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A COMPARATIVE THERMODYNAMIC AND ECONOMIC ANALYSES OF CONVENTIONAL AND VRF HVAC APPLICATIONS IN BUILDINGS: A SOCIAL AND CULTURAL CENTER CASE STUDY

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Assoc. Prof. Dr. Emrah ÖZAHĶ

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Oĵ uzhan DAĴ CI

ABSTRACT

A COMPARATIVE THERMODYNAMIC AND ECONOMIC ANALYSES OF CONVENTIONAL AND VRF HVAC APPLICATIONS IN BUILDINGS: A SOCIAL AND CULTURAL CENTER CASE STUDY

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HVAC systems in buildings are composed of many subcomponents which consume energy in huge amounts. In recent years, energy consumption becomes very important aspect due to the lack of energy resources and environmental reasons. In terms of energy consumption, the components of HVAC systems account for large portion of total energy used in buildings. Therefore optimization of energy consumption in HVAC applications is an essential requirement in terms of thermodynamic and economic point of views. Besides, the efforts for decreasing energy consumption turn into economic recovery. It also provides benefits for health of people and cleaner environment. In this thesis, thermodynamic and economic analysis of an existing social and cultural building in Kilis 7 Aralik University, which has a heating and cooling area of 8852 m^2, is presented conducting by comparison of VRF and conventional HVAC systems. Then, the obtained results were compared. Due to these results, the advantages and disadvantages of both systems were discussed in view of thermodynamic and economic aspects.

Keywords: HVAC, VRF, Thermodynamic analysis, Economic analysis

ÖZET

BİNALARDA GELENEKSEL VE VRF KARŞILAŞTIRMALI TERMODİNAMİK VE EKONOMİK ANALİZİ: BİR SOSYAL VE KÜLTÜREL MERKEZİN SAHA ÇALIŞMASI

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Binalardaki HVAC sistemleri çok büyük miktarlarda enerji tüketen birçok alt bileşenlerden oluşmaktadır. Son yıllarda, enerji kaynaklarının azalmasından ve ekolojik nedenlerden dolayı, enerji tüketimi çok önemli bir husus haline gelmektedir. Enerji tüketimi bakımından, binalarda kullanılan toplam enerjinin büyük bir kısmı HVAC sistem bileşenlerinden kaynaklanmaktadır. Bu yüzden, termodinamik ve ekonomik bakış açısından, HVAC uygulamalarındaki enerji tüketiminin optimizasyonu hayati önem taşımaktadır. Bunun yanı sıra, enerji tüketiminin düşürülmesi için harcanan çaba ekonomik geri kazanıma dönüşmektedir. Bu çaba ayrıca insan sağlığı ve daha temiz bir çevre için yarar sağlamaktadır. Bu tezde, Kilis 7 Aralık Üniversitesi'nde mevcut 8852 m^2 ısıtma ve soğutma alanına sahip bir sosyal tesis için VRF ve geleneksel HVAC sistem karşılaştırması yapılarak termodinamik ve ekonomik bir analiz sunulmaktadır. Daha sonra elde edilen sonuçlar karşılaştırılmıştır. Bu sonuçlar sayesinde, her iki sisteminde avantaj ve dezavantajları termodinamik ve ekonomik açıdan tartışılmıştır.

Anahtar Kelimler: HVAC, VRF, Termodinamik analiz, Ekonomik analiz.

To My Family

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TABLE OF CONTENTS

LIST OF TABLES

Page

LIST OF FIGURES

Page

LIST OF SYMBOLS/ABREVIATIONS

A	Area, m^2
A_{DD}	External wall area, m^2
a	Leaking factor, m^3/mh
c	Thermal conductance
$C_{p,air}$	Specific heat of air, kj/kg
f	Thermal emittance of the heater
F_C	Correlation factor, [Fc = 1 ï 0.0116.K_T]
g_i	Solar passing factor
g_{sw}	Condensation water vapor mass
h	Heat transfer coefficient, W/m^2 K
h_0	Heat transfer coefficient of convection, $W/m^2.C$
h_{fg}	The enthalpy of vaporization of water, kj/kg
H_v	Heat loss by ventilation, W /K
H_c	Heat loss by conduction, W /K
H_T	Total heat loss, W/K
I	Intensity from the vertical surface, W/m^2
k	Thermal conductivity, Btu/h
KKO	Heat gain loss ratio

l	Length, m
m_{vapor}	The rate of evaporation from the body, kg/s
n_h	Number of air changes
n	Expansion coefficient of the water
P	Pressure, Pa
Qô	Maximum permissible heat losses for buildings, KWh/m^2
Q	Heat flow through the walls etc., Btu/h
Q_{AHU}	AHU heating capacity, W
Q_{inf}	Infiltration heat loss, W
$Q_{fancoil}$	Fan coil heating capacity, W
q_{solar}	Solar radiation incident on the surface, W/m^2
R	Thermal resistance, W/m.K
r	Shading factor
S_d	Equivalent air layer thickness, m
U	Overall heat transfer coefficient, W/m^2.K
U_{DD}	External wall heat transfer coefficient, W/m^2.K
V_h	The volume of building ventilating area, m^3
V_s	Total amount of water in the installation, m^3
W_{motor}	Power rating, W
w	Humidity ratio
φT	Difference in outside and inside temperature, K
T_s	Surface temperature
T_f	Fluid temperature, K

Z	Total increase coefficient [$Z = (1 + \%Z_D + \%Z_W + \%Z_H)$]
Z_g	Daily working time
Z_y	Annual working time, h/day

Subscripts

Con	Conduction
sur	Surface
conv	Convection
rad	Radiation
surr	Surrounding
cw	Curtain wall
ew	External wall
sr	Sandwich panel roof
uc	Uncovered ceiling
gb	Ground engaging base
d	Door
w	Window

Greek letters

Ŭ	Solar absorptivity
ὲ $_i$	Internal heat gain, W
ὲ $_g$	Solar energy gain, W
Ǎ	Heat gain loss ratio
ɖ$_{month}$	Gain utilization factor
ɖ$_{motor}$	Motor efficiency

ɚ Thermal conductivity coefficient, W/m.K

ŭ Thickness of materials, m

µ Vapor diffusion resistance factor

ẽ$_{yi}$ Indoor surface temperature, K

Abbreviations

AHU Air Handling Unit

SHGF Solar Heat Gain Factor

SCHX Sub Cooling Heat Exchanger

COP Coefficient of Performance

CLF Cooling Load Factors

CLTD Cooling Load Temperature Difference, °C

HVAC Heating Ventilating Air Conditioning

VAV Variable Air Volume

VRV Variable Refrigerant Volume

VRF Variable Refrigerant Fluid

ZD The increase coefficient of operating

ZW The increase coefficient height of the floor

ZH The increase coefficient from the direction

CHAPTER 1

INTRODUCTION

1.1 Background

Each building could be heated and cooled with all known systems. But the important thing is to determine which system would be most appropriate scenario for the building. According to the chosen system it can be used an average of 20 years or more, operating cost, quiet and smooth operation, ventilation capability, and so on. The expectations of the system can meet the selection criteria should be examined. It is very important how the system will be affected by choice facade of the building. Architects; must know the system and architectural design should take into account these factors [1].

Architectural aesthetic as well as population growth, depletion of natural resources, international competitiveness, and reasons such as increased energy costs increased the energy efficiency in building design to the forefront. We are largely dependent on foreign countries in terms of energy, so applications in developing countries have been used in our country. For example, Turkey's Energy Efficiency Law, EU Law, such as the Renewable Energy Law 2002/91/EC Energy Efficiency in Buildings Energy Performance Directive for compliance with the Regulation was issued [2].

In this thesis, based on the above-mentioned energy efficiency, Social and Cultural Facility, which is designed according to the conventional system is compared with the VRF system, based initial investment, operation and maintenance costs. First, re-examined the insulating properties of the building, heating and cooling accounts capacities are reconstructed and redefined. Found theoretical results are compared also with enterprises in various tourist resorts.

1.2 Scope and Outline of the Study

In this thesis, firstly the literature survey of HVAC systems is investigated. Then building isolation, heating and cooling calculations are evaluated. As a result of study, the conventional HVAC system and VRF system are compared. Both systemsô advantages and disadvantages are mentioned. The outline of the study with respect to chapters is as follows:

In chapter 2, a comprehensive literature survey on HVAC systems are presented. Both systems usage areas are examined and received information about the way work. The available studies are classified with respect to HVAC systems, VRF systems and both of HVAC - VRF systems comparison.

In chapter 3, the conventional HVAC system used in the Social and Cultural Building in Kilis 7 Aralik University is investigated. Heating and cooling systems are classified and the related basic definitions and terminologies are given. HVAC system in building is presented. The modeled system VRF is identified in this section.

In chapter 4, the isolation standards, TS 825 is identified. Vapor diffusion calculations are evaluated. It is identified that the building is eligibility standards or not. It is seen that insulation standards are acceptable. As a result, evaporation and condensation calculation are evaluated and building materials are shown no risk of mold.

In chapter 5, heat loss by conduction and infiltration are evaluated. Total heat loss of the building is evaluated and detailed information and accounts are given about the auxiliary equipment.

In chapter 6, heat gain calculations are evaluated. AHU and fan coil cooling capacities are calculated and detailed information and accounts are given about the auxiliary equipment.

In chapter 7, conventional systems and VRF systems investment, operation and maintenance costs are evaluated. Both systems advantages and disadvantages are mentioned. It is seen that VRF systems is more economical and applicable system. Initial investment, operating and maintenance of all costs are lower than conventional system.

CHAPTER 2

LITERATURE SURVEY

2.1 Introduction

In this chapter, the conventional HVAC systems and the variable refrigerant fluid (VRF) technologies and the related studies are presented. The available studies are classified with respect to HVAC systems, VRF systems and both of HVAC - VRF systems comparison. The aim here is firstly to give a view on this research and to have an idea for the current study and further studies.

2.2 HVAC Applications

Murat Koru [3] studied the controlling, modeling and optimization in the HVAC systems. The automatic control methods of HVAC systems which are developed and appeared by the comfortable conditions of peoples have been discussed. The running of systems components and how they can provide control operations have been stated. During the work of building at normal or minimum loads, the optimization methods which can be used are stated. How the controls of environment conditions are provided and a mathematical model which is belonging to HVAC systems are constituted. The results showed that the control equipment are so necessary for the energy saving.

Murat Özer [4] studied the design, thermodynamic tests and analysis of an operating room air handling unit. According to some initial values and design criterions, equipment of operating room air handling unit were calculated and selected. After that calculation, a test stand prepared to see the effects of various supply air conditions on the cooling capacity and check the theoretical calculations. The cooling capacity from air side and refrigeration side were compared. The results showed that both measurements are consistent with each other and with the theoretical calculation.

Ayhan Doĵan [5] investigated evaluation of sports hall in case of ventilation. The appropriate ventilation system to provide a healthy condition for sportsmen and spectators at sports halls was researched. Consequently, how the ventilation system of a sports hall should be and determination factors for ventilation systems are emphasized. Also negative effects of insufficient ventilation at sports hall on sportsmen and spectators health and efficiency of sportsmen were considered. Designing this system usability, air conditioning and economical life are also taken into account.

Daĵhan Yalçēn [6] studied comparisons and economic analysis of heating, cooling and ventilation installations in a building. The purpose of this thesis was to provide a source for mechanical installation systems while comparing the present systems with the systems of todayôs improving technology. For the present building heating, cooling and ventilation installations were considered completely and compared with the alternative installations within the same perspective. The advantages and disadvantages of the alternative systems were evaluated.

Uĵur Özcan [7] surveyed HVAC systems used in modern architecture, their relations with architecture and application samples in high tech buildings. Smart building examples such as Sabancē Center, Holiday Inn Crown Plaza, Maya Akar Center, EGS Business Park, Türkiye Ķk Bankasē General Directorate Complex and Metrocity complex were examined using theoretical calculations. Isolation and automation advantages were examined.

Ezgi Baba [8] studied energy economy in HVAC systems. Selected from different climatic regions of our country in Istanbul, Ankara, Antalya and Erzurum provinces located in a shopping center in central air conditioning and heating capacity with hourly temperature and humidity values were calculated. The new systems as a whole for review by the design of economy have been analyzed. As a conclusion the using heat exchanger is getting higher system efficiency. Especially in Ankara energy saving is higher because of the climate.

Nurullah Arslanoĵlu [9] investigated optimization and economic analysis of indoor air conditions in HVAC systems. Energy consumption under different COP values of a clinic project which is cooled by an air conditioner were calculated and monetary cost calculations were obtained by using the present value method. As a result, it has been seen that insulation of building has earned us energy costs. And also in the HVAC system, COP is very effective for energy consumption.

Liang Yang and Chun-Lu Zhang [10] studied the analysis on energy saving potential of integrated supermarket in which HVAC and refrigeration systems with multiple subcoolers were used. The study presents a model-based analysis on the energy saving potential of supermarket HVAC and refrigeration systems using multiple subcoolers among the high-temperature HVAC system, the medium-temperature refrigeration system, and the low-temperature refrigeration system. The principle of energy reduction is to have the higher COP system which generates more cooling capacity to increase the cooling capacity or reduce the power consumption of the lower COP system. The optimal sequence of adding subcoolers was also proposed. As a result, it has been found that when adding multiple subcoolers into the integrated supermarket system, the sequence should be ML (medium-low), HM (high medium) and HL (high low) subcooler. The analysis showed that at a small HVAC energy demand supermarket, adding a ML subcooler saved 10.5% energy consumption.

Gökhan Ünlü [11] investigated HVAC system selection, design and energy analysis for sustainable buildings. An office building was examined which would be constructed in Ankara. Heat load calculations and energy consumption simulation calculations were performed for this building with the aim of finding best HVAC system. Hourly Analysis Program was used for all these calculations. HVAC system alternatives for an office building were examined and energy consumption rates of these alternatives were calculated.

Merve Yenice [12] surveyed project preparation and application of cooling and ventilation systems in terminal stations. The aim of this dissertation was to provide a source for the selection of appropriate heating, cooling and ventilation systems in public buildings. At the preparation period of the dissertation, heating and air conditioning systems were classified and system selection criteria were evaluated. In the conclusion, with the comparison of the systems, heating, cooling and ventilation systems to be used in terminal stations and similar buildings considering the criteria of comfort, maintenance and operation easiness, investment cost, operation cost and environmental factors were considered. The result showed that the central heating system was more applicable then zonal heating system.

Luis Pérez-Lombard, et al. [13] presented a review of HVAC systems requirements in building energy regulations. This study analyzed the development of building energy codes concerning heating, ventilation and air-conditioning (HVAC) energy efficiency, along with their scope and compliance paths. The study focused on the synthesis of energy efficiency requirements on HVAC systems of non-residential buildings in different regulations. Critical issues for the development of prescriptive and performance regulatory paths for this type of systems in non-residential buildings were discussed in order to improve the understanding of HVAC energy efficiency topics and to provide policy makers with a menu of options to strengthen the HVAC section of building energy codes.

Lun Zhang, et al. [14] surveyed the application of entransy in the analysis of HVAC systems in buildings. The main task of HVAC systems in the cooling condition is to remove heat from indoor environment to outdoor environment. Single analysis method or thermal parameter could hardly describe all the processes in an HVAC system. When the purpose of heat transfer was cooling or heating, entransy analysis was a direct method for optimizing heat transfer processes. Loss of HVAC system was mainly in heat transfer process. The entransy dissipation extremum principle or the minimum thermal resistance principle was suitable for analyzing heat transfer process in HVAC system. The main conclusions can be summarized as; entransy was defined as heat transfer ability. When the purpose of heat transfer was cooling or heating, entransy analysis was a clear and direct method for optimizing heat transfer processes. The entransy dissipation extremum principle or the minimum thermal resistance principle was suitable for analyzing heat ex-changers, indoor cooling processes.

XueTao Cheng and XinGang Liang [15] studied the analyses and optimizations of thermodynamic performance of an air conditioning system for room heating. This study presented the optimization of an air conditioning system, in which the heat from the hot stream was transported to do work first, and then the work was used to pump heat from the environment into the room. It was shown that the system could deliver more heat into the room than the direct heating by the hot stream. A function was set up to optimize the distribution of the total thermal conductance for the system, and the numerical results showed that the function was effective. The system was also discussed with the concepts of entropy and entransy. It was found that the minimum values of the entropy generation rate, entropy generation number, revised entropy generation number, entransy loss and entransy loss coefficient corresponded to the maximum heat flow rate into the room with prescribed heat flow rate from the hot stream, but the values of the parameters increased with increasing heat flow rate from the hot stream and that into the room.

Zhaoxia Wang, et al. [16] analyzed energy efficiency retrofit schemes for heating, ventilating and air-conditioning systems in existing office buildings based on the modified bin method. Poor thermal performance of building envelop and low efficiencies of heating, ventilating and air-conditioning (HVAC) systems can always be found in the existing office buildings with large energy consumption. This paper adopted a modified bin method to propose and optimize the energy efficiency retrofit (EER) schemes. An existing office building in Tianjin was selected as an example to demonstrate the procedures of formulating the design scheme. Pertinent retrofit schemes for HVAC system were proposed after the retrofit of building envelop. With comprehensive consideration of energy efficiency and economic benefits, the recommended scheme that could improve the overall energy efficiency by 71.20% was determined.

2.3 Variable Refrigerant Flow (VRF) Applications

Mustafa Sabri ķamdan [17] studied the building heating and cooling systems. To minimize operating costs with a correct study of estimated automation project values and resulting values in the application was the subject of the thesis. It was concluded that the VRF system used in automation caused to have low installation costs and easy application process.

Xiaobing Liu and Tianzhen Hong [18] compared the energy efficiency between VRF systems and ground source heat pump systems. In this article, a preliminary comparison of energy efficiency between the air-source VRF and GSHP systems was presented. The computer simulation results showed that, GSHP system was more energy efficient than the air-source VRF system for conditioning a small office building in two selected US climates. In general, GSHP system was more energy efficient than the air-source VRF system, especially when the building had significant heating loads. For the buildings with less heating loads, the GSHP system could still perform better than the air-source VRF system in terms of energy efficiency.

Yue Ming Li and Jing Yi Wu [19] studied the energy simulation and analysis of the heat recovery VRF system in winter. In order to evaluate the energy features of the system, a new energy simulation module was developed and embedded in the dynamic energy simulation program, Energy Plus. Using the program with the newly developed module, the dynamic energy simulation was performed for a simplified typical commercial building. The indoor thermal comfort of the building in winter and the setting temperature of the system were analyzed. Based on the simulation results, the energy characteristics of the system were investigated, and it was indicated that different methods of the temperature control and the percentage of the heat recovery had influence on the relative ratio of the energy saving. If the HR-VRF system adopted the same temperature control method as the heat pump VRF (HP-VRF) system, the HR-VRF system promised 15ï 17% energy-saving potential, when compared to the HP-VRF system.

Tolga N. Aynur [20] analyzed VRF systems. This study presented a detailed overview of the configurations of the outdoor and indoor units of a multi-split VRF system, and its operations, applications, marketing and cost. Besides, a detailed review about the experimental and numerical studies associated with the VRF systems was provided. The aim was to put together all the diversified information about the VRF systems in a single source. It was concluded that VRF system not only consumed less energy than the common air conditioning systems such as variable air volume, fan-coil plus fresh air under the same conditions, but also provided better indoor thermal comfort as long as it was operated in the individual control mode. It was found that even though the main drawback of the VRF system was the high initial cost compared to the common air conditioning systems, due to the energy saving potential of the VRF system, the estimated payback period of the VRF system compared to an air cooled chiller system in a generic commercial building could be about 1.5 year.

Laeun Kwon, et al. [21] investigated the field performance measurements of a VRF system with sub-cooler in educational offices for the cooling season. The effects of the subcooling heat exchanger (SCHX) on the performance of the multi-split variable refrigerant flow (VRF) system with long pipe were investigated in a field test during the cooling season. By varying the refrigerant mass flow rate via the electronic expansion valve, the degree of subcooling was controlled. It was found that VRF system with SCHX improved the cooling performance factor (CPF) about 8.5% under similar outdoor temperature profiles, as compared to the baseline without SCHX. As a result the employing a SCHX in a VRF system with long pipe was found to be an effective method for improving the system performance and reliability.

Laeun Kwon, et al. [22] investigated the experimental investigation of multifunctional VRF system in heating and shoulder seasons. The MFVRF system was installed in an office building and fully instrumented to measure the performance of the system under a wide range of outdoor weather conditions. The effects of a part-load ratio (PLR), a hot water demand and a heat recovery operation mode on the performance of the MFVRF system were investigated in a field test for the heating and shoulder seasons. As the hot water demand for the MFVRF system increased, the PLR was improved, which resulted in an increase in system heating performance. In the heat recovery operation mode, the heat absorbed from the indoor units operating in the cooling mode was transferred to other indoor units operating in the heating mode. The daily performance factor was 2.14 and 3.54 when the ratio of Daily total cooling energy to daily total energy was 13.0% and 28.4%, respectively, at the similar outdoor weather conditions. This enhancement was attributed to the waste heat recovered during the heat recovery operation mode and the decrease in pressure ratio, which is a result of the improvement of the compressor efficiency. The performance of the MFVRF system for the heating and shoulder seasons was improved by transferring the recovered energy to the indoor space and supplying the hot water.

Tianzhen Hong, et al. [23] surveyed a new model to simulate energy performance of VRF systems. The main improvement of the new model was the introduction of the evaporating and condensing temperature in the indoor and outdoor unit capacity modifier functions. The new approach allowed compliance with different specifications of each indoor unit so that the modeling accuracy was improved. In this study computer based building energy modeling and simulation was demonstrated and the VRF system performance was then compared with three other types of HVAC systems. Calculated energy savings from the VRF systems were significant.

2.4 HVAC- VRF Comparison

Adnan Etem [24] investigated comparison of classical fancoil and VRF system and determining the convenient system. The usual split additional circuitry system that used over electricity ad fuel ad had separate heating and cooling units and additional water circulation, and the combined heating and cooling systems with variable gas flow (split air conditioner) were compared with regard to technical and economic aspects. In conclusion, it was seen that VRF system was more economical systems.

Tolga N. Aynur and Yunho Hwang [25] compared the simulation comparison of VAV and VRF air conditioning systems in an existing building for the cooling season performance. Two widely used air conditioning (AC) systems, variable air volume (VAV) and variable refrigerant flow (VRF), in an existing office building environment under the same indoor and outdoor conditions for an entire cooling season was simulated by using two validated respective models and compared. As a consequently it was found that the VRF AC system promised 27.1ï 57.9% energy-saving potentials depending on the system configuration, indoor and outdoor conditions, when compared to the VAV AC system.

Ahmet Teke and Oĵ uzhan Timur [26] investigated assessing the energy efficiency improvement potentials of HVAC systems considering economic and environmental aspects at the hospitals. In this study, the research papers and practical studies on energy efficiency and energy saving potentials on HVAC systems at the hospitals were presented. VRF technology enabled greater energy efficiency and cost savings compared with conventional HVAC systems was also introduced. This detailed review also focused on the payback periods of some projects on HVAC including the installation of cogeneration, trigeneration, chiller, new burners, heat exchangers and steam trap systems.

Yonghua Zhu, et al. [27] studied optimal control of combined air conditioning system with VRF and VAV for energy saving. An optimal control strategy for minimizing the energy consumption of VRF and VAV combined air conditioning systems was presented. The combined system was proposed to take advantages of VAV systems to solve the ventilation problem of VRF systems. To determine set-point of the optimal control variable, i.e. outdoor air supply temperature, this study proposed an optimal control strategy based on adaptive predictive model and recursive least squares estimation technique. Results indicated that the optimal control strategy reduced energy consumption of the combined system.

2.5 Conclusion

In this chapter, some essential studies related to HVAC and VRF systems are outlined. The studies are generally related with efficiency of the systems. Economic comparisons have been taken into account rather than comfort conditions. It was noted that the importance of isolation. As a result of research, it was generally founded that VRF system was more economical then other HVAC systems.

CHAPTER 3

THE CURRENT CASE: HVAC APPLICATIONS USED IN THE SOCIAL AND CULTURAL BUILDING IN KILIS 7 ARALIK UNIVERSITY AND MODELLING OF A VRF HEAT PUMP SYSTEM

3.1 Introduction

In this chapter, the conventional HVAC system used in the Social and Cultural Building in Kilis 7 Aralik University will be investigated. Heating and cooling systems are classified and the related basic definitions and terminologies are given. Also, VRF system is modeled and analyzed for the same building. The optimum conditions and system are determined as a result of the study.

3.2 Descriptions of HVAC systems

HVAC (heating, ventilating, and air conditioning) is the technology of indoor and environmental comfort. Its goal is to provide thermal comfort and acceptable indoor air quality. HVAC system design is a subdiscipline of mechanical engineering, based on the principles of thermodynamics, fluid mechanics, and heat transfer.

HVAC is important in the design of medium to large industrial and office buildings such as skyscrapers and in marine environments such as aquariums, where safe and healthy building conditions are regulated with respect to temperature and humidity, using fresh air from outdoors [28].

3.2.1 Heating

Heat is a form of energy transferred, from a hotter body to a colder one, other than by work or transfer of matter. It occurs spontaneously whenever a suitable physical pathway exists between the bodies by virtue of temperature difference. The pathway can be direct, as in conduction and radiation, or indirect, as in convective circulation. Heating is a dissipative process. Heat is not a state function of a system.

3.2.2 Ventilating

Ventilating is the process of changing or replacing air in any space to provide high indoor air quality. Ventilation is used to remove unpleasant smells and excessive moisture, introduce outside air, to keep interior building air circulating, and to prevent stagnation of the interior air.

Ventilation includes both the exchange of air to the outside as well as circulation of air within the building. It is one of the most important factors for maintaining acceptable indoor air quality in buildings. Methods for ventilating a building may be divided into mechanical/forced and natural types.

3.2.3 Air-Conditioning

An air conditioning system, or a standalone air conditioner, provides cooling, heating and humidity control for all or part of a building. Air conditioned buildings often have sealed windows, because open windows would work against the system intended to maintain constant indoor air conditions. Outside, fresh air is generally drawn into the system by a vent into the indoor heat exchanger section, creating positive air pressure. The percentage of return air made up of fresh air can usually be manipulated by adjusting the opening of this vent. Typical fresh air intake is about 10% [29-30].

Air conditioning and refrigeration are provided through the removal of heat. Heat can be removed through radiation, convection, or conduction. Refrigeration conduction media such as water, air, ice, and chemicals are referred to as refrigerants. A refrigerant is employed either in a heat pump system in which a compressor is used to drive thermodynamic refrigeration cycle, or in a free cooling system which uses pumps to circulate a cool refrigerant (typically water or a glycol mix). The following factors should be considered in the selection of air conditioning systems:

- Initial investment cost
- Comfort conditions
- Noise
- Aesthetics

- Operating cost
- Ease of installation
- Energy expenditure
- Ease of operation

3.3 Classification of Air Conditioning Systems

The purpose of classifying air conditioning systems is to distinguish one type from another and to provide a background for selecting the optimum air conditioning system based on building requirements. A classification of air conditioning systems should include the classification of air and refrigeration systems in order to define a more specific system. Air conditioning systems can be classified into three categories corresponding to their related equipment as follows [31],

3.3.1 Individual Systems

An individual air-conditioning system normally employs either a single, self-contained, packaged room air conditioner (installed in a window or through a wall) or separate indoor and outdoor units to serve an individual room. ñSelfcontained, packagedò means factory assembled in one package and ready for use (see in Figure 3.1).

Figure 3.1 Individual systems

3.3.2 Packaged Systems

Similar in nature to individual systems but serve more rooms or even more than one floor, have an air system consisting of fans, coils, filters, ductwork and outlets (e.g. in small restaurants, small shops and small cold storage rooms)

These systems are installed with either a single selfcontained, factory-assembled packaged unit (PU) or two split units; an indoor air handler, normally with ductwork, and an outdoor condensing unit with refrigeration compressor(s) and condenser. In a packaged system (is shown in Figure 3.2), air is cooled mainly by direct expansion of refrigerant in coils called DX coils and heated by gas furnace, electric heating, or a heat pump effect, which is the reverse of a refrigeration cycle.

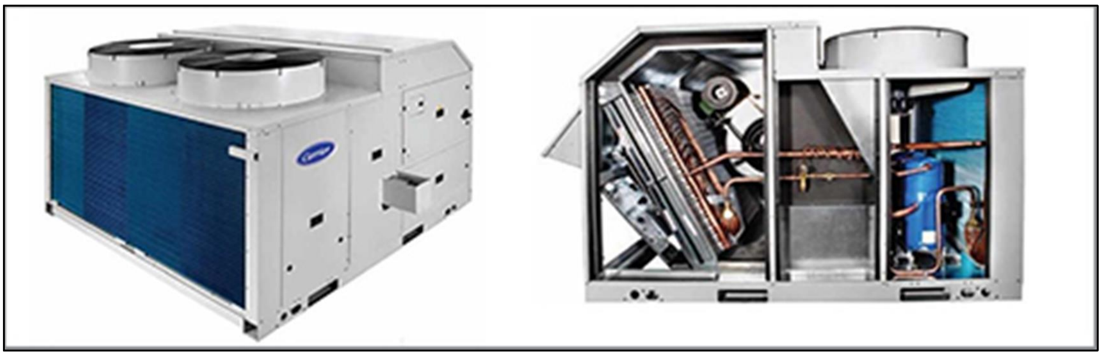


Figure 3.2 Packaged systems

3.3.3 Central Systems

These systems are used primarily for the conditioning of larger buildings. Air handling units, air ducts, vents and/or fan coil units etc. constitute the system components. Heating, cooling, ventilation and humidity control is achieved by circulating water, air or coolant within the tubes and ducts of the system. Central systems can be classified in three main groups;

a) All-Air Systems: These are systems where air is used as heat transfer fluid. These systems provide sensible and latent cooling by transferring air conditioned by cooling-de-humidification processes to the living space through ducts and provide heating by conditioning heated air and transferring it to the living space through ducts.

b) All-Water Systems: The living space is heated or cooled by fan coil unitös places in each living space. The cold water required for cooling is transferred to the installation by pumps from a central cooling group (a chiller), while the hot water required for heating is transferred to the installation by pumps from the central boiler system. The comfort conditioning of spaces is achieved by thermostat.

c) Air-Water Systems: Air and water systems employ the process of cooling or heating spaces in a way that will also meet peopleôs fresh air requirement, by transferring clean air that is conditioned in a central unit or water that is cooled in a central cooling group to fan coil units.

The existing central systems are fan coils, AHUs and VRF systems. If briefly examined:

3.3.3.1 Fan Coil

A fan coil unit (FCU) is a simple device consisting of a heating or cooling coil and fan. It is part of an HVAC system found in residential, commercial, and industrial buildings. Typically a fan coil unit is not connected to ductwork, and is used to control the temperature in the space where it is installed, or serve multiple spaces. It is controlled either by a manual on/off switch or by thermostat.

Due to their simplicity, fan coil units are more economical to install than ducted or central heating systems with air handling units. However, they can be noisy because the fan is within the same space. Unit configurations are numerous including horizontal (ceiling mounted) or vertical (floor mounted). In Social & Cultural building ceiling mounted types of fan coils are used. The component of FCU is shown in Figure 3.3.

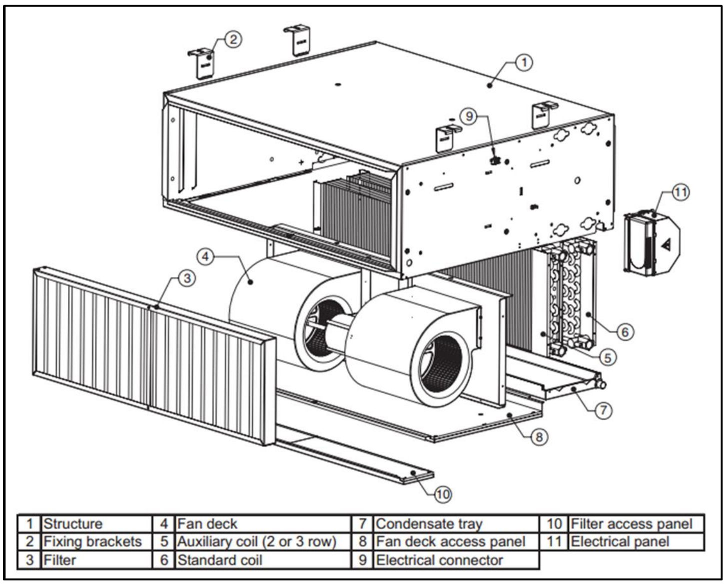


Figure 3.3 Fan coil parts

3.3.3.2 Air Handling Units (AHUs)

Air handling units (AHUs) also known as air handlers, are used to supply and circulate air around a building, or to extract stale air as part of a buildingös heating, ventilating and air conditioning (HVAC) system.

Essentially, an AHU system comprises a large insulated metal box that contains a fan, heating and/or cooling elements, filters, sound attenuators and dampers. In most cases, the AHU is connected to air distribution ductwork; alternatively, the AHU can be open to the space it serves.

Supply air passing through the AHU is filtered and is either heated or cooled, depending on specified duty and the ambient weather conditions. In some buildings, AHU systems are used only to supply fresh air for ventilation and extract stale air.

For heating or cooling, AHUs may be connected to central plant such as boilers or chillers, receiving hot or chilled water for heat exchange with the incoming air. Alternatively, heating or cooling may be provided by electric heating elements or direct expansion refrigeration units built into the air handler.

When AHU systems are used to extract stale air from the building, a controlled proportion of this air may be recirculated to avoid having to condition all supplied air. AHUs can also incorporate heat recovery mechanisms to extract heat from the air being expelled and use it to heat incoming supply air.

AHU systems vary considerably in size, capacity and complexity, depending on the job they are designed to perform. AHUös cross section is given in Figure 3.4

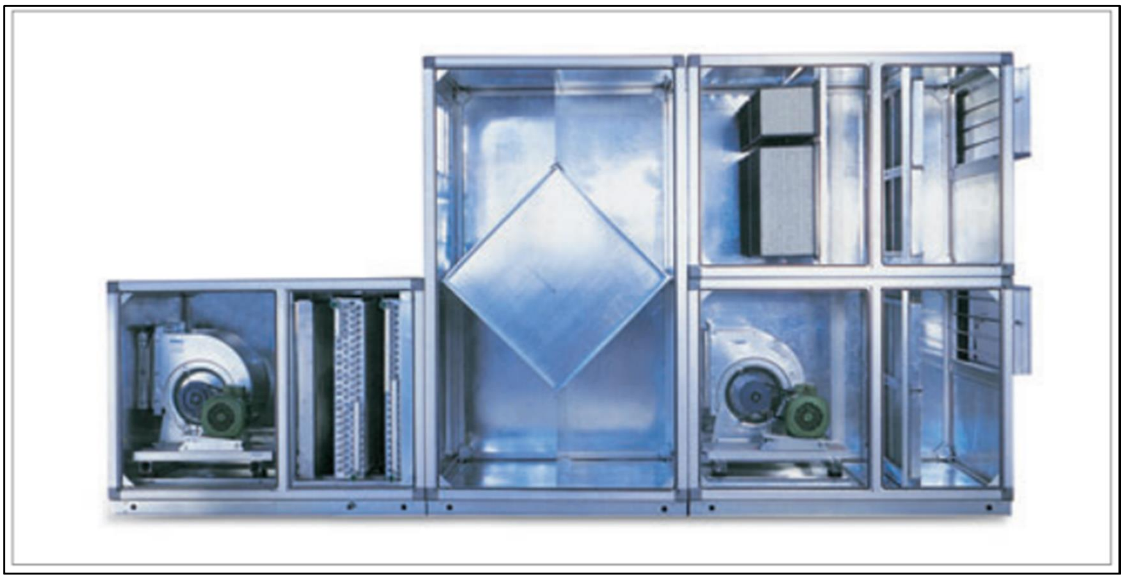


Figure 3.4 AHU߀s cross section

The systems such as AHU and fan coils are heated and cooled by respectively boiler and chiller. Let߀s examine this auxiliary equipment.

3.3.3.3 Boiler

A boiler is a closed vessel in which steam is generated from water by the application of heat (see in Figure 3.5).

The simple boiler is like a barrel, consisting of a cylindrical steel shell, with the ends closed by flat steel heads. It is partly filled with water and then sealed, after which a fire is started beneath it. The fire and hot gases rise around the lower outside of the shell, the heat being conducted through the steel into the water. This heats the water on the bottom of the boiler first. Hot water being lighter than cold water, it rises, while the colder water in the upper part, being heavier, sinks down to replace it and is in turn heated. These are convection currents, and the process is known as circulation, which goes on continually while a boiler is in service. Circulation is good in some boilers and poor in others, depending upon the design.

The water gradually reaches the temperature where steam is given off, which accumulates in the space above the water known as the steam space. As the steam accumulates, a pressure is built up which would create a very dangerous condition with the simple boiler. As pressure is exerted in every direction, the flat heads would bulge outward because a flat surface cannot support itself. The boiler would hold very little pressure and would be useless.

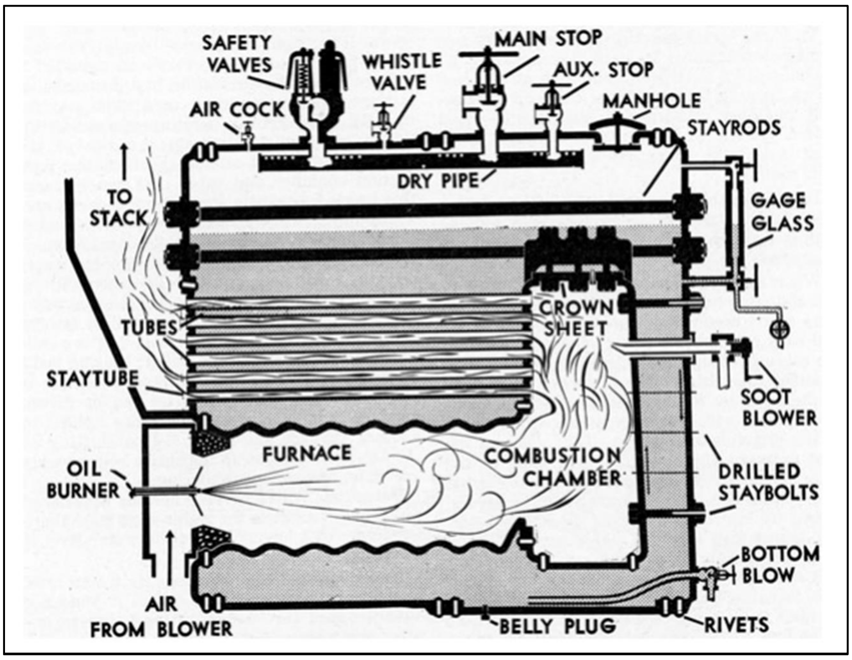


Figure 3.5 View of boiler

3.3.3.4 Chiller

A chiller is a machine that removes heat from a liquid via a vapor or absorption refrigeration cycle (see in Figure 3.6). This liquid can then be circulated through a heat exchanger to cool air or equipment as required. As a necessary by product, refrigeration creates waste heat that must be exhausted to ambient or, for greater efficiency, recovered for heating purposes. Concerns in design and selection of chillers include performance, efficiency, maintenance, and product life cycle environmental impact.

Figure 3.6 Water chiller

3.3.4 Variable Refrigerant Flow (VRF) Systems

Variable refrigerant flow (VRF) is an air-condition system configuration where there is one outdoor condensing unit and multiple indoor units. The term variable refrigerant flow refers to the ability of the system to control the amount of refrigerant flowing to the multiple evaporators (indoor units), enabling the use of many evaporators of differing capacities and configurations connected to a single condensing unit. The arrangement provides an individualized comfort control, and simultaneous heating and cooling in different zones.

The term VRF refers to the ability of the system to control the amount of refrigerant flowing to each of the evaporators, enabling the use of many evaporators of differing capacities and configurations, individualized comfort control, simultaneous heating and cooling in different zones, and heat recovery from one zone to another. VRF systems operate on the direct expansion (DX) principle meaning that heat is transferred to or from the space directly by circulating refrigerant to evaporators located near or within the conditioned space. Refrigerant flow control is the key to many advantages as well as the major technical challenge of VRF systems [29, 32, 41].

VRF systems are similar to the multi-split systems which connect one outdoor section to several evaporators. However, multi-split systems turn OFF or ON completely in response to one master controller, whereas VRF systems continually adjust the flow of refrigerant to each indoor evaporator. The control is achieved by continually varying the flow of refrigerant through a pulse modulating valve (PMV) whose opening is determined by the microprocessor receiving information from the thermistor sensors in each indoor unit. The indoor units are linked by a control wire to the outdoor unit which responds to the demand from the indoor units by varying its compressor speed to match the total cooling and/or heating requirements.

VRF systems promise a more energy-efficient strategy (estimates range from 11% to 17% less energy compared to conventional units) at a somewhat higher cost.

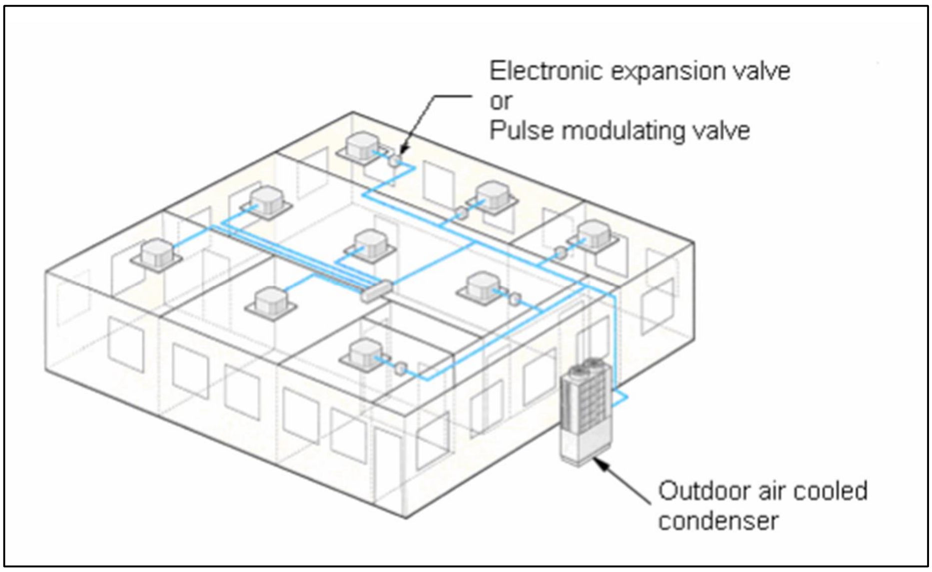


Figure 3.7 VRF systems with multi indoor unit

The modern VRF technology uses an inverter-driven scroll compressor and permits as many as 48 or more indoor units to operate from one outdoor unit (varies from manufacturer to manufacturer). The inverter scroll compressors are capable of changing the speed to follow the variations in the total cooling/heating load as determined by the suction gas pressure measured on the condensing unit. The capacity control range can be as low as 6% to 100%. Refrigerant piping runs of more than 200 ft. are possible, and outdoor units are available in sizes up to 240,000 Btu/h. A schematic VRF arrangement is indicated below in Figure 3.7 and Figure 3.8.

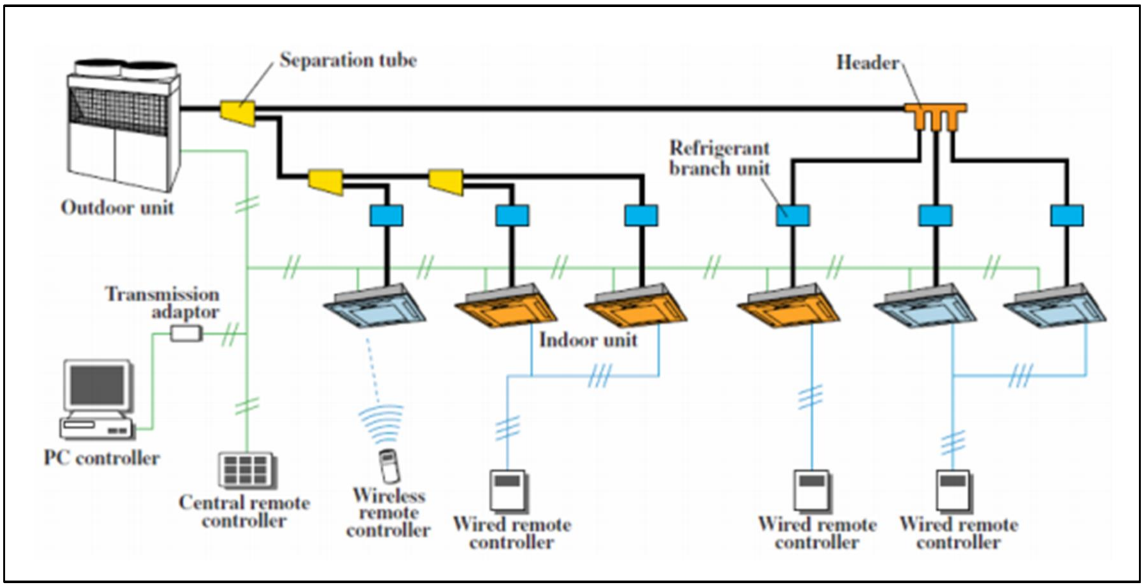


Figure 3.8 Heat pump VRF systems with multi indoor unit

3.3.4.1 Types of VRF

VRF systems can be used for cooling only, heat pumping or heat recovery. On heat pump models there are two basic types of VRF system: heat pump systems and energy recovery [32].

3.3.4.1.1 VRF Heat Pump Systems

VRF heat pump systems permit heating or cooling in all of the indoor units but NOT simultaneous heating and cooling. When the indoor units are in the cooling mode, they act as evaporators; when they are in the heating mode, they act as condensers. These are also known as two-pipe systems are shown in Fig. 3.9.

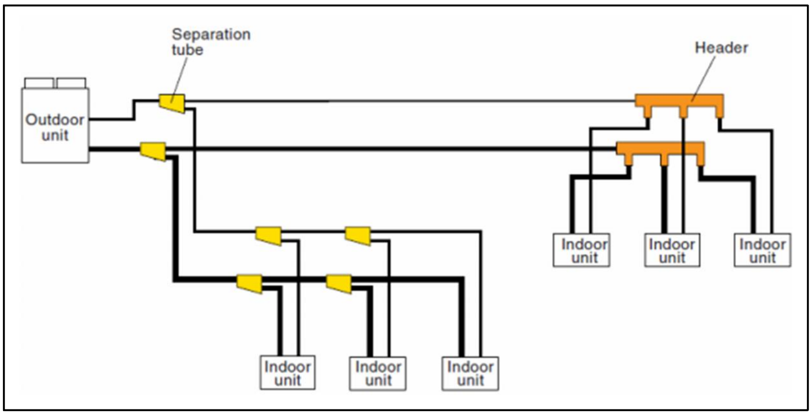


Figure 3.9 Cooling type VRF systems

VRF heat pump systems are effectively applied in open plan areas, retail stores, cellular offices and any other area that require cooling or heating during the same operational periods.

3.3.4.1.2 Heat Recovery VRF system (VRF-HR)

VRF systems with heat recovery (VRF-HR) capability can operate simultaneously in heating and/or cooling mode, enabling heat to be used rather than rejected as it would be in conventional heat pump systems. VRF-HR systems are equipped with enhanced features like inverter drives, pulse modulating electronic expansion valves and distributed controls that allow system to operate in net heating or net cooling mode, as demanded by the space (see in Figure 3.10).

Each manufacturer has its own proprietary design (2-pipe or 3-pipe system), but most uses a three-pipe system (liquid line, a hot gas line and a suction line) and special valving arrangements. Each indoor unit is branched off from the 3 pipes using solenoid valves. An indoor unit requiring cooling will open its liquid line and suction line valves and act as an evaporator. An indoor unit requiring heating will open its hot gas and liquid line valves and will act as a condenser.

Typically, extra heat exchangers in distribution boxes are used to transfer some reject heat from the superheated refrigerant exiting the zone being cooled to the refrigerant that is going to the zone to be heated. This balancing act has the potential to produce significant energy savings.

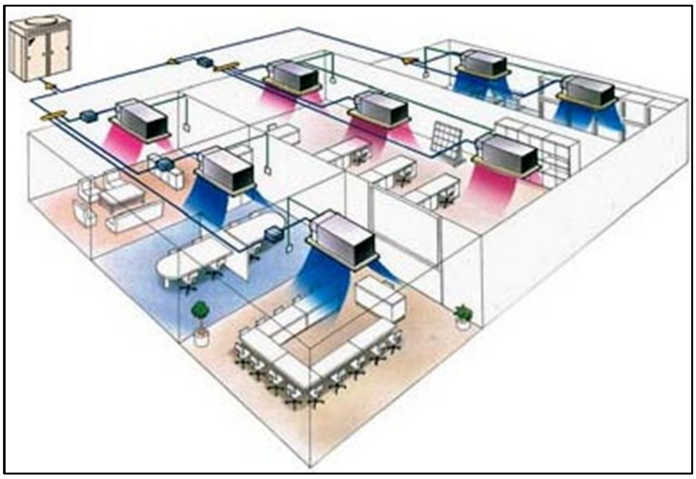


Figure 3.10 Heat recovery VRF system

VRF-HR mixed mode operation leads to energy savings as both ends of the thermodynamic cycle are delivering useful heat exchange. If a system has a cooling COP (Coefficient of Performance) of 3, and a heating COP of 4, then heat recovery operation could yield a COP as high as 7. COP is defined as follows;

$$COP_R = \frac{Q_L}{Q_H - Q_L} \tag{3.1}$$

$$COP_{HP} = \frac{Q_H}{Q_H - Q_L} \tag{3.2}$$

It should be noted that this perfect balance of heating and cooling demand is unlikely to occur for many hours each year, but whenever mixed mode is used energy is saved. Units are now available to deliver the heat removed from space cooling into hot water for space heating, domestic hot water or leisure applications, so that mixed mode is utilized for more of the year [33].

VRF-HR systems work best when there is a need for some of the spaces to be cooled and some of them to be heated during the same period. This often occurs in the winter in medium-sized to large sized buildings with a substantial core or in the areas on the north and south sides of a building.

3.4 Utilized HVAC Application Existing in Social and Cultural Building in Kilis 7 Aralik University

In this section, the existing HVAC application in the Social and Cultural building in Kilis 7 Aralik University is introduced giving the details. The available components are given for introducing before the analyses.

3.4.1 General Information about Social and Cultural Building

The building was constructed in 12.01.2011. The building has a capacity of 700 student and 400 staff and academic refectory, multi-purpose hall which has 300 capacities and also, it has many official rooms, cafeteria, computer lab and 22 guest house room. The building closed area is 8852 m^2. (See Figure 3.11 and 3.12)

Figure 3.11 Social and Cultural Building 3D appearance

The building is reinforced-concrete and has 3 floor and 1 ground. Each floor height is 4.85 m. The roof is sandwich panels covering. The exterior side of the building is pink andesite stone covering.

At basement, there are kitchen, toilet and ladies locker room.

In the ground floor, there are foyer, refectory, red tape, travel agency, telephonist, shop, hairdresser, dry cleaning, internet cafe, cafeteria and toilet

In the first floor, there are floor hall, administrative staff cafeteria, academics cafeteria, workshop, computer room, cinema, cinema foyer and toilet

In the second floor, there are art gallery, multi-purpose hall, offices, director's room, secretary, archive, doctor's room, nurse room, clinic and toilet.

In the third floor, there are floor hall, rest area, rooms, service, auditorium and toilet.

Figure 3.12 Social and Cultural Building

3.4.2 Heating Systems Used in Social and Cultural Building

There are 2 boilers which have 3 pass structures in the building as can be seen from Figure 3.13. These boilers supply hot water throughout all of the heating system in the building.

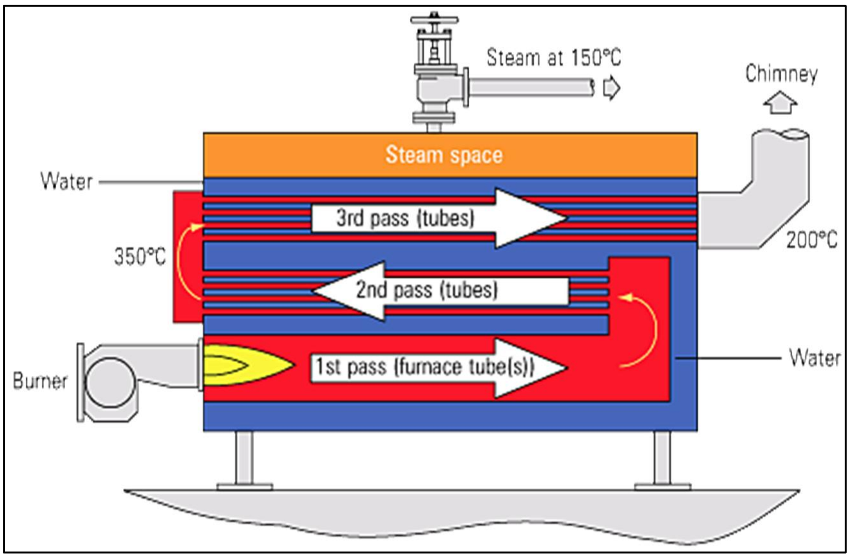


Figure 3.13 3 Pass hot water boilers [34]

There is a natural gas burner in the boiler. Naturel gas is burned in the combustion chamber and water in boiler is warmed up. The details of the boiler are given in the Table 3.1.

Table 3.1 Heating boiler technical data

HEATING BOILER			
Symbols	-	IK-01	IK-02
Location	-	Stokehold	Stokehold
Heating Capacity	Kw-h	1500	1500
Water Inlet-Outlet Temp.	C	90/70	90/70
Pressure	PN	6	6
Fuel Type	-	Naturel Gas	Naturel Gas
Efficiency	%	95	95
Material	-	Steel	Steel
Way of Working	-	Superior	Superior

The hot water is then transferred into the hot water line. The line has 3 zones (see in Figure 3.14);

- AHU zone line
- Fan coils zone line
- Exchanger zone line

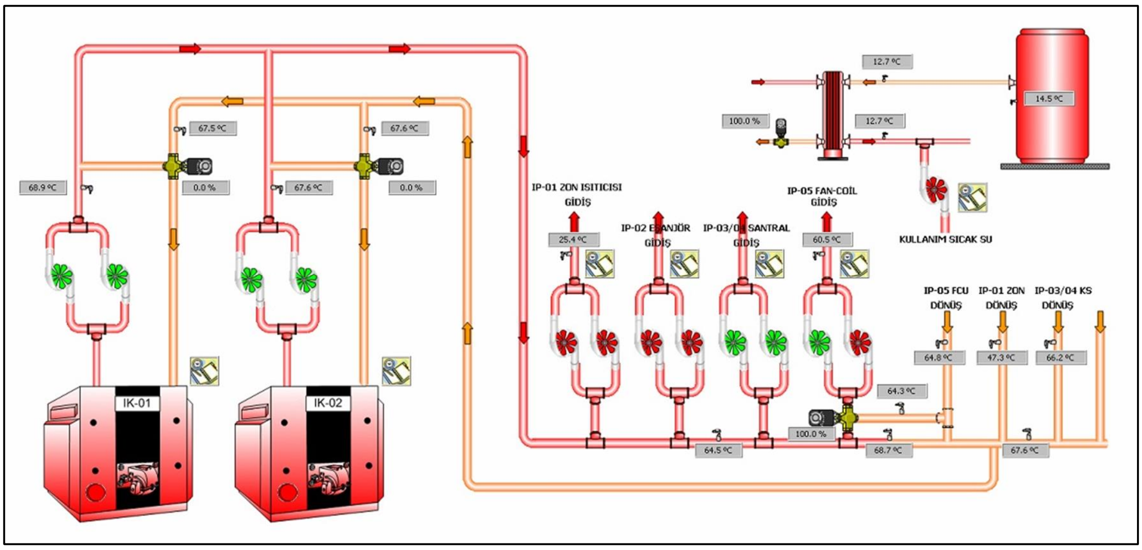


Figure 3.14 Boiler and hot water lines [35]

The boiler and heat exchanger zone line pictures and detailed hot water line project are given in respectively Figure 3.15 and Figure 3.16.

Figure 3.15 Boiler and heat exchanger zone line

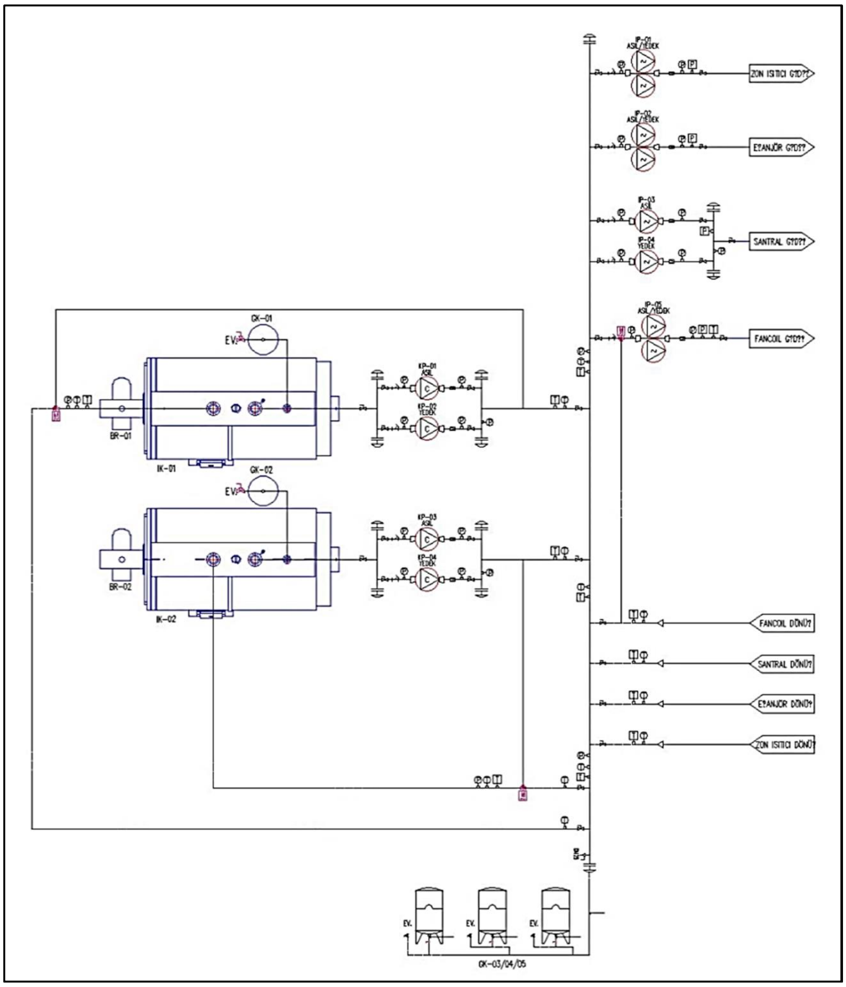


Figure 3.16 Detailed presentation of hot water line

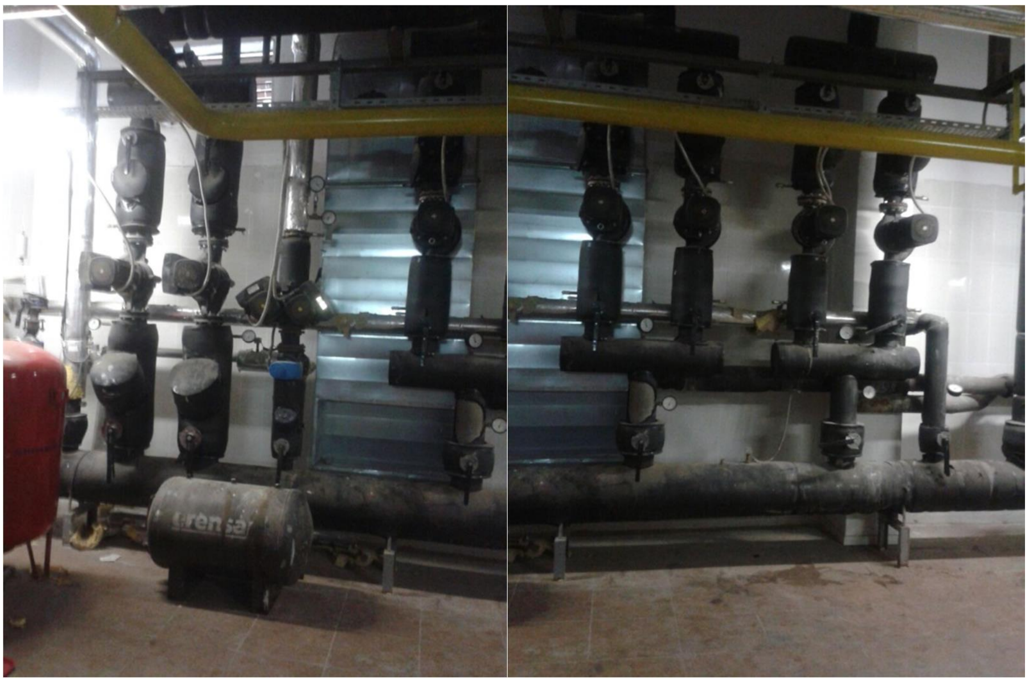


Figure 3.17 Hot water collector

3.4.2.1 AHU Zone Line

The hot water which comes from the boiler is separated into three lines from the collector. One of them is AHU zone line. This line has the biggest volume in hot water line, such that 100 m^3 hot water is circulated here in. There are 15 AHU systems in the building. One of them is heat recovery unit. There are heating coils in the AHU (see in Figure 3.18).

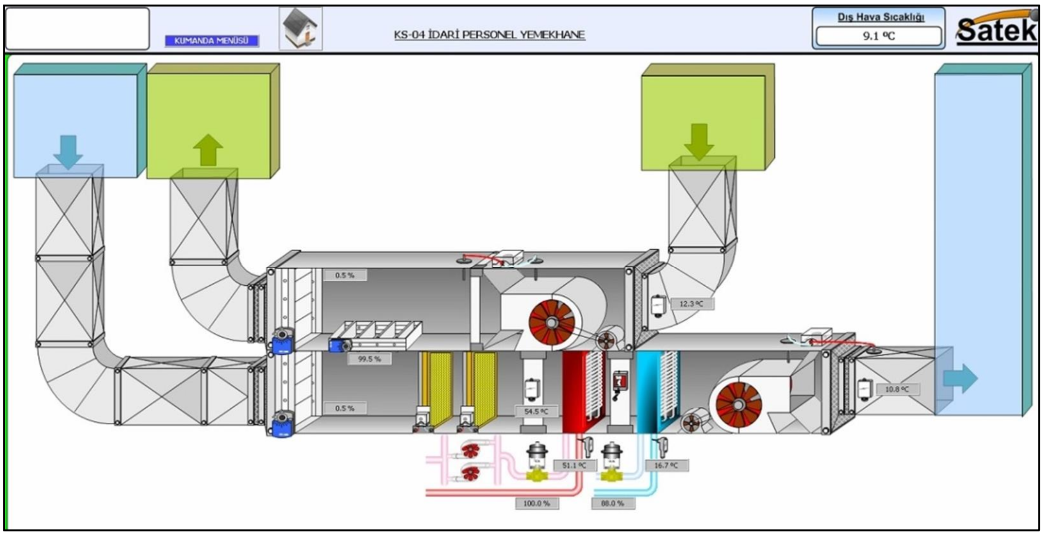


Figure 3.18 Details of AHU system [35]

The hot water inlet and outlet temperature from the heating coils are respectively 80 and 60 °C. When the water is passing through the coils, the air also passes from the coils. So the area is heated by the blowing air.

A typical AHU system connection schema is given in Figure 3.19 and Figure 3.20 respectively. In hot water line there are 2 freeze pumps. These pumps prevent the cracking of the coils in the cold days. Hot water inlet is settled by 3 way vessel.

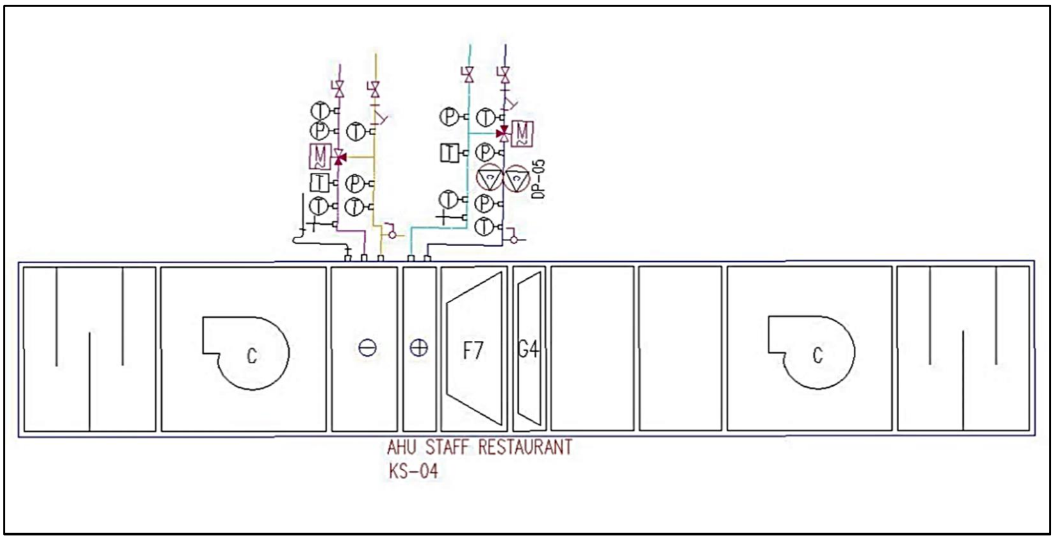


Figure 3.19 Connection schema of AHU coils

Figure 3.20 Connection of AHU units

3.4.2.2 Fan-Coil Zone Line:

There are totally 70 fan coils in the building which are generally used in the office rooms in the building. The fan-coil inlet and outlet temperatures are respectively 80-60 °C. Circulation water volume is 25 m^3/h in the system. Heating and cooling with fan coils are provided by the hot water which comes from the boiler and/or the chiller.

Working principle of fan coil is given in the Figure 3.21. The coil receives hot or cold water from a central plant, and removes heat from or adds heat to the air through heat transfer. Conventionally fan coil units can contain their own internal thermostat, or can be wired to operate with a remote thermostat. Fan coil units circulate hot or cold water through a coil in order to condition a space.

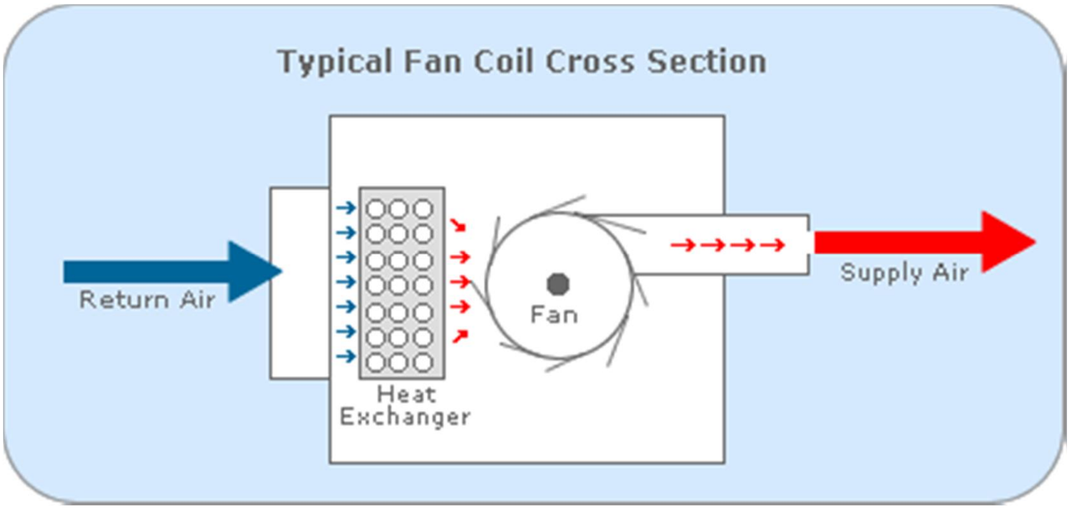


Figure 3.21 Working principle of fan coils

3.4.2.3 Exchanger Zone Line:

The plate type heat exchanger is used in the building for the utilization of water. 10 m3/h hot water from the boiler is circulated in the heat exchanger. The technical heat exchanger data is given in the Table 3.2.

Table 3.2 Heat exchanger technical data

HEAT EXCHANGER			
SYMBOLS		-	Ex-01
CAPACITY		KW-H	230
PRESSURE		PN	10
PASS.NUMBER		-	Single
PRIMER CIRCUIT	Water İnlet	°C	80
	Water Outlet	°C	60
	Max.Pressure Lost	kPa	50
SECONDER CIRCUIT	Water İnlet	°C	10
	Water Outlet	°C	60
	Max. Pressure Lost	kPa	10

The working principle of heat exchanger is shown in Figure 3.22

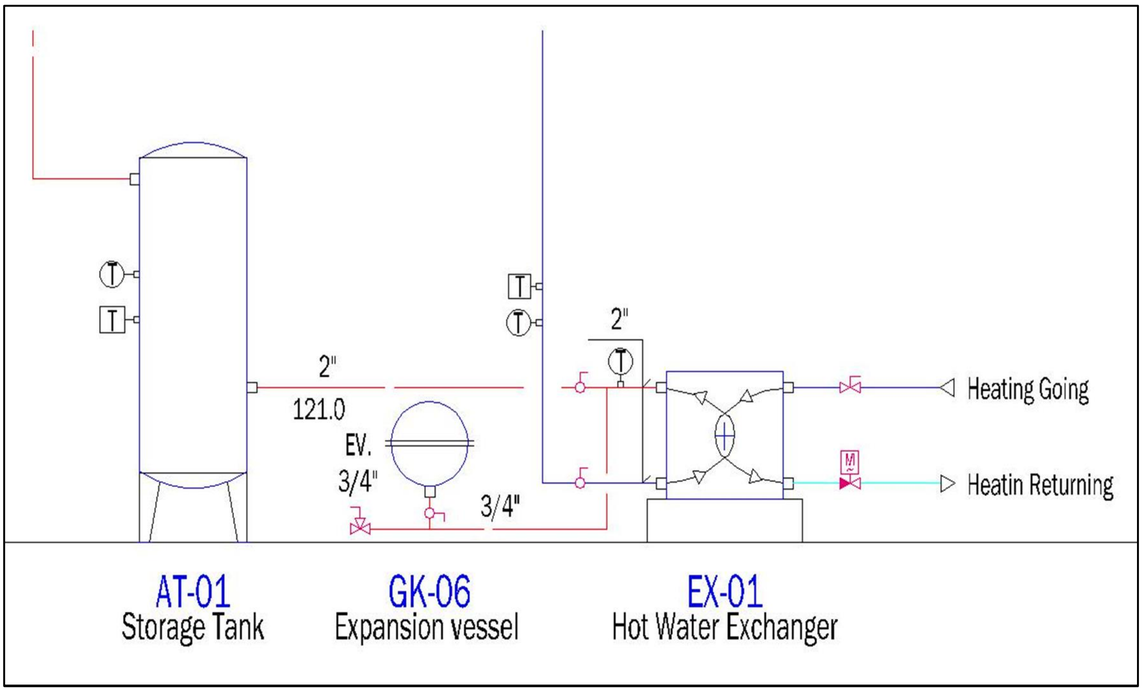


Figure 3.22 Working principle of heat exchanger

A heat exchanger is a piece of equipment built for efficient heat transfer from one medium to another. The media may be separated by a solid wall to prevent mixing or they may be in direct contact. There is a 2 way valve on the system (see in Figure 3.23). This valve is a device that regulates and controls the hot water by opening, closing, or partially obstructing various passageways.

In the system there is one epoxy insulated storage tank (accumulation tank) which has 1 m^3 volume which is used for the storage of the utilization water.

Figure 3.23 Heat exchanger and accumulation tank

3.4.3 Cooling Systems Used in Social and Cultural Building

The cooling is provided by the chiller unit in the building. There are 2 x 1000 KW screw compressor chillers. Both of them are the air cooled condenser units. So there is no need for cooling towers. 300 m^3/h of the chilled water is circulated in the cooling system. The type of refrigerant gas used in the system is 410 A. Evaporator inlets and outlet temperature is respectively 12 and 6 °C. The COP of chiller is 2.5. Outdoor dry bulb is 40 °C. The technical data of the chiller unit is given in Table 3.3

Table 3.3 Technical details of the chiller unit

AIR COOLED CHILLER			
SYMBOL		**CH-01**	**CH-02**
COOLING CAPACITY	Kw-r	1000	1000
EVAPORATOR OUTLET TEMP.	°C	6	6
EVAPORATOR INLET TEMP.	°C	12	12
REFRIGERANT	-	410 A	410 A
MIN EER (Energy Efficiency Ratio)	-	2.5	2.5
KOMPRESSOR NUM.	-	2	2
CAPACITY CONTROL	-	Proportional	Proportional
TOTAL ELECTRIC POWER	Kw-e	400	400
VOLTAGE	V	3x400	3x400
START UP	-	Soft Starter	Soft Starter

The working principle of the chiller unit is given in Figure 3.24. The chilled water is sent to the AHU system and the fan coil system. Basically the cooling and heating systems are similar. Chilled or hot water from the collector is sent to the coils and then with the aid of air, the area is cooled or heated.

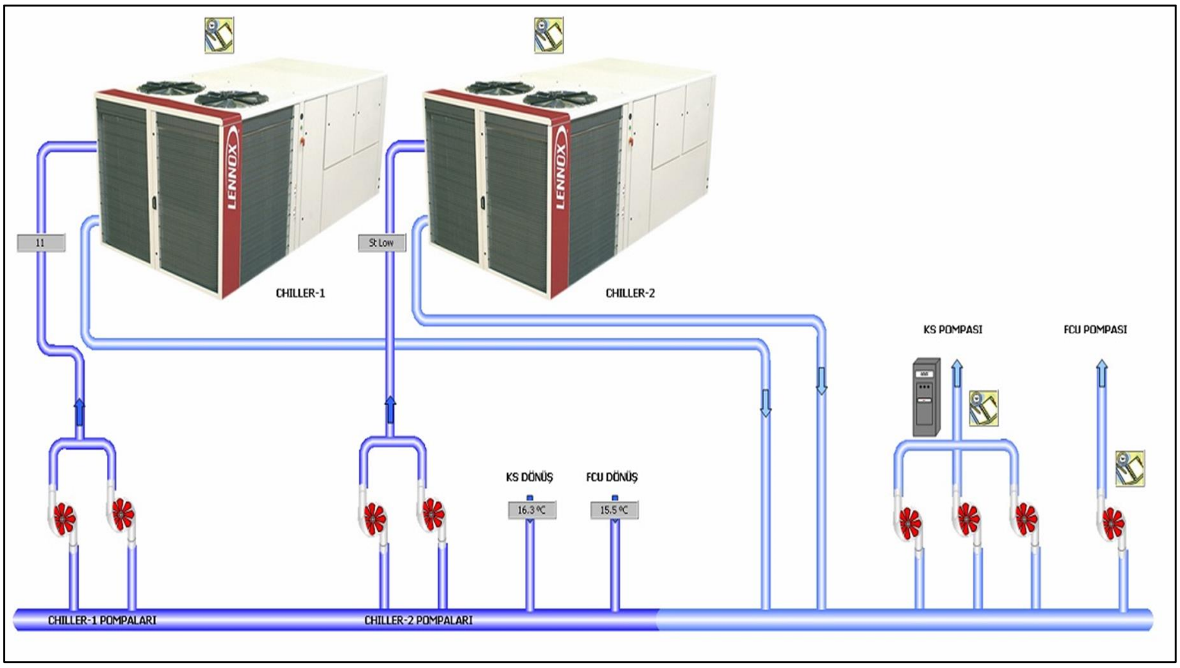


Figure 3.24 Chilled water cycle [35]

The chiller and technical drawing of the cooling circulation are shown in Figure 3.25 and Figure 3.26 respectively.

Figure 3.25 Air cooled water chiller

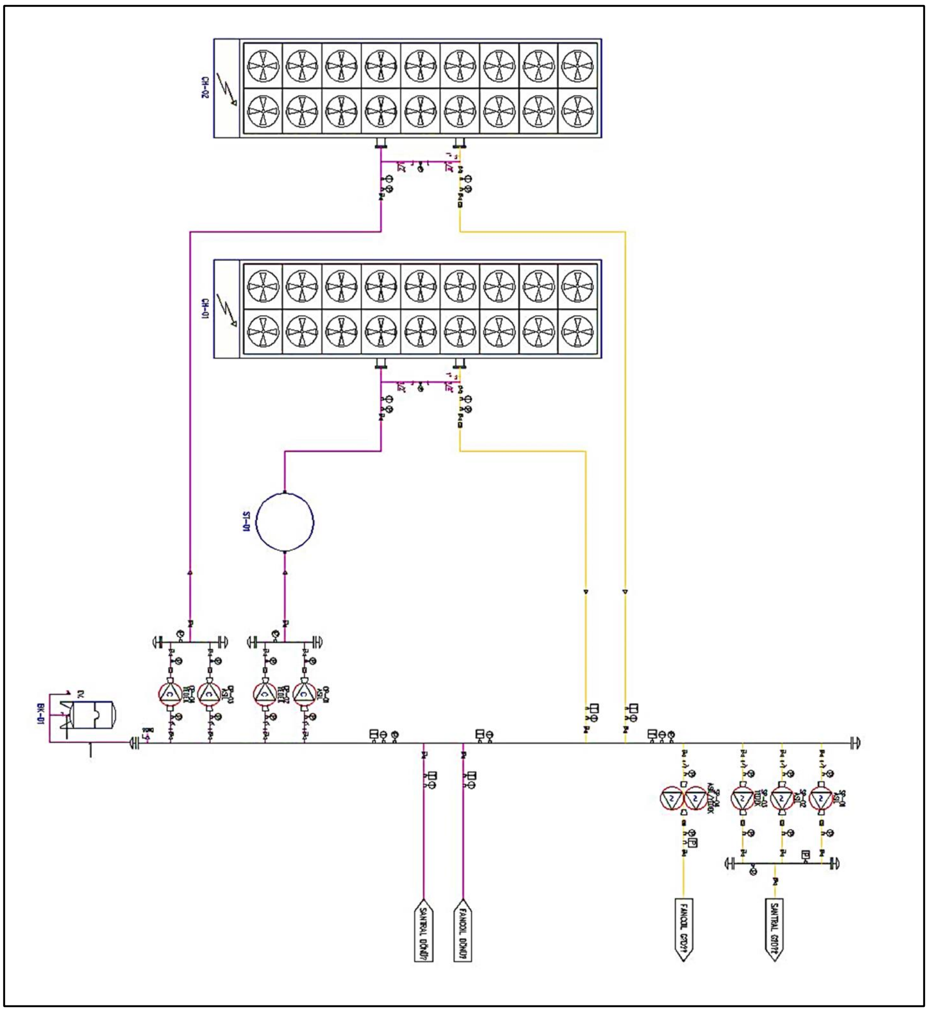


Figure 3.26 Detailed cooling processes

There is a storage tank before the chiller unit, in the cooling system. The storage tank is used to supply and storage much cold water in the cooling system. The storage tank prevents to chiller is often running on. The technical data of the storage tank is given in Table 3.4

Table 3.4 Chiller storage tank technical data

CHILLER STORAGE TANK			
SYMBOLS		-	ST-01
VOLUME		m^3	8
DIAMETER		mm	2000
HIGH		mm	2500
PRESSURE		PN	6
MATERIAL		-	Sheet Iron
INSULATION	**Material**	-	Rubber based flex
	Thickness	mm	40

3.4.4 VRF System Modeling in the Social and Cultural Building in Kilis 7 Aralik University

In this thesis, VRF system is modeled and analyzed to compare existing conventional HVAC systems Social and Cultural building. Variable Refrigerant Flow (VRF) systems have generally a large outdoor unit which serves the conditioned air to multiple indoor units. Each indoor unit uses an electronic liquid expansion valve to control its refrigerant supply to match the demand of the space it serves. The outdoor unit also varies its output to match the communal demands of the indoor units it serves. Thus, at any point in a system there will be a variable volume of refrigerant flowing. Various strategies are used to vary the output of the outdoor units (see in Figure 3.27).

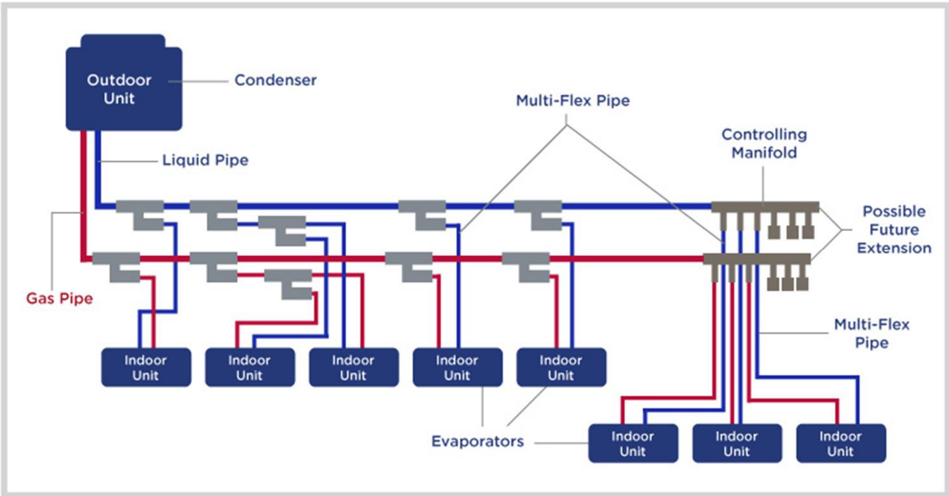


Figure 3.27 Heat pump VRF systems [33]

The chosen indoor unit capacities of the building are given in Tables 3.5

Table 3.5 VRF system indoor unit capacities

BASEMENT			
SYMBOLS	**LOCATION**	**NUMBER**	**CAPACITY**
BK 01	Kitchen	35	7.1 KW
BK 04	Ladies Locker Room	2	7.1 KW
GROUND FLOOR			
ZK 1	Foyer	17	7.1 KW
ZK 2	Refectory	36	7.1 KW
ZK 3	Red Tape	2	4.5 KW
ZK 4	Travel Agency	1	4.5 KW
ZK 5	Telephonist	1	4.5 KW
ZK 6	Shop	1	4.5 KW

ZK 7	Shop	1	4.5 KW
ZK 8	Shop	1	4.5 KW
ZK 9	Hairdresser	1	4.5 KW
ZK 10	Dry Cleaning	2	4.5 KW
ZK 11	Internet Cafe	2	7.1 KW
ZK11-17	Others	-	-
ZK18	Cafeteria	17	7.1 KW
1. FLOOR			
K 101	Floor Hall	15	7.1 KW
K 102	Administrative Staff Cafeteria	18	7.1 KW
K 103	Academics Cafeteria	18	7.1 KW
K 104	Workshop 1	2	7.1 KW
K 105	Workshop 2	2	7.1 KW
K 106	Workshop 3	2	7.1 KW
K 107	Computer Room	15	7.1 KW
K 107-116		-	-
K 116	Cinema	3	7.1 KW
K 115	Cinema Foyer	3	7.1 KW
K 114	Cinema	3	7.1 KW
2. FLOOR			
K 201	Art Gallery	22	7.1 KW
K 202	Multi-Purpose Hall	24	7.1 KW
K 203	Office	2	7.1 KW
K 204	Directorôs Room	1	4.5 KW
K 205	Secretary	1	5.6 KW
K 206	Office	2	4.5 KW
K 207	Directorôs Room	1	5.6 KW
K 208	Directorôs Room	1	5.6 KW
K 209	Archive	2	4.5 KW
K 210	Office	2	5.6 KW
K 210-216	-	-	-
K 216	Doctorôs Room	2	7.1 KW
K 217	Doctorôs Room	1	5.6 KW
K 218	Doctorôs Room	1	5.6 KW
K 219	Nurse Room	1	5.6 KW
K 220	Clinic	1	5.6 KW
K 221	Clinic	1	5.6 KW
K 222	Clinic	1	5.6 KW
K 223	Archive	2	4.5 KW
K 224	Archive	2	5.6 KW
3. FLOOR			
K 301	Floor Hall	18	7.1 KW
K 302	Rest Area	2	3.6 KW
K 302-1	Room 1	2	4.5 KW
K 304	Room 2	1	3.6 KW
K 306	Room 3	1	4.5 KW
K 308	Room 4	1	4.5 KW
K 310	Room 5	1	4.5 KW

K 312	Room 6	1	4.5 KW
K 314	Room 7	1	4.5 KW
K 316	Room 8	1	4.5 KW
K 318	Room 9	1	3.6 KW
K 320	Room 10	1	4.5 KW
K 322	Room 11	2	6.4 KW
K 324	Service	1	3.6 KW
K 328	Auditorium	8	7.1 KW
K 329	Auditorium	6	7.1 KW

The technical representation of the VRF system used in the social & Cultural building is shown in Figure 3.28. As can be seen from the figure, 81 indoor units are used in ground floor.

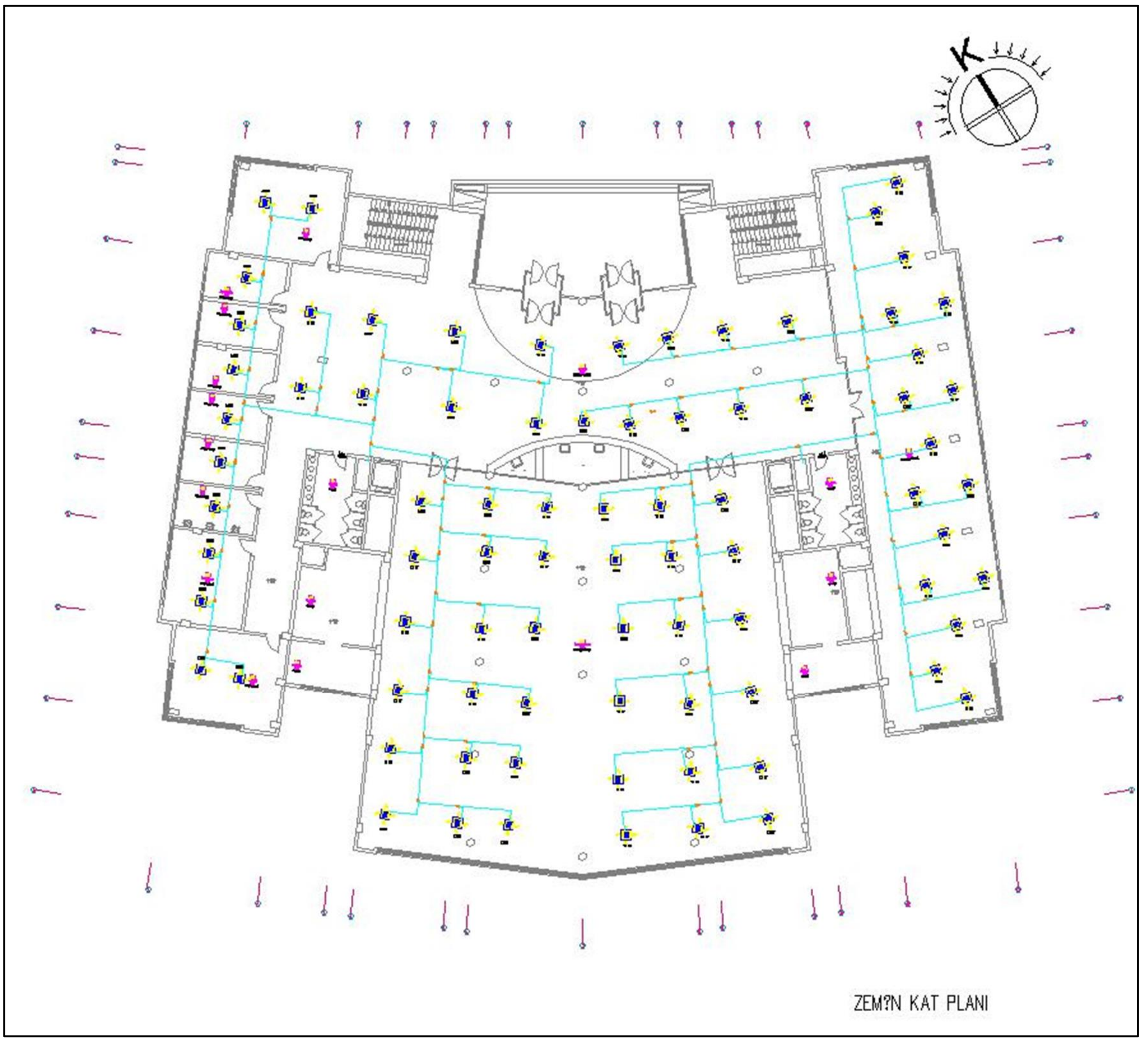


Figure 3.28 VRF systems project in Social and Cultural building

3.5 Conclusion

In this chapter, the existing conventional HVAC system utilized in the Social and Cultural building in the University is introduced with its all components. The conventional HVAC system is composed of fan coils and AHU units. In this chapter, also, a VRF system is modeled for the Social and Cultural building to compare it with the existing conventional HVAC system by analyzing in the next chapters.

CHAPTER 4

HVAC ANALYSES OF THE EXISTING BUILDING IN KILIS 7 ARALIK UNIVERSITY

4.1 Introduction

In this chapter, the Social and Cultural building in Kilis 7 Aralik University is analyzed in terms of HVAC application considering the heating and cooling capacity. At this time, TS 825 standards are used for analyses. The total heating and cooling loads for the existing building are evaluated. Utilizing the insulation concept by considering heat and mass (vapor diffusion) transfers.

4.2.1 Evaluation of Heat Transfer Coefficient for the Building

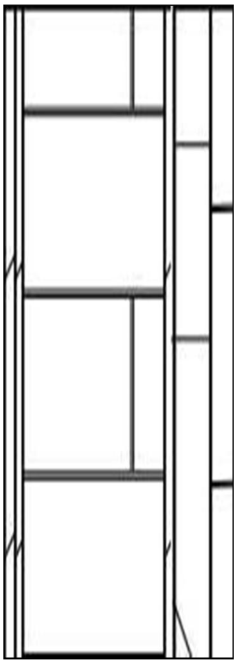
In this section, all building structures with their technical specifications are tabulated giving the structure dimensions, thermal conductivity and their resistance of each structure in the building as shown in Table 4.1.

The HVAC analyses of the building is performed considering the structure components of external curtain wall, ground basement, sandwich panel roof and uncovered ceiling. In this section, the external curtain wall as a component of the building is analyzed in terms of HVAC application.

As can be seen in Table 4.1, the external curtain wall is constructed with many subcomponents. As an evaluation, it is found that the total overall heat transfer coefficient of external curtain wall is about 0.347 W/m^2K. The technical specifications of the external curtain wall can be seen in Table 4.1.

Throughout this chapter, all thermal resistance and overall heat transfer coefficients are evaluated by using the Equations (4.1) and (4.2) [36].

Table 4.1 Thermal conductivity coefficients and thickness of external curtain wall



Materials	δ(m)	λ(W/mK)	$R(m^2K/W)$
Internal Surface Thermal Conductivity Resistance			0.130
Plaster	0.015	0.700	0.021
Cement	0.010	1.600	0.006
Brick Wall	0.190	0.190	1
Cement	0.010	1.600	0.006
Extruded Polystyrene Foam	0.050	0.035	1.429
Igneous Rocks	0.030	2.300	0.013
External Surface Thermal Conductivity Resistance			0.040
$\mathbf{R_{TOTAL}}$	$2.883\ m^2K/W$		
$\mathbf{U_{TOTAL}}$	$0.347\ W/m^2K$		

$$\frac{1}{U}=\frac{1}{\alpha_i}+\frac{\delta_1}{\lambda_1}+\frac{\delta_2}{\lambda_2}+\frac{\delta_3}{\lambda_3}+\frac{\delta_4}{\lambda_4}+\frac{\delta_5}{\lambda_5}+\frac{\delta_6}{\lambda_6}+\frac{1}{\alpha_0} \tag{4.1}$$

$$\frac{1}{U_{CW}}=\frac{1}{0.13}+\frac{0.015}{0.70}+\frac{0.01}{1.60}+\frac{0.19}{0.19}+\frac{0.01}{1.60}+\frac{0.05}{0.035}+\frac{0.03}{2.30}+\frac{1}{0.04} \tag{4.2}$$

Thermal resistance is;

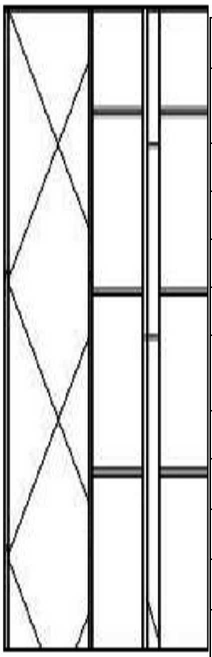
$R = 1/U = 2.883\ m^2K/W$

Overall heat transfer coefficient is;

$U_{CW} = 0.347\ W/m^2K$

The other part of the building where heat transfer takes place is the external wall which contacts with the ground. The sub-structure of the external wall which contacts with the ground is shown in Table 4.2. As can be seen from the table, the total thermal resistance is found as $2.279\ m^2K/W$.

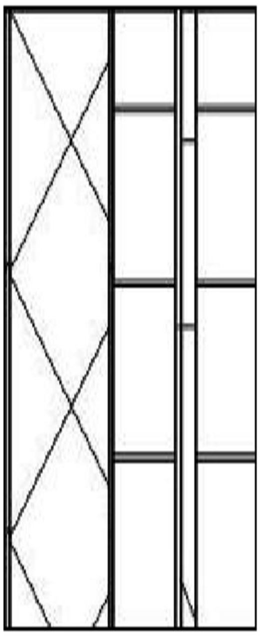
Table 4.2 Thermal conductivity coefficients and thickness of external wall which contacts with the ground [36]



Material	**δ(m)**	**λ(W/mK)**	**R(m²K/W)**
Internal Surface Thermal Conductivity Resistance			0.130
Plaster	0.015	0.70	0.021
Reinforced Concrete	0.300	2.50	0.120
Cement	0.010	1.60	0.006
Brick Wall (Vert. Hole Cli.)	0.190	1.40	0.136
Polymer Waterproofing Membrane	0.012	0.19	0.063
Extruded Polystyrene Foam	0.050	0.03	1.667
Brick Wall(Vert. Hole Clin.)	0.190	1.40	0.136
Ex.Surf. Thermal Cond. Res.			0.00
$\mathbf{R_{TOTAL}}$	2.279 m^2K/W		
$\mathbf{U_{TOTAL}}$	0.439 W/m^2K		

The other major component of the Social and Cultural building is ground basement. It is composed of many layers as can be seen from the schematic representation in Table 4.3 and the thermal resistance for all sub components is given. The total thermal resistance and overall heat transfer coefficient are evaluated as 2.137 m^2K/W and 0.468 W/m^2K, respectively.

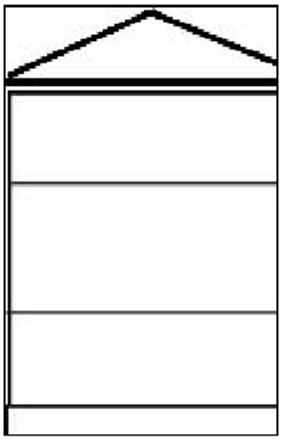
Table 4.3 Thermal conductivity coefficients and thickness of the ground basement



Material	**δ(m)**	**λ(W/mK)**	**R(m²K/W)**
Internal Surface Thermal Conductivity Resistance			0.170
Reinforced Concrete	0.170	2.50	0.068
Aggregate Unreinforced Concrete	0.050	0.44	0.114
Polymer Waterproofing Membrane	0.012	0.19	0.063
Polyisobutylene Cover	0.003	0.26	0.012
Cement Mortar Screed	0.025	1.40	0.018
Extruded Polystyrene Foam	0.050	0.03	1.667
Cement Mortar Screed	0.025	1.40	0.018
Granite	0.020	2.80	0.007
External Surface Thermal Conductivity Resistance			0.00
$\mathbf{R_{TOTAL}}$	2.137 m^2K/W		
$\mathbf{U_{TOTAL}}$	0.468 W/m^2K		

At the top of the Social and Cultural building, there is a sandwich panel roof as a structure. At the roof, 4 major structures are used for thermal and moisture insulation. In Table 4.4, the technical specifications of the components of the sandwich panel roof are given. As can be understood from the tablet the total thermal resistance and the overall heat transfer coefficient are found as 2.388 m^2K/W and 0.419 W/m^2K, respectively.

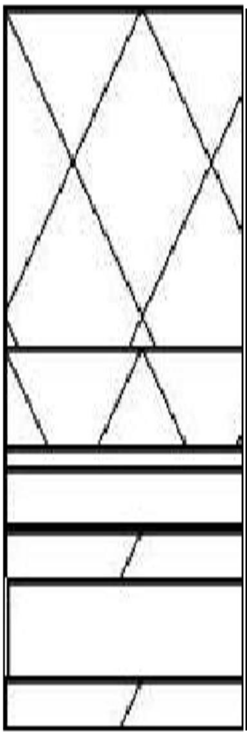
Table 4.4 Thermal conductivity coefficients and thickness of the sandwich panel roof



Material	**δ(m)**	**λ(W/mK)**	**R(m^2K/W)**
Int.Surf. Thermal Cond. Res.			0.13
Polymer Waterproof.Membr.	0.012	0.19	0.063
Polyisobutylene Cover	0.003	0.26	0.012
Stone Wool Insulation Panel	0.075	0.035	2.143
Ex.Surf. Thermal Cond. Res.			0.04
$\mathbf{R_{TOTAL}}$	2.388 m^2K/W		
$\mathbf{U_{TOTAL}}$	0.419 W/m^2K		

The major part of the roof of the building is uncovered. In Table 4.5 the sub-structures of the uncovered ceiling are shown with their technical specifications.

Table 4.5 Thermal conductivity coefficients and thickness of the uncovered ceiling



Material	**δ(m)**	**λ(W/mK)**	**R(m^2K/W)**
Internal Surf. Ther.Cond.Res.			0.13
Reinforced Concrete	0.17	2.5	0.068
Aggregate Unreinforced Concrete	0.05	0.44	0.114
Mastic Asphalt Covering	0.01	0.7	0.014
Polymer Waterpr.Membrane	0.012	0.19	0.063
Polyisobutylene Cover	0.003	0.26	0.012
Cement Mortar Screed	0.025	1.4	0.018
Extruded Polystyrene Foam	0.05	0.03	1.667
Cement Mortar Screed	0.025	1.4	0.018
Ext. Surf. Thermal Cond. Res.			0.04
$\mathbf{R_{TOTAL}}$	2.144 m^2K/W		
$\mathbf{U_{TOTAL}}$	0.466 W/m^2K		

Although the area of the uncovered ceiling is (1196 m^2) twice times of the area of sandwich panel roof (478 m^2), the thermal resistance of the uncovered ceiling is almost the same with flat of the sandwich panel roof. As can be understood from this statement, the technical property of the insulation materials used in uncovered ceiling are very similar to those used in the sandwich panel roof. The overall heat transfer coefficient of the windows and doors are given directly in Table 4.6.

After the evaluation of overall heat transfer coefficients of each main structure, specific heat losses by conduction, infiltration and heat gains are evaluated considering the effects of in terms of TS 825 standards.

In this part, the overall heat loss through the building structure is considered and the evaluation results are tabulated in Table 4.6.

As can be seen from Table 4.6, the most heat loss occurs from the glass surface. Because all surfaces of the building are covered by glass material. The magnitudes of heat loss for all building structures are evaluated using the Eq. (4.3)

$$q = U \times A \tag{4.3}$$

Table 4.6 Heat loss evaluation of all building structures [36-37]

Location	**Area (m^2)**	**Heat Transfer Coefficient (W/m^2 K)**	**Formula**	**Total Heat Loss (q)**
Heat Loss From the External Curtain Wall	1859	0.347	q = U x A	645.1 W/K
Heat Loss From External Wall Which Contacting the Soil	626	0.439	q = U x A	274.8 W/K
Heat Loss From Ground Engaging Base	1996	0.468	q = U x A	934.1 W/K
Heat Loss From Sandwich Panel Roof	478	0.419	q = U x A	200.3 W/K
Heat Loss From Uncovered Ceiling	1196	0.466	q = U x A	557.3 W/K
Heat Loss From Exterior Doors	55	5.5	q = U x A	302.5 W/K
Heat Loss From Glass Surface	1664	3	q = U x A	4992 W/K
$H_C = \Sigma$ AU Total Heat Loss by Conduction	$U_{CW}A_{CW} + U_{EW}.A_{EW} + 0.8\ U_{SR}.A_{SR} + 0.8\ U_{UC}.A_{UC} + 0.5\ U_{GB}.A_{GB} + U_d.A_d + U_W.A_W$			7301.7 W/K

Then, the overall total heat loss through the building by conduction is evaluated by using Eq. (4.4). In Eq.(4.4) same coefficients such as 0.8 and 0.5 are used according to TS 825 standards, if the uncovered ceiling directly contacts with the outside air, the coefficient before U_{SR} is taken 1 instead of 0.8

$$\Sigma AU = U_{CW}A_{CW} + U_{EW}.A_{EW} + 0.8U_{SR}.A_{SR} + 0.8U_{UC}.A_{UC} + 0.5U_{GB}.A_{GB} + U_d.A_d + U_W.A_W \quad (4.4)$$

4.2.2 Evaluation of Heat Transfer Due to Ventilation

Besides, heat transfer by conduction due to building structure, there is also heat transfer by means of the ventilation of the building. In this section, the heat loss due to natural air ventilation is given by using the Eq. (4.5) through Eq. (4.9)

Heat loss occurs through the natural air ventilation in buildings [36-37];

$$H_i = 0.33 \times n_h \times V_h \quad (4.5)$$

$$V_h = 0.8 \times V_{brüt} \quad (4.6)$$

$$V_h = 0.8 \times 43706$$

$$V_h = 34964 \text{ m}^3$$

$n_h = 0.8$ (The number of air changes)

$$H_i = 0.33 \times 0.8 \times 34964$$

$$H_i = 9230.1 \text{ W/K}$$

The specific heat loss is;

$$H = H_c + H_i \quad (4.7)$$

$$H = 7301.7 + 9230.1$$

$$H = 16531.8 \text{ W/K}$$

4.2.3 Evaluation of Heat Transfer Monthly Due to Domestic Benefits

Internal gains include the following:

- Metabolic heat gain caused by people,
- Heat gain from hot water systems,
- Heat gain resulting from the cooking process
- Heat gain resulting from the lighting system
- Heat gain resulting from electrical appliances used in the building

Inside the building, the estimated average monthly heat gain due to domestic benefits is evaluated the Eq. (4.7) through Eq. (4.8).

$$\Phi_{i\ ay} = A_n \times 5 \tag{4.8}$$

Building area of use is;

$$A_n = V_{br} \times 0.32 \tag{4.9}$$

$$A_n = 43706 \times 0.32$$

$$A_n = 13985.92\ m^2$$

$$\Phi_{i\ ay} = A_n \times 5$$

$$\Phi_{i\ ay} = 69929.6$$

4.2.4 Evaluation of Heat Transfer Monthly Due to Solar Energy

There is also heat transfer to the building due to solar radiation through glass surface of the building. The calculation of heat transfer due to solar gain is given by Eq. (4.10). The gains from passive solar energy system have been neglected [36].

$$\Phi_{g\ month} = \Sigma\ r_{i\ month} \times g_{i\ month} \times I_{i\ month} \times A_i \tag{4.10}$$

The coefficient is selected as;

$r_{i\ month}$ = Shading Factor 0.8 (separate and low-rise buildings)
$g_{i\ month}$ = Solar Passing Factor 0.5 (colored glazing)

Monthly average solar radiation intensity from the vertical surface ($I_{i\ month}$) is selected from the table;

$I_{South,\ january} = 72\ W/m^2$, $I_{North,january} = 26\ W/m^2$ $I_{East/West,january} = 43\ W/m^2$

Windows surface area

$A_{south} = 934\ m^2$
$A_{north} = 208\ m^2$
$A_{east} = 208\ m^2$
$A_{west} = 314\ m^2$

Table 4.7 gives the annual solar gain through the glass surface of the building. The most heat gain is seen in June.

Table 4.7 Annual solar gain [36]

January	February	March	April	May	June
30433 W	37090 W	42326 W	46326 W	51798 W	54297 W
July	**August**	**September**	**October**	**November**	**December**
52898 W	50361 W	43924 W	37026 W	28971 W	26773 W

4.2.5 Evaluation of Monthly Heat Gain-Lost Ratio

For the evaluation of permitted annual heating energy requirement (Qô), monthly gain-lost ratio (KKO_{month}), gain utilization factor (η_{month}) and monthly total heating energy needs (Q_{month}) must be evaluated.

$$KKO_{month} = (\Phi_{g\ month} + \Phi_{i\ month}) / [H \times (T_{i\ month}\ ï\ T_{d\ month})] \quad (4.11)$$

If the KKO_{month} ratio is equal or over the 2.5 we accept there is no heat lost in the system. In this eqn;

$T_{i,\ month}$: For houses and offices 19 °C is acceptable.

In Table 4.8, the monthly heat gain/loss ratio is given from January to December.

Table 4.8 Monthly heat gain-loss ratio

Months	$T_{i\ month}$ ï $T_{d\ month}$	KKO
January	16.1	0.38
February	14.6	0.44
March	11.7	0.58
April	6.2	1.13
May	1.0	7.36
June	High	0.00
July	High	0.00
August	High	0.00
September	High	0.00
October	4.9	1.32
November	10.5	0.57
December	15.2	0.38

4.2.6 Evaluation of Heat Gain Utilization Factor

Monthly average heat gain is calculated using the following eqn.

$$\eta_{month} = 1\text{-}e^{-1/KKO_{month}} \tag{4.12}$$

Table 4.9 gives the average heat gain utilization factor monthly.

Table 4.9 Gain Utilization Factor

Months	η_{month}
January	0.93
February	0.90
March	0.82
April	0.59
May	0.13
June	High
July	High
August	High
September	High
October	0.53
November	0.83
December	0.93

4.2.7 Evaluation of Monthly Total Heating Energy Needs (Q_{month})

After evaluation of annual heating energy requirement, monthly heat gain/loss ratio and heat gain utilization factor, the monthly total heating energy need is calculated by using Eq. (4.13). The evaluated values for each month are tabulated in Table 4.10. The annual heating energy need for the Social and Cultural building is found as 526458 kWh.

$$Q_{year} = [(H \times (T_{i\,ay}-T_{d\,ay}) - \eta_{ay} \times (\Phi_{i\,ay}-\Phi_{g\,ay}))\ ] \times t \tag{4.13}$$

Then the maximum permissible annual heating energy need is evaluated and found as 12300 kWh/m^3 according to TS 825.

Because of the room length > 2.6 m. volumetric calculation method is used.

$$Q = Q_{year}/V_{brut}\ (kWh/\ m^3) \tag{4.14}$$

$$Q = 526458kWh\ /\ 43706\ m^3$$

$$Q = 12.045\ kWh/m^3$$

The comparison of evaluated annual heating energy needs (12.045 kWh/m^3) and the maximum permissible annual heating energy need (12.300 kWh/m^3) shows that the insulation of the Social and Cultural building is acceptable as shown followings:
The maximum permissible annual heating energy needs Qô=12.300 kWh/m^3

Q<Qô(12.045 kWh/m^3 < 12.300 kWh/m^3) insulation is acceptable [42, 43, 45].

Table 4.10 Annual heating energy needs [36]

Months	**Heat Loss**			**Heat Gain**					
	Specific Heat Loss	***Temperature Difference***	***Heat Loss***	***Internal Heat gain***	***Solar Energy Gain***	***Total***	***KKO***	***Gain Utilization Factor***	***Heating Energy Needs***
	$H=H_i+H_h$ (W/K)	T_i-T_d (K,°C)	$H(T_i- T_d)$ (W)	ϕ_i (W)	ϕ_g (W)	$\phi_T=\phi_i+\phi_g$	γ (-)	η_{month} (-)	Q_{month} (kJ)
January	16532	16.1	266171	69929	30433	100363	0.38	0.93	447984561
February		14.6	241372		37090	107020	0.44	0.90	375981497
March		11.7	193428		42326	112256	0.58	0.82	262773849
April		6.2	102500		46326	116256	1.13	0.59	0
May		1.0	16532		51798	121728	7.36	0.13	0
June		High	0		54297	124227	0.00	High	0
July		High	0		52898	122828	0.00	High	0
August		High	0		50361	120291	0.00	High	0
September		High	0		43924	113854	0.00	High	0
October		4.9	81008		37026	106956	1.32	0.53	63042442
November		10.5	173589		28671	98601	0.57	0.83	237818695
December		15.2	251292		26773	96703	0.38	0.93	418240506
$Q_{month} = [H (T_i - T_d) - \eta (\phi_{i, ay} + \phi_{g, ay})]. t$ (J) (1k J = 0.278×10^{-3} kWh)							$Q_{year} = \Sigma Q_{month}$ = 1893735365 kj = 526458 kWh		

4.2 Evaluation of Mass Transfer Through the Social and Cultural Building

In previous section evaluations for heat transfer through the building is performed. In this section, the evaluations on the vapor diffusion mass transfer through the building are introduced. If the vapor diffusion occurs, the components should be changed with new one.

Between two sides of a construction element, the different vapor pressures arising due to the fact that the temperatures and relative humidity are different happen. When we consider the winter season which is heating period, generally there is high vapor pressure on the internal side and the water vapor being available in the form of gas in the internal environment moves in the same direction with the heat flow and tries to reach to the external environment. In case the water vapor reaches to the external environment as gas, there is no problem in terms of both the usage life and thermal performance of the construction element. However, depending on the resistance that the materials forming the construction element show to the transition of the water vapor and sequence of the materials, the possibility that the water vapor is changed into the liquid form from the gas form while passing through the construction element, namely its condensation is available [45].

4.3.1 Vapor Diffusion for External Curtain Wall

The technical data is given about the external curtain wall components. Water vapor diffusion resistance factor (μ), equivalent air layer thickness (Sd), thermal conductivity **(** λ_h) and thermal resistance (R) are seen in Table 4.11.

Table 4.11 Equivalent air layer thickness and thermal resistance due to vapor diffusion for external curtain wall [36]

Material	**δ(m)**	**μ**	**Sd (m)**	**Sd (Total)**	**Thermal conduct. (λ_h)**	**R (m^2K/W)**	**R_T (m^2K/W)**
External Surface Thermal Conductivity Resistance			0.00	0.00		0.04	0.04
Igneous Rocks	0.03	250	7.50	7.50	2.30	0.01	0.05
Extruded Polystyrene Foam	0.05	80	4.00	11.50	0.03	1.67	1.72
Cement	0.01	15	0.15	11.65	1.60	0.01	1.73

Brick Wall	0.19	5	0.95	12.60	0.19	1.00	2.73
Cement	0.01	15	0.15	12.75	1.60	0.01	2.74
Plaster	0.015	10	0.15	12.90	0.70	0.02	2.76
Internal Surface Thermal Conductivity Resistance			0.00	12.90		0.25	3.01
			Sdt	12.90		1/U	3.01
						U	0.332 W/ m^2K

Equivalent air layer thickness is calculated using the following Eqn;

$$S_d = \mu \times \delta \tag{4.15}$$

Total equivalent air layer thickness is calculated as follows;

$$s^{I}_{d,T} = \sum_{j=1}^{N} s_{d,j} \tag{4.16}$$

$S_{d,\,T} = 3.01\ m^2K/W$

The equivalent air layer thickness is used for evaluation of condensation in building components. In Table 4.12 indoor and outdoor temperature and humidity ratio in Kilis are given.

Table 4.12 Indoor unit temperature and the humidity value in Kilis

	Outdoor Unit Temp.(°C)	**Outdoor Unit Humidity Ratio (%)**	**Indoor Unit Temp. (°C)**	**Indoor Unit Humidity Ratio (%)**
November	8.5	0.60	20	65
December	3.8	0.70		
January	2.9	0.68		
February	4.4	0.64		
March	7.3	0.61		
April	12.8	0.57		
May	18.0	0.49		
June	22.5	0.46		
July	24.9	0.50		
August	24.3	0.51		
September	19.9	0.48		
October	14.1	0.49		

First the total heat transfer from the components, after indoor and outdoor surface temperature should be calculated. For January, from the Table 4.11 the 2. area outdoor temperature and indoor unit temperature is respectively taken 2.9 °C and 20 °C.

Total heat loss [26, 47];

$$q = U.(\theta_i - \theta_e) \tag{4.17}$$

$$q = 5.6772\ W/m^2$$

The specific heat transfer is found to be 5.6772 W/m^2.

$$\theta_{yi} = \theta_i - R_i.q \tag{4.18}$$

$$\theta_{yd} = \theta_d + R_e.q \tag{4.19}$$

$$\theta_{yi} = 18.6°C$$

According to the temperature in the intermediate surface, the water vapor saturation pressures are obtained from Table 4.13. According to the vapor diffusion equivalent air layer thickness Sd of each layer, the section of the construction element is drawn. The saturated vapor pressures at each intermediate surface between the materials are drawn by being combined with straight lines. The month of January, May and July condensation graphics are given in Figure 4.1, Figure 4.2 and Figure 4.3 respectively. In the graphics, the red and bold line indicates the saturated vapor pressure and the green and light line shows the water vapor partial pressure. The intersection of the both lines indicates the point at which condensation begins. If this line does not exceed the saturation pressure in any intermediate surface, the condensation does not happen [44-45].

As can be seen from the Figure 4.3, in summer months, the lines do not intersect. This means that there is no condensation occurred in the summer.

Table 4.13 Surface temp and saturated water vapor pressure of external curtain wall

	January		**February**		**March**		**April**		**May**		**June**	
	°C	**Pa**	**°C**	**Pa**	**°C**	**Pa**	**°C**	**Pa**	**°C**	**Pa**	**°C**	**Pa**
Ambient	2.90	753	4.40	837	7.30	1023	12.8	1479	18.0	2065	22.5	2727
Outer Surf.	3.13	765	4.61	849	7.47	1038	12.9	1488	18.0	2065	22.4	2727
1.Interface	3.10	765	4.50	843	7.60	1045	13.0	1498	18.0	2065	22.4	2711
2.Interface	12.6	1460	13.2	1518	14.6	1663	17.0	1937	19.1	2212	21.0	2487
3.Interface	12.7	1470	13.3	1528	14.6	1663	17.0	1937	19.1	2212	21.0	2487

4.Interface	18.4	2119	18.5	2132	18.8	2172	19.4	2254	19.8	2310	20.2	2369
5.Interface	18.5	2132	18.6	2145	18.8	2172	19.4	2254	19.8	2310	20.2	2369
Inner Surf.	18.6	2145	18.7	2158	18.9	2185	19.4	2254	19.8	2310	20.2	2369
Indoor	20.0	2340	20.0	2340	20.0	2340	20.0	2340	20.0	2340	20.0	2340
	July		**August**		**September**		**October**		**November**		**December**	
	°C	**Pa**	**°C**	**Pa**	**°C**	**Pa**	**°C**	**Pa**	**°C**	**Pa**	**°C**	**Pa**
Ambient	24.9	3151	24.3	3040	19.9	2324	14.1	1610	8.5	1110	3.8	803
Outer Surf.	24.8	3132	24.2	3021	19.9	2324	14.1	1621	8.65	1125	4.02	813
1.Interface	24.7	3114	24.2	3021	19.9	2324	14.2	1621	8.7	1125	4.0	813
2.Interface	22.0	2645	21.8	2613	20.0	2340	17.5	2001	15.1	1717	13.0	1498
3.Interface	22.0	2645	21.8	2613	20.0	2340	17.5	2001	15.1	1717	13.0	1508
4.Interface	20.4	2399	20.4	2399	20.0	2340	19.5	2268	18.9	2185	18.5	2132
5.Interface	20.4	2399	20.4	2399	20.0	2340	19.5	2268	18.9	2185	18.6	2145
Inner Surf.	20.4	2399	20.4	2399	20.0	2340	19.5	2268	19.0	2197	18.7	2158
Indoor	20.0	2340	20.0	2340	20.0	2340	20.0	2340	20.0	2340	20.0	2340

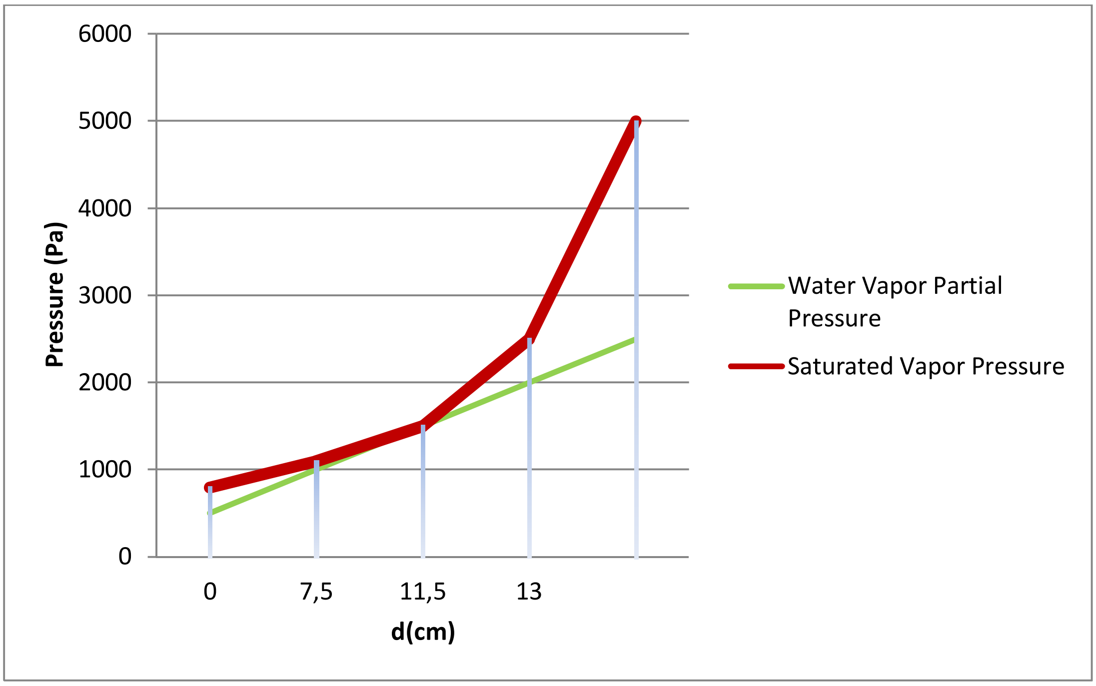


Figure 4.1 Condensation graphic in January

In January, the condensation is realized in the single point.

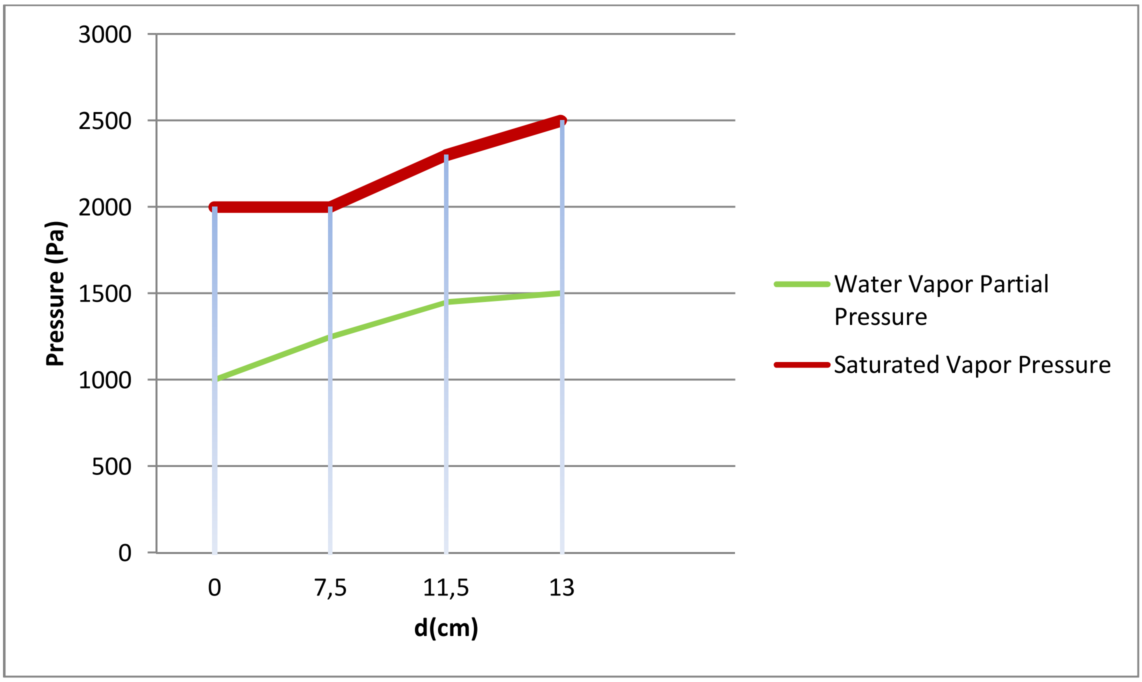


Figure 4.2 Condensation graphic in May

In Figure 4.2 and Figure 4.3 have been seen there is no condensation in May and July respectively.

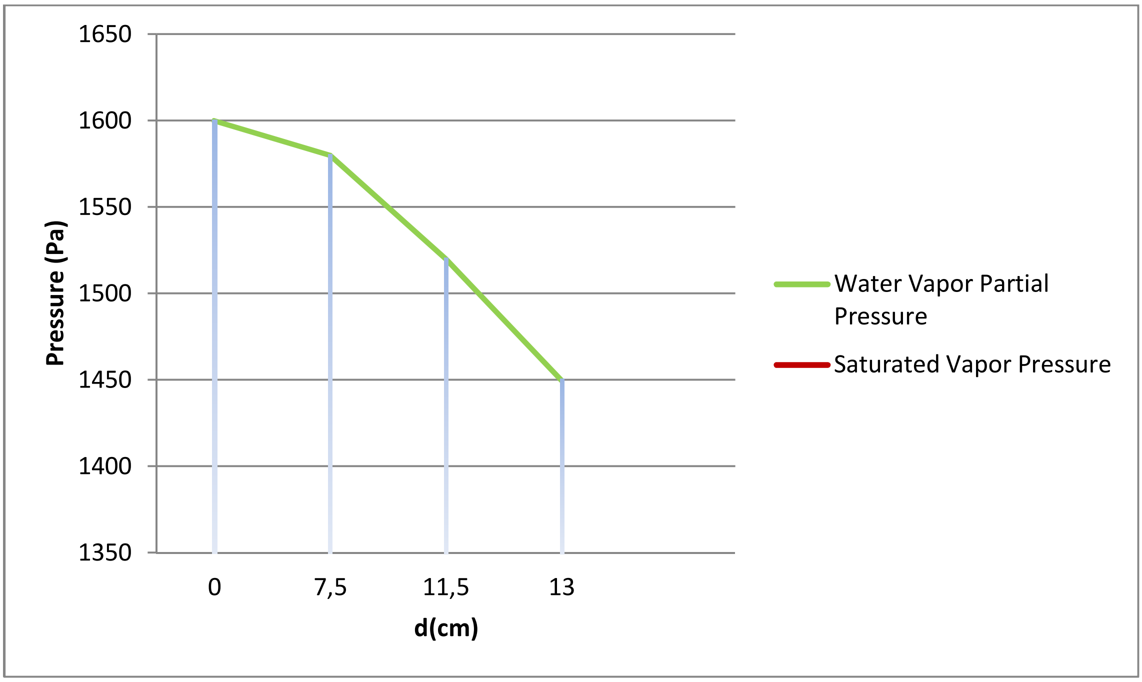


Figure 4.3 Condensation graphic in July

The evaluation of moisture per unit area is performed using the following equations and the results are tabulated in Table 4.14 [43].

$$g = \delta_o \left(\frac{p_i - p_{sw}}{s'_{d.T} - s'_{d,sw}} - \frac{p_{sw} - p_d}{s'_{d,sw}} \right) \tag{4.20}$$

In the table, as can be seen, the positive value indicates condensation and negative one indicates vaporization.

$$p_i = \varphi_i . p_{si} \tag{4.21}$$

$$m_y = g.t \tag{4.22}$$

Table 4.14 Condensation and evaporation amount in the construction components of external curtain wall

	Outdoor Unit Temp.(°C)	Outdoor Unit Humidity Ratio (%)	m_y (kg/m^2)	m_y (kg/m^2) m_y (cumulative)
November	8.5	0.60	0.0063	0.0063
December	3,8	0.70	0.0506	0.0569
January	2.9	0.68	0.0551	0.1120
February	4.4	0.64	0.0438	0.1558
March	7.3	0.61	0.0166	0.1724
April	12.8	0.57	-0.0431	0.1293
May	18.0	0.49	-0.1250	0.0043
June	22.5	0.46	-0.2149	-0.2106
July	24.9	0.50	-0.2593	0
August	24.3	0.51	-0.2456	0
September	19.9	0.48	-0.1606	0
October	14.1	0.49	-0.0671	0

In January, the maximum condensation takes place. In this case study, there is no risk of mold. The risk of mold occurs when the difference between the designed indoor temperature and the inner surface temperature is more than 3 °C. The maximum amount of the cumulative value should not exceed the limit of 1.0 kg/m^2.

As can be seen from Table 4.14, the maximum value of cumulative value of m_y is 0.1724 kg/m^2. Therefore there is no crack on the wall due to condensation.

4.3.2 Vapor Diffusion for External Wall Which Contacts with Ground

The technical data is given about the external wall which contacts with ground components. Water vapor diffusion resistance factor (µ), equivalent air layer thickness (S_d), thermal conductivity (λ_h) and thermal resistance R is seen in Table 4.15.

Table 4.15 Equivalent air layer thickness and thermal resistance due to vapor diffusion for external wall which contacts with ground

Material	δ(m)	μ	Sd (m)	Sd (Total)	Thermal conduct. (λ_h)	R (m^2K/W)	R_T (m^2K/W)
Ext. Surf. Thermal Conduc. Resistance			0	0		0.04	0.04
Brick Wall	0.190	5	0.95	0.95	1.4	0.14	0.18
Extruded Polystyrene Foam	0.050	80	4	4.95	0.03	1.67	1.85
Polymer Wat.Membrane	0.012	20000	240	244.95	0.19	0.06	1.91
Brick Wall	0.190	5	0.95	245.90	1.4	0.14	2.05
Cement	0.010	15	0.15	246.05	1.6	0.01	2.06
Reinforced Concrete	0.300	80	24	270.05	2.5	0.12	2.18
Plaster	0.015	10	0.15	270.20	0.7	0.02	2.20
Int. Surf. Thermal Cond. Resistance			0	270.20		0.25	2.45
			Sdt	270.20		1/U	2.45
						U	0.408 W/m^2K

The equivalent air layer thickness is used for evaluation of condensation in building components. Pressures are taken from Table 4.16

Table 4.16 Surface temp. and saturated water vapor pressure for external wall which contacts with ground

	January		**February**		**March**		**April**		**May**		**June**	
	°C	**Pa**	**°C**	**Pa**	**°C**	**Pa**	**°C**	**Pa**	**°C**	**Pa**	**°C**	**Pa**
Ambient	2.9	753	4.4	837	7.3	1023	12.8	1479	18.0	2065	22.5	2727
Outer S.	3.18	770	4.65	854	7.5	1038	12.9	1488	18.03	2065	22.4	2727
1.Interface	4.2	825	5.5	902	8.2	1088	13.3	1528	18.2	2091	22.3	2695
2.Interface	15.9	1806	16.1	1830	16.9	1926	18.2	2091	19.6	2283	20.6	2428
3.Interface	16.3	1854	16.5	1878	17.2	1963	18.4	2119	19.6	2283	20.5	2413
4.Interface	17.3	1976	17.4	1988	17.9	2052	18.8	2172	19.7	2297	20.4	2399
5.Interface	17.4	1988	17.5	2001	18.0	2065	18.8	2172	19.7	2297	20.4	2399
6.Interface	18.2	2091	18.3	2105	18.6	2145	19.2	2227	19.8	2310	20.3	2384
Inner s.	18.3	2105	18.4	2119	18.7	2158	19.3	2241	19.8	2310	20.3	2384
Indoor	20.0	2340	20.0	2340	20.0	2340	20.0	2340	20.0	2340	20.0	2340
	July		**August**		**September**		**October**		**November**		**December**	
	°C	**Pa**	**°C**	**Pa**	**°C**	**Pa**	**°C**	**Pa**	**°C**	**Pa**	**°C**	**Pa**
Ambient	24.9	3151	24.3	3040	19.9	2324	14.1	1610	8.5	1110	3.8	803
Outer S.	24.8	3132	24.2	3021	19.9	2324	14.2	1621	8.6	1125	4.1	819

1.Interface	24.4	3059	23.8	2950	19.9	2324	14.7	1674	9.3	1171	5.0	872
2.Interface	21.1	2504	20.9	2473	20.0	2340	18.7	2158	17.1	1950	16.0	1818
3.Interface	21.0	2487	20.8	2457	20.0	2340	18.8	2172	17.4	1988	16.4	1866
4.Interface	20.7	2443	20.6	2428	20.0	2340	19.1	2212	18.1	2079	17.3	1976
5.Interface	20.7	2443	20.6	2428	20.0	2340	19.1	2212	18.1	2079	17.4	1988
6.Interface	20.5	2413	20.4	2399	20.0	2340	19.4	2254	18.7	2158	18.2	2091
Inner S.	20.5	2413	20.4	2399	20.0	2340	19.4	2254	18.8	2172	18.3	2105
Indoor	20.0	2340	20.0	2340	20.00	2340	20.0	2340	20.0	2340	20.0	2340

The evaluation of moisture per unit area is performed using the following equations and the results are tabulated in Table 4.17.

Table 4.17 Condensation and evaporation quantity of structural components of external wall which contacts with ground

	Outdoor Unit Temp.(°C)	**Outdoor Unit Humidity Ratio (%)**	**m_y (kg/m^2)**	**m_y (kg/m^2) m_y (cumulative)**
November	8,5	0,60	0,0008	0,0008
December	3,8	0,70	0,0014	0,0022
January	2,9	0,68	0,0014	0,0037
February	4,4	0,64	0,0013	0,0050
March	7,3	0,61	0,0010	0,0060
April	12,8	0,57	0,0001	0,0061
May	18,0	0,49	0,0010	0,0071
June	22,5	0,46	0,0023	0,0094
July	24,9	0,50	0,0031	0,0125
August	24,3	0,51	0,0029	0,0154
September	19,9	0,48	0,0015	0,0169
October	14,1	0,49	-0,0002	0,0167

Because of the inlet temperature is suitable (θ_i >17°C) there is no risk of mold.

Total positive m_y = 0, 0169 < 1 calculation is suitable.

All the condensed water is not evaporated during the summer, so is **not valid** for TSE 825. Because of the condensed water, cracks are formed in the wall.

Figure 4.4 Deformed wall due to condensed water

4.3.3 Vapor Diffusion for Ground Basement

The technical data is given about the uncovered ceiling components. Water vapor diffusion resistance factor (μ), equivalent air layer thickness (Sd), thermal conductivity (λ_h) and thermal resistance (R) is seen in Table 4.18.

Table 4.18 Equivalent air layer thickness and thermal resistance of ground basement

Material	δ(m)	**μ**	**Sd (m)**	**Sd (Tot.)**	**Thermal conduct. (λ_h)**	**R (m^2K/W)**	**R_T (m^2K/W)**
External Sur. Thermal Conduc. Resistance			0.000	0.00		0.04	0.04
Granite	0.020	10000	200	200	2.8	0.01	0.05
Cement Mortar Screed	0.025	15	0.375	0.38	1.40	0.02	0.07
Extruded Polystyrene Foam	0.050	80	4.000	4.38	0.03	1.67	1.74
Cement Mortar Screed	0.025	15	0.375	4.76	1.40	0.02	1.76
Polyisobutylene Cover	0.003	300000	900	904.76	0.26	0.01	1.77
Polymer Waterproofing Membrane	0.012	20000	240	1144.76	0.19	0.06	1.83
Aggregate Unreinforced Concrete	0.050	70	3.500	1148.27	0.44	0.11	1.94
Reinforced Concrete	0.170	80	13.600	1161.87	2.50	0.07	2.01
Internal Surf. Thermal Conduc.Resistance			0.000	1161.87		0.25	2.26
			Sdt	1161.87		1/U	2.26
						U	0.442 W/ m^2K

Table 4.19 Surface temp. and saturated water vapor pressure of ground basement

	January		**February**		**March**		**April**		**May**		**June**	
	°C	**Pa**	**°C**	**Pa**	**°C**	**Pa**	**°C**	**Pa**	**°C**	**Pa**	**°C**	**Pa**
Ambient	2,9	753	4,4	837	7,3	1023	12,8	1479	18,0	2065	22,5	2727
Outer S.	3,13	765	4,61	849	7,47	1038	12,9	1488	18,0	2065	22,47	2727
1.Interface	3,1	765	4,5	843	7,6	1045	13,0	1498	18,0	2065	22,4	2711
2.Interface	12,6	1460	13,2	1518	14,6	1663	17,0	1937	19,1	2212	21,0	2487
3.Interface	12,7	1470	13,3	1528	14,6	1663	17,0	1937	19,1	2212	21,0	2487
4.Interface	18,4	2119	18,5	2132	18,8	2172	19,4	2254	19,8	2310	20,2	2369
5.Interface	18,5	2132	18,6	2145	18,8	2172	19,4	2254	19,8	2310	20,2	2369
Inner S.	18,6	2145	18,7	2158	18,9	2185	19,4	2254	19,8	2310	20,2	2369
Indoor	20,0	2340	20,0	2340	20,0	2340	20,0	2340	20,0	2340	20,0	2340

	July		August		September		October		November		December	
Ambient	24,9	3151	24,3	3040	19,9	2324	14,1	1610	8,5	1110	3,8	803
Outer S.	24,8	3132	24,2	3021	19,9	2324	14,1	1621	8,65	1125	4,02	813
1.Interface	24,7	3114	24,2	3021	19,9	2324	14,2	1621	8,7	1125	4,0	813
2.Interface	22,0	2645	21,8	2613	20,0	2340	17,5	2001	15,1	1717	13,0	1498
3.Interface	22,0	2645	21,8	2613	20,0	2340	17,5	2001	15,1	1717	13,01	1508
4.Interface	20,4	2399	20,4	2399	20,0	2340	19,5	2268	18,9	2185	18,5	2132
5.Interface	20,4	2399	20,4	2399	20,0	2340	19,5	2268	18,9	2185	18,6	2145
Inner S.	20,4	2399	20,4	2399	20,0	2340	19,5	2268	19,0	2197	18,7	2158
Indoor	20,0	2340	20,0	2340	20,0	2340	20,0	2340	20,0	2340	20,0	2340

Table 4.20 Condensation and evaporation quantity of structural components of ground basement

	Outdoor Unit Temp.(°C)	Outdoor Unit Humidity Ratio (%)	m_y (kg/m^2)	m_y (kg/m^2) (cumulative)
November	8,5	0,60	-0,0002	0,0020
December	3,8	0,70	-0,0003	0,0017
January	2,9	0,68	-0,0003	0,0014
February	4,4	0,64	-0,0003	0,0011
March	7,3	0,61	-0,0002	0,0009
April	12,8	0,57	0,0000	0,0000
May	18,0	0,49	0,0002	0,0002
June	22,5	0,46	0,0005	0,0007
July	24,9	0,50	0,0006	0,0013
August	24,3	0,51	0,0006	0,0019
September	19,9	0,48	0,0003	0,0022
October	14,1	0,49	-0,0000	0,0022

Because of the inlet temperature is suitable (θ_i >17°C) there is no risk of mold.

Total positive m_y = 0, 0022 < 1 calculation is suitable.

All the condensed water is evaporated during the summer, so is valid for TSE 825.

4.3.4 Vapor Diffusion for Uncovered Ceiling

The technical data is given about the uncovered ceiling components. Water vapor diffusion resistance factor (µ), equivalent air layer thickness (Sd), thermal conductivity (λ_h) and thermal resistance (R) is seen in table 4.21

Table 4.21 Vapor equivalent thickness and thermal resistance of uncovered ceiling

Material	δ(m)	μ	Sd (m)	Sd (Total)	Thermal conduct.	R (m^2K/W)	R_T (m^2K/W)
External Surf.Thermal Conduc.Resistance			0.00	0.00		0.04	0.04
Cement Mortar Screed	0.025	15	0.375	0.38	1.40	0.02	0.06
Extruded Poly. Foam	0.05	80	4.000	4.38	0.03	1.67	1.73
Cement Mortar Screed	0.025	15	0.375	4.76	1.40	0.02	1.75
Polyisobutylene Cover	0.003	300000	900	904.76	0.26	0.01	1.76
Poly.Water.Membrane	0.012	20000	240	1144.76	0.19	0.06	1.82
Mastic Asphalt Covering	0.01	1	0.010	1144.77	0.70	0.01	1.83
Aggregate Unreinforced Concrete	0.05	70	3.500	1148.27	0.44	0.11	1.94
Reinforced Concrete	0.17	80	13.600	1161.87	2.50	0.07	2.01
Internal Surf.Thermal Conduc.Resistance			0.00	1161.87		0.25	2.26
			Sdt	1161.87		1/U	2.26
						U	0.442 W/m^2K

Table 4.22 Outdoor temp. and saturated water vapor pressure uncovered ceiling [36]

	January		February		March		April		May		June	
	°C	**Pa**	**°C**	**Pa**	**°C**	**Pa**	**°C**	**Pa**	**°C**	**Pa**	**°C**	**Pa**
Ambient	2,9	753	4,4	837	7,3	1023	12,8	1479	18,0	2065	22,5	2727
Outer S.	3,2	770	4,6	854	7,5	1038	12,9	1488	18,0	2065	22,4	2727
1.Interface	3,3	776	4,8	861	7,6	1045	13,0	1498	18,0	2065	22,4	2711
2.Interface	15,9	1806	16,3	1853	17,0	1937	18,3	2105	19,5	2268	20,6	2428
3.Interface	16,1	1830	16,4	1866	17,1	1950	18,4	2119	19,5	2268	20,6	2428
4.Interface	16,2	1841	16,5	1878	17,2	1963	18,4	2119	19,5	2268	20,6	2428
5.Interface	16,7	1901	16,9	1926	17,5	2001	18,6	2145	19,6	2283	20,5	2413
6.Interface	16,8	1914	17,0	1937	17,6	2014	18,6	2145	19,6	2283	20,5	2413
7.Interface	17,6	2014	17,8	2039	18,2	2091	19,0	2197	19,7	2297	20,4	2399
Inner S.	18,1	2079	18,3	2105	18,6	2145	19,2	2227	19,8	2310	20,3	2384
Indoor	20,0	2340	20,0	2340	20,0	2340	20,0	2340	20,0	2340	20,0	2340
	July		**August**		**September**		**October**		**November**		**December**	
	°C	**Pa**	**°C**	**Pa**	**°C**	**Pa**	**°C**	**Pa**	**°C**	**Pa**	**°C**	**Pa**
Ambient	24,9	3151	24,3	3040	19,9	2324	14,1	1610	8,5	1110	3,8	803
Outer S.	24,8	3132	24,2	3021	19,9	2324	14,2	1621	8,7	1125	4,1	819
1.Interface	24,6	3095	24,1	3003	19,9	2324	14,1	1610	8,6	1117	4,2	825
2.Interface	21,0	2487	20,9	2473	20,0	2340	18,5	2132	17,1	1950	16,2	1841
3.Interface	21,0	2487	20,9	2473	20,0	2340	18,6	2145	17,2	1963	16,3	1854
4.Interface	21,0	2487	20,9	2473	20,0	2340	18,6	2145	17,3	1976	16,4	1866
5.Interface	20,9	2473	20,8	2457	20,0	2340	18,8	2172	17,6	2014	16,8	1914

6.Interface	20,9	2473	20,8	2457	20,0	2340	18,8	2172	17,7	2027	16,9	1926
7.Interface	20,7	2443	20,6	2428	20,0	2340	19,1	2212	18,3	2105	17,7	2027
inner s.	20,5	2413	20,5	2413	20,0	2340	19,3	2241	18,7	2158	18,2	2091
Indoor	20,0	2340	20,0	2340	20,00	2340	20,0	2340	20,0	2340	20,0	2340

Table 4.23 Condensation and evaporation quantity of structural components uncovered ceiling

	Outdoor Unit Temp.(°C)	**Outdoor Unit Humidity Ratio (%)**	**m_y (kg/m^2)**	**m_y (kg/m^2) (cumulative)**
November	8,5	0,60	0,0000	0,0000
December	3,8	0,70	0,0000	0,0000
January	2,9	0,68	0,0000	0,0000
February	4,4	0,64	0,0000	0,0000
March	7,3	0,61	0,0000	0,0000
April	12,8	0,57	0,0000	0,0000
May	18,0	0,49	0,0000	0,0000
June	22,5	0,46	0,0000	0,0000
July	24,9	0,50	-0,0000	0,0000
August	24,3	0,51	-0,0000	0,0000
September	19,9	0,48	0,0000	0,0000
October	14,1	0,49	0,0000	0,0000

Because of the inlet temperature is suitable; there is no risk of mold.

No vaporization and condensation

4.3 Conclusion

In this chapter building insulated calculations is examined. The calculations are compared with standards if it is acceptable or not. It was found that the insulation is suitable. Then, heat flow density, temperature and saturated vapor pressure distribution are evaluated. As a result, evaporation and condensation calculation are evaluated and building materials are shown no risk of mold.

CHAPTER 5

EVALUATION OF HEATING CAPACITY OF THE SOCIAL AND CULTURAL BUILDING

5.1 Introduction

In the previous chapter, the calculation of heat transfer considering insulation concept is performed. However, the evaluation of heating capacity of the building is considered here in.

5.2 Evaluation for Overall Heat Transfer Coefficient

Floor K221 clinic room is selected as an example of calculation of heat loss. The roomsô heat transfer coefficient is evaluated and given in Table 5.1 [47].

Table 5.1 Heat transfer coefficient

Location	Overall Heat Transfer Coefficient
External Curtain Wall	U= 0.347 W/m^2K
Up and Down Floor	U= 0.299 W/m^2K
Internal Curtain Wall	U= 2.451 W/m^2K

5.2.1 Heat Loss by Conduction

Heat loss by conduction formula is given Eq. (5.1)

$$q = U.A.\Delta T \tag{5.1}$$

Heat loss surface by conduction is evaluated and given Table 5.2

Table 5.2 Heat loss by conduction in K 221 room [37]

Location	Area (m^2)	Heat Transfer Coefficient (W/m^2 K)	Formula and Calculations	Total Heat Loss (q)
Heat Loss From the External Curtain Wall	8.24	0.347	$q = U_{CW}$ x A x (T_{in}-T_{out}) 0.347 x 8.24 x (22-(-6))	80.06 W
Heat Loss From Up Floor	15.93	3.344	$q = U_{EW}$ x A x (T_{in}-T_{out}) 3.344 x 15.93 x (22-20)	106.54 W
Heat Loss From Down Floor	15.93	3.344	$q = U_{EW}$ x A x (T_{in}-T_{out}) 3.344 x 15.93 x (22-18)	213.08 W
Heat Loss From Internal Wall	10.63	2.451	$q = U_{EW}$ x A x (T_{in}-T_{out}) 2.451 x 10.63 x (22-20)	52.1 W
Heat Loss From Internal Doors	2.2	2	$q = U_d$ x A x (T_{in}-T_{out}) 2.2 x 2 x (22-20)	8.8 W
Heat Loss From Windows	4.59	3	$q = U_W$ x A x (T_{in}-T_{out}) 3 x 4.59 x 22-(-6)	385.56 W
Q_{TOTAL}				**846.14 W**

Combined increase coefficient should be added to total heat loss. It is shown with Z. And it included operation, height and direction coefficient [37, 46].

$$Z = (1 + \%Z_D + \%Z_W + \%Z_H) \tag{5.2}$$

Z_D, Z_W and Z_H values are given as 0.15, o and 0.05 respectively, according to TS 2164 standards. Then Z value is found as Z=1.2. The total increased heat loss is evaluated by using Eq. (5.3)

$$Q = Q_T \, x Z \tag{5.3}$$

And found as

$$Q = 846.14 \text{ x } 1.2 = 1015.37 \text{ W}$$

5.2.2 Infiltration Heat Loss

Infiltration means that the heat loss with air leakages by door and windows. Infiltration formulated as [37];

$$Q_{inf} = a.l.R.Z.\Delta T \tag{5.4}$$

$$Q_{inf} = 1.5x12.2x0.7x1x28$$

$$Q_{inf} = 358.68\, kCal\ /\ h = 417.14\, W$$

$$Q_{Total} = Q + Q_{inf}$$

$$Q_{Total} \approx 1500W$$

with same method, all the room heat loss is calculated. Table 5.3 show the heating capacity of each room in the Social and Cultural Building. The device codes in the first column are designated according to their own project codes.

Table 5.3 Heating capacity of fan coils

SYMBOLS		**LOCATION**	**ROOM TEMP.**	**HEATING CAPACITY**
			(°C)	**TOTAL(W)**
GROUND FLOOR				
FC-Z-01	ZK03	STATIONERY	19	4.400
FC-Z-02	ZK03	STATIONERY	19	4.400
FC-Z-03	ZK04	TRAVEL AGENCY	19	2.800
FC-Z-04	ZK05	TELEPHONIST	19	3.200
FC-Z-05	ZK06	SHOP	19	3.200
FC-Z-06	ZK07	SHOP	19	3.200
FC-Z-07	ZK08	SHOP	19	4.000
FC-Z-08	ZK09	HAIR DRESSER	19	4.000
FC-Z-09	ZK10	DRY CLEAN	19	3.200
FC-Z-10	ZK10	DRY CLEAN	19	3.200
FC-Z-11	ZK11	INTERNET CAFE	19	2.800
FC-Z-12	ZK11	INTERNET CAFE	19	2.800
FC-Z-13	ZK11	INTERNET CAFE	19	2.800
1.FLOOR				
FC-1-01	K104	WORKSHOP	16	4.400
FC-1-02	K104	WORKSHOP	16	4.400
FC-1-03	K105	WORKSHOP	16	2.800
FC-1-04	K105	WORKSHOP	16	2.800
FC-1-05	K106	WORKSHOP	16	2.800
FC-1-06	K106	WORKSHOP	16	2.800
FC-1-07	K107	COMPUTER ROOM	19	2.000
FC-1-08	K107	COMPUTER ROOM	19	2.000
FC-1-09	K107	COMPUTER ROOM	19	2.000
FC-1-10	K107	COMPUTER ROOM	19	2.000
FC-1-11	K107	COMPUTER ROOM	19	2.000
FC-1-12	K107	COMPUTER ROOM	19	2.000
FC-1-13	K107	COMPUTER ROOM	19	2.000
FC-1-14	K107	COMPUTER ROOM	19	2.000
FC-1-15	K107	COMPUTER ROOM	19	2.000
FC-1-16	K107	COMPUTER ROOM	19	2.000
2.FLOOR				
FC-2-01	K203	OFFICE	19	4.400

FC-2-02	K203	OFFICE	19	4.400
FC-2-03	K204	MANAGER ROOM	19	3.200
FC-2-04	K205	SECRETARY	19	3.600
FC-2-05	K206	OFFICE	19	2.800
FC-2-06	K206	OFFICE	19	2.800
FC-2-07	K207	MANAGER ROOM	19	3.600
FC-2-08	K208	MANAGER ROOM	19	3.600
FC-2-09	K209	ARCHIVE	19	2.800
FC-2-10	K209	ARCHIVE	19	2.800
FC-2-11	K210	OFFICE	19	4.000
FC-2-12	K210	OFFICE	19	4.000
FC-2-13	K216	DOCTOR ROOM	22	3.600
FC-2-14	K216	DOCTOR ROOM	22	3.600
FC-2-15	K217	DOCTOR ROOM	22	2.800
FC-2-16	K218	DOCTOR ROOM	22	3.200
FC-2-17	K219	NURSE ROOM	22	3.200
FC-2-18	K220	CLINIC	22	3.200
FC-2-19	**K221**	**CLINIC**	**22**	**1.500**
FC-2-20	K222	CLINIC	22	3.200
FC-2-21	K223	ARCHIVE	19	2.400
FC-2-22	K223	ARCHIVE	19	2.400
FC-2-23	K224	ARCHIVE	19	4.000
FC-2-24	K224	ARCHIVE	20	4.000
3.FLOOR				
FC-3-01	K302	REST AREA	19	2.800
FC-3-02	K302	REST AREA	19	2.800
FC-3-03	K302-1	ROOM	19	4.800
FC-3-04	K302-1	ROOM	19	4.800
FC-3-05	K304	ROOM	19	3.600
FC-3-06	K306	ROOM	19	3.600
FC-3-07	K308	ROOM	19	3.600
FC-3-08	K310	ROOM	19	3.600
FC-3-09	K312	ROOM	19	3.600
FC-3-10	K314	ROOM	19	3.600
FC-3-11	K316	ROOM	19	3.600
FC-3-12	K318	ROOM	19	3.600
FC-3-13	K320	ROOM	19	4.400
FC-3-14	K320	ROOM	19	4.400
FC-3-15	K322	ROOM	19	2.800
FC-3-16	K322	ROOM	19	2.800
FC-3-17	K324	SERVICE	19	2.800

The heating capacity of AHU is evaluated and the Table 5.4 is given.

Table 5.4 Heating capacity of AHU

SYMBOLS	LOCATION	ROOM TEMP. (°C)	HEATING CAPACITY TOTAL
KS-01	STUDENT CAFETERIA	19	287.759
KS-02	1.AND GROUND FLOOR FOYER	19	308.834
KS-03	GROUND FLOOR CAFETERIA	19	159.596
KS-04	ADMINISTRATIVE STAFF REFECTORY	19	138.652
KS-05	ACADEMICIAN REFECTORY	19	138.652
KS-06	CINEMA-1	19	43.702
KS-07	CINEMA FOYER	19	43.702
KS-08	CINEMA-2	19	43.702
KS-09	EXHIBITION HALL	19	159.596
KS-10	MULTI-PURPOSE ROOM	19	219.393
KS-11	3.FLOOR HALL	19	127.899
KS-12	3.FLOOR CONFERENCE HALL-1	19	57.708
KS-13	3.FLOOR CONFERENCE HALL-2	19	43.700
			1.772.895

$$Q_T = Q_{FAN\_COIL} + Q_{AHU} \tag{5.5}$$

$$Q_T = 226.000W + 1.772.895W$$

$$Q_T \approx 2000\,KW$$

Total heating capacity of the building is found as 1.772.895W by air-handling unit and 226.000 W by fan-coil, total heat loss by the building is evaluated as 2000 KW.

5.3 Auxiliary Equipment Selection

In this section auxiliary equipment are selected. Auxiliary equipments are used for initial investment and operation costs. So the capacity of equipment should be known. Selected equipment comprise from heating boiler, pumps and expansion tank.

5.3.1 Heating Boiler Account

The total heating capacity is 2000 KW. 2 boilers are selected against the possibility of failure. Total capacity of the heating cannot be supply with one boiler, so half capacity is added. Boiler capacities are evaluated as;

Q_T = 2000 KW x 3 / 2 = 3000 KW

2 x 1500 KW heating boiler is selected and technical data are given in Table 5.5

Table 5.5 Heating boiler technical data

HEATING BOILER			
Symbols	-	IK-01	IK-02
Location	-	Stokehold	Stokehold
Heatıng Capacity	Kw-h	1500	1500
Water Inlet-Outlet Temp.	C	90/70	90/70
Pressure	PN	6	6
Fuel Type	-	Fuel Oil	Fuel Oil
Efficiency	%	95	95
Material	-	Steel	Steel
Way of Working	-	Superior	Superior

5.3.2 Boiler, AHU and Fan-Coil Heating Pumps Selection

It is important flow and pressure for pump selection. Boiler circulation pumps flow capacity is evaluated with Eq. (5.6)

$$Q = m.c.\Delta T \tag{5.6}$$

$$1.290.000 = m.1000.(90-70)$$

$$m \approx 70\, m^3/h$$

Pressure is evaluated as;

$$H_P = 0.04[\,width + length + height\,] \tag{5.7}$$

$$H_P \approx 4.32\, mSS$$

AHU and fan coil pumps are evaluated with same method and Table 5.6 is given.

Table 5.6 The capacity of the pump

NAME	FLOW CAPACITY	PRESSURE
Boiler Circulation Pumps	70 m^3/h	45 kPa
AHU Pumps	85 m^3/h	80 kPa
Fan Coil Pumps	10 m^3/h	100 kPa

Expansion tank is evaluated as;

Static Pressure = $h_{building}$ = 26 m = 2.6 bar

Front gas pressure

P = 2.6 + 0.4 = 3 bar

$$V_s = Q \times f \tag{5.8}$$

V_s = 10.320 lt

$$V_G = V_s \times n \tag{5.9}$$

V_G = 370 lt

$$V_n = V_G / K \tag{5.10}$$

V_n å 1.500 lt is selected

5.3.3 Burner Selection

Burner is calculated by the following formula [42].

$$B_y = \frac{3.6\,Q_k\,Z_g\,Z_y}{2\,H_u\,\eta_k} \tag{5.11}$$

$$B_y = \frac{3.6\,x\,1500\,x\,1000\,x\,9\,x\,108}{2\,x\,41.860\,x\,0.95}$$

$$B_y = 65994\ kg\ /\ year$$

5.4 Conclusion

In this chapter, the clinic room is selected as an example room and heat loss by conduction and infiltration are evaluated. Then the other rooms are calculated with same method and the Social and Cultural building heat loss calculations are evaluated. Fan coils and AHU capacities are found. In this chapter also, detailed information and accounts are given about the auxiliary equipment.

CHAPTER 6

EVALUATION OF COOLING CAPACITY OF THE SOCIAL AND CULTURAL BUILDING

6.1 Introduction

In the previous chapter, the calculation of heating capacity of the building is performed. The evaluation of cooling capacity of the building is considered here in.

K221 manager room is selected as an example of calculation of cooling capacity. Kilis dry bulb, wet bulb temperatures and relative humidity are selected respectively 39 °C and 23 °C and % 50 [43].

6.2 Heat Gain Evaluations of Social and Cultural Building

For heat gain evaluation of the building, correction factor (F_c) should be known. F_c constitutes the cooling load heat rate of a volume gain. F_c is evaluated by the Eq. (6.1) and Eq. (6.2) [37].

$$K_T = (1/ L_F).\ (U_c.A_c + U_d.A_d + U.A) \tag{6.1}$$

$$K_T = (1/2.9)\ .\ (3 \times 4.76 + 0.347 \times 9.015)$$

$$K_T = 6$$

$$F_c = 1 - 0.0116 \times K_T \tag{6.2}$$

$$F_c = 1 - 0.0116 \times 6$$

$$F_c = 0.93$$

Table 6.1 Total cooling accounts [37, 38]

Location	**Area (m^2)**	**Heat Transfer Coefficient (W/m^2 K)**	**Formula and Calculations**	**Total Heat Loss (W)**
Heat Gain from Glass Surfaces	4.76	3	$Q_{dc} = F_ç . U_c . A_c . (CLTD)$ $Q_{dc} = 0.93 \times 3 \times 4.76 \times 8$	106.24
Heat Gain From Exterior Wall	9.015	0.347	$Q_{dd} = F_ç . U_d . A_d . (CLTD)$ $Q_{dd} = 0.93 \times 0.347 \times 9.015 \times 9$	26.183
Heat Gain From Roof	-	-	$Q_{dç} = F_ç . U_ç . A_ç . (CLTD)$ $Q_{dç} = 0$	0
Heat Gain Caused By Direct Solar Radiation From The Outer Glass Surface	4.76	0.82	$Q_{dcs} = F_c.A.SC.SHGF.CLF$ $Q_{dcs} = 0.93x4.76x0.60x681x0.82$	1483.2
Heat Gain From The Inner Wall By Conduction	-	-	$Q_{iç} = U . A . TD$ $Q_{iç} = 0$	0
Heat Gain From Lighting Elements	-	0.82	$Q_{light} = N . CLF . F_c$ $Q_{light} = 480 \times 0.82 \times 0.93$	366.048
Heat Gain From Human Body	-	0.84	$Q_{human} = Q_{sensible} + Q_{latent}$ $Q_{sensible} = S . SHG . CLF . F_c$ $Q_{latent} = S . LHG$ $Q_{human} = 2x65x0.84x0.93+2x55$	211.56
Heat gain from Infiltration and Ventilation	-	-	$V_{enf} = S . V = 2 \times 40 = 80\ m^3/h$ $Q_D = V_{enf} \times C_{ph} \times r_h \times (T_d - T_i)$ $Q_D =$ 80(m^3/h)x1,22(kg/m^3)x0,24 (kcal/kg.K)x(40 ï 20) °C	544.74
Heat gain from devices	-	-	$Q_{eq} = Q_{device1} + Q_{device2}$ $Q_{device} = P_s \times S$ 1 computer + 1 printer 116W + 290 W	406
Q_{Total}				**3144 W**

Total cooling account is;

Q_{Total} = 3144 W= with %15 safety factor å 3600 W

With the same method, other roomsô cooling capacities are calculated and given in Table 6.2

Table 6.2 Buildings heat gain chart

DEVICE		LOCATION	COOLING CAPACITY (W)		
			SENSIBLE	LATENT	TOTAL
GROUND FLOOR					
FC-Z-01	ZK03	STATIONERY	4.240	160	4.400
FC-Z-02	ZK03	STATIONERY	4.240	160	4.400
FC-Z-03	ZK04	TRAVEL AGENCY	4.040	360	4.400
FC-Z-04	ZK05	TELEPHONIST	4.450	350	4.800
FC-Z-05	ZK06	SHOP	4.450	350	4.800
FC-Z-06	ZK07	SHOP	4.450	350	4.800
FC-Z-07	ZK08	SHOP	4.450	350	4.800
FC-Z-08	ZK09	HAIR DRESSER	4.450	350	4.800
FC-Z-09	ZK10	DRY CLEAN	3.650	350	4.000
FC-Z-10	ZK10	DRY CLEAN	3.650	350	4.000
FC-Z-11	ZK11	INTERNET CAFE	4.180	220	4.400
FC-Z-12	ZK11	INTERNET CAFE	4.180	220	4.400
FC-Z-13	ZK11	INTERNET CAFE	4.180	220	4.400
1.FLOOR					
FC-1-01	K104	WORKSHOP	4.010	390	4.400
FC-1-02	K104	WORKSHOP	4.010	390	4.400
FC-1-03	K105	WORKSHOP	3.190	410	3.600
FC-1-04	K105	WORKSHOP	3.190	410	3.600
FC-1-05	K106	WORKSHOP	3.220	380	3.600
FC-1-06	K106	WORKSHOP	3.220	380	3.600
FC-1-07	K107	COMPUTER ROOM	3.600	400	4.000
FC-1-08	K107	COMPUTER ROOM	3.600	400	4.000
FC-1-09	K107	COMPUTER ROOM	3.600	400	4.000
FC-1-10	K107	COMPUTER ROOM	3.600	400	4.000
FC-1-11	K107	COMPUTER ROOM	3.600	400	4.000
FC-1-12	K107	COMPUTER ROOM	3.600	400	4.000
FC-1-13	K107	COMPUTER ROOM	3.600	400	4.000
FC-1-14	K107	COMPUTER ROOM	3.600	400	4.000
FC-1-15	K107	COMPUTER ROOM	3.600	400	4.000
FC-1-16	K107	COMPUTER ROOM	3.600	400	4.000
2.FLOOR					
FC-2-01	K203	OFFICE	3.840	160	4.000
FC-2-02	K203	OFFICE	3.840	160	4.000
FC-2-03	K204	MANAGER ROOM	2.670	130	2.800
FC-2-04	K205	SECRETARY	3.070	150	3.200
FC-2-05	K206	OFFICE	3.230	370	3.600
FC-2-06	K206	OFFICE	3.230	370	3.600
FC-2-07	**K207**	**MANAGER ROOM**	**3.470**	**130**	**3.600**
FC-2-08	K208	MANAGER ROOM	3.070	130	3.200
FC-2-09	K209	ARCHIVE	3.030	170	3.200
FC-2-10	K209	ARCHIVE	3.030	170	3.200
FC-2-11	K210	OFFICE	4.630	170	4.800
FC-2-12	K210	OFFICE	4.630	170	4.800

FC-2-13	K216	DOCTOR ROOM	4.630	170	4.800
FC-2-14	K216	DOCTOR ROOM	4.630	170	4.800
FC-2-15	K217	DOCTOR ROOM	2.670	130	2.800
FC-2-16	K218	DOCTOR ROOM	3.460	140	3.600
FC-2-17	K219	NURSE ROOM	3.460	140	3.600
FC-2-18	K220	CLINIC	3.460	140	3.600
FC-2-19	K221	CLINIC	3.460	140	3.600
FC-2-20	K222	CLINIC	3.460	140	3.600
FC-2-21	K223	ARCHIVE	3.040	160	3.200
FC-2-22	K223	ARCHIVE	3.040	160	3.200
FC-2-23	K224	ARCHIVE	3.840	160	4.000
FC-2-24	K224	ARCHIVE	3.840	160	4.000
3.FLOOR					
FC-3-01	K302	REST AREA	3.240	360	3.600
FC-3-02	K302	REST AREA	3.240	360	3.600
FC-3-03	K302-1	ROOM	3.960	40	4.000
FC-3-04	K302-1	ROOM	3.960	40	4.000
FC-3-05	K304	ROOM	3.520	80	3.600
FC-3-06	K306	ROOM	3.930	70	4.000
FC-3-07	K308	ROOM	3.930	70	4.000
FC-3-08	K310	ROOM	3.930	70	4.000
FC-3-09	K312	ROOM	3.930	70	4.000
FC-3-10	K314	ROOM	3.930	70	4.000
FC-3-11	K316	ROOM	3.930	70	4.000
FC-3-12	K318	ROOM	3.520	80	3.600
FC-3-13	K320	ROOM	4.370	30	4.400
FC-3-14	K320	ROOM	4.370	30	4.400
FC-3-15	K322	ROOM	3.160	40	3.200
FC-3-16	K322	ROOM	3.160	40	3.200
FC-3-17	K324	SERVICE	2.870	330	3.200

The location which are cooled by AHUs are calculated and given the Table 6.3

Table 6.3 Heat gain chart by AHU

DEVICE	LOCATION	COOLING CAPACITY (W)		
		SENSIBLE	LATENT	TOTAL
KS-01	STUDENT CAFETERIA	214.200	65.548	279.748
KS-02	1.AND GROUND FLOOR FOYER	233.587	63.276	296.863
KS-03	GROUND FLOOR CAFETERIA	125.268	34.630	159.898
KS-04	ADMINISTRATIVE STAFF REFECTORY	104.662	32.002	136.664
KS-05	ACADEMICIAN REFECTORY	104.662	32.002	136.664

KS-06	CINEMA-1	32.938	8.431	41.369
KS-07	CINEMA FOYER	32.938	8.431	41.369
KS-08	CINEMA-2	32.938	8.431	41.369
KS-09	EXHIBITION HALL	125.268	34.630	159.898
KS-10	MULTI-PURPOSE ROOM	149.753	36.408	186.161
KS-11	3.FLOOR HALL	101.295	27.511	128.806
KS-12	3.FLOOR CONFERENCE HALL-1	42.401	10.740	53.141
KS-13	3.FLOOR CONFERENCE HALL-2	32.937	8.430	41.367
		TOTAL		1.703.3

$$Q_T = Q_{FAN\text{-}COIL} + Q_{AHU} \quad (6.3)$$

$$Q_T = 274.8\text{ KW} + 1703.3\text{ KW}$$

$$Q_T = 1978.1\text{ KW}$$

Total cooling capacity of the building is found as 1.703.300 W by air-handling unit and 274.800 W by fan-coil. Total heat gain by the building is evaluated as 2000 KW.

6.3 Auxiliary Equipment Selection

In this section auxiliary equipment are selected. Auxiliary equipment are used for initial investment and operation costs. So the capacity of equipment should be known. Selected equipment comprise from chiller and pumps.

6.3.1 Chiller Selection

The total cooling capacity is 2000 KW before. 2 chillers are selected against the possibility of failure.

2 x 1000 KW air cooled chiller is selected technical data are given in Table 6.4

Table 6.4 Chiller technical data

AIR COOLED CHILLER			
SYMBOL		**CH-01**	**CH-02**
COOLING CAPACITY	Kw-r	1000	1000
EVAPORATOR OUTLET TEMP.	°C	6	6
EVAPORATOR INLET TEMP.	°C	12	12
REFRIGERANT	-	410 A	410 A
MIN COP.	-	2.5	2.5
KOMPRESSOR NUM.	-	2	2

CAPACITY CONTROL	-	Proportional	Proportional
TOTAL ELECTRIC POWER	Kw-e	400	400
VOLTAGE	V	3x400	3x400
START UP	-	Soft Starter	Soft Starter

6.3.2 Chiller, AHU and Fan-Coil Pumps Selection

Chiller pumps is evaluated the Eq. (6.4)

$$Q = m.c.\Delta T \tag{6.4}$$

$$860000\, kcal / h = m.(12 - 6).1000$$

$$m \approx 145 m^3 / h$$

AHU and fan coil pumps are evaluated with same method and Table 6.5 is given.

Table 6.5 The capacity of the pump

NAME	**FLOW CAPACITY**	**NUMBER (Duty + Standby)**
Chiller Pumps	145 m3/h	(1+1)
AHU Pumps	170 m3/h	(2+1)
Fan Coil Pumps	45 m3/h	(1+1)

6.4 Conclusion

In this chapter, the manager room is selected as an example room and heat gain calculations are evaluated. Then the other rooms are calculated with same method and the Social and Cultural building total heat gain calculations are evaluated. Fan coils and AHU capacities are found. In this chapter also, detailed information and accounts are given about the auxiliary equipment.

CHAPTER 7

SYSTEMS INVESTMENT, OPERATION AND MAINTENANCE COSTS ACCOUNT

7.1 Introduction

In this chapter, two different systems conventional and VRF systems, which are respectively used and modeled in Social and Cultural building are investigated. Conventional systems and VRF systems investment, operation and maintenance costs are evaluated. Both systems advantages and disadvantages are mentioned.

7.2 Initial Investment Costs for Conventional HVAC and VRF Systems

In this section, both systems material purchase prices are evaluated. Conventional system unit prizes are obtained from Ministry for Environment and Urbanization. VRF system unit prizes are obtained from the companies. The conventional system initial costs are evaluated and Table 7.1 is given.

Table 7.1 Conventional systems initial investment cost

Exposure Number	Material	Quantity	Measure	Unit Price (USD*)	Total Price (USD*)
Special 1	Boiler and Burner	2	Number	31124	62248
Special 2	Chiller	2	Number	69228	138456
Special 3	Chiller Storage Tank	1	Number	1618	1618
Special 4	Variable Flow Pumps (Chiller)	4	Number	6394	25576
Special 5	Var. Pumps(Boiler)	4	Number	1888	7552
Special 6	Variable Flow Pumps (Fan Coil)	1	Number	1652	1652

Special 7	Variable Pumps (AHU Cooling)	3	Number	3533	10599
Special 8	Variable Pumps (AHU Heating)	3	Number	4577	13731
Special 9	Fan Coil (2800W)	2	Number	313	626
Special 10	Fan Coil (3200 W)	10	Number	318	3180
Special 11	Fan Coil (3600 W)	15	Number	378	5670
Special 12	Fan Coil (4000 W)	24	Number	398	9552
Special 13	Fan Coil (4400 W)	10	Number	398	3980
Special 14	Fan Coil (4800 W)	9	Number	497	4473
173.107	Collector Pipe	180	Meter	36	6480
201.108	Screw Welded	4516	Meter	4	18064
231.202	Pipe Painted with Oil Paint	4516	Meter	0.3	1355
241.423	Pipe Insulation	4516	Meter	2.5	11290
210.708	Ball Valve	373	Number	91	33943
Special 15	Room Thermostat	70	Number	20	1400
174.615	Expansion Tank	4	Number	393	1572
412.108	Fuel Tank	2	Number	4196	8392
412.202	Daily Fuel Tank	2	Number	159	318
Special 16	AHU	14	Number	10933	153062
261.151	Duct (0.75 mm)	8194	m^2	26	213044
261.152	Duct (0.90 mm)	554	m^2	29	16066
261.153	Duct (1 mm)	201	m^2	33	6633
221.307	Strainers	38	Number	56	2128
228.106	Check Valves	12	Number	52	624
Special 17	60 x 60 Sq. Vent	893	Number	60	53580
241.438	Rubber Foam Ins.	2218	Meter	4	8872
Special 18	Automation Eq.	1	Group	36868	36868
TOTAL					**862604**

*Indicative Exchange Rates Announced at 15:30 on 17/12/2015 by the Central Bank of Turkey 1 USD=2.9364 TL

The VRF system initial costs are evaluated and Table 7.2 is given.

Table 7.2 VRF initial investment cost

Location-Cooling Capacity	**Material**	**Quantity**	**Unit Price (*USD)**	**Total Price (*USD)**
Indoor Unit (3.6 KW)	FDT36KXE6D	5	437	2185
Indoor Unit (4.5 KW)	FDT45KXE6D	27	444	11988
Indoor Unit (5.6 KW)	FDT56KXE6D	13	459	5967

Indoor Unit (6.4 KW)	FDT64KXE6D	2	469	938
Indoor Unit (7.1 KW)	FDT71KXE6D	272	479	130288
Outdoor Unit 20HP	FDC560KXZE1	36	5362	193032
Outdoor Unit 17HP	FDC475KXZE1	3	4835	14505
All Offices	Indoor command	319	35	11165
Security Room	General command	3	1538	4614
	DIS- 22-1-Joint	145	21	3045
	DIS- 180-1-Joint	57	35	1995
	DIS- 371-1-Joint	40	48	1920
	DIS-540-2-Joint	38	94	3572
TOTAL DEVICE COST				385214
COPPER PIPING AND ASSEMBLY				91488
TOTAL				**476702**

*Indicative Exchange Rates Announced at 15:30 on 17/12/2015 by the Central Bank of Turkey 1 USD=2.9364 TL

7.3 Operating Cost for Conventional HVAC and VRF Systems

In this section, the operating cost for Conventional HVAC system and VRF are evaluated. The energy consumption of conventional system consists of electricity and fuel oil. In order to calculate operating costs of conventional system, electric consumption of fan coils, AHUs, chiller and boiler fuel oil consumption should be known. But VRF system consumes only electricity. For calculating operating cost it is enough to known indoor and outdoor unit electric consumption. The heating and cooling dates of season are given below.

Boiler heating date (15 November ï 15 April) = 108 work day/year
Chiller cooling date (15 May ï 15 September) = 87 work day/year

Electric consumption of fan coils are evaluated and the Table 7.3 is given.
Total working hours of fan coils: 9 h/day x 195 day/year = 1755 h/year

Table 7.3 Electric consumption of fan coil units [40]

Cooling Capacity	**Fan-Coil Type**	**Total Working Time**	**Power Consumption of Fan**	**Quantity**	**TOTAL POWER CONSUMPTION**
W	Type	Hours/Year	W	Number	KW/Year
2800-3200W	MKT2 400	1755	60	12	1263
3600-4000-4400W	MKT2 500	1755	75	49	6450
4800-5200W	MKT2 600	1755	96	9	1516
TOTAL					9229 KW/Year

Electric consumption of chiller

Total working hours of chiller (15 May ï 15 September) å 87 days x 9 hours = 783 h / season. Electric consumption of chiller is obtained from Table 7.4

Table 7.4 Chiller technical data

AIR COOLED CHILLER		
RHOSS TCAVBZ 21020 SERIES		
SYMBOL		**CH-01**
COOLING CAPACITY	Kw-r	1015.5
EVAPORATOR OUTLET TEMP.	°C	7
EVAPORATOR INLET TEMP.	°C	12
REFRIGERANT	-	134 A
MIN EER (Energy Efficiency Ratio)	-	3.05
KOMPRESSOR NUM.	-	2
CAPACITY CONTROL	-	Proportional
TOTAL ELECTRIC POWER	Kw-e	333.4
VOLTAGE	V	400 x 3 x 50
START UP	-	Soft Starter

The chiller is not always working with full capacity. So for evaluation of electric consumption IPLV (Integrated Part Load Value) method is used. Working times and capacities are given in Table 7.5

Table 7.5 Chiller average load and working time [39]

Load	Time of Working
% 100	% 1
% 75	% 42
% 50	% 45
% 25	% 12
Commonly %58 capacity	

Estimated Spending Power = 783 hour/season x 333.4 KW x 0.58 = 151410 KW

Electric consumption of AHUs are evaluated and the Table 7.6 is given. The devices runtimes are depending on where they work. So the working hours have been taken into account.

Table 7.6 Electric consumption of AHUs [40]

DEVICE	LOCATION	TOTAL WORKING TIME	POWER CONS.	TOTAL POWER CONS.
			KW/h	**KW/year**
KS-01	STUDENT REFECTORY	9h x 195 day/year =1755 hour/year	27.64	48087
KS-02	1.AND GROUND FLOOR FOYER	9h x 195 day/year =1755 hour/year	29.51	51790
KS-03	GROUND FLOOR CAFETERIA	9h x 195 day/year =1755 hour/year	16.41	28800
KS-04	ADMINISTRATIVE STAFF REFECTORY	2h x 195 day/year =390 hour/year	14.93	5822
KS-05	ACADEMICIAN REFECTORY	2h x 195 day/year =390 hour/year	14.93	5822
KS-06	CINEMA-1	4h x 8 day x 12m =384 hour/year	5.35	2054
KS-07	CINEMA FOYER	4h x 8 day x 12m =384 hour/year	5.35	2054
KS-08	CINEMA-2	4h x 8 day x 12m =384 hour/year	5.35	2054
KS-09	EXHIBITION HALL	4h x 8 day x 12m =384 hour/year	16	6144
KS-10	MULTI-PURPOSE ROOM	8h x 8 day x 12m =768 hour/year	26	19968
KS-11	3.FLOOR HALL	9h x 195 day/year =1755 hour/year	13.04	22885
KS-12	3.FLOOR CONFERENCE HALL-1	6h x 2 day x 12m =144 hour/year	7.67	1104
KS-13	3.FLOOR CONFERENCE HALL-2	6h x 2 day x 12m =144 hour/year	5.86	844
HS-01	OFFICES	9h x 195 day/year =1755 hour/year	34.7	60898
			TOTAL	183203

Electric consumptions of Fan coil, chiller and AHU pumps are given in Table 7.7, Table 7.8 and Table 7.9 respectively.

Table 7.7 Electric consumption for fan coil pump

Motor Power	3 KW
Quantity	1
Total Working Hour	1755 hours/ season
Capacity	% 100
Estimated Spending Power	5265 KW

Table 7.8 Electric consumption for chiller pumps

Motor Power	11 KW
Quantity	2
Total Working Hour	783 hours/ season
Capacity	% 100
Estimated Spending Power	17226 KW

Table 7.9 Electric consumption for AHU pumps

Motor Power	11 KW
Quantity	3
Total Working Hour	1755 hours/ season
Capacity	% 100
Estimated Spending Power	57915 KW

Total Electric Consumption = Fan coils + AHU + Chiller + Pumps

Total Electric Consumption = 424248 KW/Year

Annual fuel oil consumption of boiler is given in Eq. (7.1)

$$B_y = \frac{3.6 Q_k \, Z_g \, Z_y}{2 H_u \, \eta_k} \tag{7.1}$$

$$B_y = \frac{3.6 \, x 1500 \, x 1000 \, x 9 \, x 108}{2 \, x 41860 \, x 0.95} = 65994 \, kg \, / \, year$$

VRF is selected as a heat pump system. Both heating and cooling system can be performed. So it doesnơt need liquid fuel. Power consumption of student refectory is evaluated as,

P= Number of indoor unit x per indoor unit electric consumption x working time

P_{indoor} *= 24 x 0.08 KW x 1755 h/year = 3370 KW/year*

Total indoor unit capacities equal to total outdoor capacities. There is no diversity in devices. Power consumption of outdoor unit, *P = [(Total cooling capacity x IPLV)/EER] x working time*

$P_{outdoor}$ *= [(24 x7.1 KW x 0.58) / (3.40 EER)] x 1755 = 51015*

Electric consumption of VRF system indoor and outdoor units are evaluated and the Table 7.10 is given.

Table 7.10 Electric consumption of VRF indoor and outdoor unit [41]

DEVICE	LOCATION	Total Working Time	Power Cons. of Indoor Unit	Power Cons. Of Outdoor Unit	TOTAL
			KW	**KW**	**KW**
KS-01	STUDENT REFECTORY	9h x 195 day/year =1755 hour/year	24x0.08x1755 =3370	51015	54385
KS-02	1.AND GROUND FLOOR FOYER	9h x 195 day/year =1755 hour/year	32x0.08x1755 =4493	68019	72512
KS-03	GROUND FLOOR CAFETERIA	9h x 195 day/year =1755 hour/year	17x0.08x1755 =2387	36135	38522
KS-04	ADMINISTRATIVE STAFF REFECTORY	2h x 195 day/year =390 hour/year	18x0.08x390= 562	8502	9064
KS-05	ACADEMICIAN REFECTORY	2h x 195 day/year =390 hour/year	18x0.08x390= 562	8502	9064
KS-06	CINEMA-1	4h x 8 day x 12m =384 hour/year	3x0.08x384= 92	1395	1487
KS-07	CINEMA FOYER	4h x 8 day x 12m =384 hour/year	3x0.08x384= 92	1395	1487
KS-08	CINEMA-2	4h x 8 day x 12m =384 hour/year	3x0.08x384= 92	1395	1487
KS-09	EXHIBITION HALL	4h x 8 day x 12m =384 hour/year	22x0.08x384= 676	10227	10903
KS-10	MULTI-PURPOSE ROOM	8h x 8 day x 12m =768 hour/year	24x0.08x768= 1475	22315	23790
KS-11	3.FLOOR HALL	9h x 195 day/year =1755 hour/year	20x0.08x1755 =2808	42494	45302
KS-12	3.FLOOR CONFERENCE HALL-1	6h x 2 day x 12m =144 hour/year	6x0.08x144= 69	1046	1115
KS-13	3.FLOOR CONFERENCE HALL-2	6h x 2 day x 12m =144 hour/year	8x0.08x144= 92	1395	1487
	OFFICES and OTHER LOCATION	9h x 195 day/year =1755 hour/year	121x0.04x 1755=8494	224535	233029
	TOTAL		25264		**503634**

7.4 Maintenance Cost for Conventional HVAC and VRF Systems

Devices yearly maintenance should be done for prevent the failure. For the conventional system, AHU and chiller maintenance are the most important part of maintenance cost. The following Table 7.11 illustrates the maintenance periods and costs [42].

AHU maintenance supplies

Table 7.11 Annual AHU maintenance costs

Material	Quantity	Measure	Unit Price (*USD)	Total Price (*USD)	Maintenance Period
592x592x600mm F-7 Bag Filter	42	Number	36	1512	6 times/year
490x592x600mm F-7 Bag Filter	42	Number	31	1302	6 times/year
592x592x50mm G4 Cassette Filter	28	Number	22	616	4 times/year
490x592x50mm G4 Cassette Filter	28	Number	20	560	4 times/year
TOTAL				**3990**	

*Indicative Exchange Rates Announced at 15:30 on 17/12/2015 by the Central Bank of Turkey 1 USD=2.9364 TL

Chiller maintenance supplies

In the winter, antifreeze coolant should be added chiller to prevent freezing. Maintenance costs of chiller is given in Table 7.12

Table 7.12 Annual chiller maintenance costs

Material	Quantity	Measure	Unit Price (*USD)	Total Price (*USD)
Antifreeze	2 x 80	lt	3	480

*Indicative Exchange Rates Announced at 15:30 on 17/12/2015 by the Central Bank of Turkey 1 USD=2.9364 TL

Annual maintenance costs of other devices are given in Table 7.13. The costs are obtained from Kilis 7 Aralik Universityȏs annual contract. It includes maintenances except for consumables one.

Table 7.13 Annual maintenance costs of other devices

Material	Quantity	Measure	Unit Price (*USD)	Total Price (*USD)	Maintenance Period
AHU	16	Number	102	1632	2 times/year
Chiller	2	Number	1192	2384	1 times/year
Pump Groups	1	Group	2384	2384	2 times/year
Fan-Coils	70	Number	43	3010	2 times/year
Boiler and Brulor	2	Number	1192	2384	2 times/year
Automation	1	Group	1192	1192	1 times/year
TOTAL				**12986**	

*Indicative Exchange Rates Announced at 15:30 on 17/12/2015 by the Central Bank of Turkey 1 USD=2.9364 TL

If there is no failure in the system, VRF systems maintenance costs include only filter cleaning. VRF annual costs are given in Table 7.14

Table 7.14 Annual VRF maintenance cost

Material	Quantity	Measure	Unit Price (*USD)	Total Price (*USD)	Maintenance Period
Indoor Unit	319	Number	20	6380	1 times/year
Outdoor Unit	39	Number	75	2925	1 times/year
TOTAL				**9305**	

*Indicative Exchange Rates Announced at 15:30 on 17/12/2015 by the Central Bank of Turkey 1 USD=2.9364 TL

7.5 Comparison of Conventional HVAC System and VRF

In the previous section, conventional and VRF system initial, operating and maintenance costs are evaluated. In this section both systems are compared. In Table 7.15 comparison of the systems are given.

Table 7.15 Initial, operating and maintenance comparison of conventional system and VRF

	Conventional System	VRF
Initial Investment Costs	862604 *USD	476702 *USD
Operating Costs	Electric Consumption 424248 KW x 0.14 *USD/KW = 59395 *USD	Electric Consumption 503634 KW x 0.14*USD/KW = 70509 *USD

	Annual Fuel Consumption 65994 kg/year x 0.75 *USD (P.O. Fair Price) = 49496 *USD	No Fuel Consumption
Total Operating Costs	108891 *USD	70509 *USD
Maintenance Costs	17456 *USD	9305 *USD
TOTAL	**988951 *USD**	**556516 *USD**
	VRF system has %44 cost profit	

*Indicative Exchange Rates Announced at 15:30 on 17/12/2015 by the Central Bank of Turkey 1 USD=2.9364 TL

7.6 Conclusion

In this chapter, the initial investment, operating and maintenance costs of conventional and HVAC systems are evaluated and compared. It is seen that VRF systems is more economical and applicable system. Initial investment, operating and maintenance of all costs are lower then conventional system.

CHAPTER 8

CONCLUSIONS

Nowadays energy costs are increasing; natural resources are rapidly depleted so the importance of energy efficiency is better understood. HVAC systems aims are to provide comfortable location to people. Not only providing comfort conditions, but also need to be considered costs. Especially lighting and operating costs creates the most significant expenditure. The design of the building should be well done because of to minimize of all these costs. The design which is as important as architectural design is mechanical system design.

In this study, The Social and Cultural Building which is in Kilis 7 Aralik University is examined. The study is one of the rare studies that contain insulation, heating and cooling accounts of a building. Building is heated and cooled by conventional system. The closed area is 8852 m^2. The building insulation accounts are evaluated by MMO TS 825 program. The conditions of evaporation and condensation in the building were investigated. In places where the account was missing, has been observed to occur in cracks in the walls.

The building is heated with hot water boilers which are 3 pass, 1500 KW capacities and liquid fueled. Fan coil, AHU and heat exchanger lines are supplied hot water by the boilers. The design outdoor temperature is taken -6 °C. The overall heat transfer coefficients are evaluated for all of the structural elements in the building. It is observed that the most heat loss is from the glass surface. Also heating equipment capacities are calculated.

The building is cooled by chillers which are 1000 KW capacities each other. Fan coil and AHU lines are supplied chilled water by air cooling chiller. Dry bulb, wet bulb temperatures and relative humidity is taken respectively, 39 °C, 23 °C and %50.

In this thesis VRF system is modelled to the building for the comparison of conventional system. Initial, operating and maintenance costs are compared with conventional system. As a result it is shown that VRF is more economical and technological system. If the additional losses and leakages are added (Because of the heat exchanger and due to ventilation lost), this rate will increase. Assembly, operation maintenance is easier than conventional systems. Much different equipment is used in conventional systems, so the rate of malfunction of the system increases.

Also it is quite difficult to find a suitable location for the chiller, boiler and air handling. VRF system alone is capable of full in a little area. VRF automation is in itself also without software, but for the conventional systems must be paid for the automation equipment and software.

In VRF systems modulating capacity control facilities is between %10-%100 and checks the room temperature with precision ± 0.5°C. But in conventional systems it can be up to 2-3 °C. Energy costs will increase by 5-6% every 1°C.

VRF systemôs other advantage is volume. At a same capacity of indoor unit is measured. VRF indoor unit volume loud is 31 DB, fan coil is 43 DB.

As a result the best operators in hotels, conventional systems are changed to VRF systems. Profits from investment, operations and maintenance costs are provided.

Results:

- Conventional systems annual carbon emission value is 245771 kg CO^2/year, the VRF system emission is 0. (Derivation of the electricity is ignored)
- Andesite covering is not a good insulation material and absorbs more water.
- The cracks occur in the building material, because all of the water is not vaporized during the year.
- Energy consumption is higher in AHUs, because of the fresh air inlet in the device.
- Heat recovery in AHUs should be preferred.
- The height of the floors is planning higher due to air duct which is used by AHU system.
- The pump which is frequency converter is more efficiency.

- 3 pass boiler is more efficiency.
- Water cooled chiller is more efficiency then air cooled one. But it is not preferred due to operating difficulties.
- Heat recovery VRF system is more efficiency then heat pump one, but it is not preferred due to higher initial investment costs.
- Conventional system which is placed on the roof cause visual pollution.
- 3-way valves lose their functionality over time.
- % 15 operational cost savings can be achieved by thermal insulation in buildings. This also means a saving of 4 billion USD in the country. But insulation materials should be selected correctly. For instance, XPS is banned since 2002 for toxic gas release.
- Climate factors in city, direction, insulation materials should be taken into consideration in insulation accounts.
- Diffusion account of thermal insulation should be evaluated.
- In the outside thermal insulation applications, the insulation materials water vapor resistance must be low; in the inside applications must be high.
- Almost the half of the heat loss from the building is occurs from glass surface. So architects should be reduced the rate of glass surface.
- Colored glass should be used to prevent solar gain in summer.
- The weights of the devices as AHU, Chiller etc. should be taken into consideration in static accounts.

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