DOKUZ EYLÜL UNIVERSITY
GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES

AIR TO AIR HEAT EXCHANGER PERFORMANCE

by
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April, 2008
İZMİR

AIR TO AIR HEAT EXCHANGER PERFORMANCE

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M.Sc THESIS EXAMINATION RESULT FORM

We have read the thesis entitled **"AIR TO AIR HEAT EXCHANGER PERFORMANCE"** completed by **CİHAN ÇANGARLI** under supervision of **PROF. DR. İSMAİL HAKKI TAVMAN** and we certify that in our opinion it is fully adequate, in scope and in quality, as a thesis for the degree of Master of Science.

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Cihan ÇANGARLI

AIR TO AIR HEAT EXCHANGER PERFORMANCE

ABSTRACT

Proper ventilation is obligatory in buildings where humans spend 70% of their life. Humidity, pollutant particles, smoke and harmful gases lower indoor air quality and cause sicknesses. To create an adequate air inside, dirty indoor air shall be exhausted and fresh air shall be taken from outdoors. To obtain controlled ventilation, buildings HVAC systems are designed by using mechanical ventilation. Mechanical ventilation is done with the help of an exhaust air fan, supply air fan and proper ventilation.

Exhausted indoor air is conditioned air, for summer it is cooled and in winter it is heated to design temperatures. To transfer heat between exhaust air and supply air, air to air heat exchangers (heat recovery exchangers) are used in modern ventilation systems. By the help of air to air heat exchangers, fresh air from outdoors is pre-heated or pre-cooled by the help of exhausted air thus decreases the energy used to condition indoor air.

Air to air heat exchangers are rated according to their efficiency which varies according to indoor and outdoor air temperatures. The same heat exchanger shall be evaluated according to the climate in which it is going to be installed.

In this research, a sample heat recovery ventilator has been examined according to its energy savings. Outdoor air temperatures received from Turkish State Meteorological Service has been evaluated by means of average values and hourly values. Energy savings in heating and cooling seasons are considered in different climates including warm, cold and extremely cold climates.

Keywords : Heat recovery exchanger performance, heat recovery ventilator evaluation, by-pass effect on heat recovery ventilators, heat recovery economical analysis and payback periods.

HAVADAN HAVAYA ISI DEĞİŞTİRGECİ PERFORMANSI

ÖZ

Günümüzde insan hayatının %70’i iç mekanlarda havalandırma yapılması zorunlu hale gelmiştir. Nem, kirletici partiküller, sigara dumanı vb. etkenler iç hava kalitesini bozmaktadır. İç ortamda hava kalitesini sağlamak için, oluşan kirli hava dışarı atılırken, dış ortamdan da taze hava alınması gereklidir. Modern binalarda ısıtma soğutma havalandırma sistem tasarımlarında, bu işlem kontrollü bir havalandırma sağlayabilmek için mekanik havalandırma yöntemleri ile yapılmaktadır. Mekanik havalandırma sistemlerinde basitçe bir egzost fanı ve bir taze hava fanı bulunmaktadır..

Dış ortama atılan hava, yaz mevsiminde soğutularak, kış mevsiminde ise ısıtılarak tasarım şartlarına getirilmiş havadır. Modern havalandırma sistemlerinde egzost edilen havanın enerjisini, dış ortamdan alınan taze havaya aktaran “havadan havaya ısı geri kazanım eşanjörleri” kullanılmaktadır. “Havadan havaya ısı geri kazanım eşanjörleri yardımı ile iç ortamdan egzost edilen havanın enerjisi kullanılarak, dış ortamdan alınan taze havanın ön soğutması veya ön ısıtması gerçekleştirilir, böylece iç ortamdaki havanın iklimlendirilmesi için harcanan enerji azaltılır.

“Havadan havaya ısı geri kazanım eşanjörleri” verimlerine gore sınıflandırılmaktadır ve bu verim değişik iç/dış hava koşullarında aynı tip eşanjörde bile farklılık göstermektedir. Bu sebepten tasarım yaparken, “havadan havaya ısı geri kazanım eşanjörünün” çalışacağı ortam koşullarına göre değerlendirilmesi gerekmektedir.

Bu çalışmada, örnek bir ısı geri kazanımlı havalandırma cihazı incelenmiştir. 2005 yılına ait dış hava koşulları saatlik Devlet Meteoroloji Enstitusunden alınarak, ılıman, soğuk ve aşırı soğuk iklimlerdeki ısı geri kazanımlı havalandırma sistemlerindeki enerji tasarrufu değerlendirilmiştir.

Anahtar Sözcükler : Isı geri kazanım eşanjörü performansı, ısı geri kazanımlı havalandırma cihazı analizi, ısı geri kazanımlı havalandırma cihazlarında by-pass etkisi, ısı geri kazanımlı havalandırma cihazlarının ekonomik analizi ve geri ödeme süreleri

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CHAPTER ONE

INTRODUCTION

1.1. Introduction

A heat exchanger is a component that allows the transfer of heat from one fluid (liquid or gas) to another fluid. Reasons for heat transfer include the following;

1. To heat a cooler fluid by means of a hotter fluid
2. To reduce the temperature of a hot fluid by means of a cooler fluid
3. To evaporate a liquid by means of a hotter fluid
4. To condense a gaseous fluid by means of a cooler fluid
5. To evaporate a liquid while condensing a hotter gaseous fluid

Regardless of the function the heat exchanger fulfills, in order to transfer heat the fluids involved must be at different temperatures and they must come into thermal contact. Heat can flow only from the hotter to the cooler fluid.

In a heat exchanger there is no direct contact between the two fluids. The heat is transferred from the hot fluid to the metal isolating the two fluids and then to the cooler fluid.

1.2. Types of heat exchanger construction

Although heat exchangers come in every shape and size imaginable, the construction of most heat exchangers falls into one of two categories: tube and shell, or plate. As in all mechanical devices, each type has its advantages and disadvantages.

1.2.1 Shell and Tube Heat Exchangers

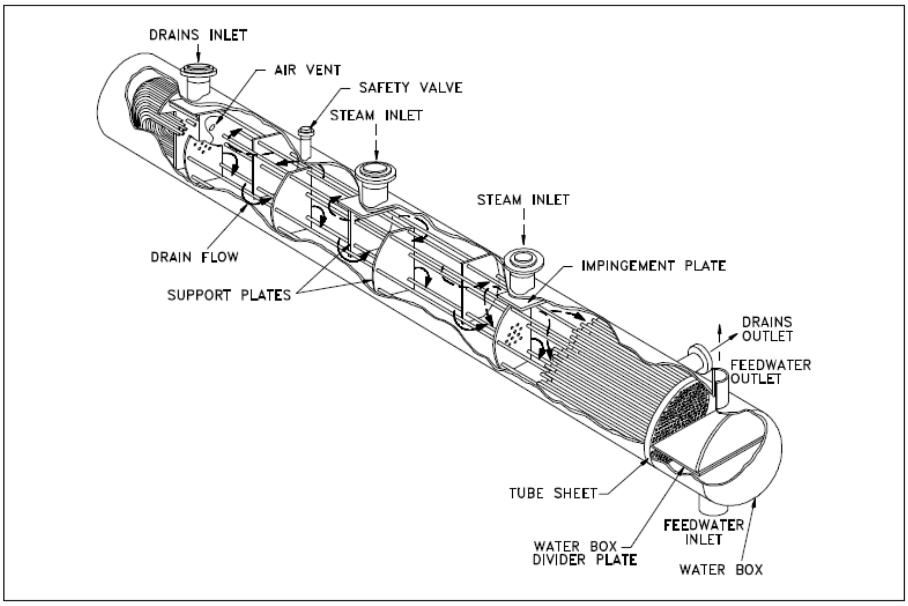


Figure 1.1 Shell and Tube Heat Exchanger

1.2.2 Plate Type Heat Exchangers

A plate type heat exchanger, as illustrated in Figure 1.2, consists of plates instead of tubes to separate the hot and cold fluids. The hot and cold fluids alternate between each of the plates. Baffles direct the flow of fluid between plates. Because each of the plates has a very large surface area, the plates provide each of the fluids with an extremely large heat transfer area. Therefore a plate type heat exchanger, as compared to a similarly sized tube and shell heat exchanger, is capable of transferring much more heat. This is due to the larger area the plates provide over tubes. Due to the high heat transfer efficiency of the plates, plate type heat exchangers are usually very small when compared to a tube and shell type heat exchanger with the same heat transfer capacity. Plate type heat exchangers are not widely used because of the inability to reliably seal the large gaskets between each of the plates. Because of this problem, plate type heat exchangers have only been used in small, low pressure applications such as on oil coolers for engines. However, new improvements in gasket design and overall heat exchanger design have allowed some large scale applications of the plate type heat exchanger. As older facilities are upgraded or newly designed facilities are built, large plate type heat exchangers are replacing tube and shell heat exchangers and becoming more common.

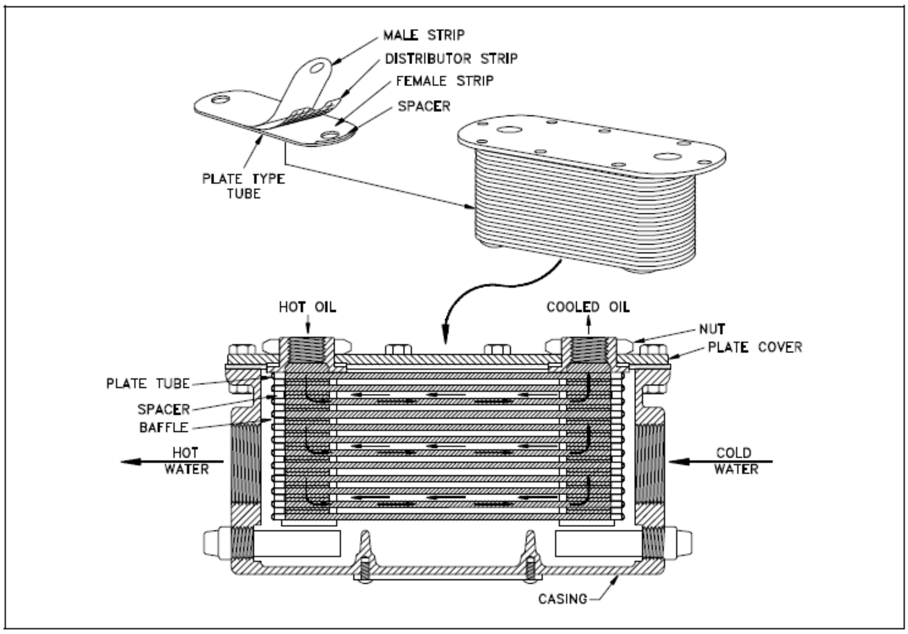


Figure 1.2 Plate Type Heat Exchanger

1.3. Types of heat exchangers

As heat exchangers come in so many shapes, sizes, makes, and models, they are categorized according to common characteristics. One common characteristic that can be used to categorize heat exchangers, is the direction of the flow of the two fluids relative to each other. The three categories are parallel flow, counter flow and cross flow.

Parallel flow, as illustrated in Figure 1.3, exists when both the tube side fluid and the shell side fluid flow in the same direction. In this case, the two fluids enter the heat exchanger from the same end with a large temperature difference. As the fluids transfer heat, hotter to cooler, the temperatures of the two fluids approach each other.

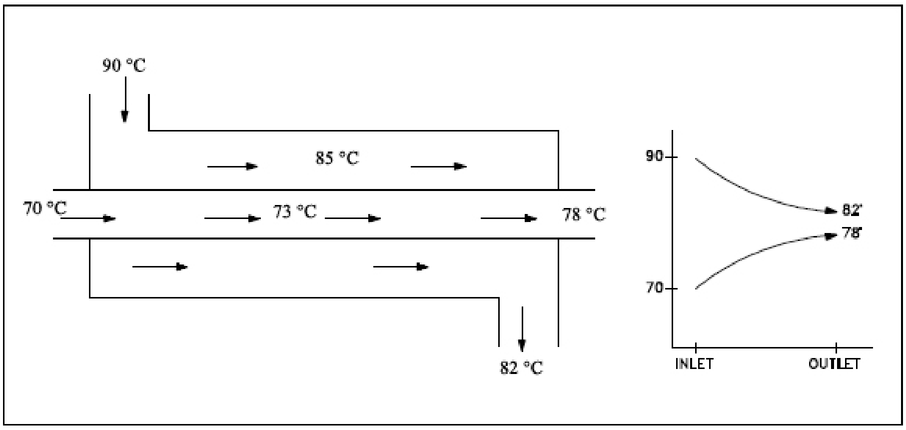


Figure 1.3. Parallel Flow Heat Exchanger

Counter flow, as illustrated in Figure 1.4, exists when the two fluids flow in opposite directions. Each of the fluids enters the heat exchanger at opposite ends. Because the cooler fluid exits the counter flow heat exchanger at the end where the hot fluid enters the heat exchanger, the cooler fluid will approach the inlet temperature of the hot fluid.

Counter flow heat exchangers are the most efficient of the three types. In contrast to the parallel flow heat exchanger, the counter flow heat exchanger can have the hottest coldfluid temperature greater than the coldest hot-fluid temperatue.

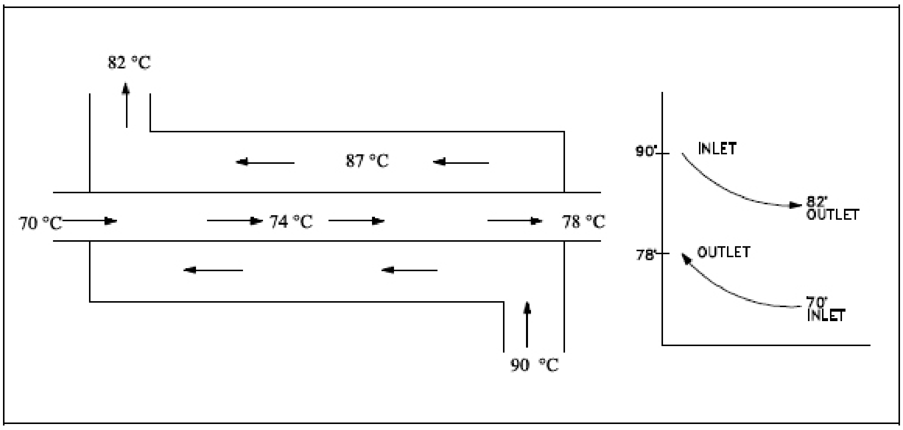


Figure 1.4 Counter Flow Heat Exchanger

Cross flow, as illustrated in Figure 1.5, exists when one fluid flows perpendicular to the second fluid; that is, one fluid flows through tubes and the second fluid passes around the tubes at 90° angle. Cross flow heat exchangers can be observed in applications where one of the fluids changes state (2-phase flow). An example is a steam system's condenser, in which the steam exiting the turbine enters the condenser shell side, and the cool water flowing in the tubes absorbs the heat from the steam, condensing it into water. Large volumes of vapor may be condensed using this type of heat exchanger flow.

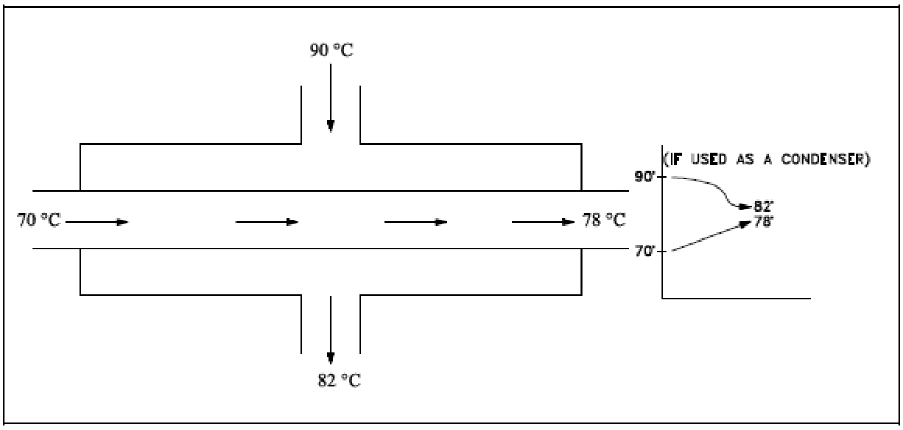


Figure 1.5. Cross Flow Heat Exchanger

1.4. Comparision of the types of heat exchangers

Each of the three types of heat exchangers has advantages and disadvantages. But of the three, the counter flow heat exchanger design is the most efficient when comparing heat transfer rate per unit surface area. The efficiency of a counter flow heat exchanger is due to the fact that the average ΔT (difference in temperature) between the two fluids over the length of the heat exchanger is maximized, as shown in Figure 1.4. Therefore the log mean temperature for a counter flow heat exchanger is larger than the log mean temperature for a similar parallel or cross flow heat exchanger. This can be seen by comparing the graphs in Figure 1.3, Figure 1.4, and Figure 1.5.

In actuality, most large heat exchangers are not purely parallel flow, counter flow, or cross flow; they are usually a combination of the two or all three types of heat exchangers. This is due to the fact that actual heat exchangers are more complex than the simple components shown in the idealized figures used above to depict each type of heat exchanger. The reason for the combination of the various types is to maximize the efficiency of the heat exchanger within the restrictions placed on the design. That is, size, cost, weight, required efficiency, type of fluids, operating pressures, and temperatures, all help determine the complexity of a specific heat exchanger.

One method that combines the characteristics of two or more heat exchangers and improves the performance of a heat exchanger is to have the two fluids pass each other several times within a single heat exchanger. When a heat exchanger's fluids pass each other more than once, a heat exchanger is called a *multi-pass heat exchanger*. If the fluids pass each other only once, the heat exchanger is called a *single-pass heat exchanger*.

Commonly, the multi-pass heat exchanger reverses the flow in the tubes by use of one or more sets of "U" bends in the tubes. The "U" bends allow the fluid to flow back and forth across the length of the heat exchanger. A second method to achieve multiple passes is to insert baffles on the shell side of the heat exchanger. These direct the shell side fluid back and forth across the tubes to achieve the multi-pass effect.

CHAPTER TWO

HEAT EXCHANGER CALCULATIONS

2.1 Heat Exchanger Design Methods

The goal of heat exchanger design is to relate the inlet and outlet temperatures, the overall heat transfer coefficient, and the geometry of the heat exchanger, to the rate of heat transfer between the two fluids. The two most common heat exchanger design problems are those of rating and sizing.

The enthalpy balance on either fluid stream is:

$$Q_c = \dot{m}_c \left(h_{c2} - h_{c1}\right) \tag{2.1}$$

and likely:

$$Q_h = \dot{m}_h \left(h_{h2} - h_{h1}\right) \tag{2.2}$$

For constant specific heats with no change of phase, we may also write :

$$Q_c = \left(\dot{m}c_p\right)_c \left(T_{c2} - T_{c1}\right) \tag{2.3}$$

and likely:

$$Q_h = \left(\dot{m}c_p\right)_h \left(T_{h2} - T_{h1}\right) \tag{2.4}$$

From energy conservation we know that Qc = Qh = Q, and that we may relate the heat ransfer rate Q and the overall heat transfer coefficient U, to the mean temperature difference ΔT_m by means of

$$Q = UA\Delta T_m \tag{2.5}$$

where A is the total surface area for heat exchange that U is based upon. Later we shall show that

$$\Delta T_m = f\left(T_{h1}\,,T_{h2}\,,T_{c1}\,,T_{c2}\right) \tag{2.6}$$

It is now clear that the problem of heat exchanger design comes down to obtaining an expression for the mean temperature difference. Expressions for many flow configurations, i.e. parallel flow, counter flow, and cross flow, have been obtained in the heat transfer field. We will examine these basic expressions later. Two approaches to heat exchanger design that will be discussed are the LMTD method and the effectiveness - NTU method. Each of these methods has particular advantages depending upon the nature of the problem specification.

2.2 Overall Heat Transfer Coefficient

A heat exchanger analysis always begins with the determination of the overall heat transfer coefficient. The overall heat transfer coefficient may be defined in terms of individual thermal resistances of the system. Combining each of these resistances in series gives:

$$\frac{1}{UA} = \frac{1}{(\eta_o hA)_i} + \frac{1}{Sk_w} + \frac{1}{(\eta_o hA)_o} \tag{2.7}$$

where η_0 is the surface efficiency of inner and outer surfaces, h is the heat transfer coefficients for the inner and outer surfaces, and S is a shape factor for the wall separating the two fluids.

The surface efficiency accounts for the effects of any extended surface which is present on either side of the parting wall. It is related to the fin efficiency of an extended surface in the following manner:

$$\eta_o = \left(1 - \left(1 - \eta_f\right)\frac{A_f}{A}\right) \tag{2.8}$$

The thermal resistances include: the inner and outer film resistances, inner and outer extended surface efficiencies, and conduction through a dividing wall which keeps the two fluid streams from mixing. The shape factors for a number of useful wall configurations are given below in Table 1. Additional results will be presented for some complex doubly connected regions.

Equation (2.7) is for clean or unfouled heat exchanger surfaces. The effects of fouling on heat exchanger performance is discussed in a later section. Finally, we should note that:

$$UA = U_o A_o = U_i A_i \tag{2.9}$$

however ,

$$U_o \neq U_i \tag{2.10}$$

Finally, the order of magnitude of the thermal resistances in the defintion of the overall heat transfer coefficient can have a significant influence on the calculation of the overall heat transfer coefficient. Depending upon the nature of the fluids, one or more resistances may dominate making additional resistances unimportant. For example, in Table 2 if one of the two fluids is a gas and the other a liquid, then it is easy to see that the controlling resistance will be that of the gas, assuming that the surface area on each side is equal.

Table 2.1 Shape Factors

GEOMETRY	**S**
Plane Wall	$\frac{A}{t}$
Cylindrical Wall	$\frac{2\pi L}{\ln\left(\frac{r_o}{r_i}\right)}$
Spherical Wall	$\frac{4\pi r_i r_o}{r_o - r_i}$

Table 2.2 *Order of Magnitude of h*

FLUID	**h (W/m^2K)**
Gases (Natural Convection)	5-25
Gases (Forced Convection)	10-250
Liquids (Non-metal)	100-1.000
Liquid Metals	5.000-250.000
Boiling	1.000-250.000
Condensation	1.000-25.000

2.3 Thermal Effectiveness

The thermal effectiveness is defined as:

$$\varepsilon = \frac{Actual\ Heat\ Transfer\ Rate}{Theoretical\ Maximum\ Heat\ Transfer\ Rate} \tag{2.11}$$

The maximum theoretical heat transfer rate occurs in counterflow with infinite heat transfer surface area. It cannot occur in parallel flow because the exit temperature must be between the two inlet temperatures.

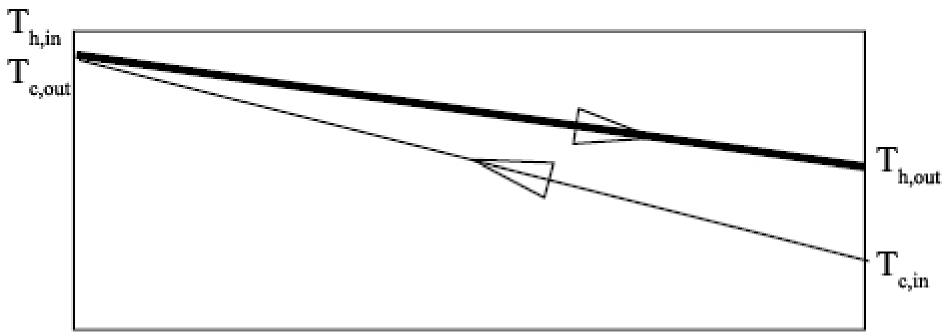


Figure 2.1 Temperature change in a counter flow heat exchanger

In the infinite surface area heat exchanger above, the thermal capacity of the cold fluid is less than that of the hot fluid. It is kept in thermal contact sufficiently for it to emerge at the hot fluid inlet temperature.

The maximum theoretical heat transfer is given by:

$$Q_{\max} = (\dot{m}c)_{\min}\left(T_{H,in} - T_{C,in}\right) \tag{2.12}$$

The actual heat transfer rate is given (as above) from:

$$Q_{\max} = (\dot{m}c)_H \left(T_{H,in} - T_{H,out}\right) = (\dot{m}c)_C \left(T_{C,out} - T_{C,in}\right) \tag{2.13}$$

It follows that if the hot fluid has the lower thermal capacity:

$$\varepsilon = \frac{(\dot{m}c)_h \left(T_{H,in} - T_{H,out}\right)}{(\dot{m}c)_h \left(T_{H,in} - T_{C,in}\right)} = \frac{\left(T_{H,in} - T_{H,out}\right)}{\left(T_{H,in} - T_{C,in}\right)} \tag{2.14}$$

And if the cold fluid has the lower thermal capacity:

$$\varepsilon = \frac{(\dot{m}c)_h \left(T_{C,out} - T_{C,\text{int}}\right)}{(\dot{m}c)_h \left(T_{H,in} - T_{C,in}\right)} = \frac{\left(T_{C,out} - T_{C,in}\right)}{\left(T_{H,in} - T_{C,in}\right)} \tag{2.15}$$

2.4 LMTD Method

The log mean temperature difference (LMTD) is derived in all basic heat transfer texts. It may be written for a parallel flow or counterflow arrangement. The LMTD has the form:

$$\Delta T_{LMTD} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\dfrac{\Delta T_2}{\Delta T_1}\right)} \tag{2.16}$$

where ΔT_1 and ΔT_2 represent the temperature difference at each end of the heat exchanger, whether parallel flow or counterflow. The LMTD expression assumes that the overall heat transfer coefficient is constant along the entire flow length of the heat exchanger. If it is not, then an incremental analysis of the heat exchanger is required. The LMTD method is also applicable to crossflow arrangements when used with the crossflow correction factor. The heat transfer rate for a crossflow heat exchanger may be written as:

$$Q = FU\Delta T_{LMTD} \tag{2.17}$$

where the factor F is a correction factor, and the log mean temperature difference is based upon the counterflow heat exchanger arrangement.

The LMTD method assumes that both inlet and outlet temperatures are known. When this is not the case, the solution to a heat exchanger problem becomes somewhat tedious. An alternate method based upon heat exchanger effectiveness is more appropriate for this type of analysis. If $\Delta T_1 = \Delta T_2 = \Delta T$, then the expression for the LMTD reduces simply to ΔT.

2.5 ε-NTU Method

The effectiveness / number of transfer units (NTU) method was developed to simplify a number of heat exchanger design problems. The heat exchanger effectiveness is defined as the ratio of the actual heat transfer rate to the maximum possible heat transfer rate if there were infinite surface area. The heat exchanger effectiveness depends upon whether the hot fluid or cold fluid is a minimum fluid. That is the fluid which has the smaller capacity coefficient $c = \dot{m}c_p$. If the cold fluid is the minimum fluid then the effectiveness is defined as:

$$\varepsilon = \frac{C_{\max}\left(T_{H,in} - T_{H,out}\right)}{C_{\min}\left(T_{H,in} - T_{C,in}\right)} \tag{2.18}$$

otherwise, if the hot fluid is the minimum fluid, then the effectiveness is defined as:

$$\varepsilon = \frac{C_{\max}\left(T_{C,out} - T_{C,in}\right)}{C_{\min}\left(T_{H,in} - T_{C,in}\right)} \tag{2.19}$$

We may now define the heat transfer rate as:

$$Q = \varepsilon\, C_{\min}\left(T_{H,in} - T_{C,in}\right) \tag{2.20}$$

It is now possible to develop expressions which relate the heat exchanger efectiveness to another parameter referred to as the nmber of transfer units (NTU). The value of NTU is defined as:

$$NTU = \frac{UA}{C_{\min}} \tag{2.21}$$

It is now a simple matter to solve a heat exchanger problem when

$$\varepsilon = f\left(NTU, C_r\right) \tag{2.22}$$

where $C_r = C_{min}/C_{max}$

Numerous expressions have been obtained which relate the heat exchanger effectiveness to the number of transfer units. The handout summarizes a number of these solutions and the special cases which may be derived from them. For convenience the ε −NTU relationships are given for a simple double pipe heat exchanger for parallel flow and counter flow:

Parallel Flow

$$\varepsilon = \frac{1 - \exp\left[-NTU\left(1 + C_r\right)\right]}{1 + C_r} \tag{2.23}$$

or

$$NTU = \frac{-\ln\left[1 - \varepsilon\left(1 + C_r\right)\right]}{1 + C_r} \tag{2.24}$$

Counter Flow

$$\varepsilon = \frac{1-\exp[-NTU(1-C_r)]}{1+C_r \exp[-NTU(1-C_r)]} \quad ,C_r < 1 \tag{2.25}$$

and

$$\varepsilon = \frac{NTU}{1+NTU} \quad , C_r = 1 \tag{2.26}$$

or

$$NTU = \frac{1}{C_r - 1} \ln\left(\frac{\varepsilon - 1}{\varepsilon C_r - 1}\right) \quad , C_r < 1 \tag{2.27}$$

and

$$NTU = \frac{\varepsilon}{1-\varepsilon} \quad , C_r = 1 \tag{2.28}$$

We would normally know ***C & NTU***, and hence can find ***E***. Hence we can find the exit temperature of the lower thermal capacity stream. Graphs are often more convenient to use than formulae.

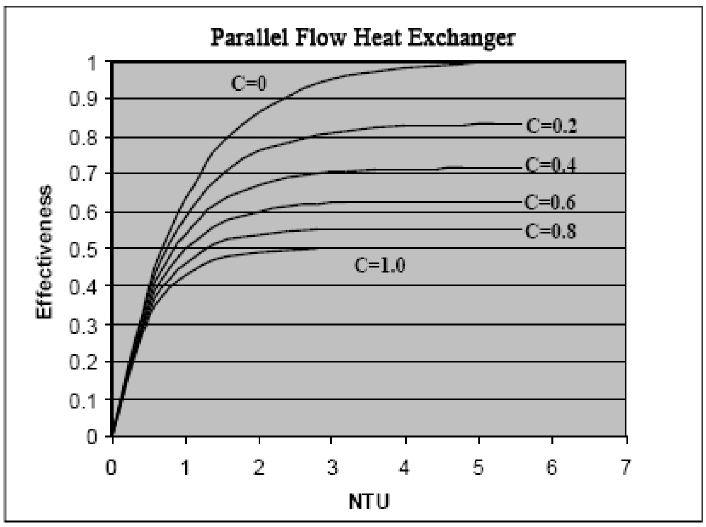


Figure 2.2 ε-NTU Diagram for Parallel Flow Heat Exchanger

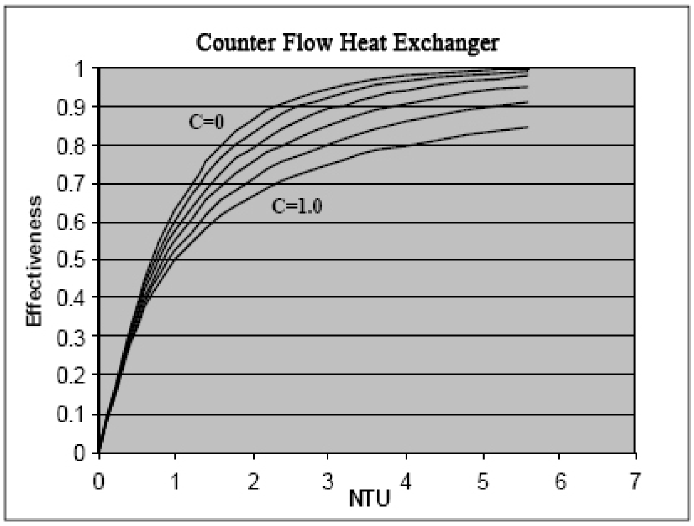


Figure 2.3 ε-NTU Diagram for Counter Flow Heat Exchanger

CHAPTER THREE

HEAT RECOVERY VENTILATION

After the energy crisis in 70's, new energy considerations have been developed. The limitation for the decreasing availability of Carbon based energy resources encouraged scientists and governments to design and use renewable resources like wind, geothermal and sun. According to the researches made, 70% of annual electrical energy consumption and 50% natural gas consumption are made add residential buildings. (*Better Building Brighter Future March 2003*.)

Investors and constructor companies nowadays pay attention not only to the installation costs and operational costs but also the advantage of creating high comfort and productive building in marketing new buildings.

With the development of new construction material, tight buildings are developed and the costs of heating/cooling systems are reduced. In such buildings infiltration between outdoor air and indoor air is minimized to ensure low energy loss to surroundings. Although this helps to reduce buildings installation and operational costs for heating/cooling system it also reduces the ventilation rate among construction material thus reducing indoor air quality. The reduction in indoor air quality forms well known "Sick Building Syndrome-SBS". The major factor for SBS is low/insufficient ventilation. As a result of the oil boycott in 1973, the ventilation demand per person is reduced 65% to 18 m3/h and the newly constructed building are designed with this criteria. (*U.S. Environmental Protection Agency*)

Most of the buildings constructed in that period are now having HVAC system revisions to fit today needs. Indoor air quality which majorly is dependant to fresh air rate shall also be regarded with filtration quality, moisture ratio and particles indoors. So, while realizing indoor air quality, ventilation, humidity control, filtration and contaminants must be evaluated in common.

The development of new construction material also affected the way of ventilation. In old design, for most buildings, infiltration through walls, windows etc. also regarded as natural ventilation, was enough, but with the reduction in infiltration, the designers changed the system to mechanical ventilation. Mechanical ventilation, basically, is removal of dirty indoor air and/or taking fresh outdoor air by means of electrically driven fans where pressure difference and/or gravity is not sufficient for ventilation. Mechanical ventilation systems can be designed with only two fans or they can consist of filters according to desired particle rate indoors, humidifiers/dehumidifiers according to desired indoor humidity ratio, heat recovery exchangers to transfer exhaust air energy from/to fresh air, heating/cooling coils to heat/cool fresh air to desired indoor temperature.

Heat recovery ventilators (HRV) are designed to have controlled ventilation for indoors and minimize energy loss due to ventilation. In winter, the principle of HRV’s is to transfer exhaust air energy to fresh air, thus decreasing the heating load indoors and in summer to transfer fresh airs energy to exhaust air and decrease the cooling load indoors.

The initial cost and the operational cost for HRV’s are studied by Nyman M. and Simonson CJ for cold climates in 2005. As a result of the research it is shown that HRV’s are energy saving solutions in HVAC systems and with the increase in the efficiency of the unit, energy savings increase significantly. (Nyman M. & Simonson C.J. , 2007)

Also Palin S.L, McIntyre D.A. and Edwards R.E., approved that HRV’s are more efficient to obtain low indoors humidity ratio than natural ventilation system and systems with only exhaust fans. (Palin S.L., McIntyre D.A. & Edwards R.E., 2007)

3.1 Ventilation

Ventilation is a necessity for the health and comfort of occupants of all buildings. Ventilation supplies air for occupants to breathe and removes moisture, odors, and indoor pollutants like carbon dioxide. Ventilation design for apartment buildings is inherently more complex than what is required for single-family homes. Most apartments have limited exposure of walls and windows to the outside environment. Additionally, the natural physical forces that move air are more pronounced in taller buildings. This includes "infiltration" and "exfiltration" the unintentional and uncontrollable flow of air through cracks and leaks in the building envelope. There are two primary forms of intentional ventilation -natural and mechanical. Low-rise buildings (3 stories and under) often utilize "natural" ventilation, that is, air supplied and vented through operable windows. High-rise buildings (over 3 stories) often use "mechanical" ventilation systems in the form of fans, air-inlets, ducts and registers, but may also rely on operable windows when mechanical systems fail to provide adequate ventilation.

Sufficient ventilation is necessary for occupant comfort and maintaining building integrity. Ventilation air is needed in all habitable spaces including common areas used for circulation, such as hallways and stairwells. Ventilation may also be needed in lobbies, storage spaces, parking areas, janitor's offices, and mechanical and equipment rooms.

3.1.1 Natural and Mechanical Ventilation

Most residences rely exclusively on infiltration and natural ventilation strategies. The main drawback to these ventilation strategies is the lack of control. Unreliable driving forces can result in periods of inadequate ventilation followed by periods of over-ventilation which can cause excessive energy waste. Good design can provide some measure of ventilation control, but normally it is necessary for the occupant to adjust ventilation openings to suit demand.

When infiltration and natural ventilation systems are inadequate (as determined either by code or experience), mechanical ventilation should be installed. Mechanical ventilation systems are capable of providing a controlled rate of air exchange and should respond to the varying needs of occupants and pollutant loads, irrespective of climate vagaries. While some systems filter supply air, others have provisions for energy recovery from the exhaust air stream. In some countries, especially in parts of Canada and Scandinavia, mechanical systems are being incorporated into virtually all new apartment building construction and are also being included in many building renovation programs.

The typical apartment building “mechanical ventilation” system has a central supply system which conditions the air (e.g., heats, cools, and filters) and individual exhaust fans serving each apartment. Both natural and mechanical ventilation systems must be installed and operated correctly to provide proper ventilation. Decisions on whether to provide natural and or mechanical supply-only, exhaust-only, or both supply and exhaust will depend on several ventilation-related factors, including:

- Weather
- Building configuration
- Access to ventilation
- Tenant behavior
- Cost.

3.2 Heat Recovery Ventilation for Comfort Applications

In principle, heat recovery ventilator, HRV systems consist of two electrically driven fans (one for exhaust air and one for fresh air), heat recovery exchanger, fresh and exhaust air filters, air ducts, diffusers and control system. As a result of HRV systems usage, it is possible to reduce indoor heat loss due to ventilation between 50-75% according to room insulation and reduce total heat loss between 20-60%. (Kragh Jesper , Rose Jørgen & Svendsen Svend Mechanical Ventilation with heat recovery in cold climates. *Proceedings of the 7th Symposium on Building Physics in the Nordic Countries 1033-1040).* In most cases exhaust diffusers are placed where there is humidity, pollutants or other contaminants and fresh air diffusers are placed where humans spend more time (i.e. office spaces, meeting rooms, living rooms etc.).

Mechanical ventilation has the disadvantages like initial cost, the space required for the ventilation system including the unit and air ducts, electrical consumption etc.

To minimize electrical consumption, in design era, the designer shall beware of pressure drops.

There are 3 well known heat exchanger type that are used in heat recovery ventilation system for comfort applications

3.2.1 Plate type heat recovery exchangers

Plate type heat recovery exchangers are made of Aluminum, plastic or paper plates which have a corrugated surface to enlarge heat transfer area. There two types according to flow configurations.

a. Cross Flow
b. Counter Flow

In the past, plate type heat recovery exchangers are made of metallic material to have higher heat transfer coefficients, nowadays with the development of new production methods, paper and plastic materials are also used. Although the heat transfer coefficient for these two elements are significantly lower than metals, they can be manufactured very thin which enhances the heat transfer coefficient and also makes it possible to have corrugated surfaces to increase heat transfer area. Also for places where there is high humidity difference between indoor and outdoor air, paper type heat exchangers (membrane type) makes it possible to transfer not only heat but mass also.

Figure 3.1 Specially joined Aluminum plates.

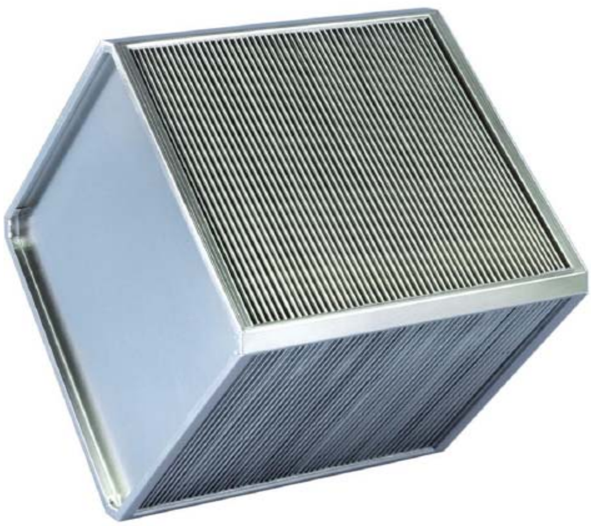


Figure 3.2 Cross flow heat recovery exchanger (Aluminum type).

Figure 3.3 Counter flow heat recovery exchanger (Aluminum type).

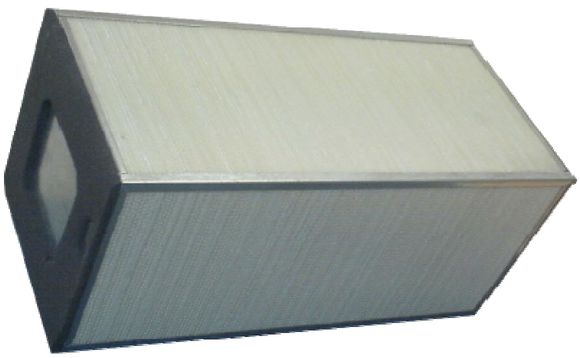


Figure 3.4 Cross flow heat recovery exchanger (Paper type).

Figure 3.5 Counter flow heat recovery exchanger (Plastic type).

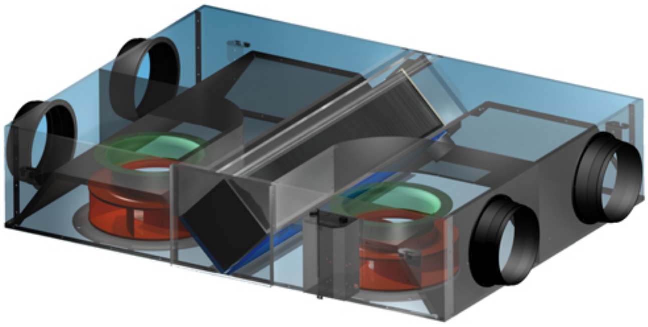


Figure 3.6 Cross flow heat recovery exchanger installation sample.

3.2.2 Rotary type heat recovery exchangers

A rotary air-to-air heat exchanger, or heat wheel, is a revolving cylinder filled with heat transfer media through which the air passes. Supply air flows through half of the wheel, exhaust air through the other half. A partition is arranged to separate the two airstreams, although some leakage and cross-contamination may occur. This may be minimized by use of a purge section between the two halves and by making the clean airstream ‘‘positive’’ with respect to the exhaust airstream. Heat transfer may be sensible only or sensible plus latent, depending on the type of heat transfer medium selected. Even the sensible-heat-only design may involve some latent heat transfer if there is a buildup of hygroscopic dust on the medium. Capacity may be controlled by varying the rotational speed.

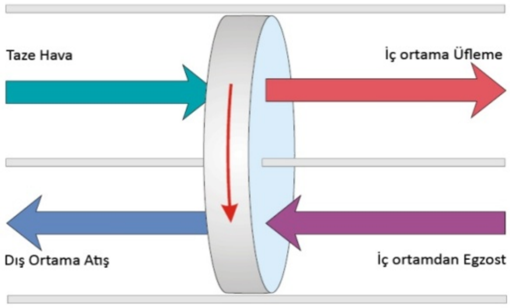


Figure 3.7 Rotary type heat recovery exchanger operation scheme.

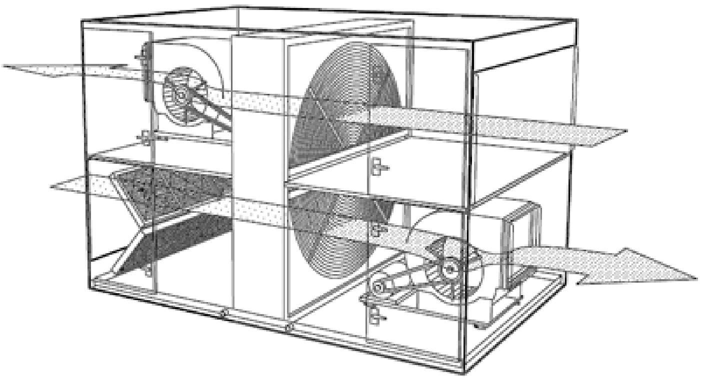


Figure 3.8 Rotary type heat recovery exchanger design sample.

Figure 3.9 Rotary type heat recovery exchanger.

3.2.3 Heat Pipe heat recovery exchangers

A heat-pipe energy recovery system consists of a bank of closed tubes, each of which operates independently. The tube is lined with a capillary wick, partially filled with a suitable refrigerant and sealed (Fig.3.10). One end of the tube is in the warm airstream, the other in the cold airstream; an external partition divides the two airstreams. The warm air vaporizes the refrigerant, and vapor migrates through the tube and is condensed in the cold end. The condensed liquid returns to the warm end through the wick. This is a passive system, being driven entirely by the temperature difference between the two airstreams. Tilting the tube to increase the liquid flow rate will increase the capacity; the capacity can be automatically controlled by varying the degree of tilt. The tube may be finned or bare.

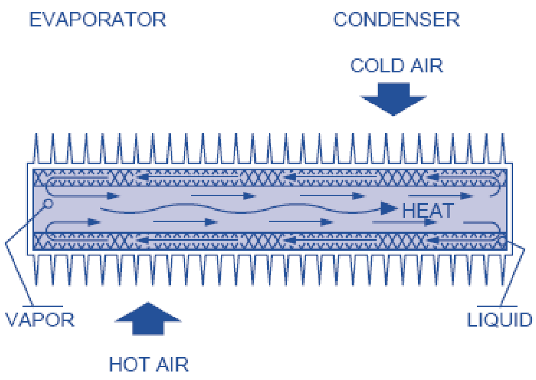


Figure 3.10 Heat pipe heat recovery exchanger operation scheme.

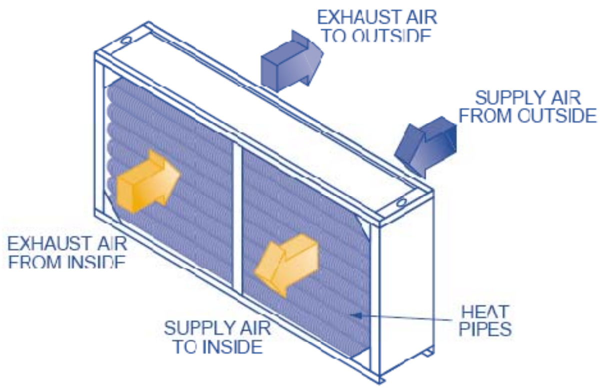


Figure 3.11 Heat pipe heat recovery operation sample

3.3 Heat and Mass Transfer Calculations

Heat transfer in an air to air heat exchanger is the energy transported from one air stream to another due to temperature differences, also known as sensible heat (W. Kays, M.Crawford and B. Weigand, 2005). The heat transfer increases with the temperature difference and the ability of membrane to make the transfer.

The heat transfer occurs in three steps.

1. Convective heat transfer on the hot side
2. Heat conduction through the Aluminum plate
3. Convective heat transfer on the cold side

The nature of the mass transfer are similar to the physics of heat transfer. Mass transfer in an air to air heat exchanger is the energy transported from an air stream to another due to differences in humidity. The mass transfer increases with the humidity difference and the ability of the membrane to make the transfer. The mass transfer analogy is the same as the heat transfer case.

1. Convective mass transfer on the humid side
2. Mass diffusion through the membrane
3. Convective mass transfer on the dry side

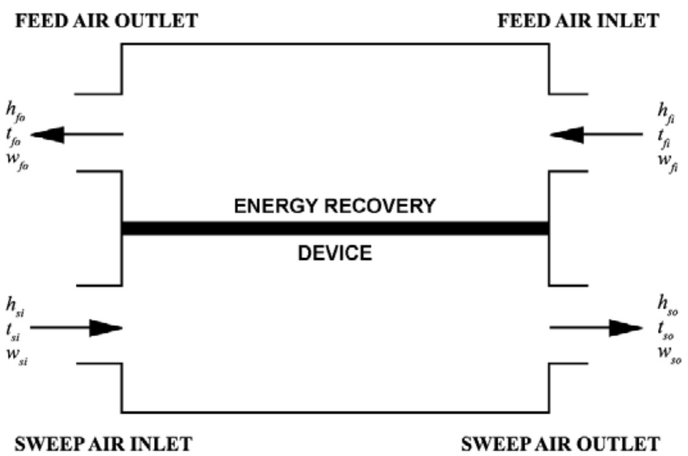


Figure 3.12 Airstream numbering convention.

3.3.1 Enthalpy

Since both sensible and latent heats are exchanged in HRV, the interesting measurement is a combination of both these measurements. This combined measurement is the sum of the internal energy of air, which is known as the enthalpy. The specific enthalpy of the air can be calculated as (L. Z. Zhang & Y. Jiang, , 2007).

$$h = 1{,}005t + \omega(2501 + 1{,}68T_{ort}) \tag{3.1}$$

Where h=specific enthalpy (kJ/kg of dry air), T= temperature(C), ω =absolute humidity (kg/kg of dry air) .

The following definitions and equations are based on the same mass flow rate of the feed and sweep channels ($\dot{m}_f$= $\dot{m}_s$).

$$\Delta T = \left|T_{fi} - T_{fo}\right| = |T_{so} - T_{si}| \tag{3.2}$$

$$\Delta t_{tot} = \left|T_{fi} - T_{si}\right| \tag{3.3}$$

$$\Delta\omega = \left|\omega_{fi} - \omega_{fo}\right| = |\omega_{so} - \omega_{si}| \tag{3.4}$$

$$\Delta\omega_{tot} = \left|\omega_{fi} - \omega_{si}\right| \tag{3.5}$$

$$\Delta h = \left|h_{fi} - h_{fo}\right| = |h_{so} - h_{si}| \tag{3.6}$$

$$\Delta h_{tot} = \left|h_{fi} - h_{si}\right| \tag{3.7}$$

Where Δt, $\Delta\omega$, Δh is the difference in temperature, humidity and enthalpy respectively between air inlet and outlet of one air stream. ΔT_{tot}, $\Delta\omega_{tot}$, Δh_{tot} is the total temperature, humidity and enthalpy difference respectively between the inlets of the HRV. The subscripts are *f*=feed, s=sweep, *i*=inlet, *o*=outlet.

The enthalpy transferred in the HRV is proportional to the mass flow rate and the differential of enthalpy between the inlet and outlet air. Since all energy is conserved and only exchanged between the air streams, the total enthalpy of feed and sweep air streams are equal. The total transferred enthalpy is (L. Z. Zhang & Y. Jiang, 2007) ;

$$H = \dot{m}\Delta h \tag{3.8}$$

Where H=total enthalpy transferred (kJ/s), $\dot{m}$=mass flow rate (kg/s).

3.3.2 Efficiency

The sensible efficiency is the amount of transferred heat in the HRV in relation to the heat difference between the inlets and is calculated as:

$$\varepsilon_S = \frac{\Delta T}{\Delta T_{tot}} \tag{3.9}$$

The latent efficiency is the amount of transferred humidity in the HRV in relation to the humidity difference between the inlets and is calculated as:

$$\varepsilon_L = \frac{\Delta\omega}{\Delta\omega_{tot}} \tag{3.10}$$

The enthalpy efficiency of the HRV is the fraction of the total enthalpy going through the unit which is exchanged between the air streams. The enthalpy efficiency is calculated as:

$$\varepsilon_{tot} = \frac{\Delta h}{\Delta h_{tot}} \tag{3.11}$$

Where ε_{tot}=the enthalpy efficiency ($0<\varepsilon_{tot}<1$). The efficiency value of 1 is when all the enthalpy is exchanged and re-tranferred. An efficiency value of 0 is when none of the enthalpy is exchanged.

3.3.2.1 Theoretical efficiency of a HRV

It is hard to construct an exact theoretical model for the HRV, but a study made by Zhang&Niu showed that the efficiency of the HRV can be approximated with the help of the dimensionless Number of Transfer Units (NTU). NTU reflects the sensible heat exchanged in HRV and is calculated (L. Z. Zhang & J. L. Niu, 2007) as;

$$NTU = \frac{A_{tot}U}{\dot{m}c_{pa}} \tag{3.12}$$

Where A_{tot}=total area of plates (m^2),U=total heat transfer coefficient (kW/m^2K), c_{pa}= specific heat (kJ/kgK).

The total heat transfer coefficient U is calculated as:

$$U = \left[\frac{1}{u_f} + \frac{\delta}{\lambda} + \frac{1}{u_s}\right]^{-1} \tag{3.13}$$

Where u_f, u_s are the convective heat transfer coefficients for feed and sweep channel (kW/m^2K). δ is the plate thickness(m) and λ is the thermal conductivity of the membrane (kW/mK).

The sensible efficiency for unmixed cross flow can be expressed as:

$$\varepsilon_s = 1 - exp\left[\frac{exp(-NTU^{0,78}) - 1}{NTU^{-0,22}}\right] \tag{3.14}$$

The number of transfer units for the latent heat is similar to the definition in sensible heat;

$$NTU_L = \frac{A_{tot}U_L}{\dot{m}} \tag{3.15}$$

Where U_L is the total mass transfer coefficient (kg/m^2s)

And the corresponding latent heat efficiency for unmixed cross flow is defined as:

$$\varepsilon_L = 1 - exp\left[\frac{exp(-NTU_L{}^{0,78})-1}{NTU_L{}^{-0,22}}\right] \quad (3.16)$$

NTU_L can also be calculated as $NTU_L=\beta.NTU$ *and* β is the ratio os total number of transfer units for latent to sensible heat.

The enthalpy efficiency can be calculated as a combination of ε_s and ε_L as:

$$\varepsilon_{tot} = \frac{\varepsilon_s+\varepsilon_L H^*}{1+H^*} \quad (3.17)$$

Where H^* is the ratio of latent to sensible energy difference between the inlets of two air streams and is approximated to:

$$H^* \approx 2501\frac{\Delta\omega_{tot}}{\Delta t_{tot}} \quad (3.18)$$

Niu & Zhang states that the sensible efficiency of an operational HRV is only affected by the mass flow rate, which is the only dynamic factor (J. L. Niu and L. Z. Zhang, 2007). Increasing mass flow rate would decrease the number of transfer units for heat and thereby decrease the sensible efficiency. The latent efficiency is more complex coupled to the operational conditions of the HRV. Unlike the thermal diffusive resistance, the moisture diffusive resistance is not a constant and that will influence β. The moisture diffusive resistance is depending on the membrane material, temperature and humidity difference. A membrane material with a linear sorption curve will give the highest latent efficiency and have the same ability to transfer heat and moisture regardless the temperature and humidity conditions (L. Z. Zhang & J. L. Niu, 2007).

3.3.3 Test Procedure of Heat Recovery Exchangers

There exist two test methods for air to air heat exchangers that are used in heat recovery ventilators for commercial use. The first one is developed by American Society of Heating, Refrigeration and Air Conditioning Engineers, ASHRAE with the standard number 84. According to this standard an evaluation program by Air Conditioning and Refrigeration Industry, ARI has been developed and the manufacturers are treated with this certification program 1060-2005. For European market, a certification program is also been evolved by European Committee of Air Handling and Air Conditioning Equipment Manufacturers, Eurovent according to the European Norm, EN308 and named “Air to Air heat exchanger performance”. Eurovent program requires design software for the whole range of manufactured heat recovery exchangers. In this program, it is required to calculate;

- Pressure drop across the heat exchanger
- Heat Recovered
- Outlet Air conditions
- Condensation in the air flows

according to given air flow rate for both air streams, air conditions (temperature and humidity).

The certification process starts with the submittal of the software to Eurovent. Eurovent randomly selects a heat recovery exchanger and log the calculation results from the software. Manufacturer is informed to deliver selected heat recovery exchanger to Eurovent laboratories. After the tests are made according to EN 308, the results that are logged by the output of the selection software of the manufacturer and the test results are compared. If the calculated data is in the range of the test data then, the selection software receives the approval from Eurovent. This test is renewed each 6 months to ensure continuity.

Test procedure for air to air heat/energy ventilation equipment has been identified in ARI Standard 2005.1060 “Performance Rating of Air-To-Air Heat Exchangers for

Energy Recovery Ventilation Equipment". The test is established to gain the data of; airflow, pressure drop, sensible effectiveness, latent effectiveness, total effectiveness, exhaust air transfer ratio, outdoor air correction factor and purge angle and tilt angle.

The tested unit is shown in below scheme.

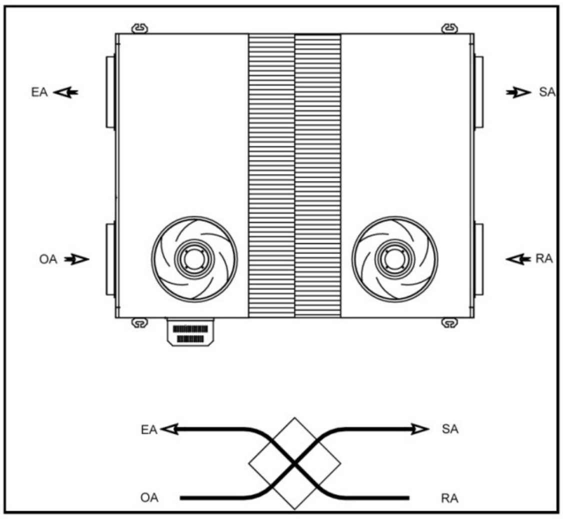


Figure 3.13. Heat Recovery Ventilator test unit.

Where;

OA : Outdoor Air

EA : Exhaust Air to Outdoors

SA : Supply Air to Indoors

RA : Return Air from the space.

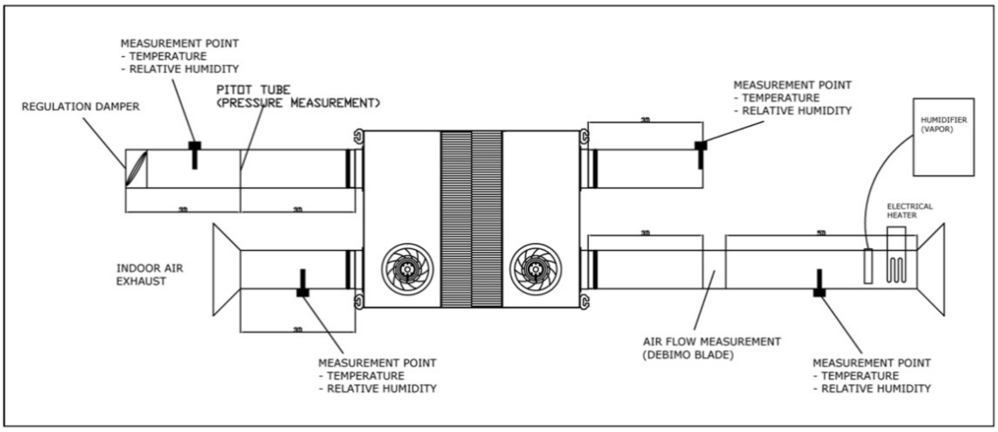


Figure 3.14 Heat Recovery Ventilator test scheme.

The test system allows gaining desired outdoor and indoor air conditions by the help of the electrical heater and humidifier installed temperature for both air streams before and after the unit and the relative humidity is also logged with the probes installed in the system. As the system is carried out to fit the demanded measurements, also a pitot tube is installed to measure available pressure after the unit. Available pressure is adjusted with the regulation damper.

3.3.3.1 Air Flow Measurement

Air flow measurement is made according to Tchebycheff method (David S. Douga, 2003) distributed pressure intakes on Debimo blades. The blades have pressure intakes for both static pressure and total pressure. The velocity and therefore the air flow is calculated with the well-known formula;

$$P = \frac{1}{2}\rho V^2 \tag{3.19}$$

Figure 3.15 Tchebycheff method blade and transmitter.

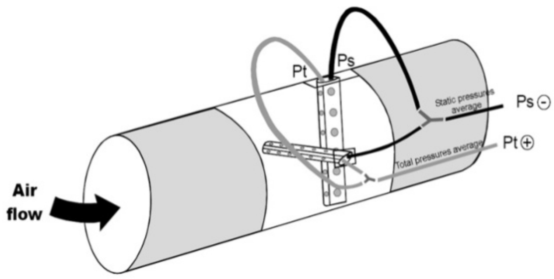


Figure 3.16 Operation of the air flow terminal.

3.3.3.2 Temperature and Humidity Measurement

The sensor information needed for the test bench is temperature and humidity of the air streams going through the heat recovery ventilator, HRV. The temperature and humidity are measured in four different locations at the heat recovery ventilator as shown in Figure 3.14.

1. Feed Inlet (OA) : Incoming fresh air from outside to the HRV
2. Feed outlet (SA) : Outgoing fresh air supply from the HRV to indoors
3. Sweep inlet (RA) : Stale room exhaust air going into the HRV
4. Sweep outlet (EA) : Exhaust air to outdoors leaving the HRV

The readings from the sensors are very sensitive to where the actual sensors are located in the air streams. To have as little influence as possible from the surrounding, both air streams are insulated.

3.3.3.3 Electrical Heater

To create a differential of temperature between outdoor air and the indoor air in the test facility, an electrical heater having a capacity of 6 kW has been used. The electrical heater is control via proportional electrical heater control unit, TTC40FX. The controller is capable of driving the electrical heater according to 0-10V signal output from the thermostat, hence creating freedom for wide range of air temperature with very high accuracy.

3.3.3.4 Humidifier

To obtain the desired humidity ratio for the outdoor air, a vaporized humidifier with a capacity of 30 kg/h is installed in the test system.

3.3.3.5 Microcontroller

The electrical heater and the humidifier are operated according to 0-10V signals from the sensors automatically. For the communication and data process Corrigo EH-E8D-H control board has been used. The desired values -set values for air conditions- for temperature and humidity are set in the control board, the control board drives both electrical heater and the humidifier according to the inputs from the sensors.

3.3.3.6 Electrical measurements

Total power consumption of the heat recovery ventilator has been measured with CA 8220 power analyzer.

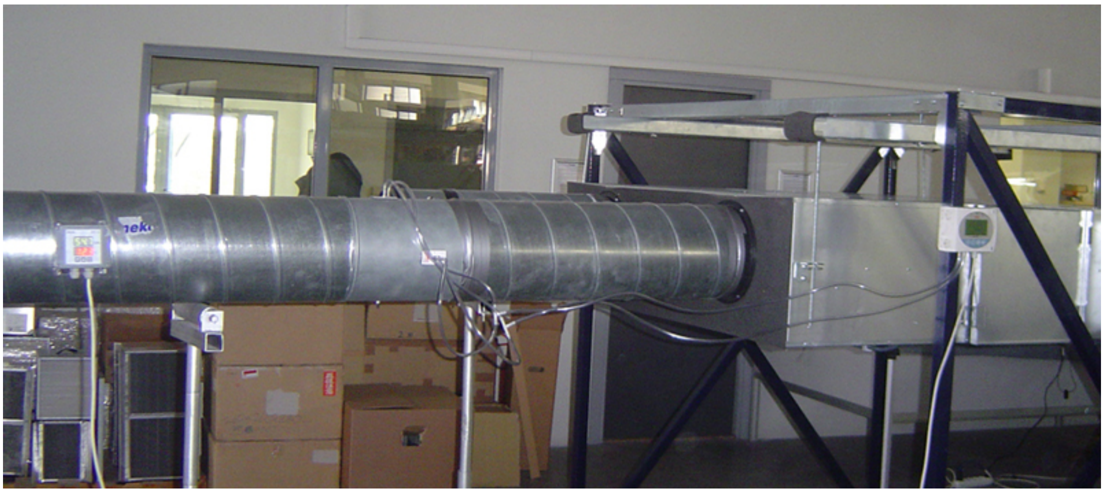


Figure 3.17 Test System.

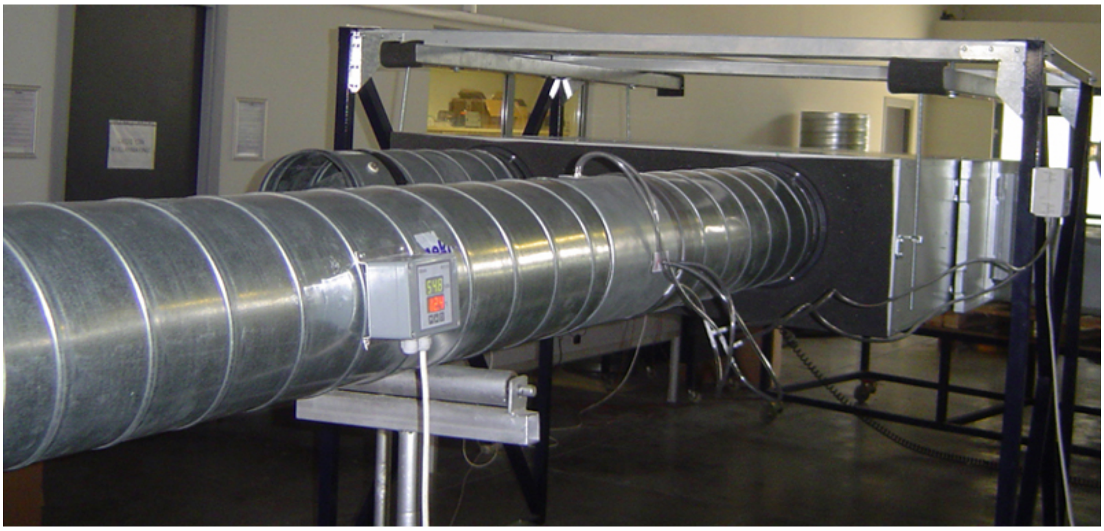


Figure 3.18 Test System.

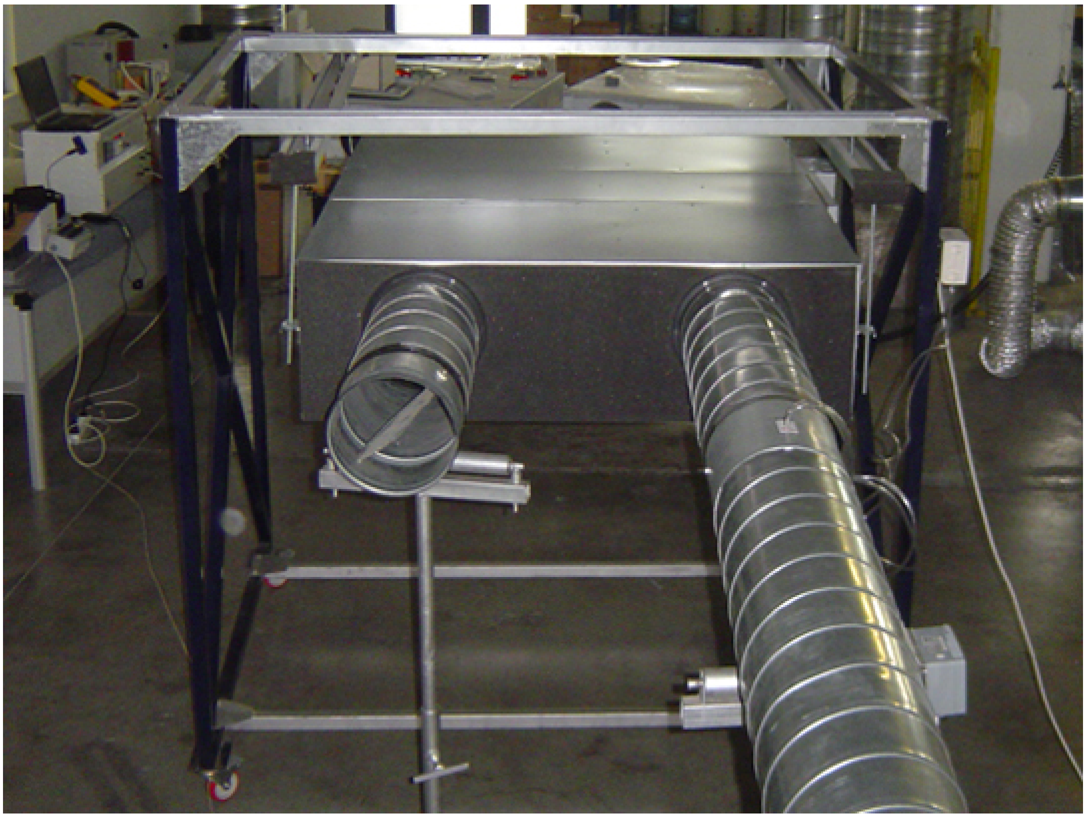


Figure 3.19 Test System.

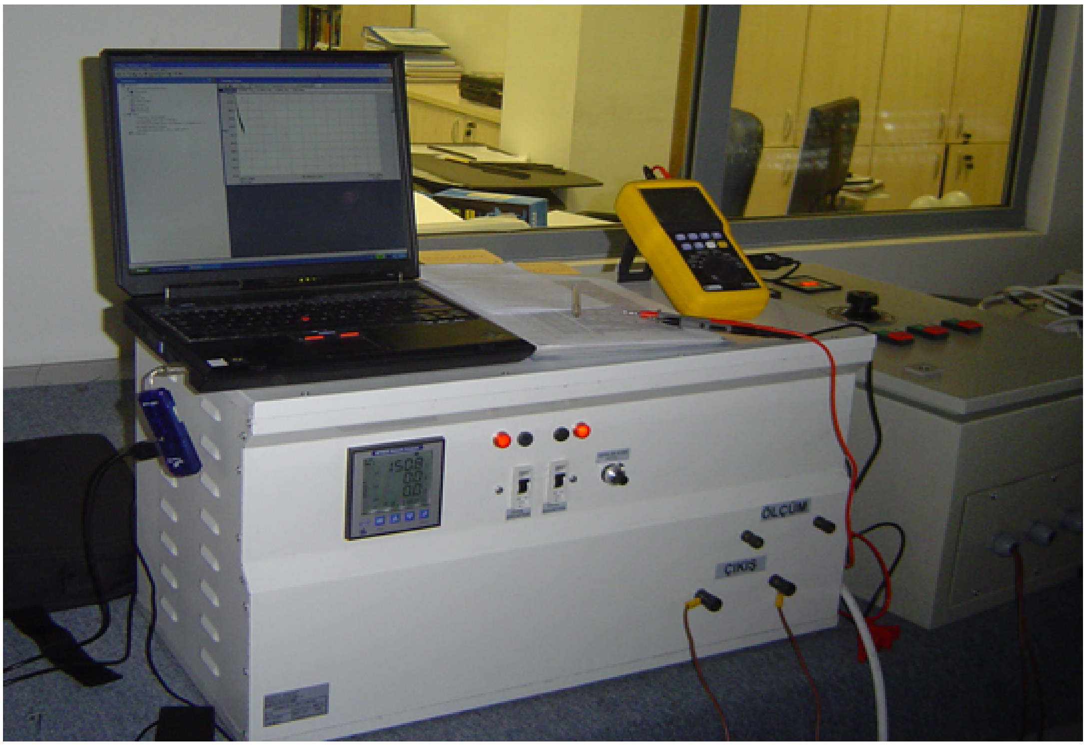


Figure 3.20 Test System.

3.4 Energy Savings

To reduce air conditioning systems energy consumption, ventilation systems are installed with heat recovery from extract air as additional energy would be demanded to warm up/cool down fresh air to indoor air temperature. The total heat recovery from the system is;

$$Q_{rec} = \dot{m}(h_2 - h_1) \tag{3.20}$$

Where;

Q_{rec}: Recovered heat (kW)

$\dot{m}$: Fresh air flow rate (kg/s)

h_2: supply air enthalpy (kJ/kg)

h_1: outdoor air enthalpy (kJ/kg)

3.4.1 Psychometric Chart

The atmosphere is a mixture of air (Oxygen, Nitrogen, CO_2 etc.) and water vapor. Psychrometry is the study of moist air and of the changes in its conditions. The psychrometric chart graphically represents the interrelation of air temperature and moisture content and is a basic design tool for building engineers and designers. Several terms must be explained before the charts can be fully appreciated.

Absolute humidity (AH) is the vapor content of air, given in g or kg of water vapor per kg of air, i.e. g/kg or kg/kg. It is also known as moisture content or humidity ratio. Air at a given temperature can support only a certain amount of moisture and no more. This is referred to as the saturation humidity.

Relative humidity (RH) is an expression of the moisture content of a given atmosphere as a percentage of the saturation humidity at the same temperature.

Wet-bulb temperature (WBT) is measured by a hygrometer or a sling psychrometer and is shown as sloping lines on the psychrometric chart. A status point on the psychrometric chart can be indicated by a pair of dry-bulb temperature (DBT) and WBT.

Specific volume (Spv) , in m^3/kg, is the reciprocal of density and is indicated by a set of slightly sloping lines on the psychrometric chart.

Enthalpy (H) is the heat content of unit mass of the atmosphere, in kJ/kg, relative to the heat content of 0 °C dry air. It is indicated on the psychrometric chart by a third set of sloping lines, near to, but not quite the same as the web-bulb lines. In order to avoid confusion, there are no lines shown, but external scales are given on two sides.

Sensible heat (Q_{sen}) is the heat content causing an increase in dry-bulb temperature. Latent heat (Q_{lat}) is the heat content due to the presence of water vapor in the atmosphere. It is the heat which was required to evaporate the given amount of moisture.

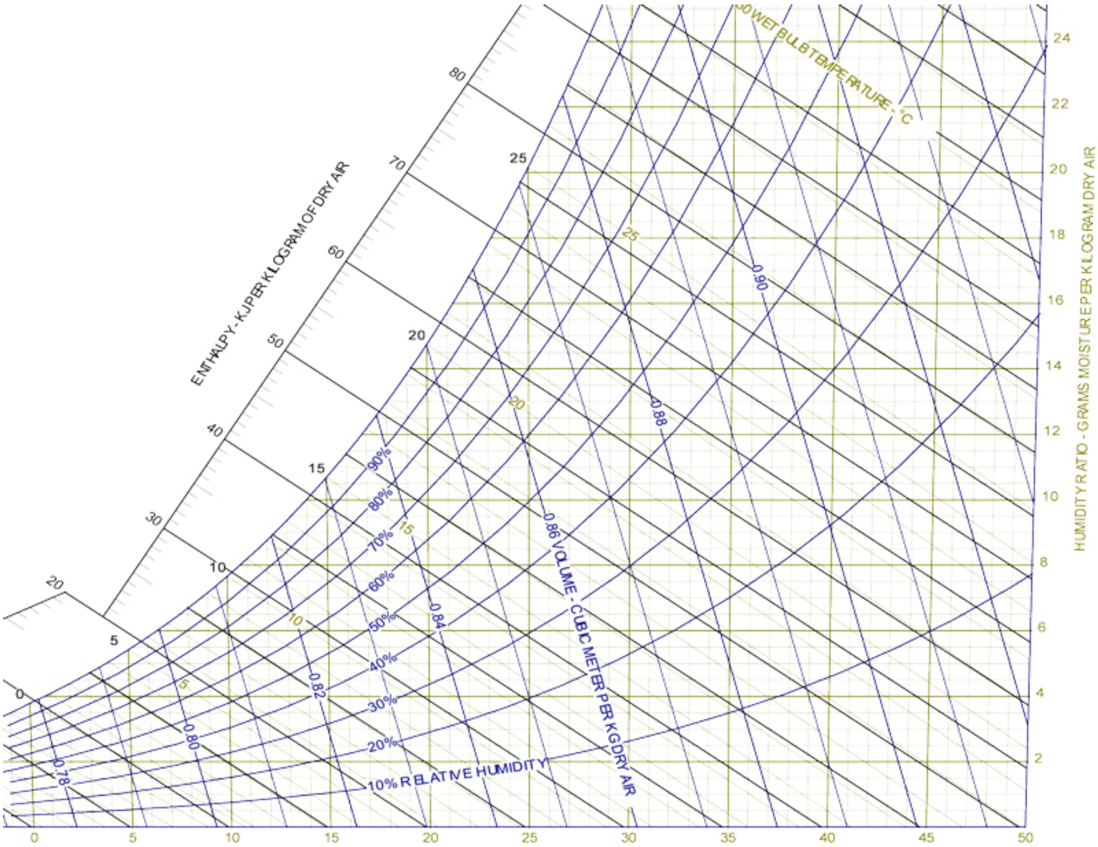


Figure 3.21 Psychrometric Chart at sea level.

3.4.1.1 Preheating with HRV

Fresh air enters heat recovery ventilator from outdoors X and by the help of extract air energy it is preheated to Y. Air conditioning system heats it to Z. Recovered heat is;

$$Q_{rec} = \dot{m}\left(h_y - h_x\right) \tag{3.21}$$

3.4.1.2 Precooling with HRV

Fresh air enters heat recovery ventilator from outdoors A and by the help of extract air energy it is precooled to B. Air conditioning system cools it to C. Recovered heat is;

$$Q_{rec} = \dot{m}(h_a - h_b) \tag{3.22}$$

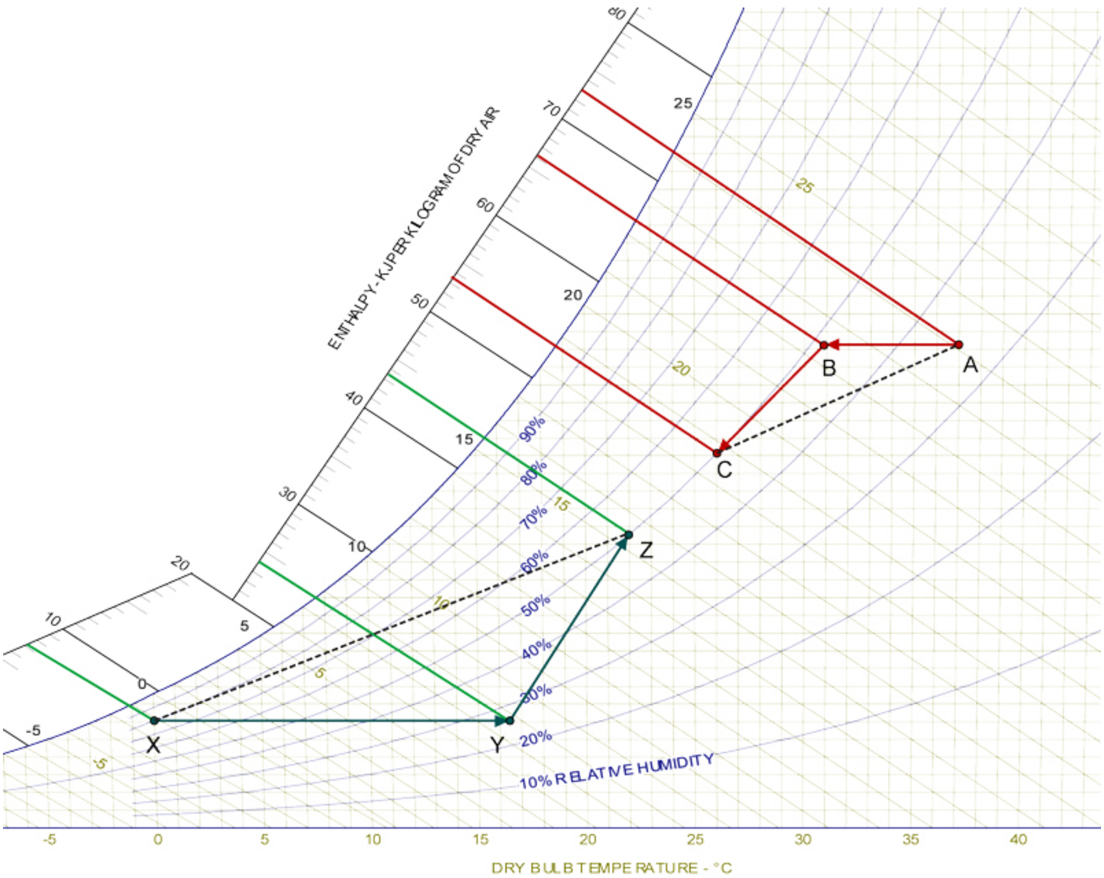


Figure 3.22 Heat Recovery process in Psychrometric chart.

3.4.2 Sample Calculation

A call-center in a building located in Ankara with 130 m^2 flow area has been considered for energy saving calculations. There are 16 people working in the office for every shift. There exist 3 shifts. Heating/cooling system has been designed for variable refrigerant volume (VRV) systems. The design conditions for summer indoor air is 26 °C and for winter indoor air is 22 °C. Fresh air demand for the office is calculated according to ASHRAE standard Ventilation for Indoor Quality 62.1.2007. The declared combined air rate for office spaces in the standard is 8,5 L/s. Person. Total population for the office is 16 people. That will result in 490 m^3/h fresh air demand.

For the calculation EVER-500 unit from Eneko A.Ş. has been used, nominal air flow for the unit is 500 m^3/h.

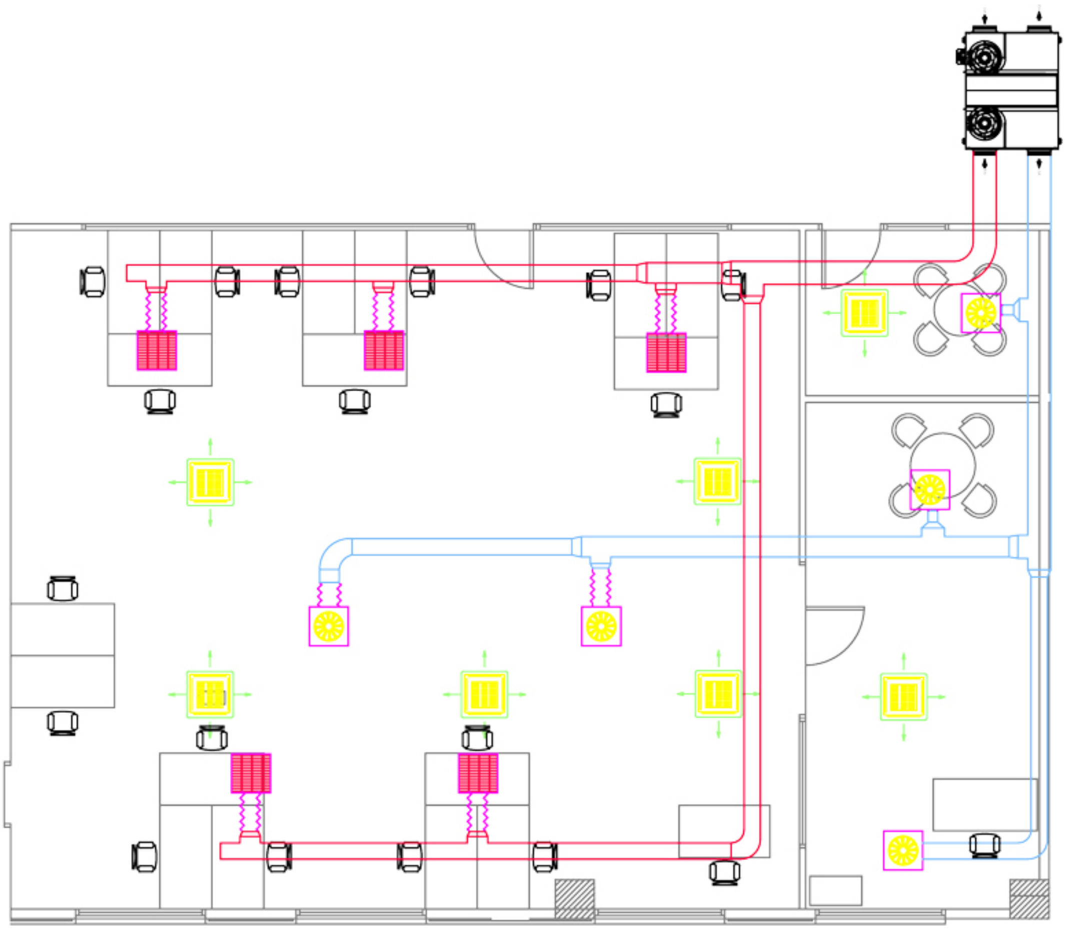


Figure 3.23 Heat Recovery Ventilator installation design.

Table 3.1.Sample HVAC data

Space	:	Office	
Area	:	130	m^2
Population	:	16	people
Cooling Load (without HRV)	:	19	kW
Ventilation Demand	:	489,6	m^3/h
Outdoor Air Conditions (Summer)	:	Various each hour	
Indoor Air Conditions (Summer)	:	26 °C DB, 50% RH	
Outdoor Air Conditions (Winter)	:	Various each hour	
Indoor Air Conditions (Winter)	:	22 °C DB, 50% RH	

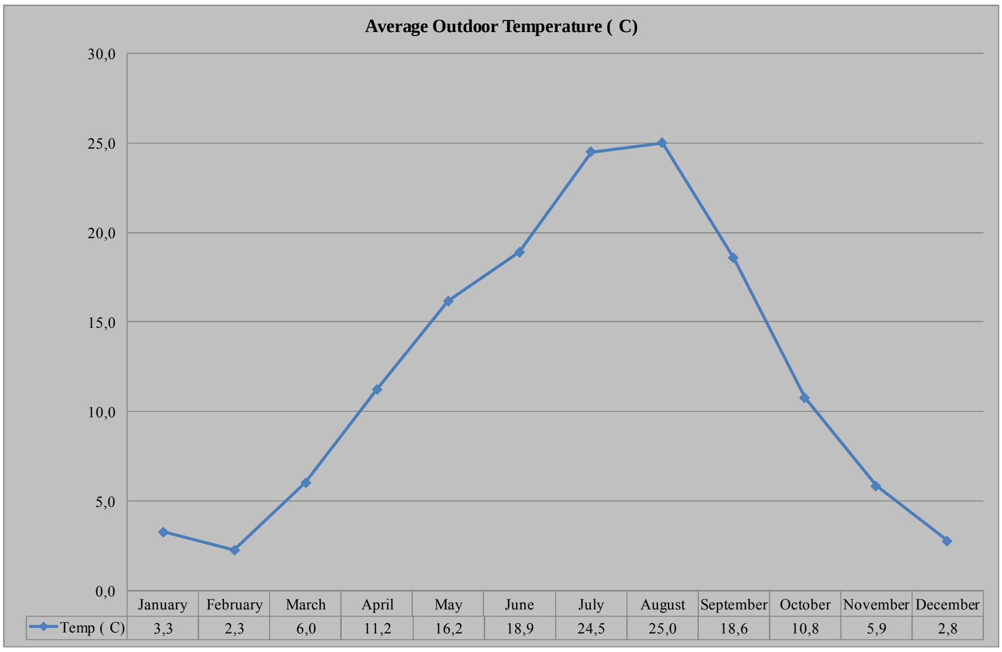


Figure 3.24 Outdoor Air Temperature Monthly Average.

For the calculation of recovered heat across the heat recovery ventilator, average monthly outdoor temperatures have been considered with Klingenburg selection software. The results are based on average temperatures, not hourly values.

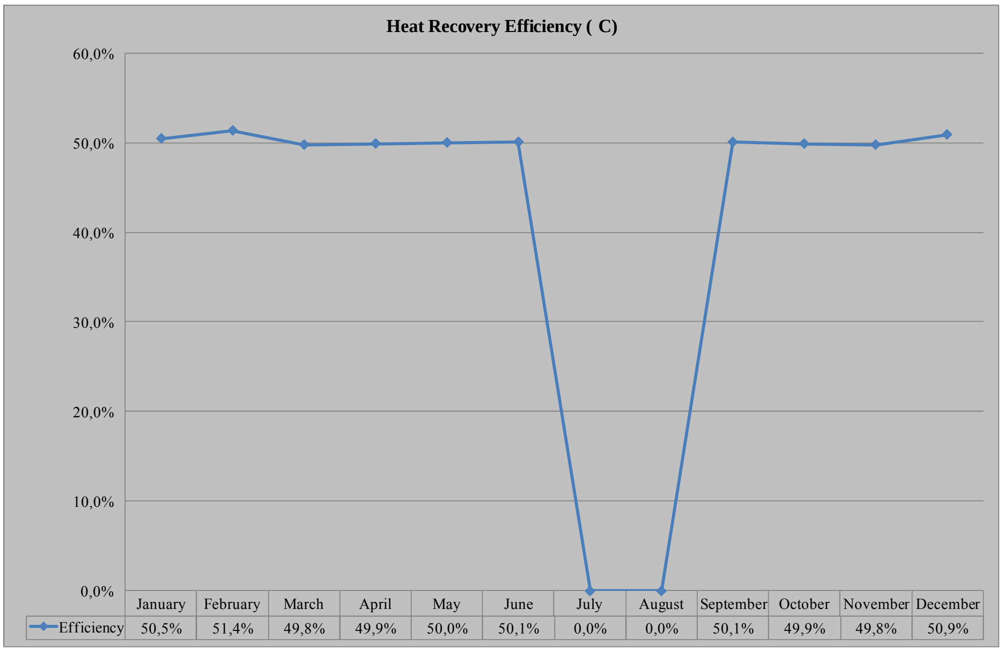


Figure 3.25 Heat Recovery Efficiency through months in Ankara.

As seen on Figure 3.25, all months except July and August are cooling seasons for Ankara. The efficiency of the heat exchanger is very close for each month although average temperatures are quite different. For July and August the average temperatures are close to indoor air temperature thus, heat recovery ventilator is by-passed and fresh air is taken inside directly.

Although heat recovery efficiency is very similar in each Month, recovered heat differs as a function of average outdoor air temperature as seen in Figure 3.26. For months July and August recovered heat amount is zero as fresh air is by-passed through the heat recovery ventilator. Maximum heat recovery is observed in February as minimum average temperature is for February as seen in Figure 3.24.

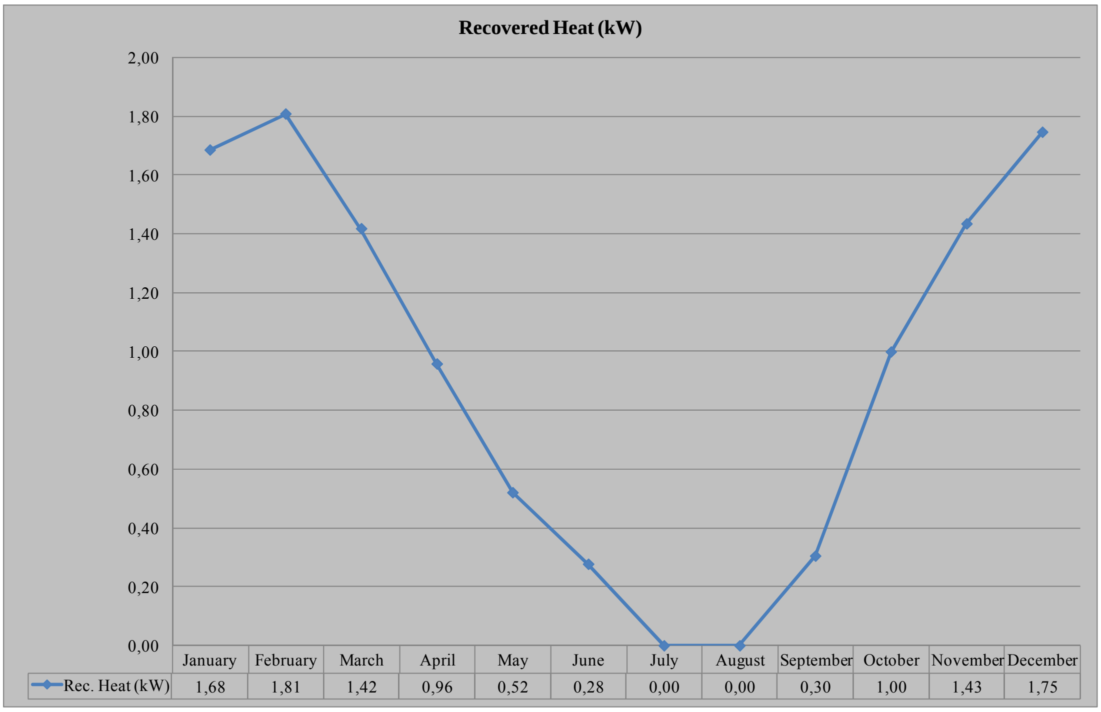


Figure 3.26 Hourly Recovered Heat through months in Ankara.

To calculate yearly energy savings as a result of heat recovery ventilator, average recovered heat shall be considered in the calculation. Call-center operates for 24 hours a day and 7 days a week excluding religious vacations. Total working hours for year 2005 is 8568 hours and total operational hours for the heat recovery (subtracting by-pass ventilation for July and August 7080.

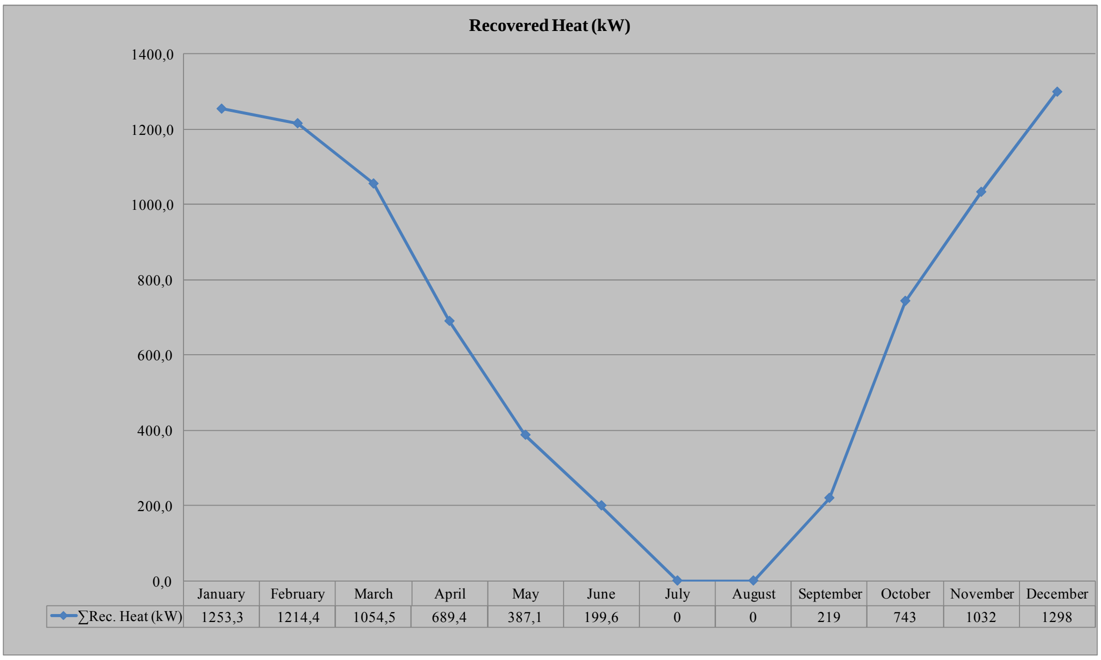


Figure 3.27 Monthly Recovered Heat through months in Ankara.

Total heat recovery for year 2005 in Ankara according to average outdoor temperature calculation method is 8091 kWh. Heat Recovery formed only in heating season. Air conditioning units inside the call-center would consume additional energy to warn fresh air and this additional energy, which is also total heat recovery, is 8091 kWh for Ankara. Air conditioning is provided by Natural gas burners and cassette type fan coils. In Ankara Natural gas cost for 1 kWh energy is 0,037 €. Yearly energy saving as a result of heat recovery usage is 299,4 €.

The sample heat recovery ventilator (EVER-500) provided by Eneko A.Ş for the calculation is 750 € including taxes. The unit supplies 500 m^3/h fresh air from outdoor air ducts with a pressure drop of 95 Pa's and removes 480 m^3/h indoor air from exhaust air ducts with a pressure drop of 110 Pa's. Instead of a heat recovery ventilator, to perform ventilation, 2 duct type fans with model number RFA 30/15 TE1 with the brand name ATC can be used which cost 160 € each including taxes. To filtrate outdoor air a filter box with model number EFB 30/15 with the brand name ATC can be used which cost 55 € including taxes. Initial cost rise in the Natural gas burner is neglected. Also the difference in the power consumption between the HRV system and the duct system is neglected.

Table 3.2 Cost comparison analysis between HRV System and Duct Fan System

		HRV System	Duct Fan System
Initial Cost	Heat Recovery Ventilator	750,0 €	-
	Exhaust Air Fan	-	160,0 €
	Supply Air Fan	-	160,0 €
	Fresh Air Filter	-	55,0 €
	Pre-heater	-	Neglected
Operational Costs	Electrical Consumption	Neglected	Neglected
	Service Costs	Neglected	Neglected
	Pre-heating cost	-	299,4 €

Heat recovery ventilators initial cost is 475 € more than the typical duct system for the sample installation. To calculate payback period, initial cost difference shall be divided into operational cost difference;

$$t_{pb} = \frac{\Delta_{Initial\ Cost}}{\Delta_{operational\ cost}} \qquad (3.23)$$

$$t_{pb} = \frac{750-(160+160+55)}{299{,}4} = \frac{375}{299{,}4} = \text{ 15 months}$$

CHAPTER FOUR

ECONOMICAL ANALYSIS OF HRV's IN TURKEY

4.1 Analysis design considerations

Heat recovery efficiency and recovered heat depends on indoor and outdoor air conditions. (Y.P. Zhoua, J.Y. Wu & R.Z. Wanga, 2008). For non-industrial applications design criteria of indoor air temperature in winter season is 22°C and 26°C for summer season. As these values are meant to be kept constant with HVAC system, the main variable for heat recovery consideration is outdoor air for the same heat recovery ventilator. In the design process heat recovery ventilators efficiency and the heat recovered is calculated with the peak points for both seasons. This calculation is not correct to calculate annual savings for the heat recovery ventilator as it represents only a few period of heating/cooling time for the system.

Also the climate for a whole continent differs in every country, even in every city (Renato M. Lazzarin & Andrea Gasperalla, 2007). Recovered heat, supply air temperature and heat recovery ventilator efficiency shall be regarded separately during economical evaluation.

Turkey, well known for its different climate conditions, is studied by means of recovered heat, supply air temperature and heat recovery ventilator efficiency for 6 cities, İzmir, İstanbul, Ankara, Antalya, Urfa and Erzurum. To calculate mentioned values, a calculation software has been used from one of the heat recovery exchanger manufacturers, Klingenburg GmBH. The software has been approved by Eurovent.

For the outdoor air conditions data has been collected from Turkish State Meteorological Service, Ankara for every hour of 365 of year 2005. The outdoor data has been used as input by the help of external software developed by the author himself. This resulted in precise computation for performance of heat recovery ventilation annually.

Heat recovery ventilator has been provided from Eneko AS, the unit that has been used in this study is EVER 500. The unit has 500 m^3/h nominal air flow rate, the same value has been used during computation. The heat recovery exchanger used in this study is manufactured by Klingenburg GmBH and the model number is PWT10/200/750-1,8. The heat recovery exchanger is cross flow type with Aluminum plates.

Figure 4.1 Plate type Heat Recovery Exchanger

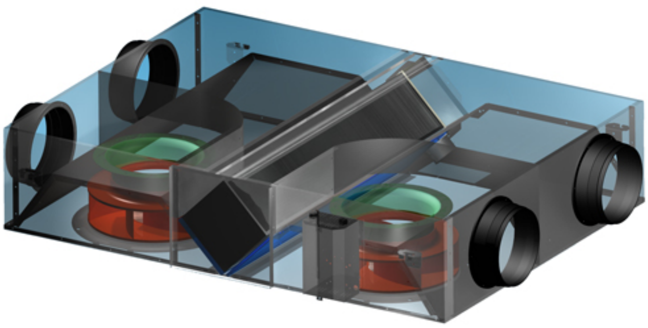


Figure 4.2 Heat Recovery Ventilator

4.2 Thermal Analysis of Heat Recovery Ventilation

In this study the winter-hour is described as every hour which is cooler than 20°C for winter and summer-hour is described as every hour which is hotter than 28°C set temperature for summer. Between 20°C and 28°C heat recovery ventilator operates in by-pass ventilation mode in which the return air is by-passed across the heat recovery exchanger so no heat is exchanged.

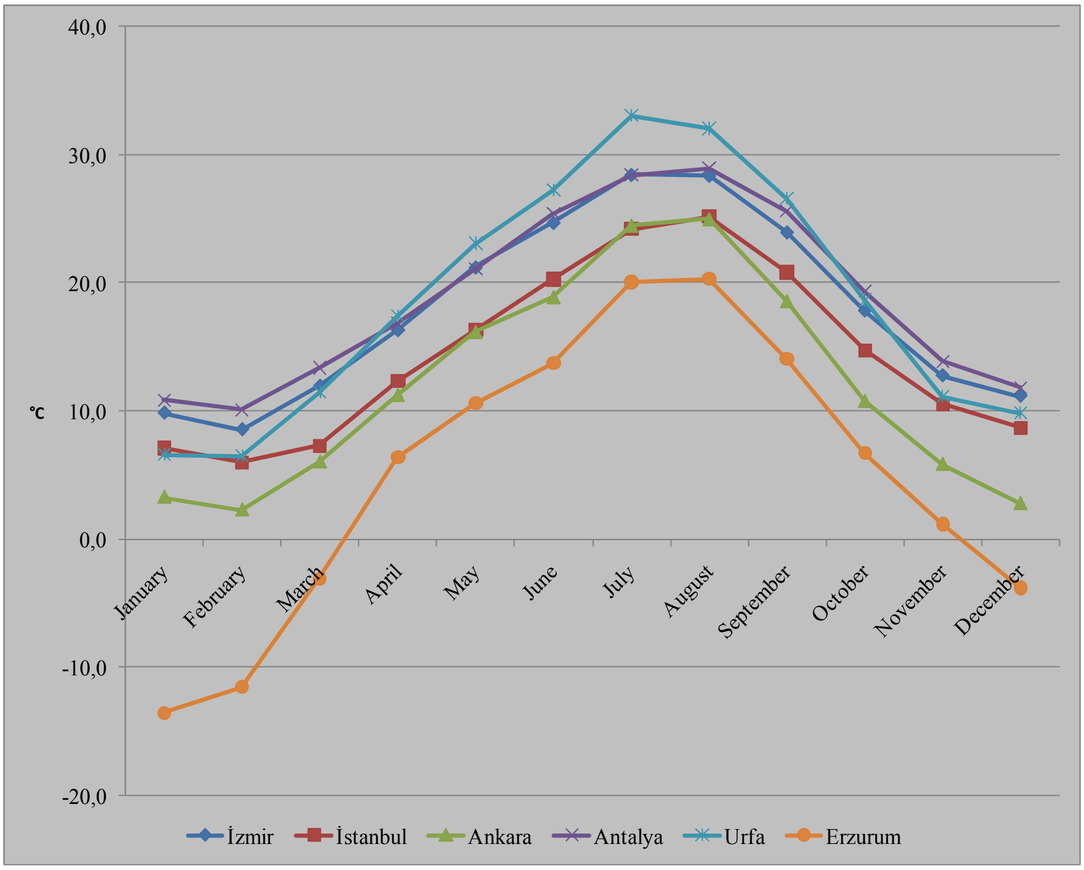


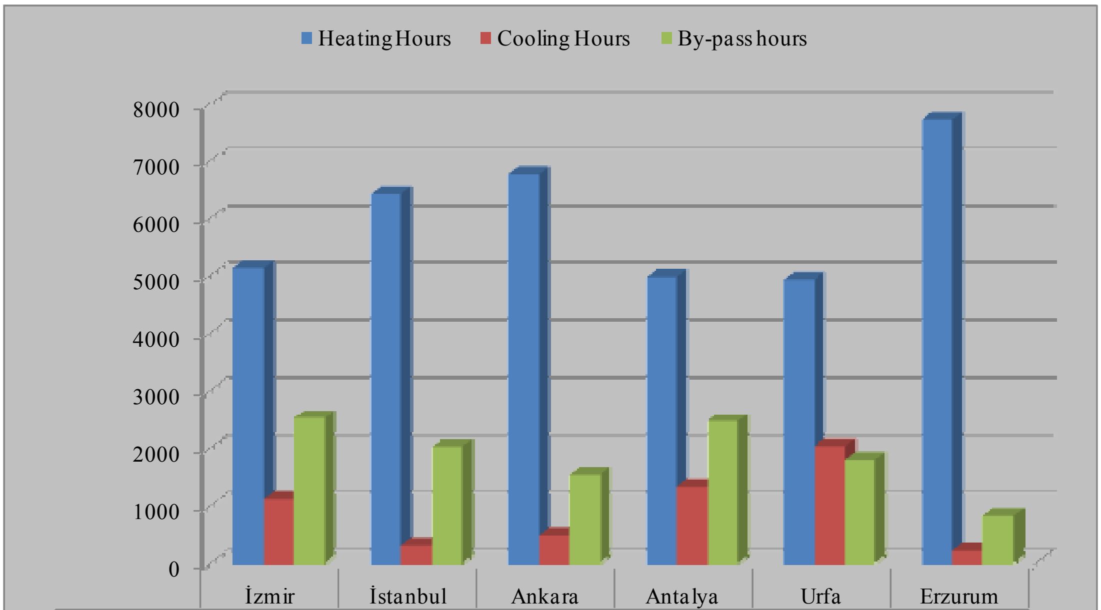
Figure 4.3 Monthly temperature averages for studied 6 cities

Table 4.1 . Monthly temperature averages for studied 6 cities

Cities	İzmir	İstanbul	Ankara	Antalya	Urfa	Erzurum
January	9,9	7,1	3,3	10,8	6,6	-13,5
February	8,6	6	2,3	10,1	6,5	-11,5
March	12	7,3	6	13,4	11,4	-3,1
April	16,3	12,3	11,2	16,8	17,4	6,4
May	21,2	16,3	16,2	21,1	23	10,6
June	24,7	20,3	18,9	25,4	27,2	13,7
July	28,4	24,2	24,5	28,4	33	20,1
August	28,4	25,1	25	28,9	32	20,3
September	23,9	20,8	18,6	25,6	26,5	14,1
October	17,8	14,7	10,8	19,3	18,6	6,7
November	12,8	10,5	5,9	13,9	11,1	1,2
December	11,2	8,7	2,8	11,8	9,8	-3,8

As shown in Figure 1 and Table 1 seaside cities like İzmir, İstanbul and Antalya have similar climate data pioneering the examples of Mediterranean climate. Urfa which is well known with its desert-like climate has the highest temperatures for summer season. Ankara located in the middle of the country is neither extremely cold nor very hot in summer, more like in between. Erzurum, located in the eastern part is the coldest climate that is studied.

With the design temperature for indoors, the hour studied can be regarded as heating hour, cooling hour or by-pass hour. If outdoor air temperature is below 20°C it is called “heating hour”, if it is above 28°C it is called “cooling hour”, if it is between these values than it is called “by-pass hour”. 8760 hours of the year has been identified and named for each city in Figure 4.4. Erzurum has the most heating hours in 6 cities, followed by Ankara and İstanbul. Warmer cities like İzmir, Antalya and Urfa has the less heating hours, but it is seen that even in the warmest city heating is required more than 55% annually. For cooling hours Urfa, with its desert-like climate leads with 2032 cooling hours and is followed by Antalya and İzmir. Although monthly average temperature for İstanbul is more than Ankara, considering cooling hours, Istanbul’s cooling demand is less than Ankara. The least cooling hours is carried out by Erzurum as it is the coldest city in 6 cities. Evaluating by-pass hours, İzmir and Antalya is with app. 28% of the year. İstanbul, Urfa and Ankara .are following and Erzurum has only 831 by-pass hours.



	İzmir	İstanbul	Ankara	Antalya	Urfa	Erzurum
Heating Hours	5124	6437	6768	4971	4927	7715
Cooling Hours	1113	299	472	1323	2032	214
By-pass hours	2523	2024	1520	2466	1801	831

Figure 4.4 Conditioning hours scatter for 6 cities.

As denoted, to perform heat recovery ventilation analysis for the cities, annual hour temperatures have been received from Turkish State Meteorological Service and computed in the calculation software. To obtain a better understanding for the temperature change annually, outdoor temperature graphs have been developed for each city. Figure 4.5 shows the data for İzmir. Two lines are indicated apart from the temperature values, heating limit line and cooling limit line. Between these two lines, by-pass hours are present. In by-pass hours no heat recovery is calculated which results in 0% efficiency, 0 kW recovered heat and also supply air temperature and outdoor air temperature are the same. The graphs also approve the relation explained for the by-pass hours ratio in 6 cities.

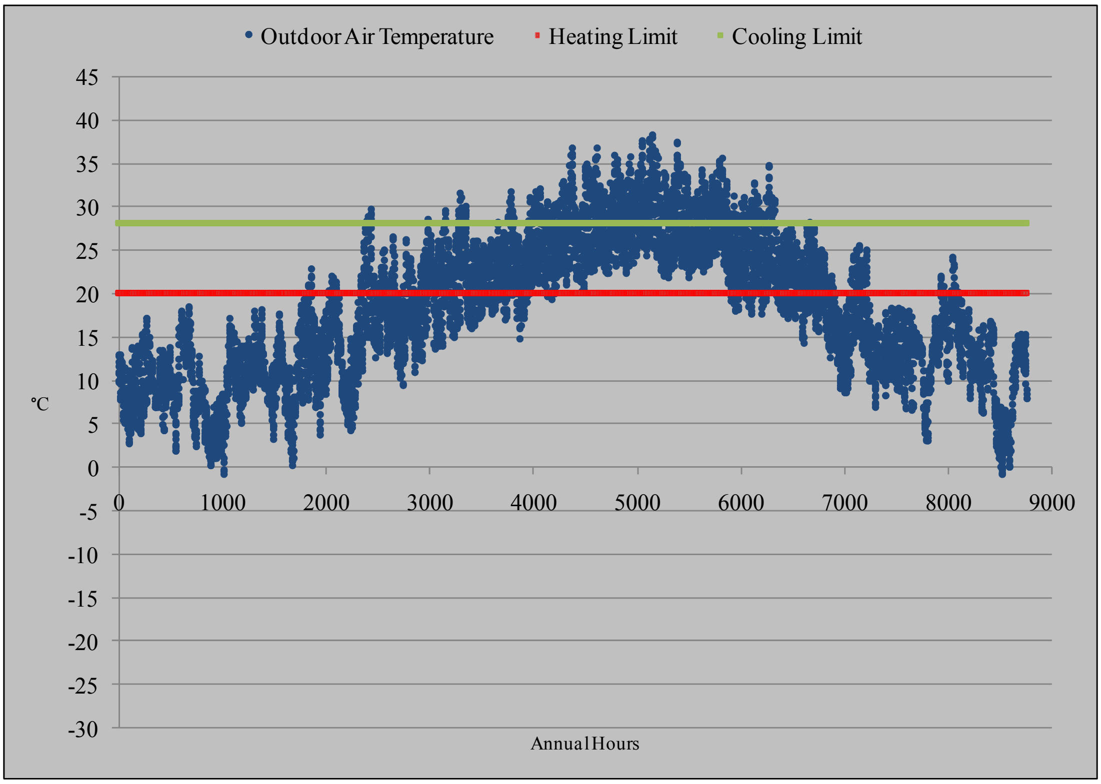


Figure 4.5 Annual hourly temperatures for İzmir.

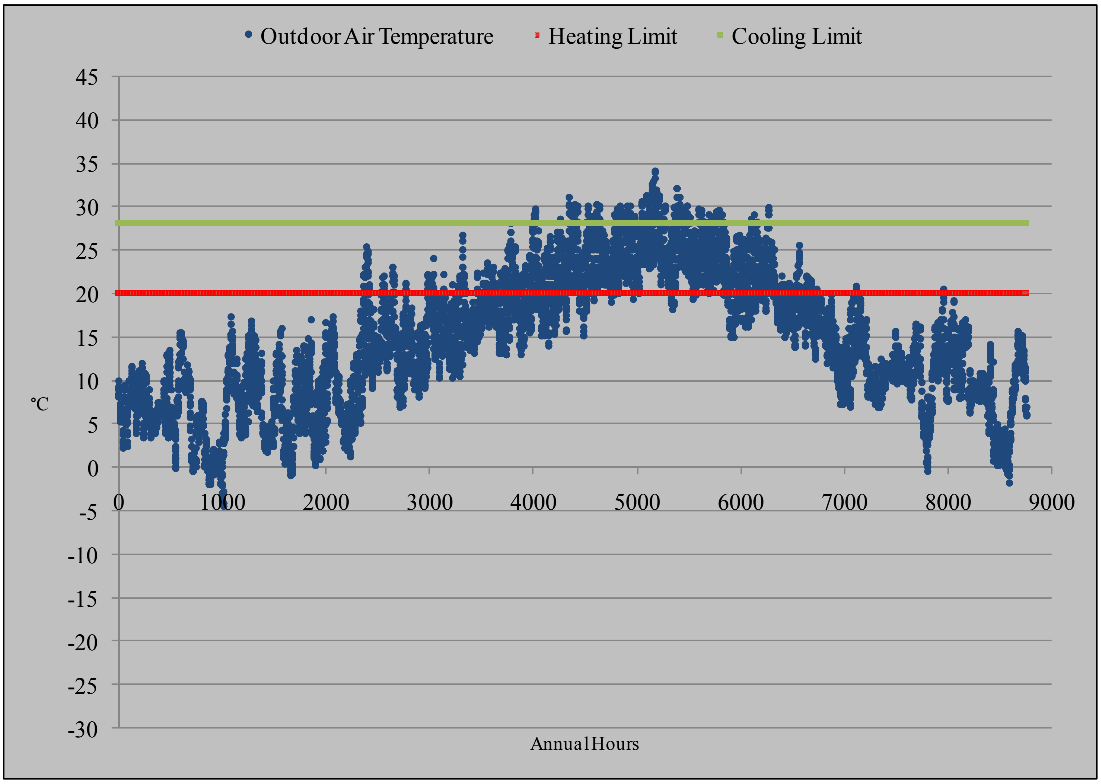


Figure 4.6 Annual hourly temperatures for İstanbul.

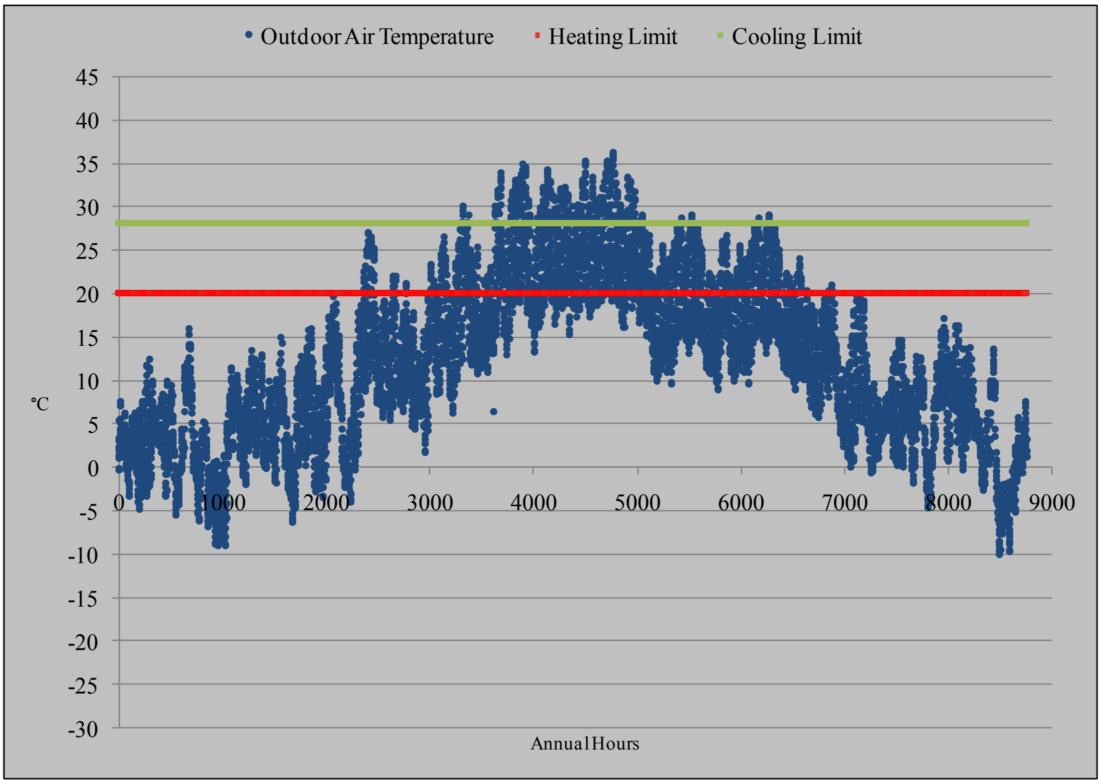


Figure 4.7 Annual hourly temperatures for Ankara.

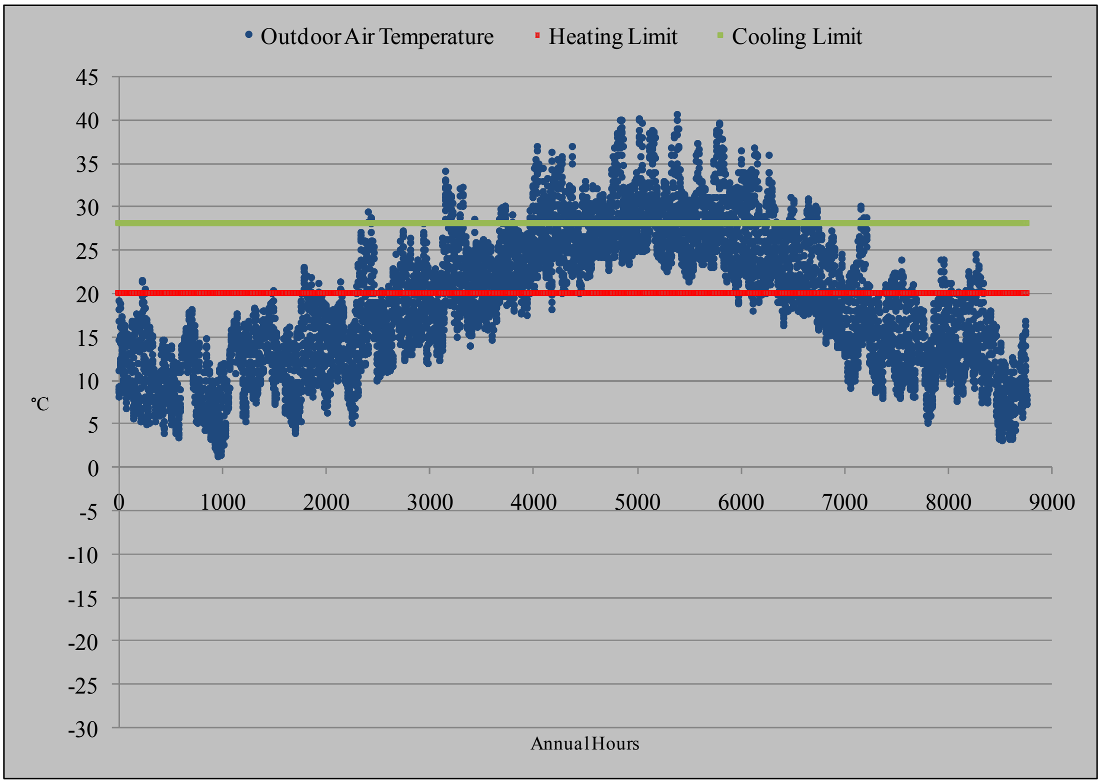


Figure 4.8 Annual hourly temperatures for Antalya.

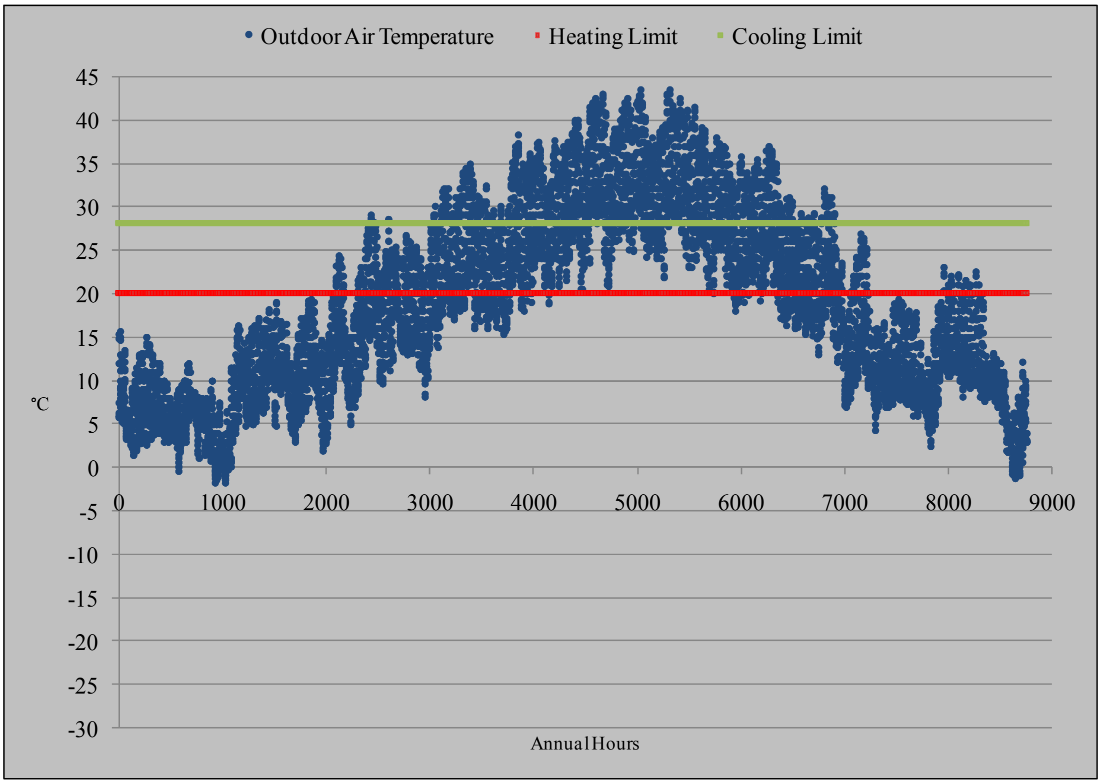


Figure 4.9 Annual hourly temperatures for Urfa.

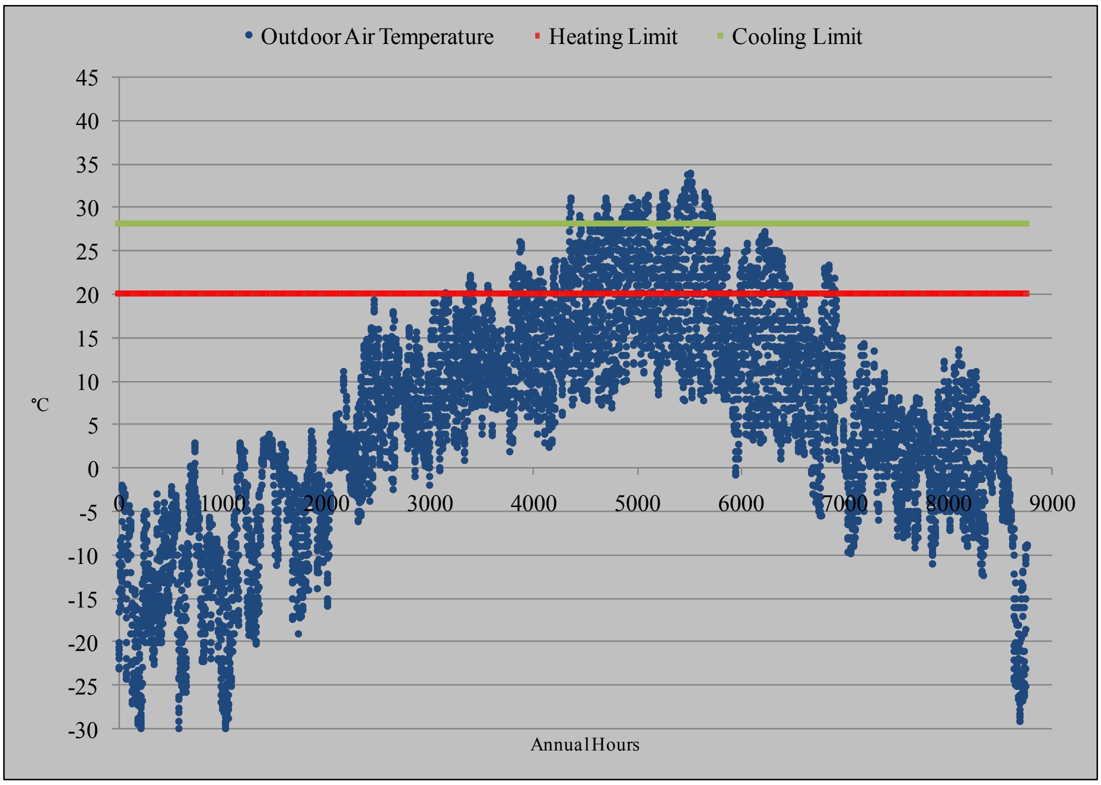


Figure 4.10 Annual hourly temperatures for Erzurum.

Considering the figures for winter and summer separately, it is shown that there are three groups for winter and three groups for summer. For winter, with warmer outdoor air conditions, Izmir, Ankara and Urfa is one of the groups, the other one is colder but still not below -10°C group including the cities İstanbul and Ankara, and finally the last group for winter includes Erzurum with its extremely cold outdoor air. This grouping is subject to change when it comes to summer, there are three groups; the first group includes Urfa with its extremely hot outdoor air which is seen in every period of the season. The second group consists of İzmir, Antalya and Ankara, Antalya has also very hot outdoor air but just for some period not the whole season, Ankara which is in the cold group for winter is in the hot group for summer season, finally the last group includes Erzurum and İstanbul with their lower outdoor air temperature than other cities. Although İstanbul is located near the seaside, summer is colder than Antalya and İzmir which are also seaside cities because it is the north side of the country.

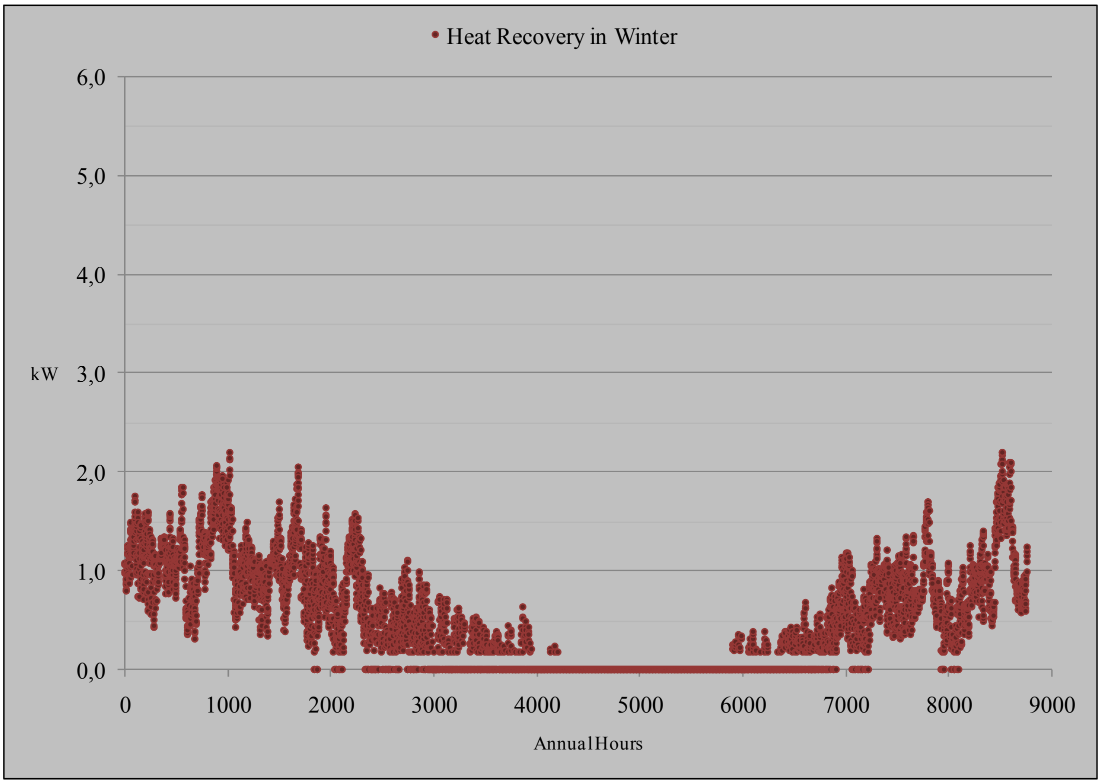


Figure 4.11 Annual hourly winter heat recovery in İzmir.

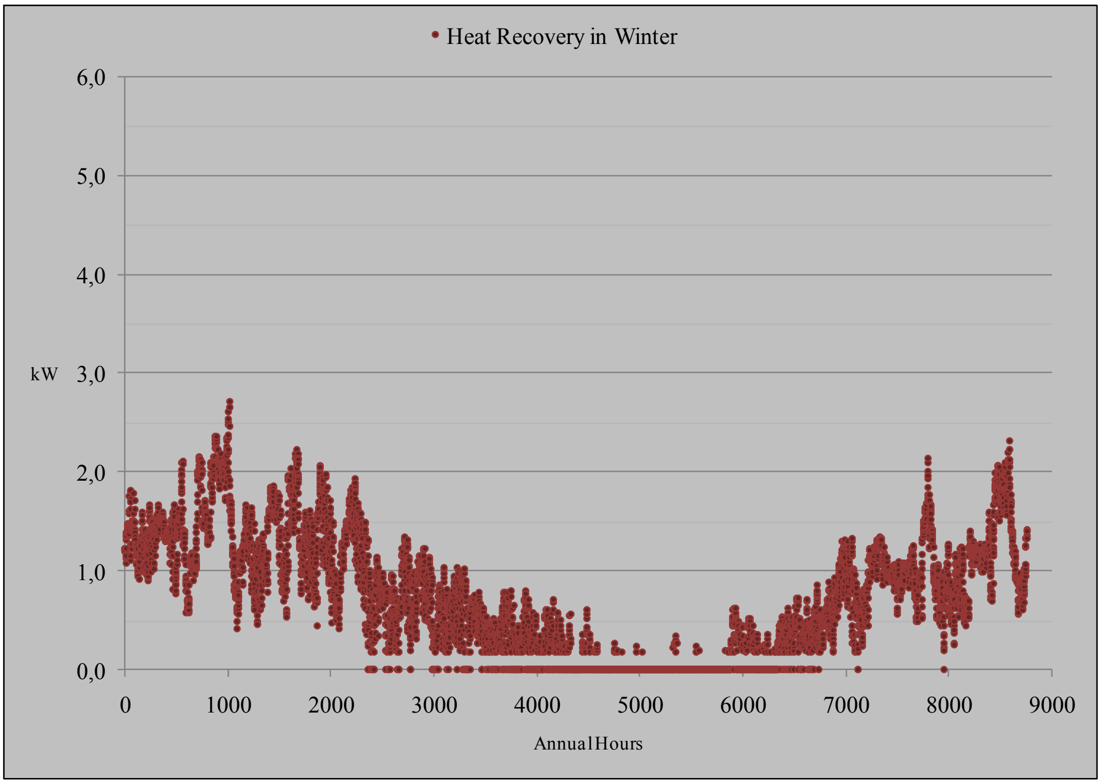


Figure 4.12 Annual hourly winter heat recovery in İstanbul.

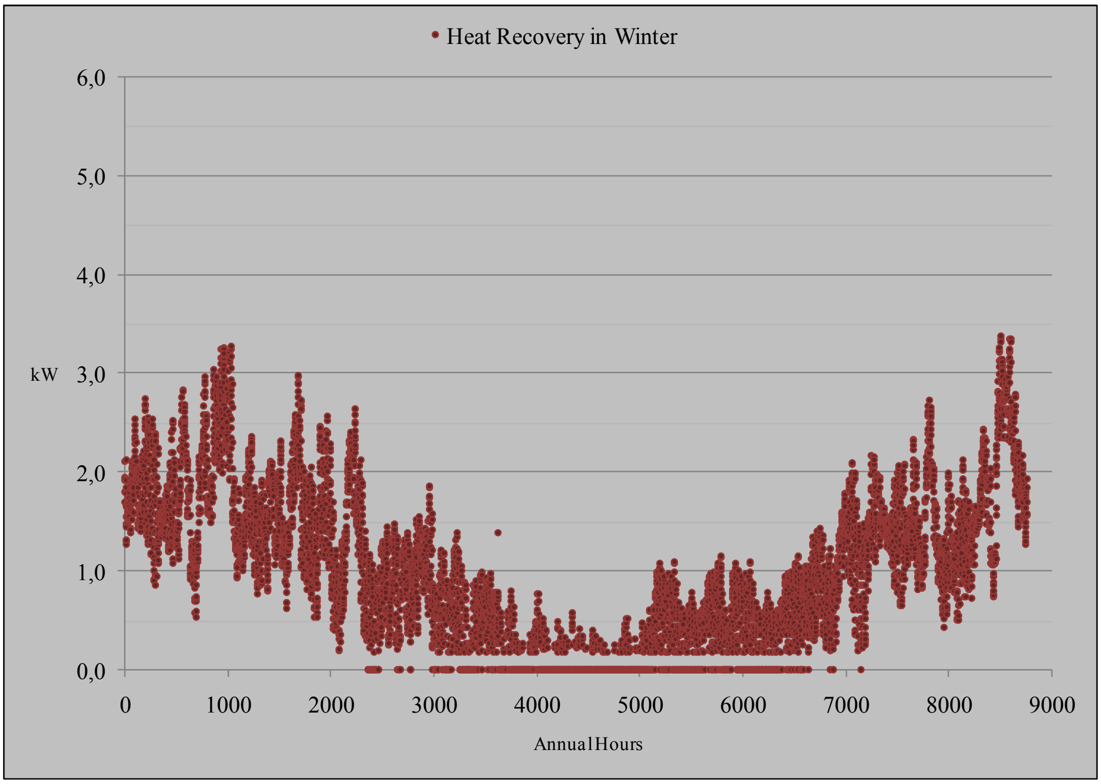


Figure 4.13 Annual hourly winter heat recovery in Ankara.

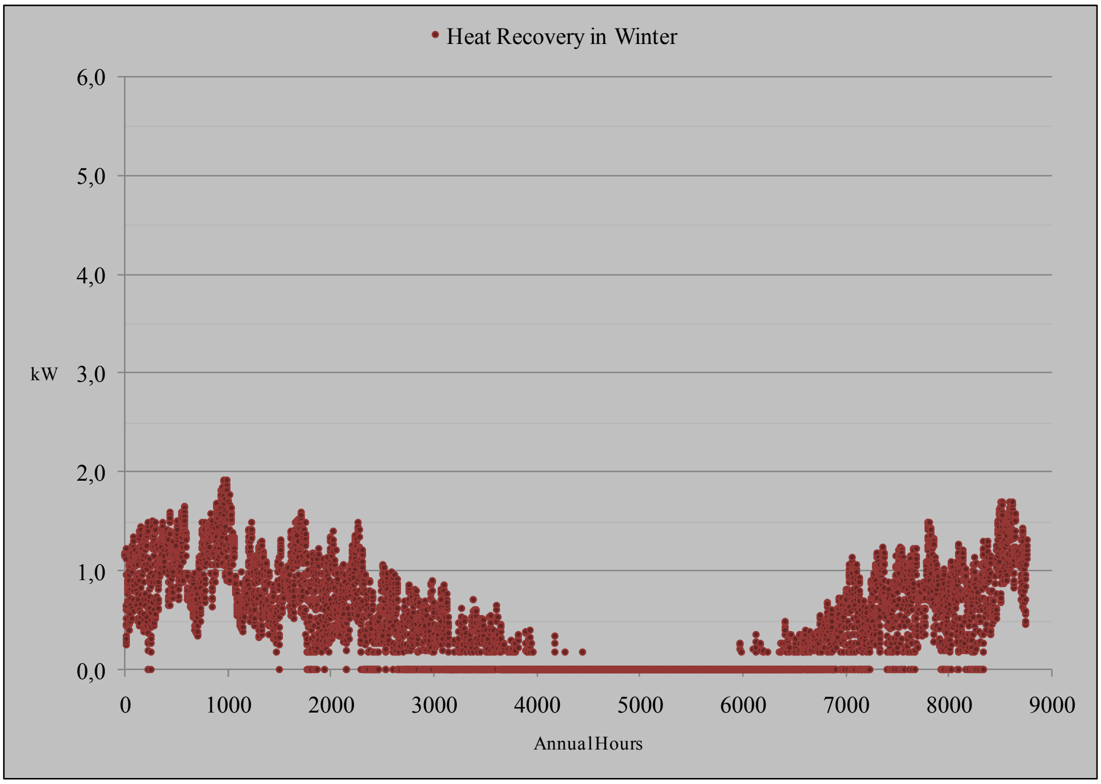


Figure 4.14 Annual hourly winter heat recovery in Antalya.

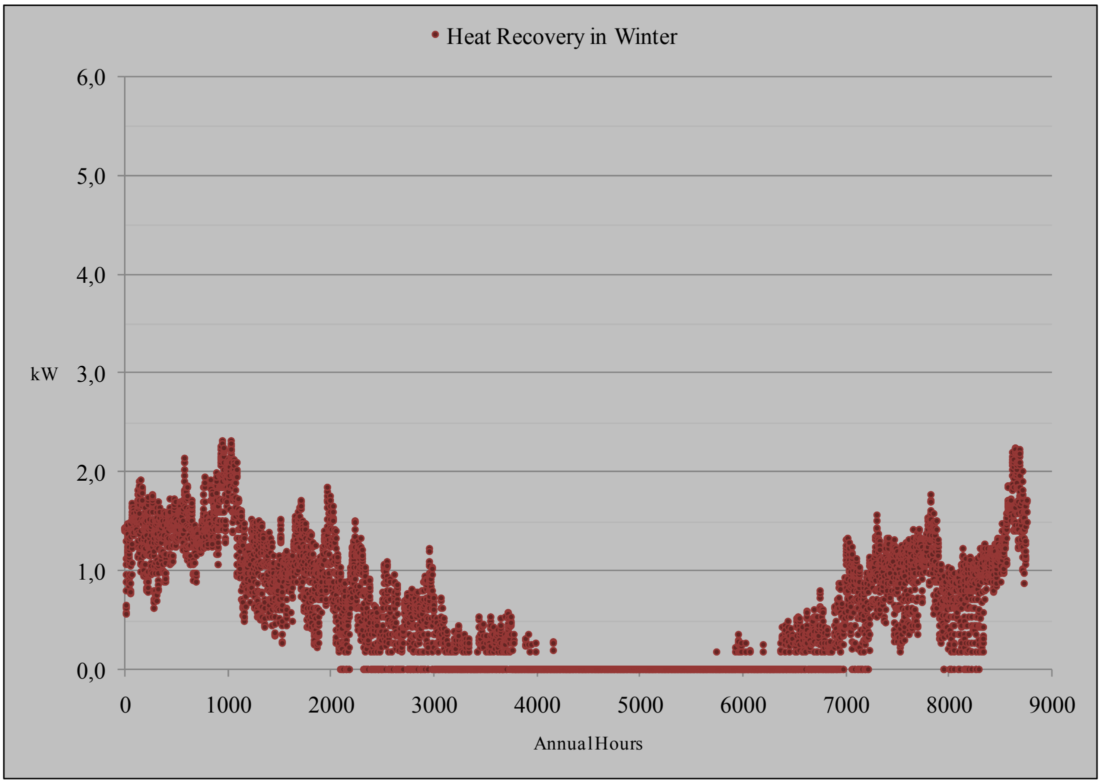


Figure 4.15 Annual hourly winter heat recovery in Urfa.

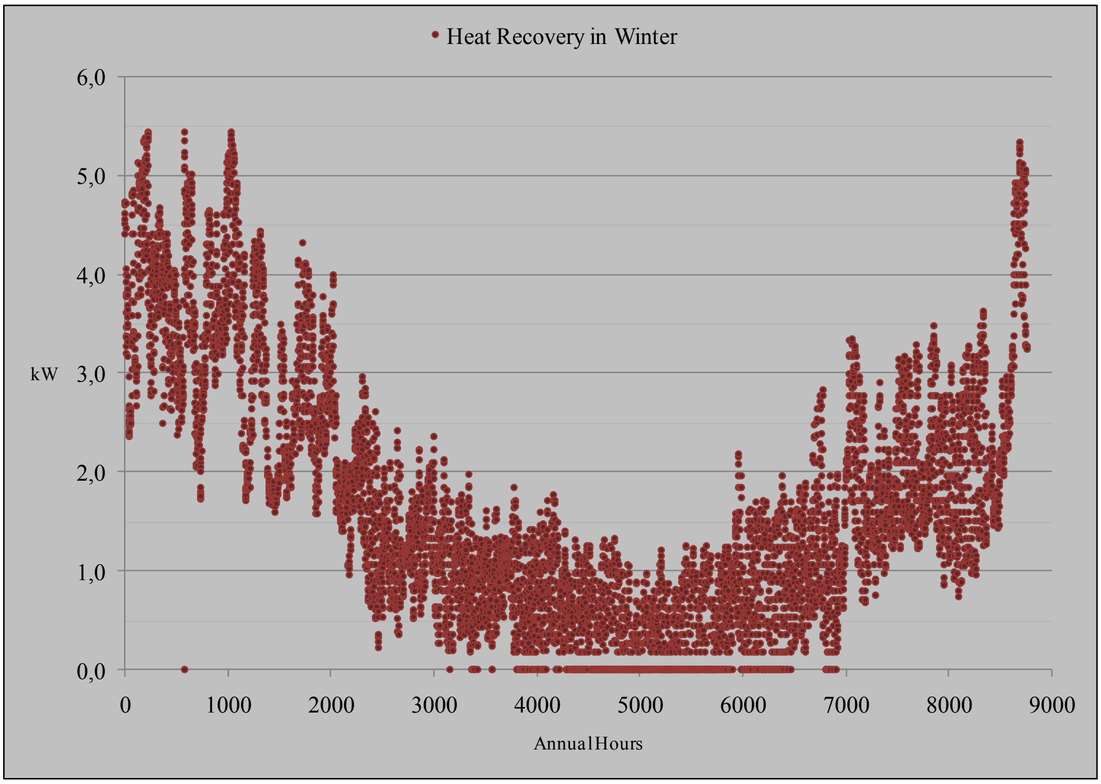


Figure 4.16 Annual hourly winter heat recovery in Erzurum.

Although winter design temperature for indoor air is the same for all cities, recovered heat changes as a function of outdoor air temperature. Heat recovery in winter season for İzmir, Antalya and Urfa are alike, where heat recovery in winter season for İstanbul and Ankara has the same characteristic. Erzurum the coldest city examined has the maximum heat recovery in winter. As a result it is considered that heat recovery in winter season increases with the decrease in outdoor air. Also it is considered that, climates that have similar outdoor air conditions result in similar heat recovery.

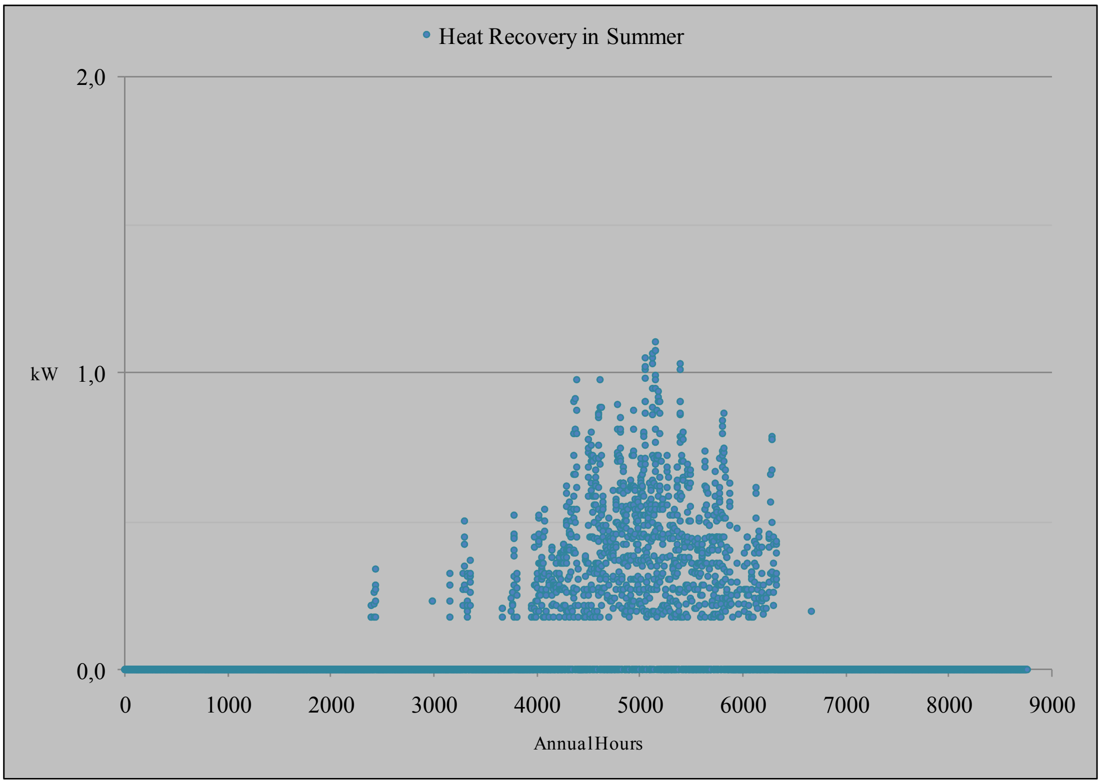


Figure 4.17 Annual hourly summer heat recovery in İzmir.

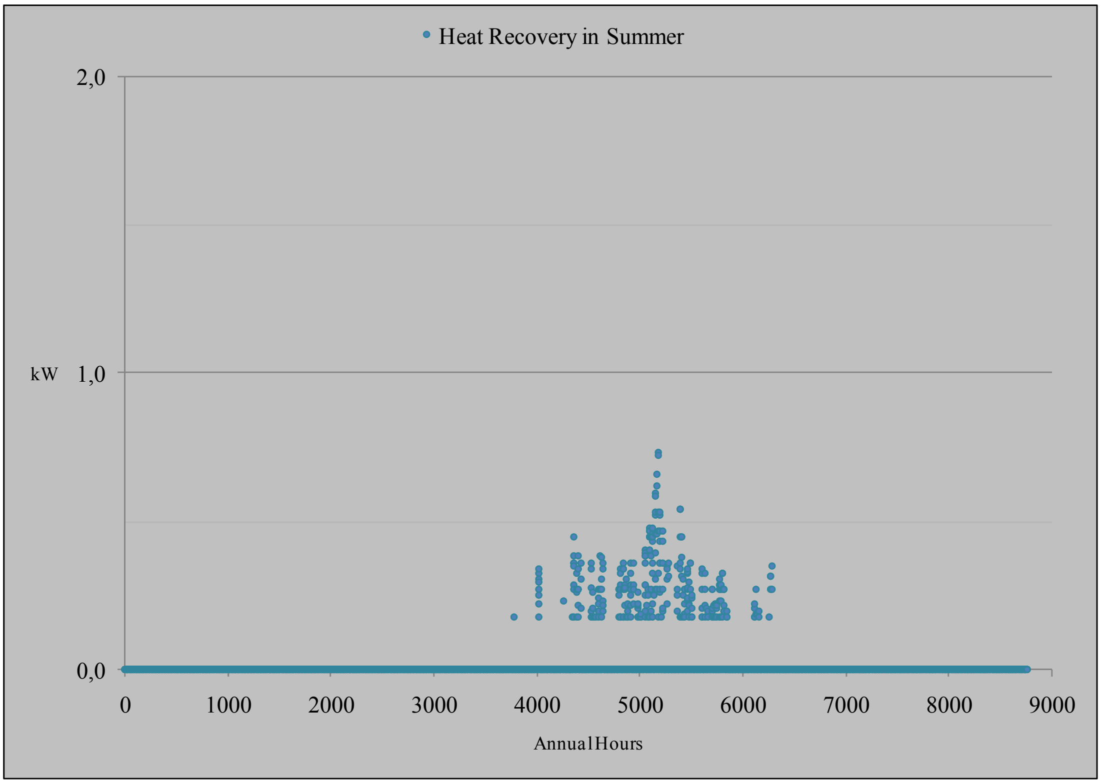


Figure 4.18 Annual hourly summer heat recovery in İstanbul.

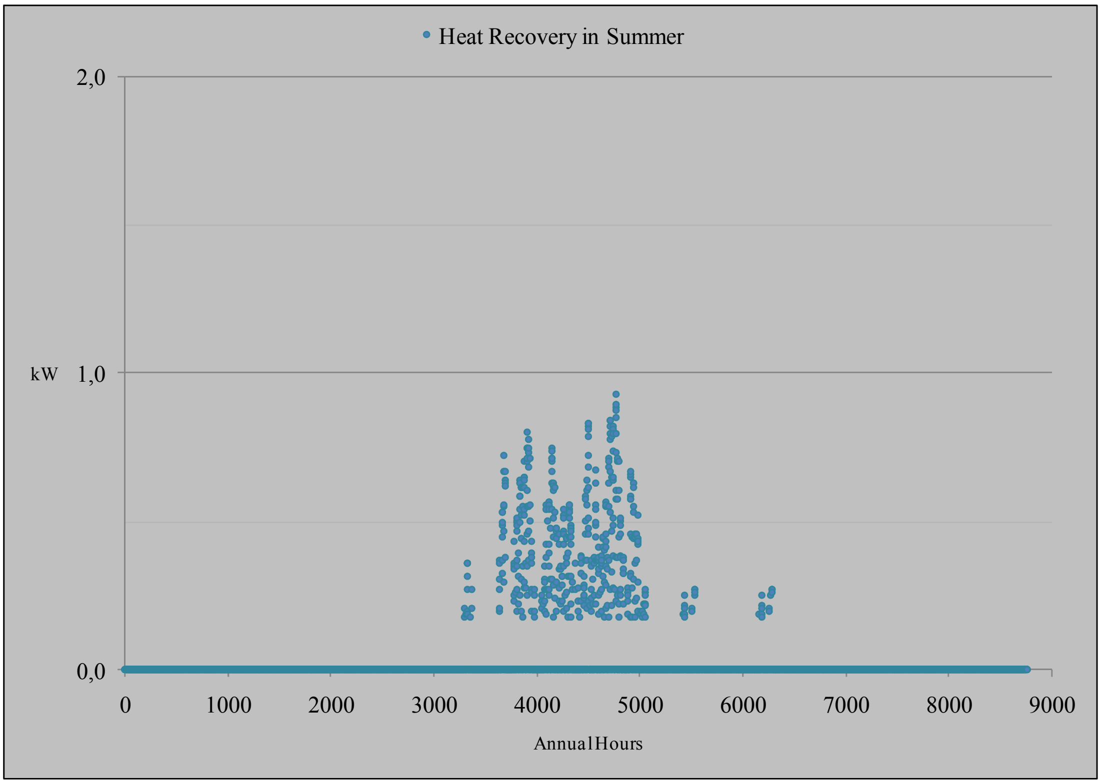


Figure 4.19 Annual hourly summer heat recovery in Ankara.

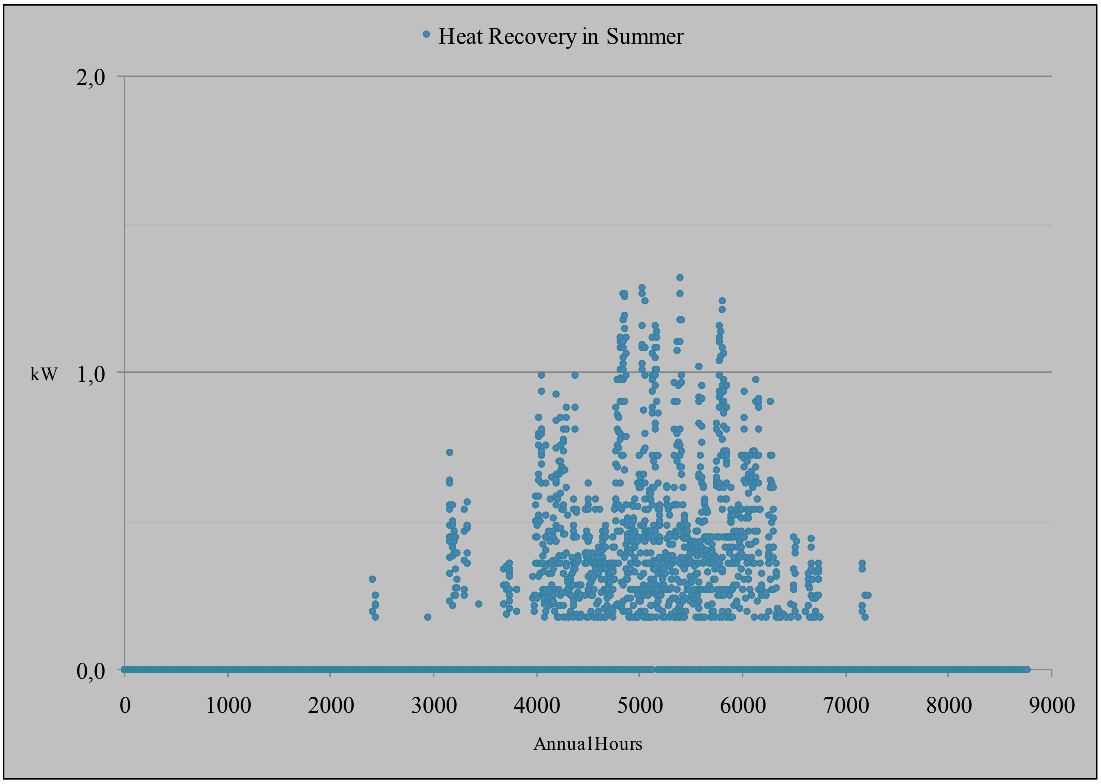


Figure 4.20 Annual hourly summer heat recovery in Antalya.

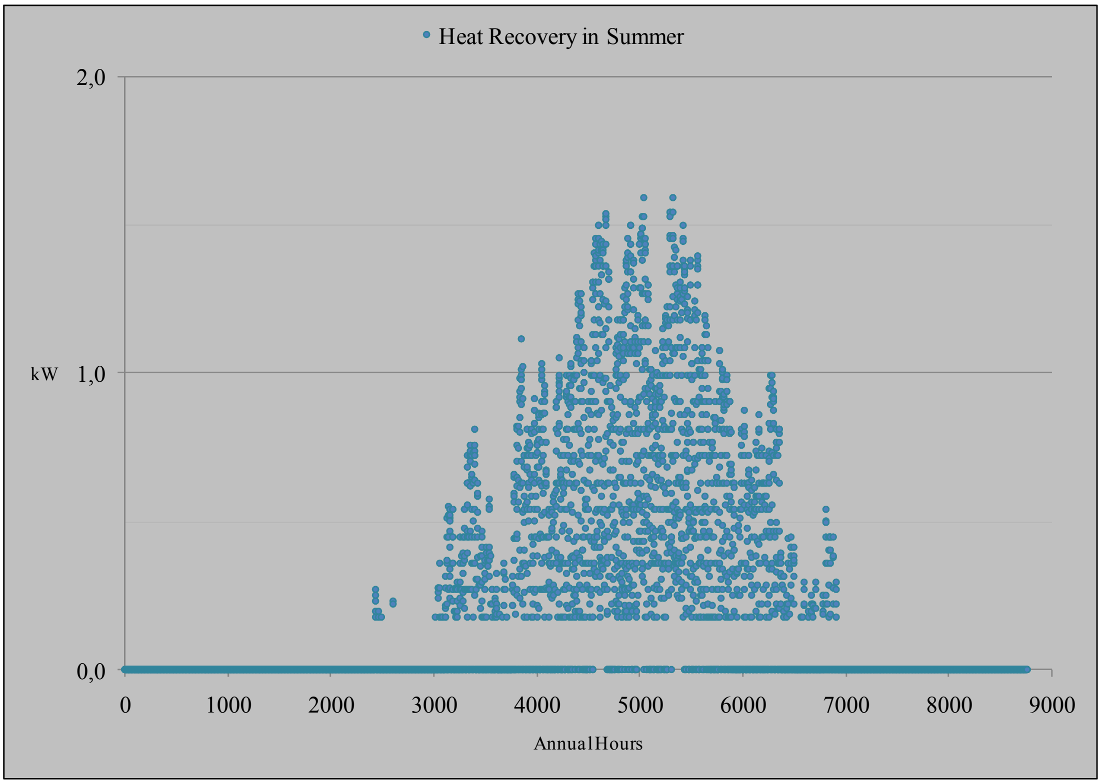


Figure 4.21 Annual hourly summer heat recovery in Urfa.

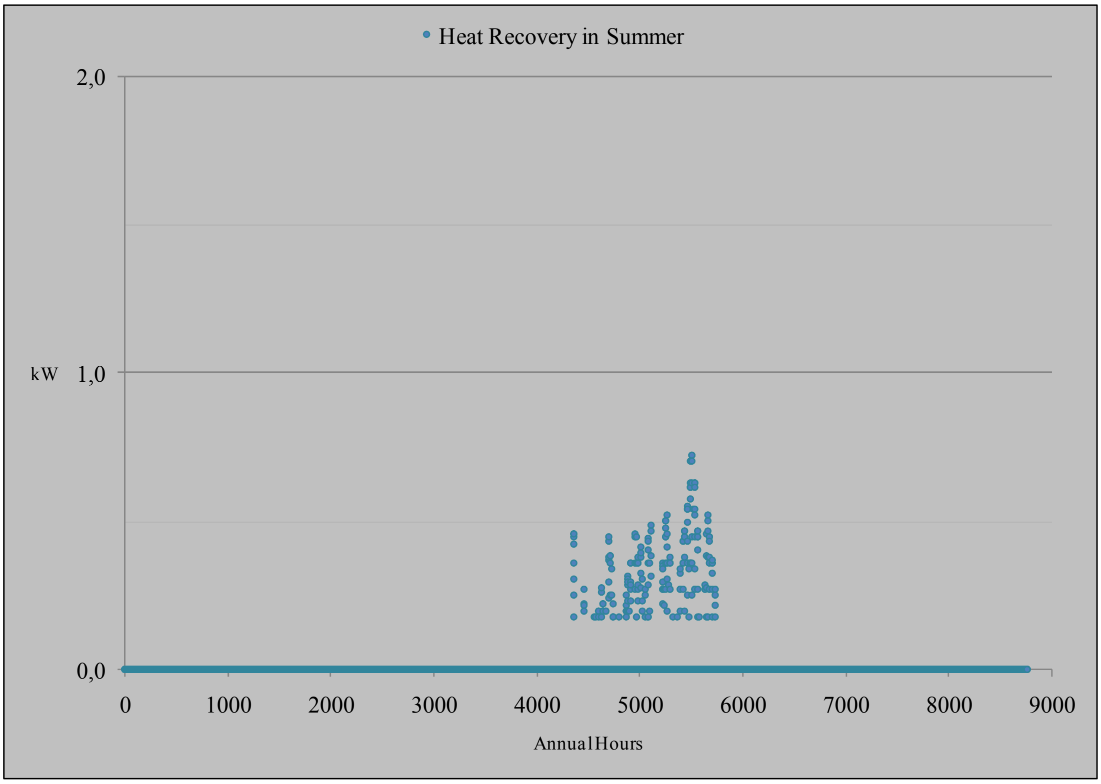


Figure 4.22 Annual hourly summer heat recovery in Erzurum.

Summer heat recovery in İzmir, Antalya and Urfa are alike as outdoor air temperature for mentioned cities are alike. İstanbul, Ankara and Erzurum's heat recovery in summer is very low due to outdoor air temperature, also outdoor air temperatures, especially in night, allow by-pass ventilation. In these cities by-pass ventilation is a must as it reduces energy consumption for air conditioning.

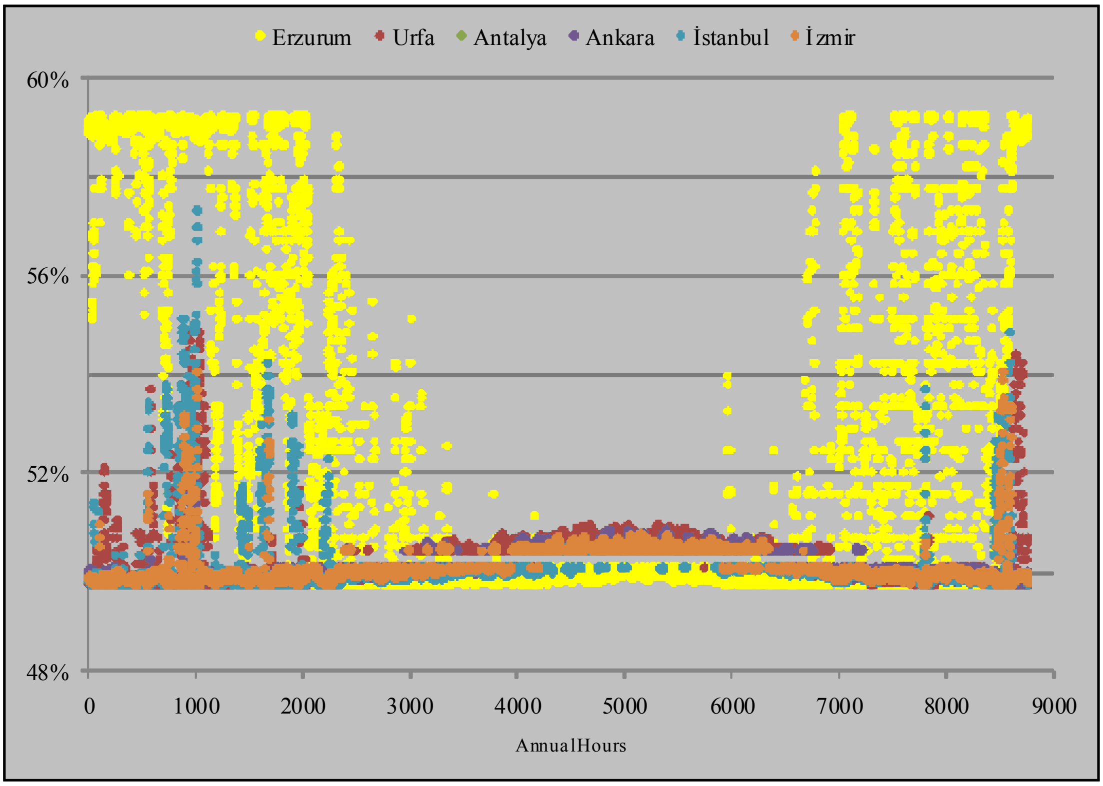


Figure 4.23 Heat recovery efficiency comparisons.

For summer season, although outdoor air temperatures vary in examined cities, it is observed that heat recovery efficiencies are alike. In winter season, heat recovery efficiency for colder cities like Erzurum, İstanbul and Ankara is considerable larger than warmer cities, Urfa, İzmir and Antalya. This is because of the condensation in exhaust air. In colder cities, exhausted indoor air is cooled down to its dew point temperature and below in the heat exchanger which results in condensation so, the latent heat comes upon the earth and increases the thermal efficiency of the heat recovery ventilator.

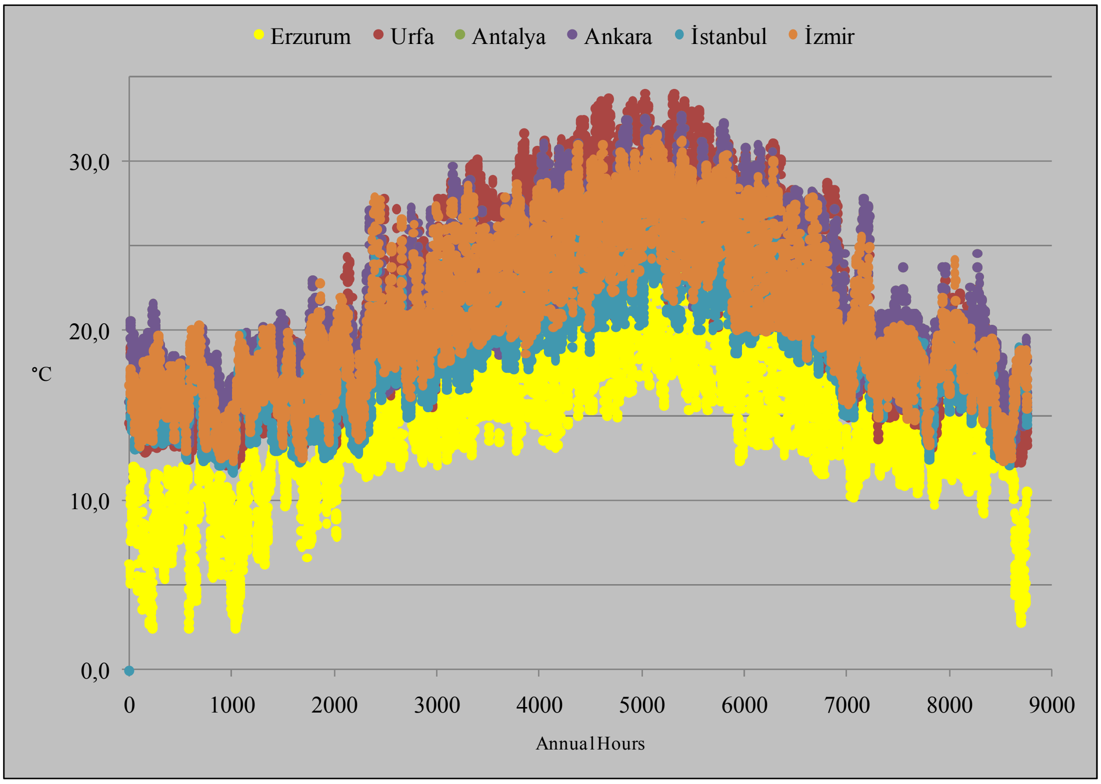


Figure 4.24 Supply air temperatures after heat recovery ventilator.

For moderate and cold climates examined supply air temperature for winter after heat recovery ventilator are close to each other, although outdoor air temperature varies significantly. For extreme climates like Erzurum it is observed that supply air temperature after heat recovery ventilator is less than all other cities considered although it has the highest thermal efficiency for the heat recovery ventilator. For winter, except Erzurum, supply air temperature is between 14-20°C.

For summer the interval between maximum and minimum supply air temperatures increases where the temperature differs from 19°C to 34°C. It is also observed in summer that although outdoor air temperature is different in examined cities, supply air temperatures are very alike.

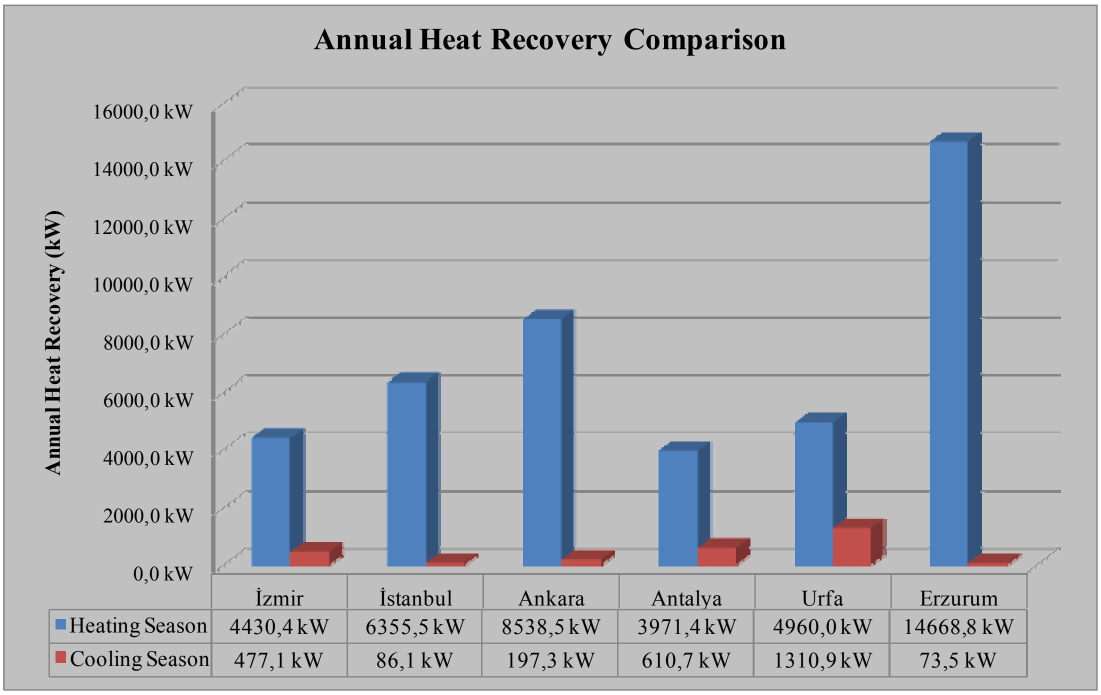


Figure 4.25 Annual heat recovery comparisons.

Annual heat recovery characteristic for İzmir, Antalya and Urfa are alike as discussed previously. Winter heat recovery for these cities is low than other cities where summer heat recovery is more. Urfa has the most heat recovery for summer over all cities. İstanbul is between these 3 cities and Ankara for winter heat recovery and has the least heat recovery for summer like Erzurum. Erzurum, with its extremely cold climate, has the biggest amount of heat recovery for winter, whereas it has the lowest heat recovery for summer.

The previous chapter, annual heat recovery for Ankara is examined according to monthly average outdoor temperatures. Annual heat recovery obtained after the calculation was 8091 kW. According to the calculation that is presented in Figure 4.25, annual heat recovery for Ankara is 8735,8 kW. As a result it shall be considered that using average temperatures for outdoor air, yearly, monthly or daily, result in calculation errors. To obtain heat recovery accurately, a wide range of outdoor air temperature shall be regarded in the calculation.

4.3 Payback period for HRV's in Different Cities

Payback period for Ankara has been considered in Chapter 3. Further analysis including both heating and cooling seasons will be examined.

Heat recovery during heating season reduces the amount of energy used for air conditioning indoors. For all cities discussed, natural gas burners are used to heat up water that is used for cassette type fan coils in the air conditioning system for heating. For cooling wall type air conditioners with an average COP of 2,7 are used.

Annual heat recovery for both heating and cooling seasons is calculated in this chapter. To calculate payback periods for each city, additional natural gas consumption demand and additional air conditioner electrical operational demand are considered. Electrical cost for residential use is 0,095 €/kW. In air conditioners thermal electrical consumption can be calculated;

$$COP = \frac{Q_{evap}}{w_{net}} \tag{4.1}$$

Where;

COP : Coefficient of performance

Q_{evap} : Cooling Capacity

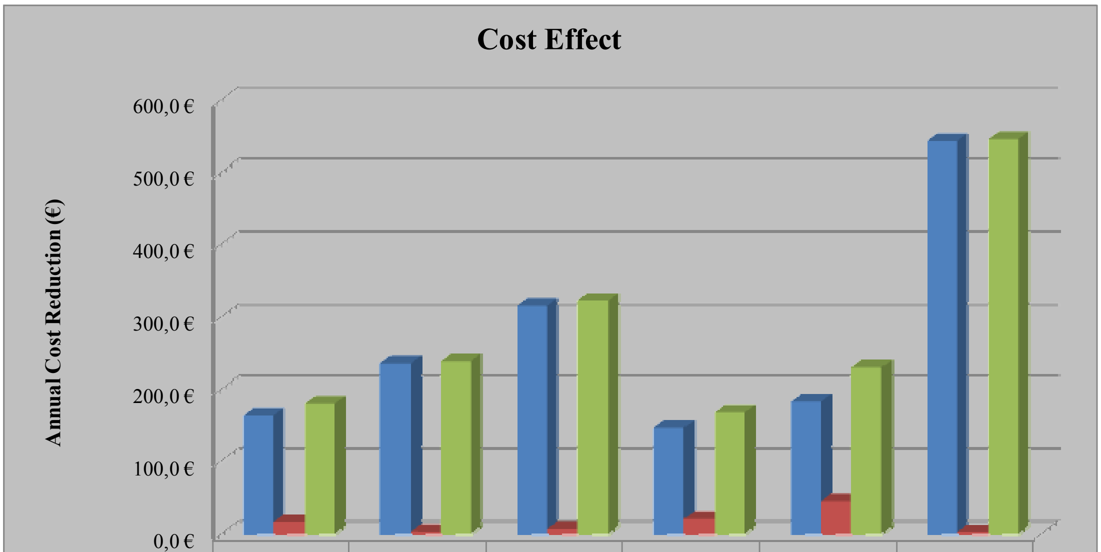
w_{net} : Net compressor work

To calculate operational cost reduction in air conditioners formula 4.1 is used and cost benefit of every kW of cooling reduction is;

$$2{,}7 = \frac{0{,}095€}{electrical\ cost} \tag{4.2}$$

$$electrical\ cost \quad = 0{,}035\ €/kW \tag{4.3}$$

Natural gas cost for the cities are 0,037 €/kW as discussed in the previous chapter.



	İzmir	İstanbul	Ankara	Antalya	Urfa	Erzurum
Heating Season	163,9 €	235,2 €	315,9 €	146,9 €	183,5 €	542,7 €
Cooling Season	16,7 €	3,0 €	6,9 €	21,4 €	45,9 €	2,6 €
Total	180,6 €	238,2 €	322,8 €	168,3 €	229,4 €	545,3 €

Figure 4.26 Annual costs reduction.

Operational cost reduction as a result of heat recovery usage is shown in Figure 4.26. Antalya and İzmir with their warm climate over the season has the least operational cost reduction due to heat recovery ventilator usage in 6 cities. Urfa, another warm city, also has low operational cost reduction characteristic in heating season, but has the biggest operational cost reduction in cooling seasons. İstanbul and Ankara follows these 3 cities with significant operational cost reduction. Erzurum, the coldest city examined, has the biggest operational cost reduction throughout the year. Figure 4.26 also indicates that, independent from the climate, heat recovery ventilator reduces operational cost significantly.

Table 4.1 Cost comparison analysis between HRV System and Duct Fan System

		HRV System	Duct Fan System
Initial Cost	Heat Recovery Ventilator	750,0 €	-
	Exhaust Air Fan	-	160,0 €
	Supply Air Fan	-	160,0 €
	Fresh Air Filter	-	55,0 €
	Pre-heater	-	Neglected
Operational Costs	Electrical Consumption	Neglected	Neglected
	Service Costs	Neglected	Neglected
	Pre-heating cost	-	Various

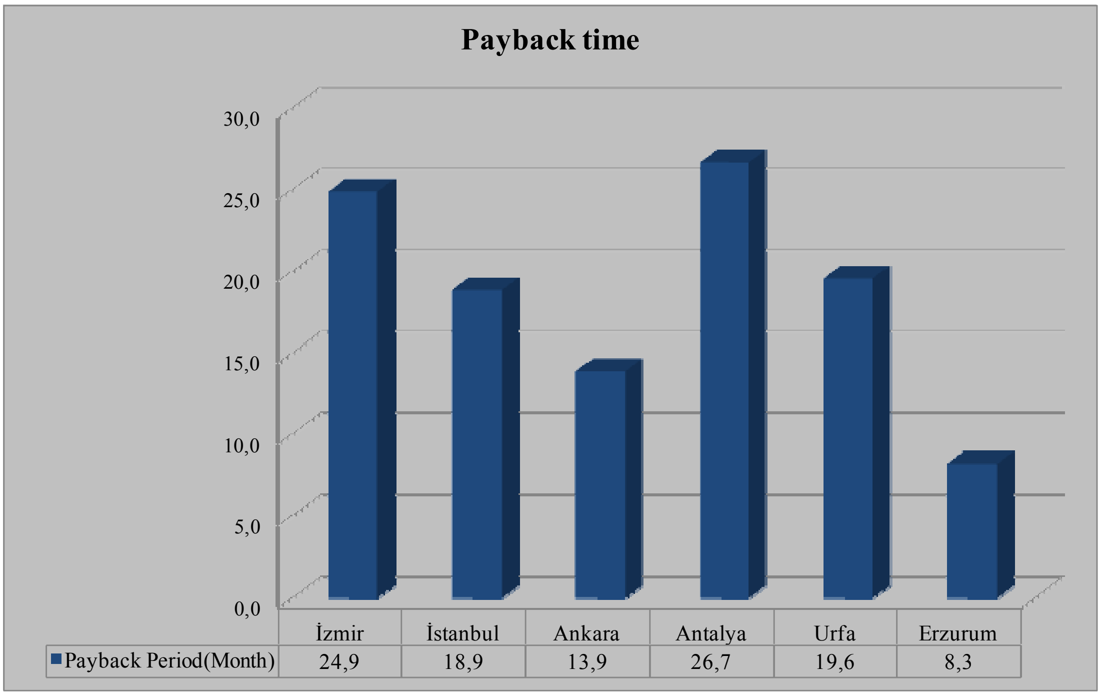


Figure 4.27 Payback time for Cities.

CHAPTER FIVE

CONCLUSIONS

5.1 Overview

In this research, outdoor air temperatures for 6 cities of Turkey have been used for energy saving calculation of heat recovery ventilators. A call center has been taken as a sample and analyzed according to average temperature method and hourly temperature method. For calculations, certified software has been modified according to multiple outdoor air temperatures input. Supply air temperature after heat recovery ventilator, heat recovery exchanger efficiency, and recovered energy has been calculated and evaluated.

In appropriate air conditions, by-passing outdoor air through the heat recovery exchanger helps to reduce air conditioning load inside, which is called by-pass ventilation or free cooling. In most summer nights by-pass ventilation occurs. In this research by-pass hours are encountered and were not put in the calculations.

5.2 Conclusions about Heat Recovery Ventilator and Future Work

The starting point of this thesis is to represent heat recovery efficiency differences due to various outdoor air conditions. 6 cities have been considered having various climate conditions, one group of warm cities, one group of cold cities and one group of extremely cold city.

It is noticed that heat recovery ventilator efficiency increases as the temperature gap between outdoor air and indoor air. As a result the highest efficiency has been observed in the extremely cold city, Erzurum and then colder cities Ankara and İstanbul and finally warmer cities Antalya, İzmir and Urfa. Although the climate conditions vary, it is observed that cities except Erzurum have likely the same supply air temperature after the heat recovery ventilator because of the efficiency difference between cities.

Recovered heat, as foreseen, has the maximum value where heat recovery ventilator efficiency is the biggest, Erzurum. The following cities are drawn up likely to the efficiency characteristics.

In Chapter 3 heat recovery calculations in the sample installation has been considered according to average outdoor air temperatures. Same calculation has been made in Chapter 4 for hourly outdoor air temperatures. In Chapter 3 only 12 temperatures have been used for calculation where 8760 temperature data have been used for calculation. This resulted in very accurate calculation and the chance to compare the results. It is shown that to have an exact result more number of temperatures for outdoor air shall be considered during calculation.

It is also shown that by-pass ventilation takes place in all cities. A future study shall be made to consider by-pass ventilation effect in heat recovery ventilation.

Also the sample was considered for 24h running office, whole calculations shall be regarded for one shift offices which are normally closed in nights which will reduce energy savings in heating season.

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