Further research may include parametric optimization techniques such as adjusting the heat exchanger size, and secondary fluid characteristics on the heat exchanger, modifying the capillary length and determining charge quantity, and developing alternative compressor applications fully compatible with R290.

Again, within the scope of this thesis, the base system design using pure refrigerant R600a offers results that are highly consistent with the experiment and the model. However, the model is open to further development for mixture refrigerants. In this direction, a single compressor model that works in pure and mixed states can be developed. In addition, by working on the refrigerant distribution model at the stopping times of the compressor, a model with realistic results can be obtained. Nonetheless, even using the mixing model, the effect of parameter changes on system performance can be examined quite realistically.

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