İSTANBUL TECHNICAL UNIVERSITY ★ ENERGY INSTITUTE

ECONOMIC ANALYSIS OF LOW TEMPERATURE WASTE HEAT RECOVERY OPPORTUNITIES AT TWO FACILITIES OF DAIMLER AG

M.Sc. Thesis by
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Department : Energy Science and Technology

Programme : Energy Science and Technology

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JANUARY 2011

İSTANBUL TEKNİK ÜNİVERSİTESİ ★ ENERJİ ENSTİTÜSÜ

DAIMLER AG'YE AİT İKİ FABRİKADA DÜŞÜK SICAKLIKTA ATIK ISI GERİ KAZANIMI OLANAKLARININ EKONOMİK ANALİZİ

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FOREWORD

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December 2010   Atilla AYATA

                                                                    Energy Science and Technology

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ABBREVIATIONS

AG	**:** Incorporated (stock) company
ASR	**:** Guideline for comfort requirements of working environment
DIN	**:** German Institute for Standardization
HVAC	**:** Heating-Ventilating-Air Conditioning
VDI	**:** Association of German Engineers

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ECONOMIC ANALYSIS OF LOW TEMPERATURE WASTE HEAT RECOVERY OPPORTUNITIES AT TWO FACILITIES OF DAIMLER AG

SUMMARY

Rational use of energy aims to enhance the prosperity and comfort of a country and to alleviate greenhouse gas emissions. Waste energy recovery is one of the methods to achieve rational use of energy. In particular, waste heat recovery is an interesting topic of industrial energy management, since industrial energy plays a key role for end use energy and process heat is the most frequently demanded sort of energy. In other words, it is a significant tool to diminish installation and operation costs in order to survive in the competitive market.

This study focuses on two facilities of Daimler AG in Stuttgart, where waste heat is rejected from mechanical production at low temperature. Therefore, opportunities of low temperature waste heat recovery are investigated to identify a bundle of equipments to fulfill the needs at best. Primary objective is to help the decision makers to select the most viable equipment.

Theoretical sections provide fundamental information about differences and similarities of low temperature waste heat recovery with the help of figures and tables. Moreover, application fields, installation and operation requirements, and subsequently exchange coefficients depending upon other variables are encountered. Sections which include practical information and calculations are based on the knowledge introduced previously. Out of the theoretical potential economical potential is derived regarding to the realistic data obtained from contemporary literature and equally important from manufactures to attain the most accurate results. Consequently an economic analysis is proceeded to compare the selected equipments.

DAIMLER AG'YE AİT İKİ FABRİKADA DÜŞÜK SICAKLIKTA ATIK ISI GERİ KAZANIMI OLANAKLARININ EKONOMİK ANALİZİ

ÖZET

Enerjinin rasyonel kullanımı toplumun refah seviyesini arttırmakta, sera gazı salınımını azaltmaktadır. Atık enerjinin geri kazanımı enerjinin rasyonel kullanımına olanak sağlayan yöntemlerden biridir. Sırasıyla enerji talebinde endüstrinin, endüstride kullanılan enerjide de ısı enerjisinin büyük pay sahibi olması, atık ısı geri kazanımını önemli kılmaktadır. Daha açık bir ifadeyle atık ısı geri kazanımıyla yatırım ve işletme maliyetleri düşürülebilir ve böylece firmaların pazarda rekabet gücünün artması sağlanabilir.

Bu çalışmanın ana konusu Daimler AG'nin Stuttgart'ta bulunan iki fabrikasında, üretim proseslerinde açığa çıkan düşük sıcaklıkta atık ısının geri kazanımıdır. Bu kapsamda, mevcut ihtiyaçları en iyi şekilde giderebilmek için düşük sıcaklıkta atık ısı geri kazanım olanakları araştırılmıştır. Esas amaç, karar alıcılara konu özelinde en uygun ekipmanı seçme konusunda yardımcı olmaktır.

Teorik bilgiyi içeren kısımlar düşük sıcaklıkta atık ısı geri kazanımında kulanılan ekipmanların benzer ve ayırt edici özelliklerini tablo ve şekiller yardımıyla özetlemektedir. Söz konusu ekipmanların kullanım alanları ve diğer bazı değişkenlere bağlı olarak farklılaşan verimliliklerine de yine bu bölümde yer verilmiştir. Hesaplamaları ve diğer bazı pratik bilgileri içeren kısımlarda ise teorik kısıma dayanarak konunun kapsamı çalışma özelinde daraltılmış teorik potansiyelden ekonomik potensiyele geçilmiştir. Bunu yaparken mümkün olan en doğru ve eksiksiz sonuca ulaşmak amacıyla, güncel literatür bilgilerinin yanı sıra firmaya ait bilgiler sunulduktan sonra üreticilerden tedarik edilen bilgilerden de yararlanılmıştır. Ekonomik analizin ikinci ve son aşamasında ise seçilen ekipmanlar arasında karşılaştırma yapılmıştır.

1. INTRODUCTION

The prosperity and comfort of today's civilization is based on the rational use of energy. Production of steel and concrete for buildings, energy supply for transportation, manufacturing of various industrial goods may be introduced to exemplify continuous needs.

Rational use of energy refers to all precautionary measures concluding to optimal consumption of national economic resources [1]. To provide the rational use of energy, four possibilities may be listed as follows:

- Avoidance of non-essential use: Unnecessary production of goods or services to provide extra comfort must be prevented.
- Decline of energy demand: Mostly technical precautions e.g. material selection, heat insulation and optimal construction are applied in order to alleviate end-use energy.
- Improving energy efficiency: Enhancement of efficiency via construction, thorough maintenance and high capacity of work load is aimed.
- Waste energy utilization: Commonly heat recovery is the foremost purpose, whereas electricity consumption also increases simultaneously whilst operating regenerators, recuperators or heat pumps.

1.1 Why is rational use of energy important?

Whilst fulfilling the needs of mankind, there exist two regarded parameters from energy point of view to measure the prosperity of a country for which the rational use of energy is intended: Energy intensity and energy consumption per person [2]. Final remarkable benefit would be reduction of greenhouse gas emissions.

Energy intensity is one of the subtopics to identify and compare energy requirement on a national base, which is the ratio of units of energy to gross domestic product (GDP). Gross domestic product or gross domestic income is the total annual market

value of goods and services produced in a country. High energy intensity means high expenditure for unit good or service.
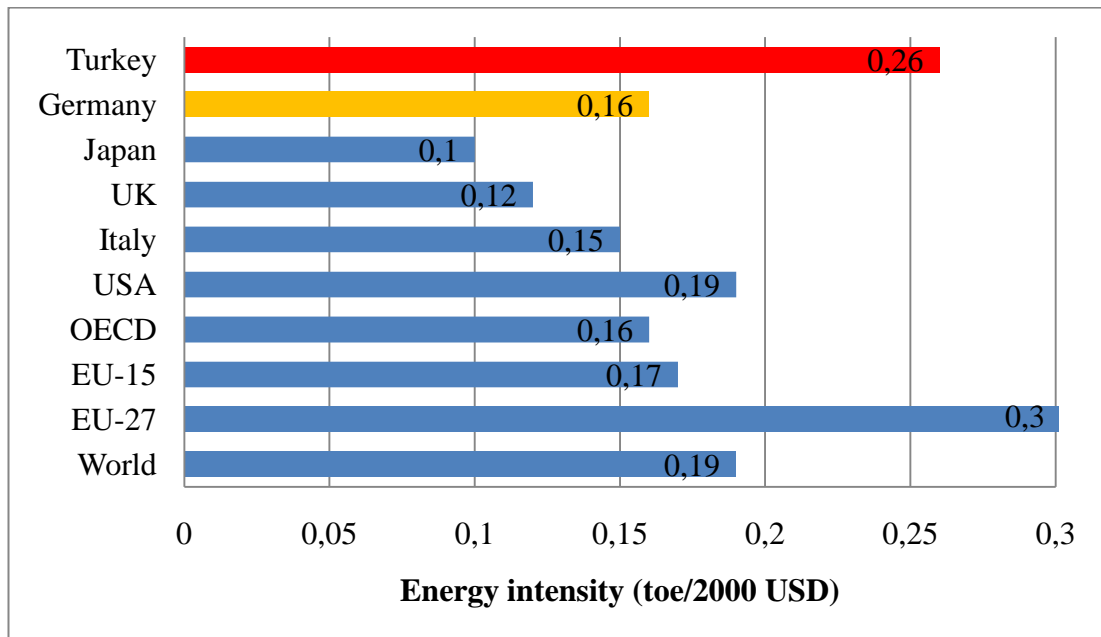
Figure 1.1: Total primary energy supply per unit of GDP [3]

Figure 1.1 demonstrates energy intensities of some countries and also average energy intensities of some international organizations. Energy intensity of Turkey is relatively higher than average energy intensities of world, OECD and EU-15 countries. The number of EU members reached 27 in 2004. New members with extreme energy intensities such as Bulgaria (1.01), Romania (0.64) and Czech Republic (0.56) increase the EU average significantly and subsequently exceed energy intensity of Turkey. To sum up, Turkey's energy expenditure for a unit good or service is high in comparison with other competitor countries in global market. Germany's energy intensity is sufficiently low to be below average within EU and OECD. On the other hand, there exist still countries such as Japan, United Kingdom, and Italy, which have achieved lower energy intensities.

Another remarkable parameter concerned with prosperity is total primary energy supply (TPES) per person. This ratio indicates how high the life standard of a person is, therefore higher values are to be attained. Figure 1.2 illustrates TPES/population ratio of some countries and organizations. TPES/Population ratio is significantly low in Turkey regarding to OECD and EU countries and also globally below average. Even though TPES/population ratio of Germany is far more than world average, it is below OECD and EU-15 average.
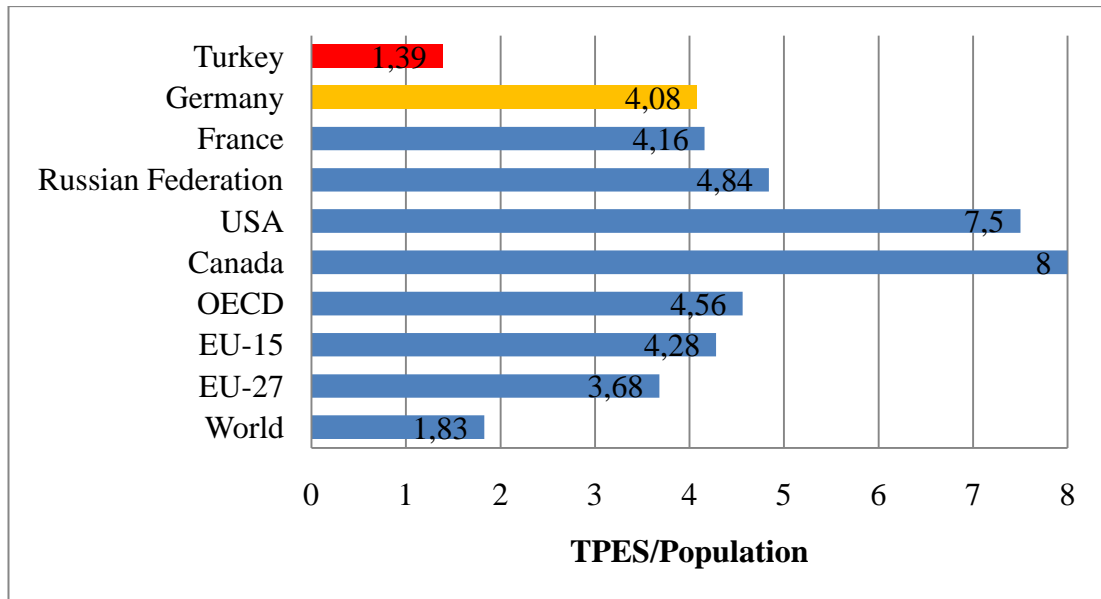
Figure 1.2: TPES/Population ratio [3]

Final remarkable outcome of rational use of energy is the reduction of greenhouse gases (GHG). Greenhouse gas profiles report of European Environment Agency (EEA) was released in October 2010 based on the data of 2008. Figure 1.3 shows several results of this report.
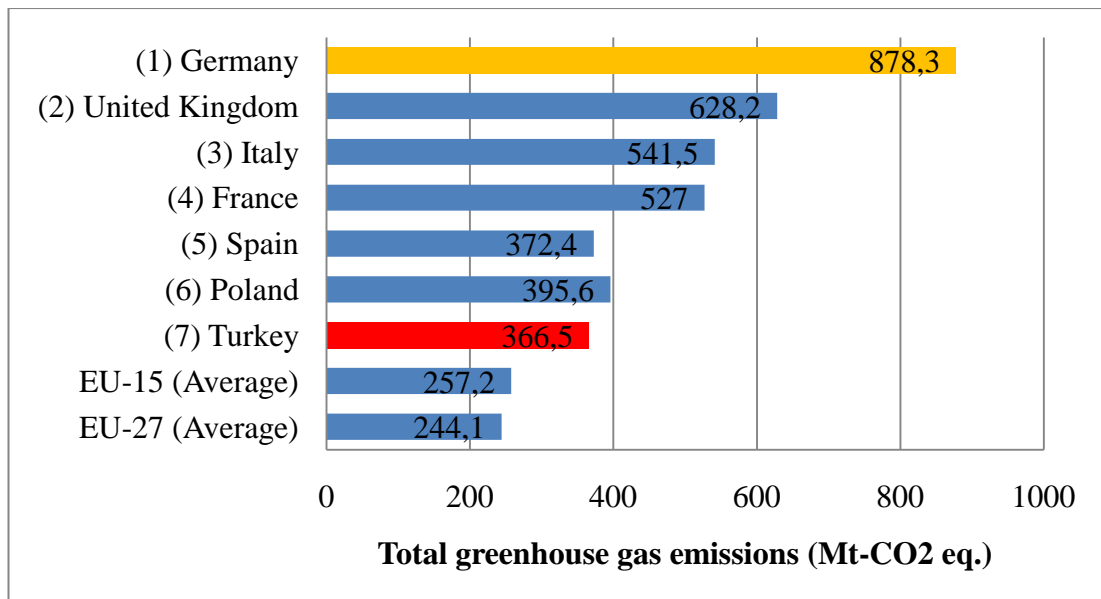
Figure 1.3: Total greenhouse gas emissions [4]

According to this report, Germany emits the highest amount of GHG in the atmosphere, which is a lot more than EU-15 and EU-27 averages. If Turkey was a member of EU, averages of both EU-15 and EU-27 were exceeded. Hence, Turkey would be at the seventh place of this ranking.

On the whole, rational use of energy is important, since it aims to enhance prosperity and comfort of a society. In other words, rational use of energy seeks for low energy intensity and high TPES/population ratio. Low energy intensity indicates that, same amount of goods and services are manufactured with less energy, where high TPES/population ratio shows better living circumstances in a country. Subject to Figure 1.2 and Figure 1.3 Turkey has considerably high energy intensity and low TPES/population ratio, which is the opposite of the desired trend. Germany attains low energy intensity but there exist countries with lower energy intensities, the economies of which acquire similar economic sizes with Germany. Nevertheless, TPES/population ratio of Germany needs to be improved in order to reach higher values. Last but not the least, GHG may be alleviated by means of rational use of energy, as well. As far as greenhouse gas profiles of Turkey and Germany concerned, possibilities of rational use of energy may be regarded as opportunities to reduce emissions of both countries, which are above average within EU.

1.2 Rational use of energy intended for industrial energy consumption

Figure 1.4 highlights how the end-use energy is distributed in different fields according to the statistics of Ministry of Energy and Natural Resources.
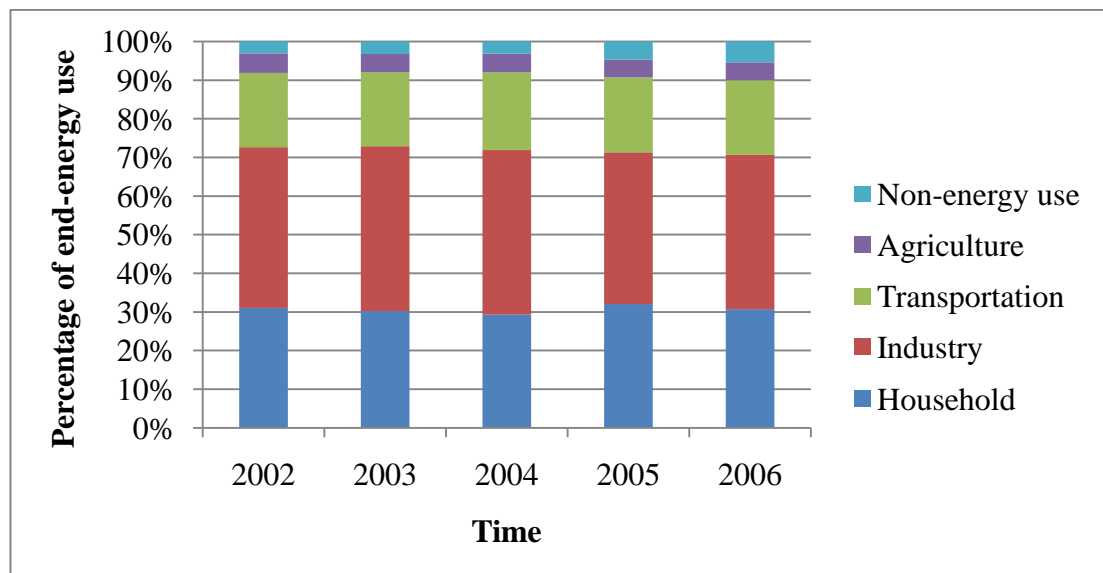
Figure 1.4: Distribution of end-use energy consumption [5]

Since 1996 end-use energy of industry is always more than households and other subgroups in Turkey and shows a clear tendency towards increase quantitatively. According to Ministry of Energy and Natural Resources, industrial energy

consumption is foreseen to be 136 and 177 Mtoe in 2015 and 2020, which are equivalent to 41% and 43% of the total energy demand, where demand of households will be 28% and 27%, respectively [6]. Hence, energy consumption of industry plays a major role for Turkey, for which 20% saving potential currently exists [7].

The economic health of Germany is also contingent upon the industrial sector, which uses 28.5% total end-use energy of 205 Mtoe. The overwhelming part with 67% of the industrial end-use energy is intended to provide heat for manufacturing processes as shown in Figure 1.5 [1].
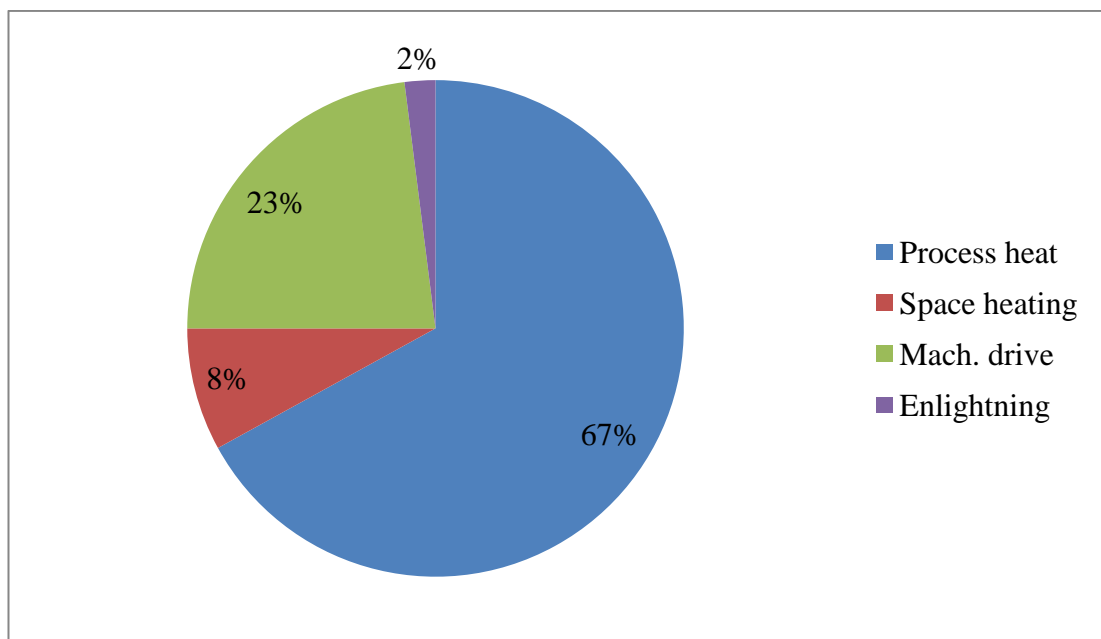
Figure 1.5: Industrial energy use in Germany

Combining waste energy utilization, which is one of the possibilities for the rational use of energy and the importance of process heat in Germany, results in recovery of waste heat as the priority of the goal setting process in waste energy utilization concept. Waste heat has been defined as heat which is rejected from a process at a temperature enough above the ambient temperature to permit the manager or engineer to extract additional value from it [8].

Another research named "Energy Use, Loss ad Opportunities Analysis: U.S Manufacturing and Mining" which is released by U.S. Department of Energy in December 2004 also confirms this determination. Table 1.1 summarizes the best opportunities for future energy savings in commodity/process manufacturing. According to this research waste heat and energy recovery leads to the largest total

energy and cost savings among all other opportunities. 35% of total energy savings equivalent to 46.1 Mtoe and 34% of total cost savings equivalent to 6408 $mill. may be achieved via waste energy and heat recovery. By way of illustration, waste heat and energy recovery from drying processes, by-product gases, metal quenching/cooling processes, gases and liquids including hot gas cleanup and dehydration of liquid waste streams are several types of opportunities.

Table 1.1: Industries' best energy opportunities for future energy savings [9].

Top R&D Opportunities for Energy Savings in Commodity/Process Manufacturing; Initiatives that Provide the Largest Energy and Dollar Savings				
Type of Opportunity	Total Energy Savings		Total Cost Savings	
	(Mtoe)	Percent of Total %	($mill.)	Percent of Total %
Waste heat and energy recovery	46.1	35	6408	34
Improvements to boilers, fired systems, process heaters and cooling systems	22.9	17	3077	16
Energy system integration and best practices opportunity	36.2	28	5655	30
Energy source flexibility and combined heat and power	20.9	16	3100	16
Improved sensors, controls, automation and robotics for energy systems	4.8	4	630	3
Totals	130.9		18870	

1.3 Purpose of the thesis

The purpose of this project is to research recovery possibilities of waste heat rejected from two factories of Daimler AG in Germany. There exists low temperature waste heat with high volumetric flow rate particularly dissipated from mechanical production.

A two phase project is suggested. At the initial phase waste heat streams are scrutinized and assessed in order to determine the theoretical waste heat recovery potential. Concrete measures related to low temperature waste heat recovery are

described and economically assessed. The most suitable applications are carried over for a detailed analysis.

In the second phase economic feasibility of selected precautions are evaluated and sought for recovery. If feasible, viable findings about costs of a potential recovery are encountered and compared in order to lead the decision maker towards the optimal solution. Conceptually, no process design is required.

2. FUNDAMENTALS OF WASTE HEAT RECOVERY

Fundamentals of waste heat recovery refer to physical terms of recovery processes, aimed human comfort requirements as a result of air conditioning processes, meteorological fundamentals and other related parameters.

2.1 Physical terms

Physical terms concerned with waste heat recovery are sensible and latent heat; moisture/water content, specific and relative humidity of air, enthalpy and finally h-x diagram demonstration of a waste heat recovery process.

2.1.1 Sensible heat

Sensible heat is energy, the transfer of which yields solely a temperature difference. Amount of transmitted sensible heat is proportional to temperature difference, mass and specific heat capacity of the material. Besides, no phase change takes place.

2.1.2 Latent heat

Latent heat is released or stored through change of the physical state e.g. evaporation and condensation of the heat transfer mediums. Unlike sensible heat, temperature remains constant during latent heat transfer.

2.1.3 Moisture/Water content

Air without vapor content is defined as dry air. In latter cases the fluid may be composed of water vapor or moisture which is of low quantity, whereas it directly affects the human comfort in HVAC applications. Atmospheric air, in other words air in the atmosphere illustrates such an example and contains water vapor. Commonly in HVAC applications, amount of dry air is assumed to be constant, where the amount of water vapor changes.

2.1.4 Specific and relative humidity of air

Specific humidity indicates the ratio of mass of water vapor to a particular mass of dry air.

$$SH = m_v / m_a \tag{2.1}$$

SH: Specific humidity
m_v: Mass of water vapor (kg)
m_a: Mass of dry air (kg)

A similar term may also be expressed as the mass of water vapor per unit volume of air, which is called absolute humidity.

$$AH = m_v / V_a \tag{2.2}$$

AH: Absolute humidity
m_v: Mass of water vapor (kg)
V_a: Volume of dry air (m^3)

Augmenting ratio of specific humidity refers to an increase in specific humidity. Nevertheless, moisture content of an air stream may not be infinite. Hence, air is saturated and subsequently excess moisture condenses. Saturation point varies subject to temperature. Sinking temperature concludes to alleviation moisture capacity.

Human comfort hinges on relative humidity rather than specific humidity. Relative humidity expresses the ratio of the moisture content of the air to moisture content of the saturated air at the same temperature. It varies from 0 to 1.

$$RH = m_v / m_s \tag{2.3}$$

RH: Relative humidity
m_v: Mass of water vapor (kg)
m_s: Mass of moisture of the saturated air (kg)

2.1.5 Enthalpy

Enthalpy stands for a thermodynamic property which is equal to the sum of internal energy and product of pressure and volume. Introduction of this term simplifies the use of variables. Via enthalpy heat transfer to/from open and closed systems may be calculated. However, it is especially useful during investigation of open systems.

$$h = u + PV \qquad\qquad (2.4)$$

h: Enthalpy
u: Internal energy
P: Pressure
V: Volume

2.1.6 h-x diagram

h-x diagrams are utilized to define the properties of atmospheric air especially in air conditioning applications. Figure 2.1 shows a typical h-x diagram according to Mollier. The horizontal axis denotes the specific humidity. The vertical axis represents the dry bulb temperature, which means the ordinary temperature of atmospheric air. Additionally there exists a curve on the right end, which is called saturation line. On the saturation line and on each curve humidity remains constant. Humidity decreases as the curves move towards the left hand side beginning from 100% on the saturation line. Constant enthalpy lines have an uphill appearance to the left.
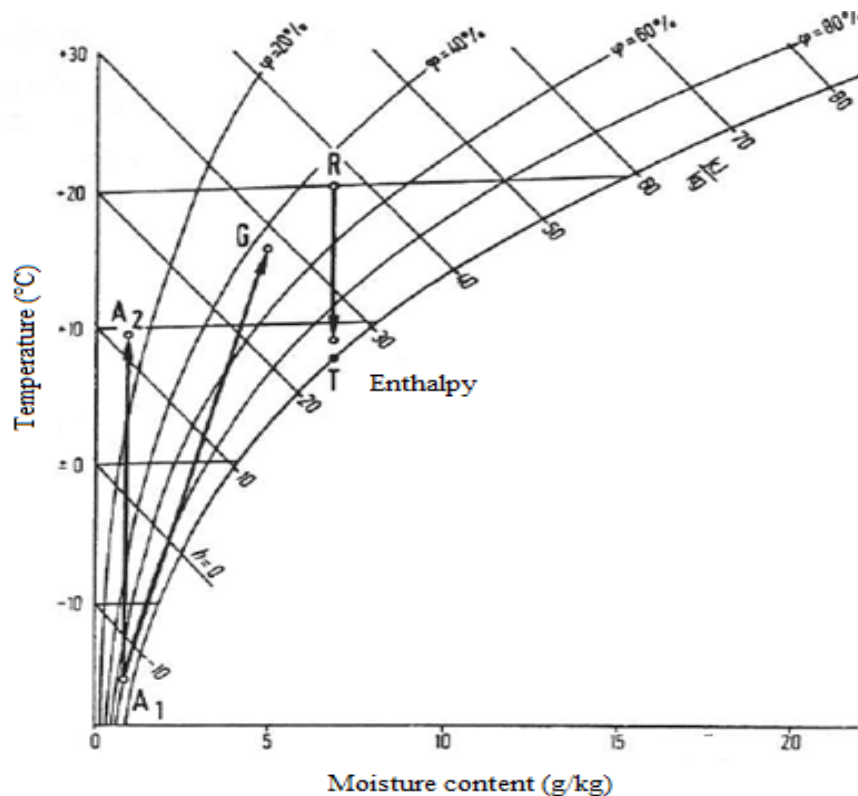
Figure 2.1: h,x diagram according to Mollier for humid air [10]

Without any latent heat transfer the state of the atmospheric air moves only in a vertical direction, in particular up for heating and down for cooling e.g. A1-A2 and

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R-T. If both latent and sensible heat is provided, the current point goes uphill to the right.

2.2. Human comfort and air conditioning processes

There exist typically four types of air conditioning processes: Heating, cooling, humidifying and dehumidifying. Those processes are either solely or together proceeded to provide the comfort of human beings. In general, cooling and dehumidifying are required in summer, whereas heating and humidifying are desired to achieve the required temperature and humidity level.

The temperature of air in air conditioning applications ranges from about -10 to 50 °C, where most people feel comfortable in an environment with a temperature between 22 and 27 °C and with a relative humidity of 40 to 60 percent [11].

During simple heating and cooling, foremost intent is to attain the desired temperature. Specific humidity remains constant. Consequently, such a heating and cooling process leads to vertical movement of the air on h-x diagram. On the other hand, heating alleviates relative humidity, since the moisture capacity of the air enhances. The same mechanism based on the same principle but proceeding in the opposite direction is applied for cooling. If needed, to avoid the problem of low relative humidity heating may be supported by humidification. Cooling and dehumidification may also take place together through attaining the dew point and condensation of surplus of water content.

2.3 Exchange coefficients

During heat recovery both sensible and latent heat may be recovered prior to the recovery equipment. Exchange coefficients are associated with the degree of recovery and divided into two subgroups. Recovered heat coefficient is encountered to define sensible heat transfer and thus recovered moisture coefficient describes the latent heat recovery process. Figure 2.2 illustrates the parameters affecting exchange coefficients. Recovered heat coefficient depends on temperature, where recovered moisture coefficient differs according to relative humidity of relevant streams.
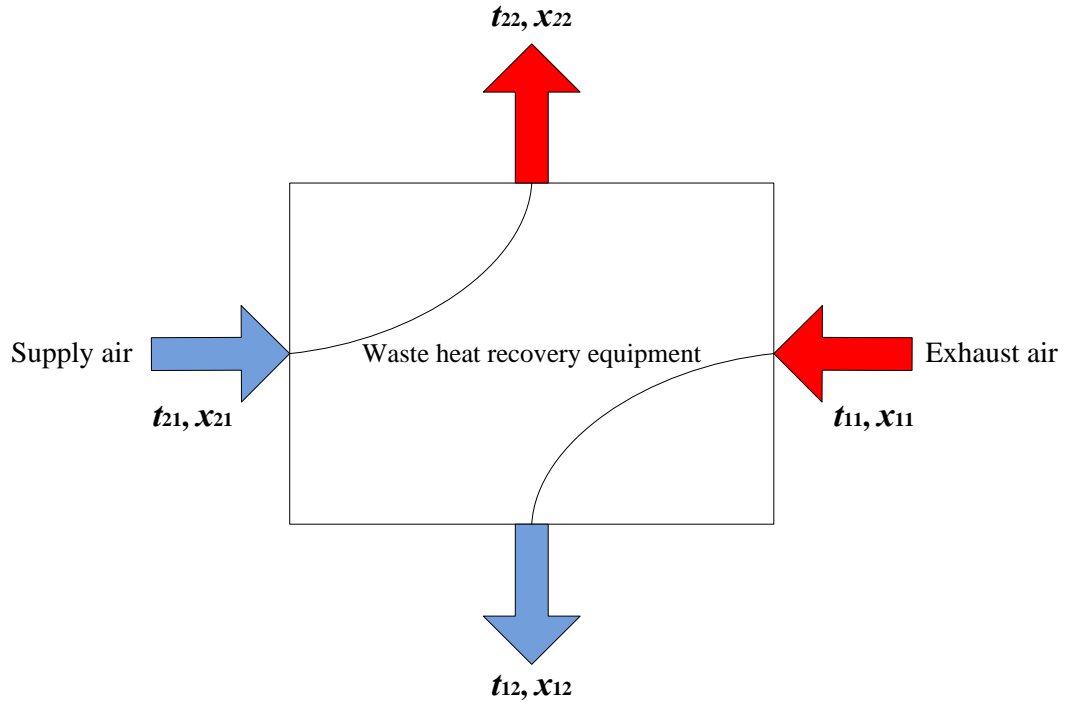
Figure 2.2: Scheme of a waste heat recovery process

2.3.1 Recovered heat coefficient

In case of an indefinitely large heat exchanger, supply air temperature may be ideally increased up to exhaust air temperature. Recovered heat coefficient expresses the ratio of absorbed heat by the supply air relative to the maximum recoverable heat under ideal circumstances of an indefinitely large heat exchanger where supply air at the exit is equal to exhaust air temperature at the entrance of a heat exchanger. The indexes of recovered heat coefficient may differ according to supply and exhaust air side, as follows [10]:

$$\Phi_A = \frac{t_{22} - t_{21}}{t_{11} - t_{21}} \qquad\qquad (2.5)$$

$$\Phi_F = \frac{t_{11} - t_{12}}{t_{11} - t_{21}} \qquad\qquad (2.6)$$

Φ_A: Recovered heat coefficient defined prior to supply air
Φ_F: Recovered heat coefficient defined prior to exhaust air
t_{11}: Exhaust air temperature at the entrance
t_{21}: Supply air temperature at the entrance
t_{12}: Exhaust air temperature at the exit
t_{22}: Supply air temperature at the exit

In case of equal mass flow rates, the following equation is valid, where Φ is overall recovered heat coefficient [10]:

$$\Phi_A = \Phi_F = \Phi \tag{2.7}$$

2.3.2 Recovered moisture coefficient

Recovered moisture coefficient is applied where latent heat in addition to sensible heat is recovered, as follows [10]:

$$\psi_A = \frac{x_{22} - x_{21}}{x_{11} - x_{21}} \tag{2.8}$$

$$\psi_F = \frac{x_{11} - x_{12}}{x_{11} - x_{21}} \tag{2.9}$$

ψ_A: Recovered moisture coefficient defined prior to supply air
ψ_F: Recovered moisture coefficient defined prior to exhaust air
x_{11}: Exhaust air moisture at the entrance
x_{21}: Supply air moisture at the entrance
x_{12}: Exhaust air moisture at the exit
x_{22}: Supply air moisture at the exit

Equal mass flow rates result in the following equation [10]:

$$\psi_A = \psi_F = \psi \tag{2.10}$$

where ψ is overall recovered moisture heat coefficient.

2.4 Meteorological fundamentals

In order to determine the annual recovery potential, meteorological fundamentals should be taken into consideration. Therefore, the progress of outside air temperature and humidity in winter and summer are observed and compared with fixed exhaust air temperature at the recovery process.

2.4.1 Supply air temperature

Temperature difference between exhaust and supply air streams directly affect the amount of recoverable energy. Besides, time period is another important factor to draw annual duration curve of outside temperature and compute specific air conditioning load.

2.4.1.1 Annual duration curve of outside temperature

Figure 2.3 shows how an annual duration curve of outside temperature is drawn. Initially, annual progress of temperature is separated into time intervals. Subsequently the number of hours composing the intervals may simply be added to state the total number of hours up to a certain temperature or within the same temperature range. As a result, annual duration curve of outside temperature is drawn. On the contrary, more viable data is required to calculate the annual recovery potential, since the operation time does not continue for 24 hours in some cases. Specialized duration curves or converting factors are applied to calculate the frequency of these temperatures regarding to place and operation time interval.
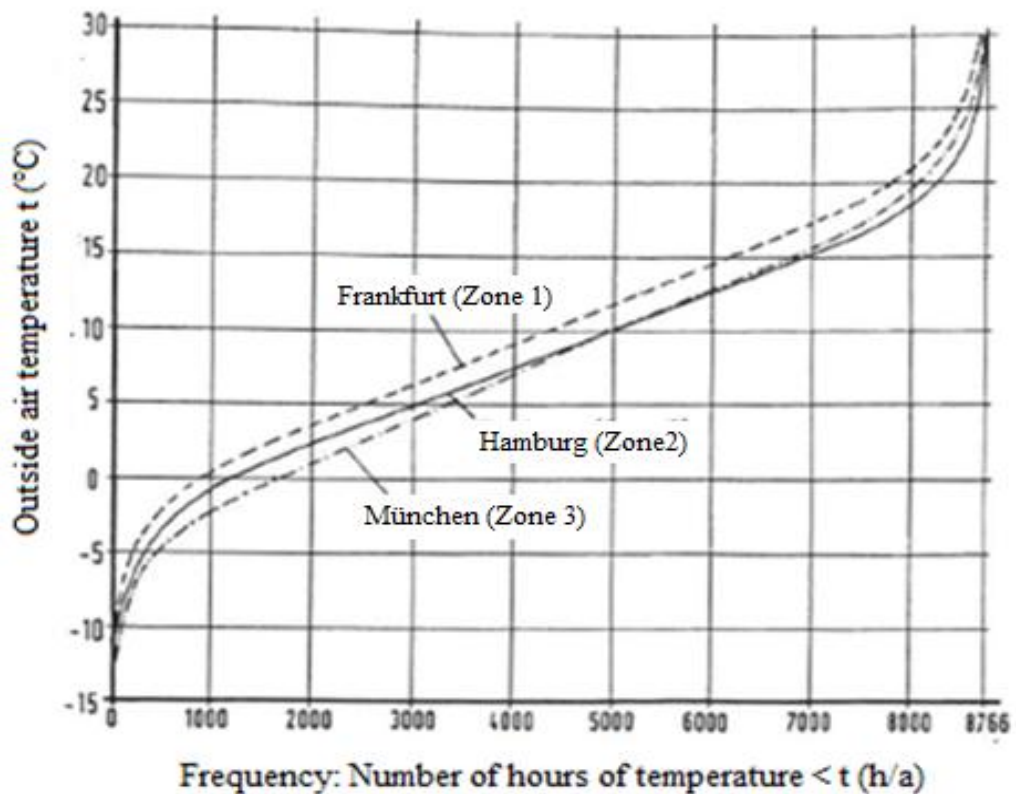
Figure 2.3: Annual duration curves for Munich, Hamburg and Frankfurt [10]

Another factor which may affect the duration curve is deviance of temperature distribution within the city. Specific data for the exact location exists seldom. Therefore, temperature deviation of ±10% may be considered during calculations.

2.4.1.2 Specific air conditioning load

Specific air conditioning load, G_t, defines annually required heating or cooling up/down to a certain temperature on an hourly basis regarding to the temperature

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difference, as well. The unit of specific air conditioning load is $\dfrac{°K \cdot h}{yr}$. It hinges on the operation time of air conditioning and temperature difference. Figure 2.4 demonstrates how specific air conditioning load is calculated. To begin with, annual duration curve of temperature is drawn. Secondly, the horizontal line BC is encountered which refers to the demanded working space temperature. Subsequently, ABC is originated, the area of which indicates specific heating load. However, ABC is converted into A'BC to calculate the specific heating load by means of a geometrically more regular shape, in this case a triangular, through inserting the transversal line A'C, where the areas F_1 and F_3 become equal to F_2. Via conversion of ABC into A'BC, the areas of which are equivalent, specific heating load may be calculated in a simpler way. The area of triangular A'BC thus implies the specific air conditioning load, which may be scaled down to a certain period of operation time by means of conversion factors.
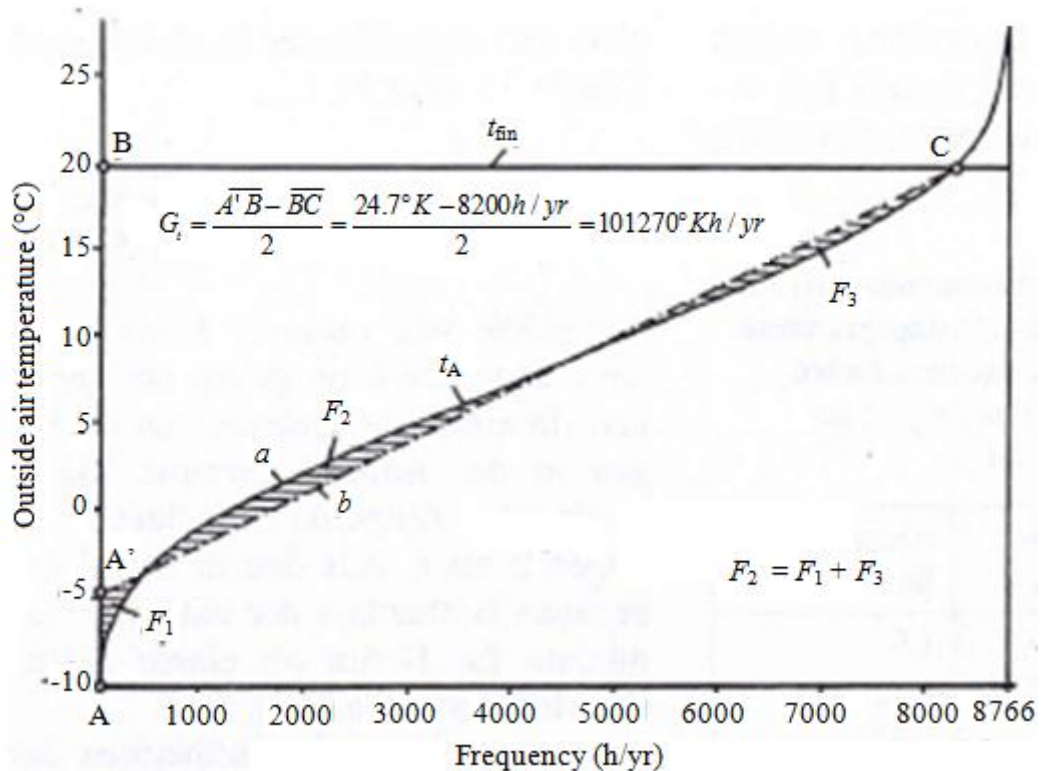
Figure 2.4: Conversion of triangular ABC into A'BC [10]

2.4.2 Supply air humidity

Similar to terms of supply air temperature, humidity is expressed through annual duration curve and specific humidifying load.

2.4.2.1 Annual duration curve of humidity

Specific humidity remains almost stable during daytime. Consequently, no conversion factor is required unlike annual duration curve of outside temperature and specific air conditioning load. The ratio of operating hours to daily amount of specific humidity may directly be applied. Besides, it strongly augments in summer on annual basis. Average annual specific humidity of Stuttgart is measured to be 5.9 g/kg dry air, where in entire Germany it varies from 5.6 to 6.2 g/kg dry air [10]. Unlike specific humidity, relative humidity varies strongly inverse proportional to temperature and achieves its maximum value in winter.

2.4.2.2 Specific humidifying load

Like specific air conditioning load, specific humidifying load, G_K, defines the number of hours in a year humidifying and dehumidifying up/down to a definite value is needed. The unit of specific humidifying load is $\dfrac{g \cdot h}{kg \cdot yr}$. Figure 2.5 illustrates the same and valid method for determining the annual humidification load, which makes use of the area of triangle. Average humidification and dehumidification load during operation may simply be calculated by means of proportioning the operation time to 24 hour load.

$$G_K = \frac{\overline{A'B} - \overline{BC}}{2} = \frac{15g/kg - 6000h/yr}{2} = 45000 \frac{g}{kg} \frac{h}{yr}$$
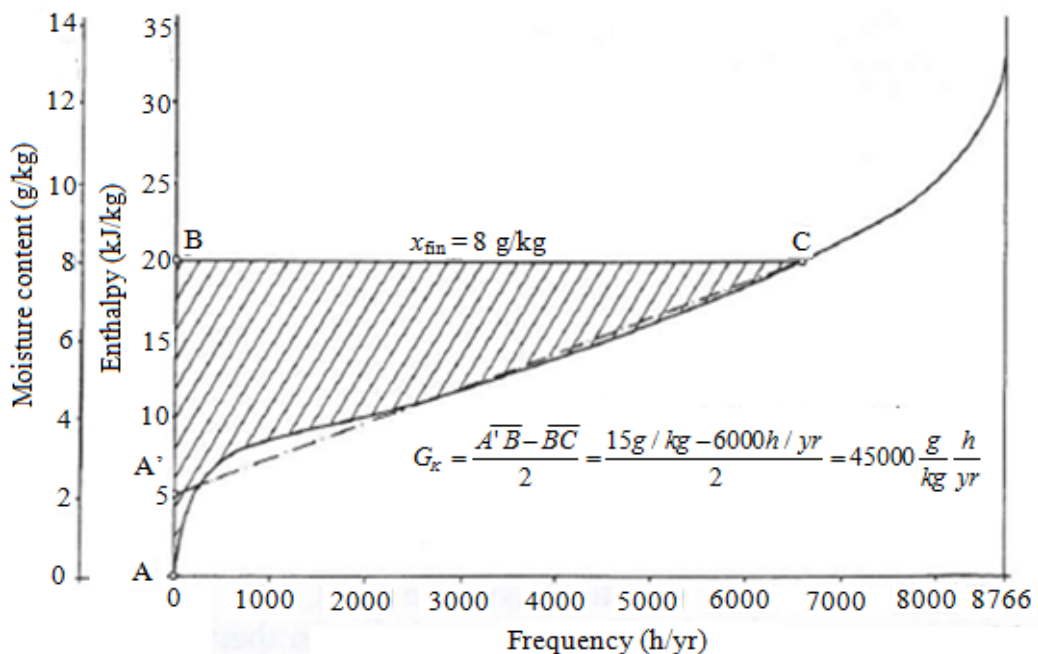
Figure 2.5: Conversion of triangular ABC into A'BC [10]

2.5 Related Parameters

Types of air, official comfort requirements defined by trade control regulations of Baden Württemberg and maximum allowable concentrations introduced by German Research foundation are other parameters related with waste heat recovery processes.

2.5.1 Types of air

Figure 2.6 illustrates main types of air at a recovery process: supply air and exhaust air. They are widely distinguished into further subgroups such as supply air at the entrance or at the exit of the waste heat recovery equipment. The same classification is valid for exhaust air, as well.
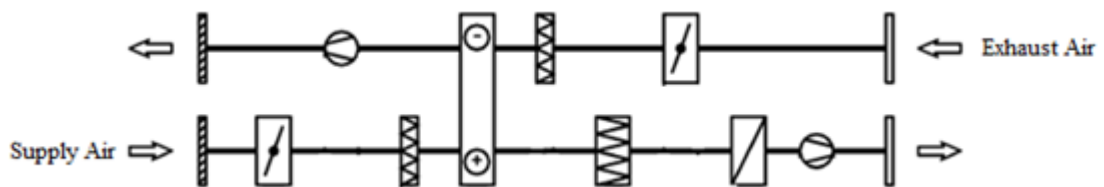
Figure 2.6: Types of air [12]

Supply air may consist of outside air and recirculated air. Outside air is simply from outside transferred air to be fed into the air conditioning or waste heat recovery equipment. Furthermore, in some applications exhaust air at the exit of recovery equipment is partially fed to recovery unit, which is called recirculated air.

After heat is absorbed or desorbed at waste heat recovery equipment it is upgraded to supply air at the air conditioned space which is either directed to air handling unit or directly to air conditioned space. Furthermore, waste heat flowing in the opposite direction of supply air is called exhaust air. Initially it leaves the process, enters the recovery unit and dissipates heat picked up by the supply air. After degradation it is released to free space.

2.5.2 Comfort requirements of working places

According to trade control regulations of Baden-Württemberg some standards are introduced to determine air conditioning restrictions. ASR 6 sets out to identify the boundaries of room temperature. Subsequently ASR 5 is intended for air conditioning.

Regarding to ASR 6 boundary temperatures are encountered based on working conditions. Three situations are classified and described as follows:

- Easy: Working with hands or arms whilst sitting and occasional moving
- Middle: Working with hands, arms or legs whilst sitting or moving
- Hard: Working with hands, arms, legs or whole body whilst standing or moving.

The maximum temperature does not have to access 26 °C. Minimum temperatures are summarized in Table 2.1 with respect to working conditions.

Table 2.1: Required minimum temperatures with respect to working conditions [13]

Predominant working conditions	Difficulty Level		
	Easy	Middle	Hard
Sitting	+20	+19	-
Standing and/or moving	+19	+17	+12

Subject to ASR 5, volumetric flow rates are again identified according to difficulty level of work, respectively:

- 20-40 m^3/h person: Working predominantly whilst sitting.
- 40-60 m^3/h person: Working predominantly whilst not sitting.
- Above 65 m^3/h person: Body work

Apart from volumetric flow rates relative moisture is also defined in ASR 5. Table 2.2 points out values, which do not have to be accessed:

Table 2.2: Maximum relative humidity subject to air temperature [14].

Air Temperature (°C)	Relative Humidity (%)
20	80
22	70
24	62
26	55

2.5.3 Maximum allowable concentrations:

Maximum allowable concentrations (MAK) have been presented by German Research Foundation to indicate the critical values of substances in form of gas, vapor or dust, access of which may cause harmful effects of people in a working environment. Table 2.3 highlights some of these concentrations which are so

calculated that being exposed to those substances for 8 hours per day and 42 hours per week will not lead any harmful effects.

Table 2.3: MAK values of several gases [15]

Harmful Substance	MAK (mg/m^3)
SO_2	1,3
CO	35
CO_2	9500
NO_2	9
N_2O	180
Ozon	0,2

2.6 Quantifying waste heat

Waste heat can be characterized in terms of following properties: (1) Quantity, (2) Quality, (3) Temporal availability [16].

Quantity is described in terms of enthalpy and mass flow as follows:

$$\dot{H} = \dot{m}\, h \qquad \qquad \text{(2.11)}$$

$$\dot{m} = \rho\, Q \qquad \qquad \text{(2.12)}$$

where $\dot{H}$ is total enthalpy flow rate of waste stream, $\dot{m}$ is mass flow rate of waste stream, h is Specific enthalpy of waste stream, ρ is density of material and Q is volumetric flow rate.

Quality is a more determining criterion for the recovery of waste heat. It is expressed in terms of temperature. Three different ranges are classified regarding to temperature at which the heat is dissipated from process equipment. Three different ranges, sources and intended use are roughly summarized in Table 2.4.

Table 2.4: Sources and uses of waste heat [17]

Temperature (°C)	Source	Use

High (650+)	Exhausts from direct fired industrial processes:
	    Cement kiln (dry), 620-730
	    Steal heating furnaces, 930-1040
	    Solid waste incinerators, 650- 980
	    Fume incinerators, 650-1430
		Cogeneration

Medium (230-650)	Exhausts:
	    Steam Boiler, 230-480
	    Gas Turbine, 370-540
	    Reciprocating Engine, 230-590
	Heat treating furnaces, 430-650
	Drying and baking ovens, 230-600
	Annealing furnace cooling systems, 430-650
		Steam Generation

Low (30-230)	Process steam condensate, 55-90
	Cooling water from:
	    Furnace doors, 55-90
	    Bearings, 30-90
	    Welding Machines, 30-90
	    Air Compressors, 25-50
	    Internal combustion engines, 65-120
	Hot processed liquids
	Hot processed solids
		Supplemental Heating

Waste heat resources in high temperature range are mostly originated from direct firing processes and cause additional costs through engineering and equipment, whereas it leads to the widest and most useful options to substitute the overall energy costs. Resources in the medium range may be utilized either via gas turbines depending upon the pressure or via steam turbines regardless of the pressure.

The use of waste heat in low temperature range is more problematic. It is ordinarily not practical to extract work directly from the waste heat source in this temperature range [16]. Preheating of liquids and gases or ventilating air may be some applications of recovering waste heat in this range. The first application requires heat pumps to achieve the essential temperature for the device, where the waste heat will be reused. The second application is set out either to cover or to alleviate the seasonal load of heating.

The final characterizing property of waste heat is temporal availability. Synchronous availability of waste heat sources and loads is a prerequisite. Although quality and quantity of loads and source fit satisfactorily, non-synchrony may still be a problem. Figure 2.7 and 2.8 demonstrate examples of synchrony and non synchrony. According to Figure 2.7, heat source and load match perfectly. However, source and load do not take place in the same period in Figure 2.8. They also do not fit each other in size.
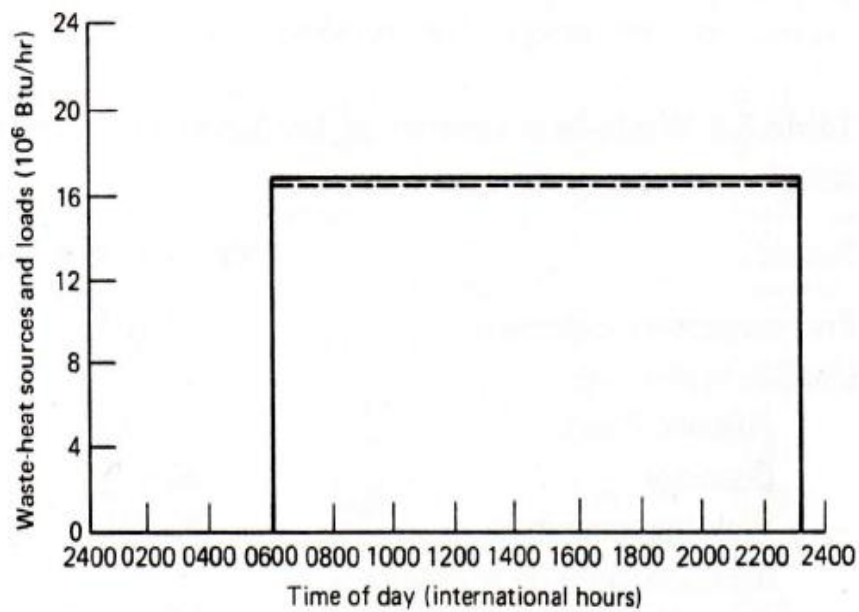
Figure 2.7: Matching waste heat resources and loads: Synchrony [16]
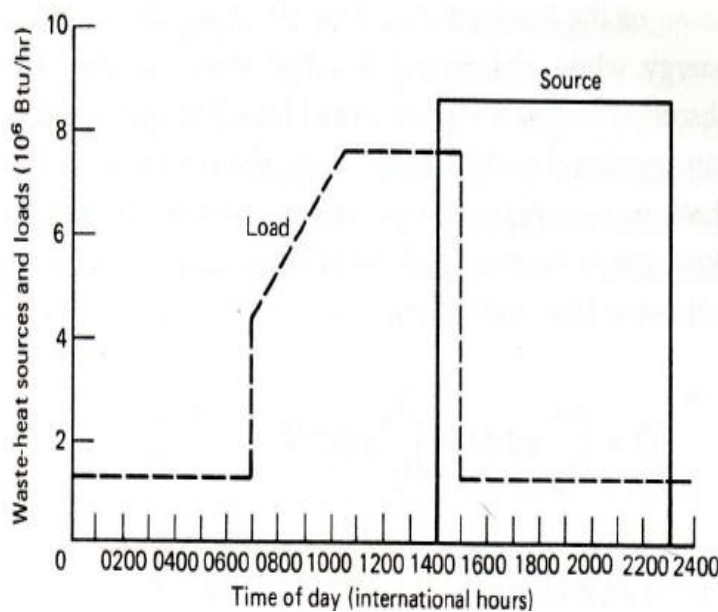
Figure 2.8: Matching waste heat resources and loads: Non-synchrony [16]

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The problem of non-synchrony may be overcome through utilization of thermal energy storage devices. There exist generally three methods of bringing load and source into time correspondence: Sensible, latent and reaction heat storage. By means of sensible heat transfer the surplus of heat is absorbed by water and then extracted when necessary. Nevertheless, the efficiency is especially low at high temperatures. Another way may be the storage in form of latent heat, which means a phase transition e.g. from solid to liquid and liquid to solid respectively. The final method requires running a reversible chemical reaction. The highest energy storage density may be attained via this option. Storage up to several months is possible.

3. COMMERCIAL OPTIONS IN WASTE HEAT RECOVERY EQUIPMENT

In this section some waste heat recovery systems; in particular gas-to-gas heat recovery systems which can be utilized in low temperature range are summarized. Even though, a categorization based on technique is also possible as recuperators and regenerators from a point of view, commercial equipments are classified and described separately in this work.

Figure 3.1 demonstrates how recuperators and regenerators function in a comparative way. Recuperators recover heat by separating continuous and steady flows through a separating wall, which result in some advantages such as stabilization of pressure difference, increase in exchange area, certain separation of flue gas and air and thus prevention of cross contamination. Moreover, they are relatively inexpensive equipments. On the other hand, the necessity of installing additional units at high temperatures may augment the capital costs. Despite some applications in low temperature range recuperators are broadly employed for recovering waste heat in medium and high temperature range.
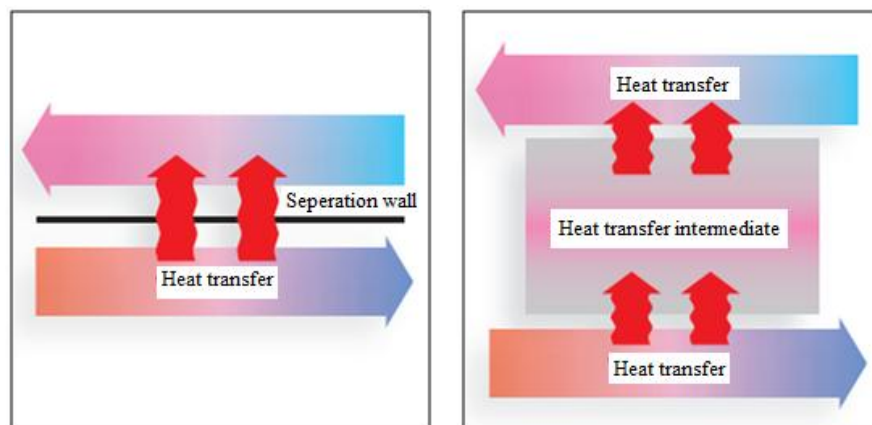
Figure 3.1: Recuperator and regenerator [18]

Regenerators function in such a way that, heat is charged to and discharged from the heat transfer intermediate periodically and cyclically as the direction of exhaust and supply streams are changed. In contrast to steady flow in recuperators there exists an

intermittent flow in regenerators. Heat pipes and run around coil illustrate examples of regenerators.

3.1 Convection (tubular) recuperators

Convection (tubular) recuperator is an option of waste heat recovery, which is commonly employed under special circumstances. Sort of fouling, facility size and temperature interval play a major role.

3.1.1 General information

Convection recuperators consist of sealed tubes, through which one stream is fed into the tubes while the other passes vertically among the tubes as illustrated in Figure 3.2.
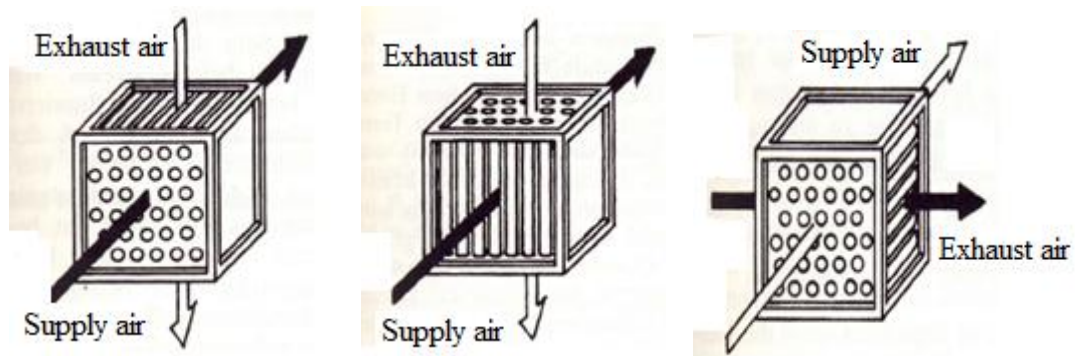
Figure 3.2: Convection recuperators with various feed options [19]

The feed-in option differs subject to condensation and pollutant content of the stream. By way of illustration, polluted streams are fed into the tubes. On the whole, cross contamination is prevented. In particular glass tube recuperators are preferred in low temperature range.

3.1.2 Exchange coefficients

Number of rows, wall thickness and inner diameter of tubes affect exchange coefficients. Increasing the pressure drop and velocity and alleviating inner diameter yield an optimization. Figure 3.3 illustrates the relation between those parameters and recovered heat coefficient. As the velocity of the stream in the tube goes down and length/inner diameter (L_R/d_i) ratio increases, higher recovered heat coefficients may be achieved.
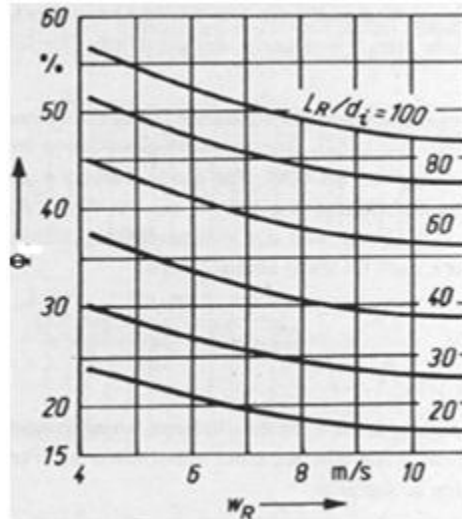
Figure 3.3: Φ as a function of the ratio L_R/d_i and W_R [20]

3.1.3 Installation and operation

Convection recuperators are utilized against fouling and corrosion. Otherwise plate heat exchanger is a more cost effective option [10]. In addition, they are easily cleaned. Prevention of cross contamination makes it also possible to recover heat from polluted streams. Convection recuperators may be operated up to 250 °C [20]. Apart from general advantages of convection recuperators, glass tube recuperators are mostly applied against ice deposition and existence of condensate. It is possible to run glass tube recuperators up to inlet temperature 250°C [10]. In most cases these units are designed to recover from 60 to 70 per cent of the heat in the exhaust gas stream [21].

3.2 Heat wheels

Rotating regenerators, rotary air preheater, rotary heat exchanger are some other names which are also used in literature instead of heat wheel. Application of heat wheels is almost possible in all temperature ranges depending upon the material. The most remarkable difference of heat wheels is that it contains a moving part.

3.2.1 General Information

Heat wheels are utilized to transfer sensible and latent heat for both heating and cooling applications [20]. As illustrated in Figure 3.4, hot and cold air streams are led to two side-by-side ducts located in heat wheel, which absorbs and desorbs waste heat continuously. While the wheel is rotating, heat is transferred from hot exhaust

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gas to cold supply gas, which may be a mixture of outdoor and circulated air. In this case, a supply-air fan must be implemented to keep the supply air constant. Predominantly, electric motors with an operating energy input from 0.1 to 1 kW drive heat wheels which rotates with a number of turns from 10 to 20 min-1 [20].
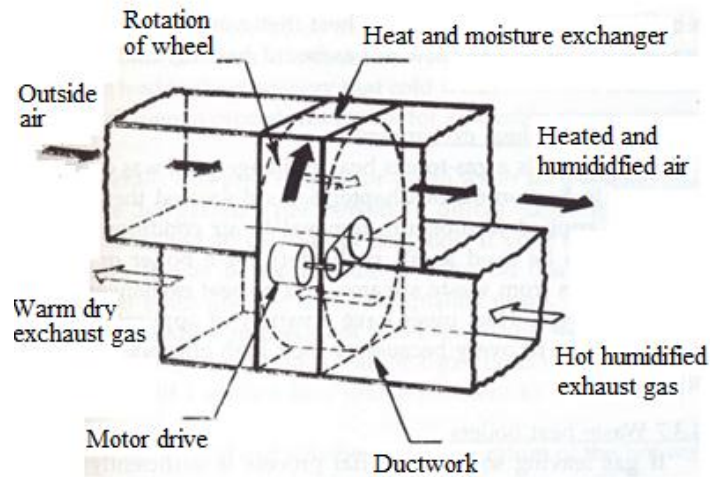
Figure 3.4: Simple heat wheel [17]

By means of heat wheels not only sensible but also latent heat can be transferred. Figure 3.5 highlights additionally a dehumidifying heat wheel intended for moisture recovery. In spite of higher costs up to 25%, hygroscopic wheels also make use of the latent heat which contains almost the same amount of energy with sensible heat [21]. Via moisture transfer recovered energy may be enhanced by 20 to 40% in comparison with a heat wheel without moisture recovery. Regarding other heat recovery equipments the additional recovery may yield from 30 to 50% [10]. A control unit is implemented to regulate the rotor speed and thus the efficiency.
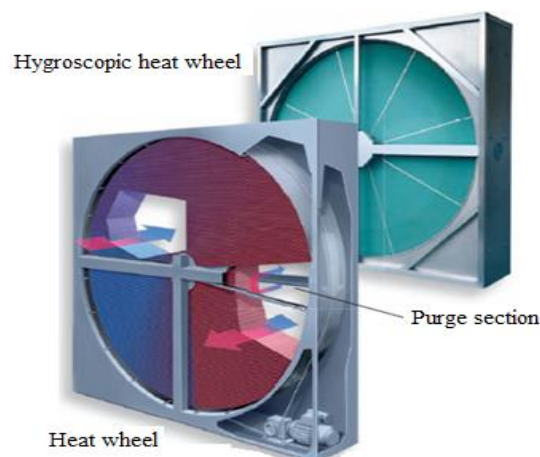
Figure 3.5: Different types of heat wheels [18]

3.2.2 Exchange coefficients

The deterministic way of thinking during regenerator selection follows this chain: Low velocity → low volumetric flow rate → large equipment → little pressure drop → high efficiency. A decision between a large, expensive unit with high exchange coefficients and small, cheap with low exchange coefficients should be made subject to application [20].

Construction is the most significant criteria affecting the exchange coefficients. The main objective is to augment area per volume. Mostly sinusoidal waves are preferred. Sort of the material is of little importance in comparison with micro geometry. Other factors are listed as operating temperature and pressure, equal/unequal flow rates of cold and hot streams and pressure drop. Moisture transfer coefficient differs subject to recovery with/without condensation. Under all circumstances, correction factors may be applied to determine exchange coefficients.

3.2.3 Installation and operation

Heat wheels are commonly implemented in HVAC systems. Precooling and preheating are two applications. Operation in low to moderately high range is common [16]. In particular, when heat exchange takes place between large masses of cold and hot air streams at slightly different temperatures and with similar flow rates. In addition, gas streams with similar pressures are mostly preferred. Owing to prevent cross contamination a slightly higher pressure of exhaust gas is required. Besides, a purge section can be installed to prevent cross contamination, example of which is shown in Figure 3.5. Under these circumstances, higher efficiency is attained in comparison with the heat pipe heat exchangers.

Diameter of a heat wheel varies from 0.6 to 5 m subject to volumetric flow rates from 1000 m^3/h to 150000 m^3/h. However, they are cost effective from 15000 m^3/h to 100000 m^3/h [10]. Pressure drop ranges from 100 Pa to 180 Pa, which are relatively small values comparing with other heat exchangers.

3.3 Plate heat exchangers

Plate heat exchanger does not only cater for gas-to-gas waste heat recovery but also liquid-to-liquid. It is a popular choice due to its simplicity.

3.3.1 General Information

Plate heat exchangers are simple recuperative heat exchangers and commonly offered in form of modules. Figure 3.6 demonstrates two sorts of plate exchanger. Independent upon different sorts, both streams pass through adjacent layers separated by thin plates made of metal or glass.

Crossflow plate heat exchanger   Counterflow plate heat exchanger
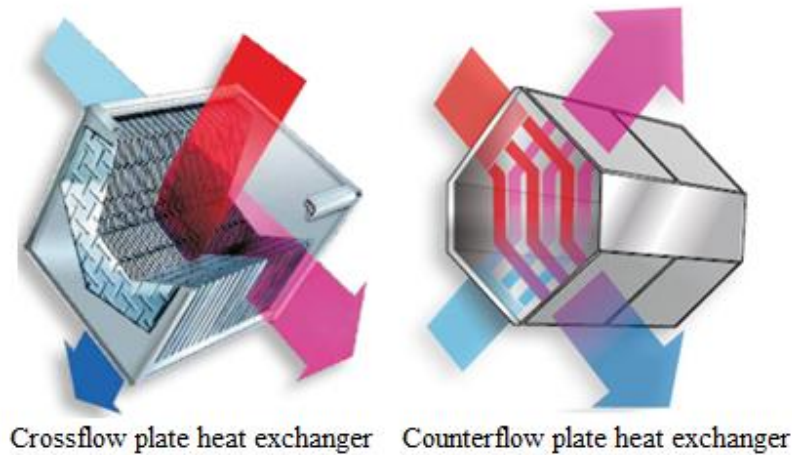
Figure 3.6: Different types of plate heat exchanger [18]

3.3.2 Exchange Coefficients

Heat transfer coefficient of a plate heat exchanger strongly depends upon the thermal conductivity coefficients, which are a function of plate length and the velocity of the stream between plates [20]. On the other hand, thermal conductivity coefficient, in other words type of the material affects the heat transfer less than other parameters. Since heat transfer takes place by means of conduction, the area by which the streams are surrounded also plays a key role. Enhancing the area results in increase of heat transfer. Moreover, dependence of efficiency upon mass flow ratio are to mention. Efficiency decreases with increasing mass flow ratio of outside air to exhaust air, $\dot{m}_A/\dot{m}_F$, since duration of heat recovery decreases [10].

Other factors affecting recovered heat coefficient are condensation and pressure drop which show a proportional tendency subject to geometrical ratio length or width relative to plate thickness, L/b (or a/b), and velocity of streams between the plates, Wsp. Figure 3.7 shows the trend of recovered heat coefficient with respect to L/b ratio and Wsp. Assuming the velocity between the plates from 4 to 8 m/s, L/b>300 is

required in order to achieve the recovered heat coefficient $\Phi > 50\%$. Furthermore, only low velocity between plates and L/b>800 may lead to $\Phi > 70\%$ [20].
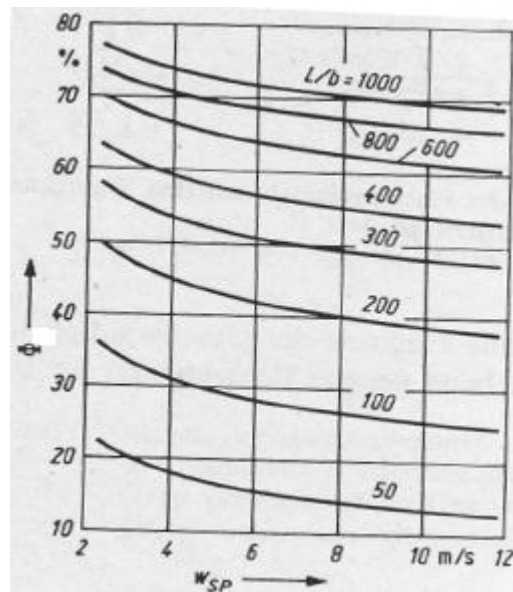
Figure 3.7: Recovered heat coefficient versus L/b and Wsp [20]

However, an increase in efficiency contributes to price strongly, where plate heat exchangers are capable of reaching the efficiency of 60% quite easily [10]. To sum up, high L/b ratio is necessary to enhance recovered heat coefficient, on the other hand due to an increase in pressure drop electrical energy load of the ventilators goes up. Final factor affecting the efficiency is condensation, which may occur if exhaust stream rejects high amount of heat, so its temperature sinks below dew point. Since, in addition to sensible heat also latent heat is recovered; recovered heat coefficient is increased by 20 %.

3.3.3 Installation and operation

Supply and exhaust air may be fed to plate heat exchanger in a standard, counter, diverted flow mode or crosswise. Figure 3.8 illustrates various types of utilization and operating options of modules.
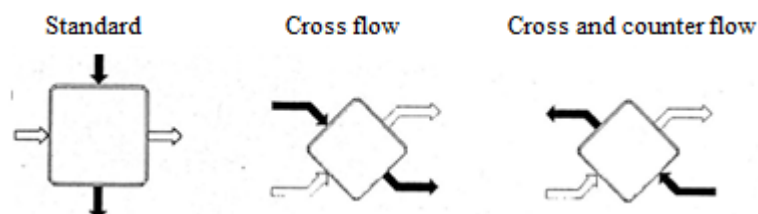
Figure 3.8: Various feeding options into a plate heat exchanger [22]

Mostly exhaust air is fed to a plate heat exchanger from top so that the condensate and cleaning chemicals may flow easily and also faster. Moreover, plate heat exchanger may be installed one after the other. Under these circumstances heat transfer area and pressure drop double. Apart from increase of the recovered heat coefficient, a cost effective solution would be installation of two diagonal flow plate heat exchangers, since the counter flow effect increases in that way [10]. Enhancement of recovered heat coefficient through this kind of an installation is illustrated in Figure 3.9.
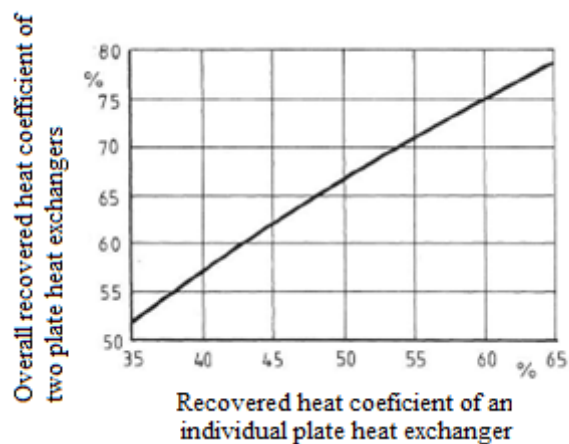
Figure 3.9: Recovered heat coefficients of two successive diagonal PHEs [10]

Plate heat exchangers can be employed up to 60000 m^3/h air [20]. Utilization with running streams of flow rates up to 20000 m^3/h would be the most cost effective option [10].

Waste heat recovery by means of plate heat exchangers are intended to HVAC systems and also industrial applications subject to temperature range. Installation of plate heat exchangers requires small space. Especially during industrial applications some precautions against corrosion such as corrosion resistant coatings or applying glass plates may be required. Other difficulties may exist in cleaning and temperature control, which may cause fouling.

3.4 Heat pipe heat exchangers

Heat pipe heat exchanger is another option which caters for process and space heating, also for seasonal preheating and precooling at HVAC applications.

3.4.1 General information

Sealed and evacuated container, working fluid and a wick lining are the basic components of a heat pipe. Figure 3.10 enlightens heat pipe operation cycle, during which hot exhaust gas evaporates the working fluid within the wick lining, which covers the interior surface. Then, the working fluid transports from one end to another, which refer to evaporation and condensation sections. Driving force is the enhancement of the pressure between sections. Both sensible and latent heats are recovered in the condensation section.
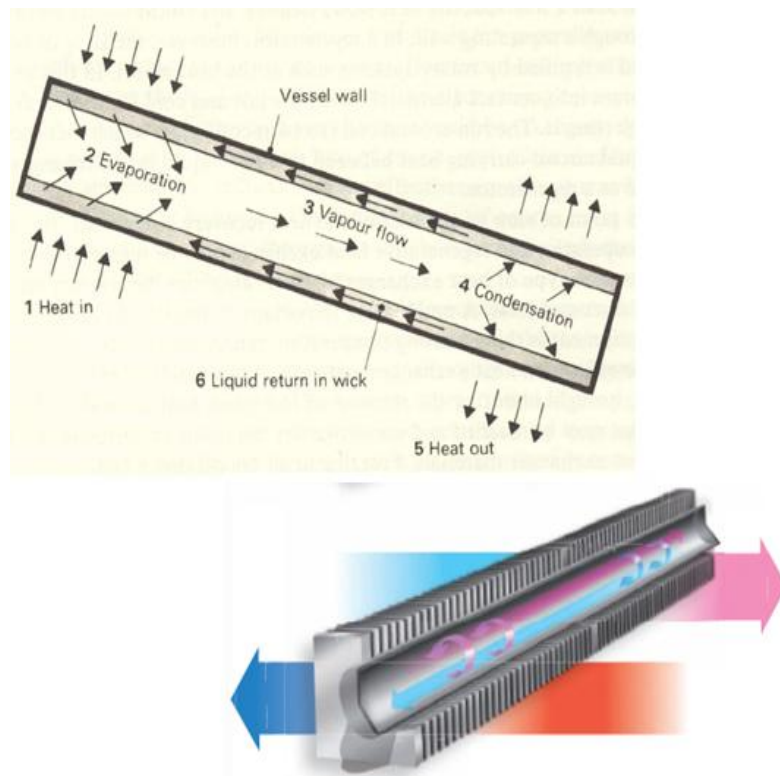
Figure 3.10: Heat pipe operation [18, 21]

Condensate is recycled in the wick through capillary action or gravitational force. Heat transfer may be controlled through angle inclination.

3.4.2 Exchange coefficients

Number of pipe rows, distance between adjacent rows, velocity of supply air, flow rate ratio of supply and exhaust streams are some variables determining the exchange coefficients and also pressure drop of a heat pipe. By way of illustration, installing 8 rows of pipes with distance of 2.3 mm and running the supply air stream with the velocity of 2.5 m/s result in efficiency from 0.6 to 0.66 and the pressure drop of 120

Pa. However, regardless of economical evaluation 90% efficiency is possible [10]. Figure 3.11 shows dependence of recovered heat coefficient upon different variables. Decreasing the distance between rows and increasing the volumetric flow rate ratio enhances the efficiency. Moreover, decreasing the velocity and augmenting the mass flow ratio in favor of exhaust air concludes to higher recovered heat coefficient.
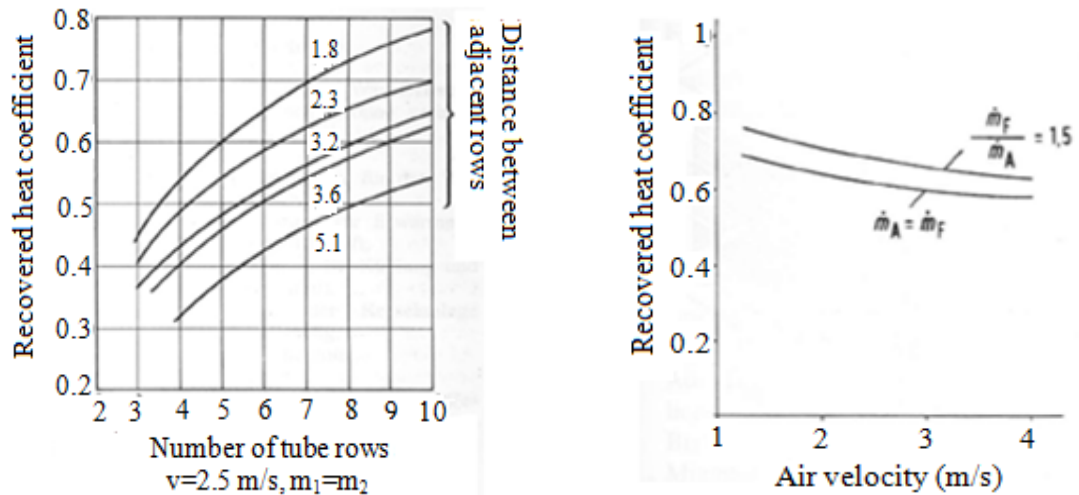
Figure 3.11: Dependence of recovered heat coefficient upon different variables [10]

Number of rows varies typically from 4 to 10. Supply stream flows optimally with a velocity from 2 to 2.5 m/s. Length of pipes are typically 1-2 m but may be extended up to 5 m [21]. Like plate heat exchangers, they may be less efficient than heat wheels under same circumstances.

3.4.3 Installation and operation

Heat pipe heat exchangers are capable of operating streams of volumetric flow rate up to 20000 m^3/h. In particular streams with temperatures from -30°C to 250°C are easier to handle [10].

Heat pipes are small in size and compact. Also cross contamination is not possible. Another merit is the lack of moving components which leads to an increase of purging costs at the same time. In addition, rerouting the duct work enhances the installation costs.

3.5 Run-around coil

Run around coil is another low temperature waste heat recovery device which permits to bring sources and loads at remote locations.

3.5.1 General information

Run around coil consists of two coils and a liquid intermediate heat transfer medium within an interconnecting pipework as shown in Figure 3.12. In summer heat is extracted from hot outdoor air by the cold medium in the heating coil. Subsequently, heat of circulating medium is recovered in the recovery coil. Thus, the temperature falls. Finally the medium is pumped back to heating coil. During heating season circulation runs in the opposite way. In Figure 3.12 coils are named according to the functions related to the medium. Due to its low freezing point, water-glycol mixture is a common intermediate medium.
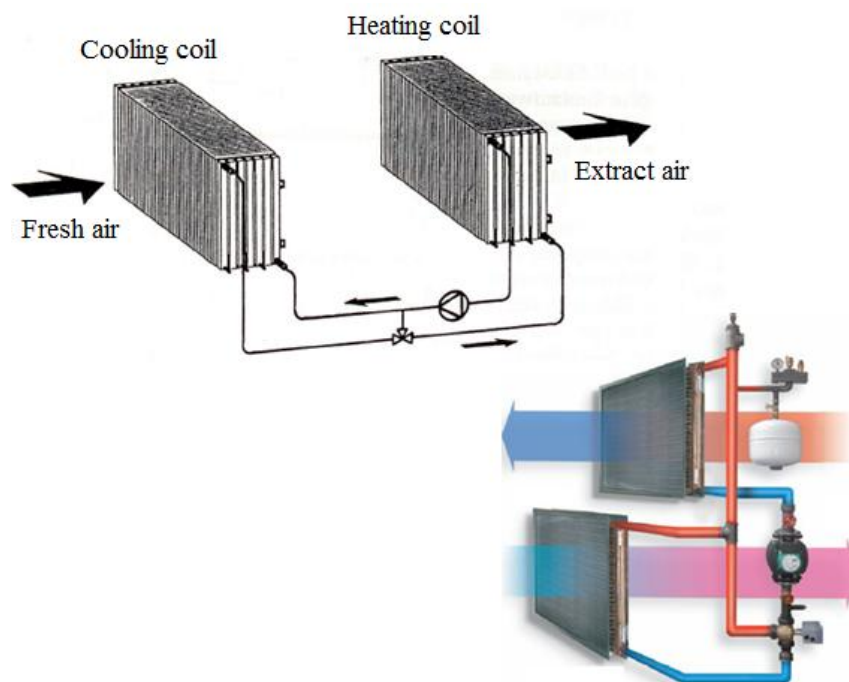
Figure 3.12: Run around coil [10, 18]

3.5.2 Exchange coefficients

Even though commonly sensible heat is recovered by run around coils, also latent heat recovery is possible in case of a condensation. The velocity of the supply stream may vary from 2 to 4.5 m/s [20]. Nevertheless, 2.5 m/s is an optimum value [10]. Furthermore, the system contains rows of finned tubes composing the coils. Number

of rows made up of tubes differs in general from 8 to 12, where 10 is optimum [10]. Apart from finned tube heat exchangers, ribbed type heat exchanger may also be used to pick up and dissipate heat.
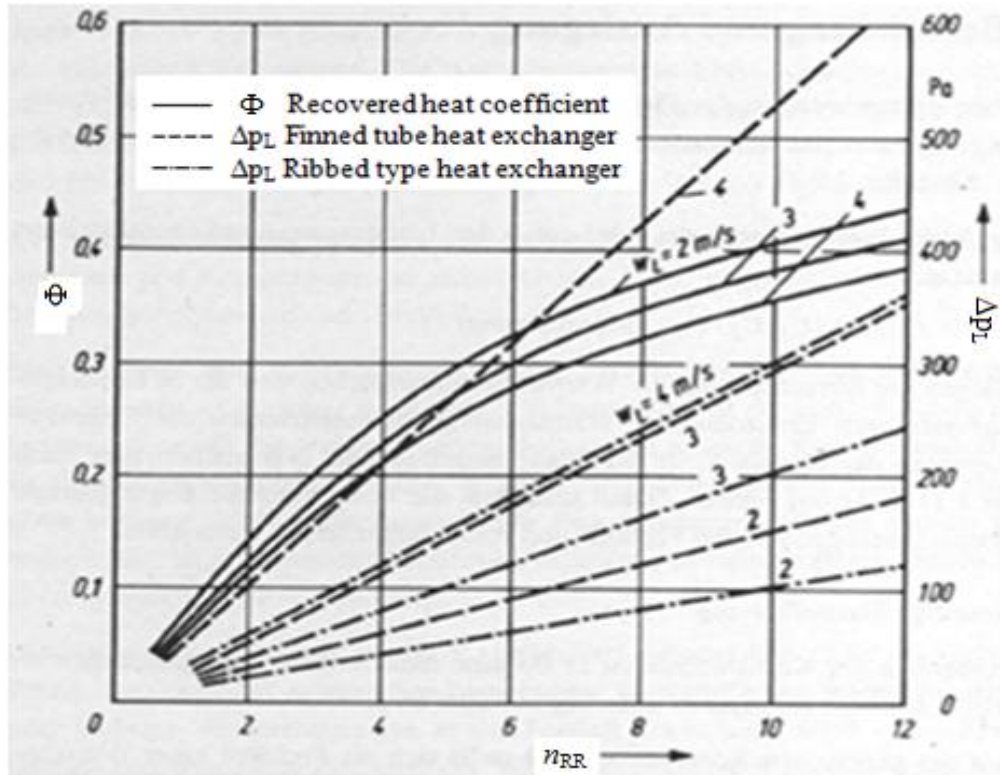
Figure 3.13: Dependence of ΔpL and ϕ upon n_{RR} and w_L [20]

Figure 3.14 illustrates the dependence of air side pressure drop Δp_L and overall recovered heat coefficient upon number of rows n_{RR} and velocity of air w_L. Increasing the number of rows enhances pressure drop and recovered heat coefficient separately for both sorts of heat exchangers. On the other hand, taking both parameters at the same time into consideration may yield a requirement of optimization.
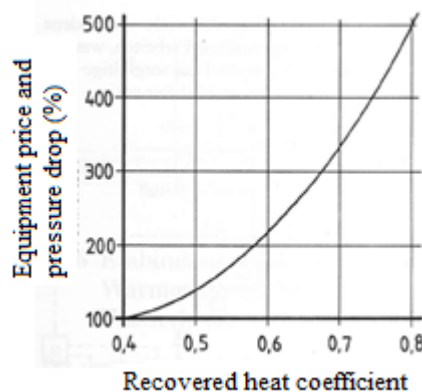
Figure 3.14: Relation between price and pressure drop with Φ [10]

Exchange coefficients from 0.4 to 0.6 can be easily attained. For instance, run around coils with the number of rows 4, 6, 8, 12 leads to efficiencies of 0.4, 0.45, 0.52, 0.62 respectively. In recent years run around coils with 0.7 to 0.85 efficiencies are stated [10]. On the other hand, price and pressure drop go up more rapidly as the recovered heat coefficient increases.

3.5.3 Installation and operation

Like most of the gas-to-gas recovery systems, run around coils are commonly intended to HVAC applications and low temperature process heat recovery.

Run around coils are mostly preferred when supply and exhaust air streams are not located adjacent to each other. Also, coils located in different places can make use of various streams at the same time. Rerouting of the duct work is not required and interconnecting pipework does not occupy much space. On the other hand, run around coils are less cost effective, when overcoming the long distance problem is not necessary. Another merit of run around coil is prevention of cross contamination. Furthermore, except pump the unit obtains no other moving part.

3.6 Heat pumps

Heat pump is utilized in common to upgrade waste heat resources, which makes it less suitable for low temperature waste heat recovery.

3.6.1 General information

Heat pump is a device which runs cyclically making use of a low temperature stream by applying external work, which means an exergy increase. On the contrary, heat pumps offer only limited opportunities for waste heat recovery simply because the cost of owning and operating the heat pump may exceed the value of the waste heat recovered [16].

Figure 3.15 demonstrates the cycle at a heat pump. It consists of an evaporator and a condenser, both of which contain own heat transfer medium to change the phase of the exhaust stream. Cooling medium runs in a closed loop. In winter supply air is fed to the condenser. The initial step may be assumed as the exit of vapor with low temperature from evaporator. Subsequently, via external work which may be provided by a compressor or a turbine the temperature of the evaporated refrigerant

augments. Also its pressure attains condenser pressure. In the next step, the stream enters the condenser with the supply air and the surplus of heat is transferred to supply air. At the same time exhaust stream is liquefied. In the next step, liquefied exhaust stream is fed to evaporator and picks up heat from the heat transfer medium. After being evaporated exhaust stream recycles the process.
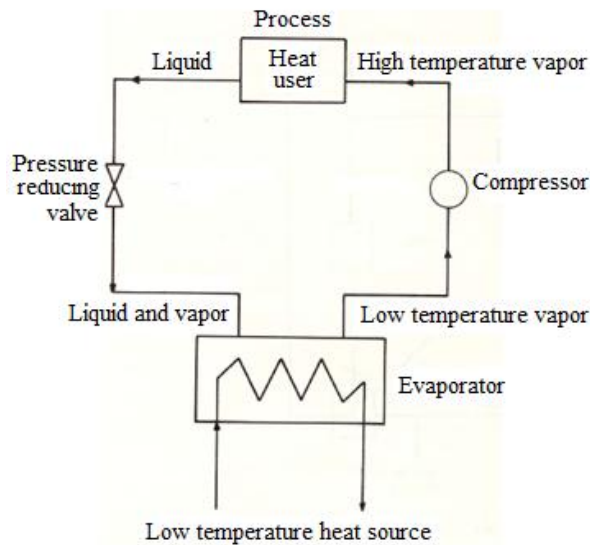
Figure 3.15: Operation of a heat pump [23]

Temperature range of heat pumps differs subject to cooling medium Recent restrictions about cooling mediums lead to a range between 60 °C and 100 °C. In terms of process application, temperature restrictions of the additional equipments belonging to the cycle should be regarded, as well. Heating power of the heat pumps varies from 1 kW to 30-40 MW depending upon the aim.

3.6.2 Coefficient of performance

The efficiency of heat pumps is identified by coefficient of performance (COP). COP is the ratio of heat supplied at the condenser by the external work. For instance COP=2 indicates that 1 kW external work leads to 2 kW heat production.

$$COP = \frac{Tc}{Tc - Te}$$

(3.1)

COP: Coefficient of performance
T_c: Condensing temperature
T_e: Evaporating temperature

However, regarding the efficiencies of additional equipment such as turbine or compressor, the overall efficiency decreases, which is about 60% at large modulations with power above 1 MW. Smaller units e.g. 3kW achieves the efficiency around 50% [10]. To sum up, practically 50 to 65% of the theoretical COP may be attained [16]. Overall efficiency goes up with alleviating temperature difference.

On the whole, it is not recommended to prefer heat pumps, unless COP is considerably greater than 3, since the cost of owning and operating the heat pump may exceed the value of the waste heat recovered [16].

3.6.3 Installation and operation

Heat pumps lead to high investment costs in comparison with regenerators and recuperators. They also require electricity. Besides, combination of heat pumps with recuperators and regenerators may be a cost effective option. On the contrary, such a mentality may be unnecessary, if the efficiency of the unit exceeds 80%. Installing heat pumps after recuperators aims to extract rest waste heat from exhaust air. Combination with regenerators seeks to make use of the heat pump to transfer heat to cold zones of the building, in particular after heat was recovered by a heat wheel. Equally important is that between two stages the temperature of the exhaust air must be increased at least 10°C for an optimum use of the modulation.

3.7 Capillary blower

In addition to heat wheel, capillary blower is the second regenerator intended for low temperature waste heat recovery.

3.7.1 General information

Capillary blower consists of a ring made of foam or steel depending on the temperature. As shown in Figure 3.16, a rotating ring extracts heat from exhaust air, which enters the unit from a baffle located across the middle of the hole. Afterwards, recovered heat and in some applications moisture is transferred to cool inlet air by the ring. Another function of the ring is filtering, when needed. Consequently, exhaust air leaves the unit. Even though it would have a restricted efficiency, cooling may also be provided.
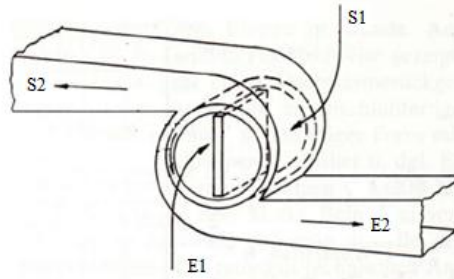
Figure 3.16: Operation of a capillary blower [10]

3.7.2 Exchange coefficients

Both heat and moisture may be recovered by capillary blower type regenerators. Moisture recovery occurs in a similar way like rotating heat exchangers. Accessing the dew point leads to existence of water drops on the ring and transferred to supply air. Under these circumstances, efficiency of capillary blower type yields up to 40-50%, including the applications where moisture is also picked up.

3.7.3 Installation and operation

Capillary blowers are implemented for HVAC applications in commercial, industrial buildings and houses subject to process configuration and modulation.

Capillary blower type regenerators rotate very fast in comparison with heat wheels. In fact, velocity varies from 15 to 34 m/s which refer to 700 to 3000 spins per minute [20]. On the contrary, heat wheels rotate with velocities from 1 to 3 m/s and from 7 to 15 spins per minute are achieved [10]. Therefore, exposition time on the ring and exchange coefficients decrease. Volumetric flow rates vary from 0.02 to 14 m^3/s, where capacity may be augmented via parallel installation. Configuration containing up to four regenerators is possible [20].

3.8 Switch-over heat exchanger

Switch-over heat exchanger is the final regenerator utilized for low temperature waste heat recovery.

3.8.1 General Information

Figure 3.17 illustrates the operation of a switch-over heat exchanger. Two separate stationary storage units e.g. aluminum plates admit periodically and respectively fed

cold and hot streams. In other words, heat transfer occurs via charge and discharge streams. Switch-over of the streams are regulated by a electrically driven damper. Two ventilators complete the equipment to feed the streams. A period including charge and discharge longs commonly 80 s [10]. However, this duration can vary from 40 s to 240 s [20].
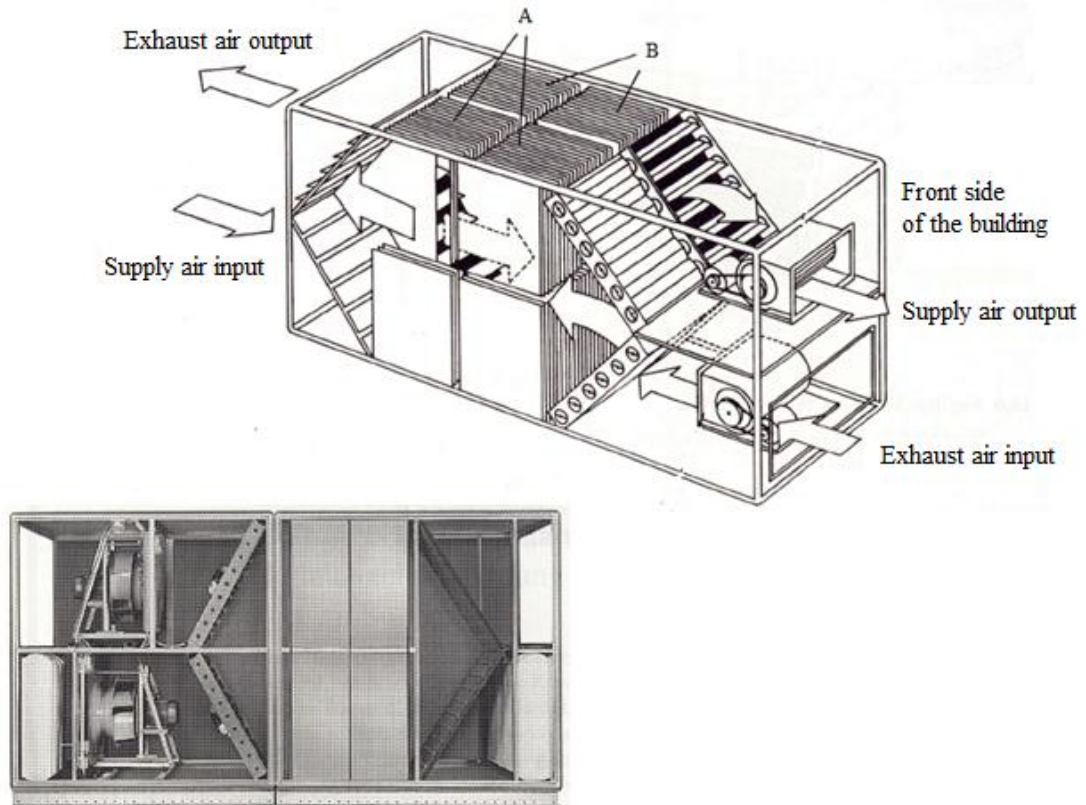
Figure 3.17: Operation of a switch over heat exchanger [10]

During charge, exhaust air heats the storage unit A as the supply air enters the preheated storage B. During discharge, hot air is directed to storage unit B by the damper, where its heat is completely picked up. At the same time, supply air is led to storage unit A, which was warmed up during charge.

3.8.2 Exchange coefficients

Under normal circumstances only sensible heat is recovered by switch-over heat exchangers. In latter cases heat transfer medium is covered with an absorbing medium to recover moisture. Hence, temperature recovery coefficient may yield up to 0.9 and moisture recovery coefficient ranges from 0.5 to 0.8 [10].

3.8.3 Installation and operation

Switch-over heat exchangers are generally installed for HVAC applications. They are easily cleaned and mostly applied, if criterion about contamination is not strict.

Despite high exchange coefficients, switch-over heat exchangers occupy much space. For instance, in order to extract heat from a stream of 2.1 m^3/s the storage volume of 0.5m×1.5m×2.5m is required [20]. In general, volumetric flow rate of fed streams range from 600 m^3/h to 39000 m^3/h [10]. Moreover, temperature varies from -25°C to 65°C [20].

3.9 Counterflow-layer heat exchanger

Counterflow-layer heat exchanger is based on a similar principle with run around coil as shown in Figure 3.18. It consists of two modulations with a number of layers, which are made up of pipes in which water is circulating. Regarding to water temperature, which may also be supplied from a separate source heating or cooling load and thus equipment size may be decreased.
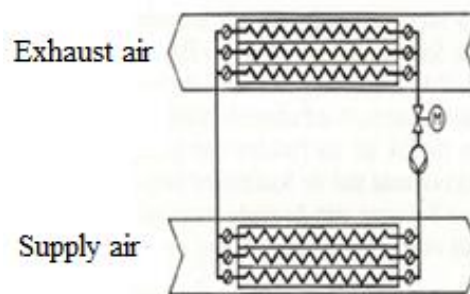
Figure 3.18: Operation of a counterflow-layer heat exchanger [10]

Generally, layers are built on each other in one modulation and made of copper layers. As the counter flow streams flow through the pipes heat is step by step charged, transferred and discharged in the complete system.

By means of counterflow-layer heat exchanger recovered heat coefficient of 0.8 may be achieved. Volumetric flow rates from 10000 to 100000 m^3/h may be employed.

4. CALCULATION OF ANNUAL HEAT RECOVERY

In this section, recoverable amount of waste heat will be calculated via graphical and single frequency method, which is the fundamental of economic analysis.

4.1 General information

Regarding to the scope of this work, economical analysis intended for low temperature waste heat recovery opportunities for two facilities of Daimler AG will be prepared.

Table 4.1 summarizes all data belonging to two facilities. In both facilities waste heat is dissipated from production at around similar temperatures. The most significant difference is that volumetric flow rate in Facility 2 is variable and relatively small scaled in comparison with Facility 1.

Process waste heat is assumed to be fresh air due to actual presence of purification equipment. This assumption may especially be considered if application of a regenerator is selected, since despite low rate mixing of supply and exhaust air take place in regenerators. Furthermore, both applications require no moisture recovery. Ambient air is fed in recovery equipments.

Table 4.1: Received process data

	Facility 1	Facility 2
Process waste heat dissipated from	Motor production: Dust exposure	Spiral conical gear production: Oil mist suction
Average temperature of process waste heat (t_{11}) (°C)	28.6	28.7
Average Velocity (m/s)	14	5
Average Volumetric flow rate (m^3/h)	144181	6072-10120
Air conditioned space temperature (t_{fin}) (°C)	19	19

At both applications exhaust air is process waste heat and supply air is outside air. Based on the theoretical section, low temperature waste heat is recovered intended for air conditioning applications. The purpose is to pick up energy from process waste heat at 28.6 °C and 28.7 °C and subsequently to augment the temperature of outside air up to 19 °C, which is the prerequisite to direct the air stream to air conditioned space.

To determine the savings in the light of these data, two different methods will be introduced: Graphical and single frequency method. Both methods will be explained in detail whilst calculating the concrete savings. Graphical method is used to compare and confirm some theoretical recovered energy data only for several recovered heat coefficients. This comparison will be figured out at particular examples in the next section. However, single frequency method is the main method to calculate theoretical recovered energy.

4.2 Graphical method

Graphical method is significant owing to its visual quality. The main purpose is to draw the curves of ambient temperature and also boundary temperature, which may be reached via heat recovery equipment. Such a graph demonstrates the data in form of temperature versus frequency on an hourly basis. Supply air temperature leaving the waste heat recovery equipment is derived from the definition of recovered heat coefficient.

Data to draw annual temperature curve of ambient air is taken from DIN 4710: Statistics on German meteorological data for calculating the energy requirement for heating and air conditioning equipment, which may be found in Appendix A1 [24]. At DIN 4710, each degree implies the interval up to next temperature. For instance, 15 °C represents the interval from 15.0 °C to 15.9 °C, which is observed with the frequency of 6525.6 h/yr. There exist also two data belonging to 0°C. One of them is positive and the other one is negative. By way of illustration, negative 0 °C covers the range between 0 °C and -0.9 °C. Since two facilities are located in Stuttgart, the data expressed in Appendix A.1 are valid for both applications.

4.2.1 Facility 1

Figure 4.1 demonstrates how initial steps of graphical method for Facility 1 are proceeded. First and foremost, curve of supply air temperature at the entrance t_{21} is drawn based on DIN 4710. Subsequently the lines of exhaust air temperature at the entrance and working space temperature are encountered.
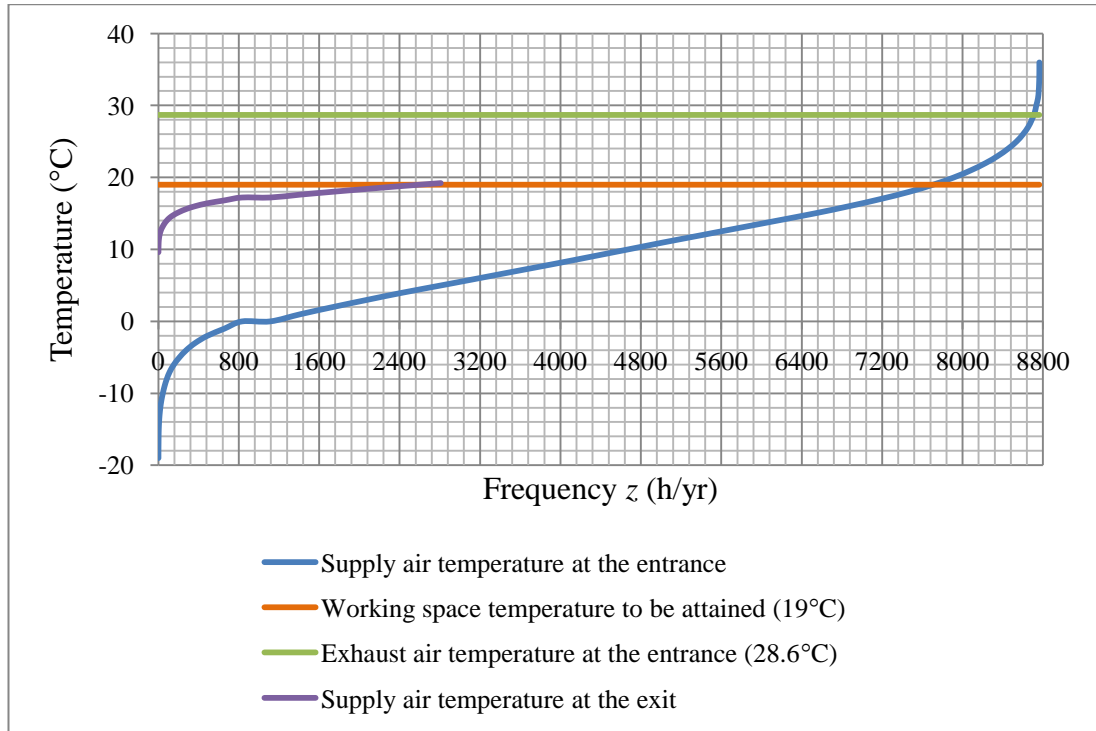
Figure 4.1: Application of graphical method for Facility 1

The second step is to calculate the boundary temperatures (t_{22}) below which heating in addition to waste heat recovery are required. Thus the definition of recovered heat coefficient is required.

In order to derive the curve of the supply air temperature at the exit, t_{22}, the recovered heat coefficient equation, Eq (2.5) is formed as:

$$t_{22} = \Phi\,(t_{11}\text{-}t_{21}) + t_{21} \tag{4.1}$$

Recovered heat coefficients of commercial equipments may vary from 0.4 to 0.8 [25, 26]. Hence, recovered heat coefficient for all example calculations of the graphical method are assumed to be 0.6, which is a reasonable value. Then, the only missing variable t_{22} in this equation may be calculated. For instance, at t_{21}=2 °C, t_{11}=28.6 °C and $\Phi = 0.6$, the equation leads to solution t_{22}=17.96 °C. Curve for supply air temperature at the exit is originated after the same calculations for each temperature

45

within the range is proceeded. Relevant temperatures t_{22} are calculated, which may be attained through waste heat recovery. If the attained temperatures are above aimed working space temperature 19 °C, they are not regarded to avoid overheating. In other words, it is known that supply air temperature can already be upgraded to working space temperature at t_{21}=10 °C. Supply air may be heated solely by heat recovery equipment up to 21.16 °C, which is above working space temperature t_{fin}=19 °C.

Next step is to originate one big and one small triangular on the graph to quantify specific air conditioning loads as illustrated in Figure 4.2. Through adding two lines three different areas are determined which match specific air conditioning loads.

- Area ABC: Specific heating load of additional equipment (°Kh/yr)
- Area ADE Specific heating load (°Kh/yr)
- Area CBDE: Specific waste heat recovery load (°Kh/yr)
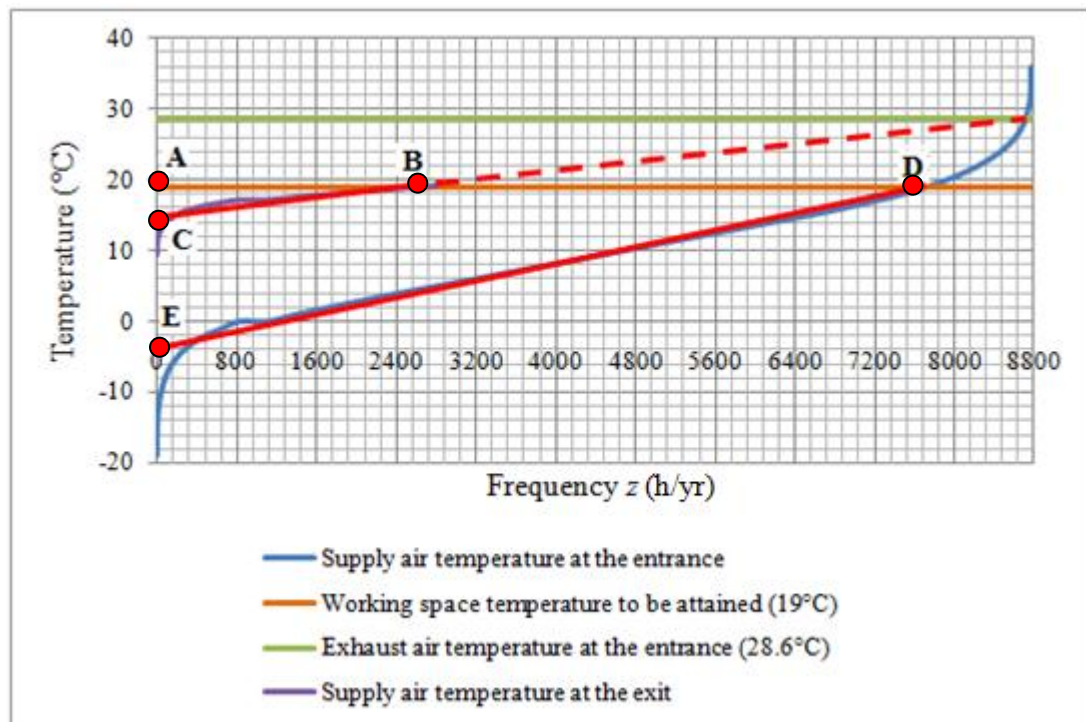
Figure 4.2: Determination of specific heating loads

After drawing triangulars, the areas are inserted to following equations, which are denoted as G_{tv}, G_{t} and G_{tR} [10]:

$$Q_{\mathrm{V}} = \dot{m}_{\mathrm{A}} \; f\mathrm{a} \; c \; G_{\mathrm{tv}} \tag{4.2}$$

$$Q = \dot{m}_{\mathrm{A}} \; f\mathrm{a} \; c \; G_{\mathrm{t}} \tag{4.3}$$

$$Q_R = \dot{m}_A \, fa \, c \, G_{tR} = Q - Q_V \qquad\qquad (4.4)$$

Q: Annual heating requirement to attain the air conditioned space temperature without heat recovery equipment

Q_R: Annual waste heat recovery

Q_V: Additional heating requirement to achieve the air conditioned space temperature

fa: Weekend restrictiveness factor

$\dot{m}_A$: Mass flow rate

c: Specific heat capacity

G_{tv} : Specific heating load of additional equipment (Area ABC)

G_t: Specific heating load (Area ADE)

G_{tR}: Specific waste heat recovery load (Area CBDE)

To convert volumetric flow rate to mass flow rate, density at the annual average ambient air temperature indicated as 10.2 °C is considered and therefore density ρ is applied as 1.247 kg/m^3. Specific heat capacity c is equal to 1.005 kJ/kg°K within the range from -50°C to 40°C. Another regarded parameter is weekend restrictiveness factor fa. Since air conditioning is switched off on weekends, no heating is necessary.

$fa = 5/7 = 0.71$

Considering the holidays in a year in Germany, weekend restrictiveness factor is equal to 0.67 [10]. Besides, the same value subject to official holidays in Turkey is calculated to be 0.69 between 2000-2010.

$fa = 0.67$

The result may also be multiplied by 2.778×10^{-7} to convert kJ to MWh. Regarding to the same a example with $\Phi = 0.6$, t_{11}=28.6 °C and t_{fin}=19 °C the following values are calculated:

$$Q_v = 144181 \frac{m^3}{h} \times 1.247 \frac{kg}{m^3} \times 0.67 \times 1.005 \frac{kJ}{kg°K} \times \left(\frac{(19-15)°K \times 2800 \frac{h}{yr}}{2} \right) \times 2.778 \times 10^{-7} \frac{MWh}{kJ}$$

Q_v = 188.337 MWh/yr

$$Q = 144181 \frac{m^3}{h} \times 1.247 \frac{kg}{m^3} \times 0.67 \times 1.005 \frac{kJ}{kg°K} \times \left(\frac{(19-(-4))°K \times 7600 \frac{h}{yr}}{2} \right) \times 2.778 \times 10^{-7} \frac{MWh}{kJ}$$

Q = 2939.402 MWh/yr

$$Q_R = 2939,402 \frac{MWh}{yr} - 188,337 \frac{MWh}{yr}$$

Q_R = 2751.065 MWh/yr

4.2.2 Facility 2

In Facility 2 process waste heat temperature is 28.7 °C. Apart from this slight difference of exhaust air temperature, the graphic is almost same with the one belonging to Facility 1. Working space temperature, location and thus ambient temperature also do not change.
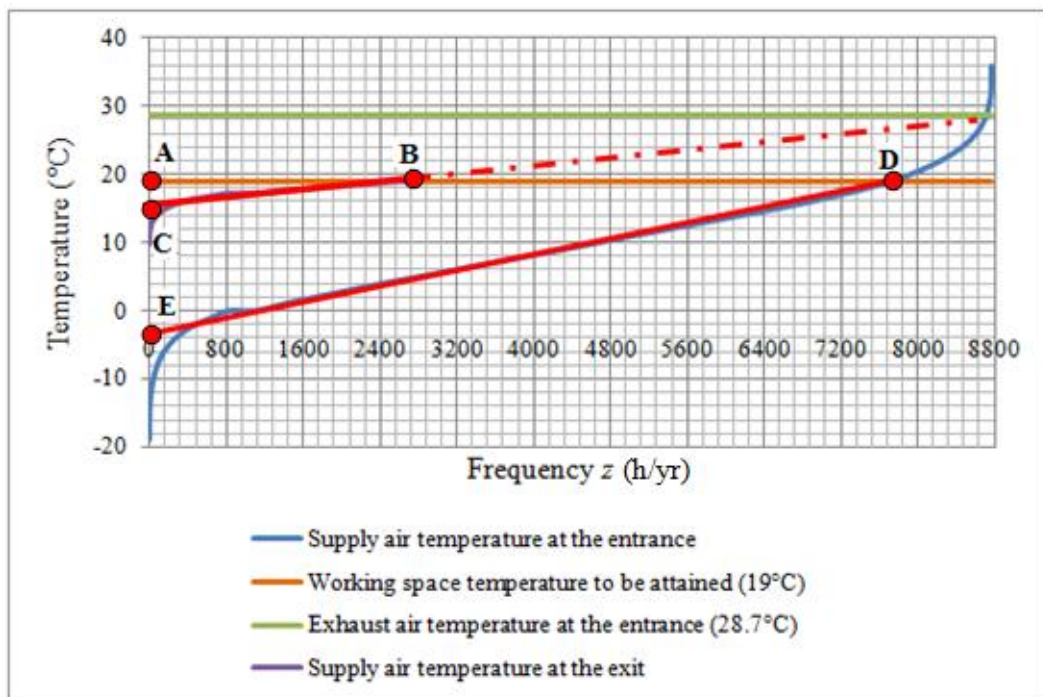
Figure 4.3: Graphical method calculation for Facility 2

- Area ABC = G_{tV} : Specific heating load of additional equipment (°Kh/yr)
- Area ADE= G_t: Specific heating load (°Kh/yr)
- Area CBDE = G_{tR}: Specific waste heat recovery load (°Kh/yr)

The main difference of Facility 2 is low and variable volumetric flow rate, which ranges between 6072 and 10120 m³/h. A minimum and maximum value is regarded to state a heating interval. On the other hand, this variation does not lead a difference in Figure 4.3, since volumetric flow rates are taken into account after originating the chart. The values related to heating are calculated as follows:

Minimum:

$$Q_{V\min} = 6072\,\frac{m^3}{h} \times 1.247\,\frac{kg}{m^3} \times 0.67 \times 1.005\,\frac{kJ}{kg\,°K} \times \left(\frac{(19-15)°K \times 2800\,\dfrac{h}{yr}}{2} \right) \times 2.778 \times 10^{-7}\,\frac{MWh}{kJ}$$

$Q_{V\min}$ = 7.932 MWh/yr

$$Q_{\min} = 6072\,\frac{m^3}{h} \times 1.247\,\frac{kg}{m^3} \times 0.67 \times 1.005\,\frac{kJ}{kg\,°K} \times \left(\frac{(19-(-4))°K \times 7600\,\dfrac{h}{yr}}{2} \right) \times 2.778 \times 10^{-7}\,\frac{MWh}{kJ}$$

$Q_{\min}$ = 123.789 MWh/yr

$$Q_{R\min} = 123.789\,\frac{MWh}{yr} - 7.932\,\frac{MWh}{yr}$$

Q_{Rmin} = 115.858 MWh/yr

Maximum:

$$Q_{V\max} = 10120\,\frac{m^3}{h} \times 1.247\,\frac{kg}{m^3} \times 0.67 \times 1.005\,\frac{kJ}{kg\,°K} \times \left(\frac{(19-15)°K \times 2800\,\dfrac{h}{yr}}{2} \right) \times 2.778 \times 10^{-7}\,\frac{MWh}{kJ}$$

$Q_{V\max}$ = 13.219 MWh/yr

$$Q_{\max} = 10120\,\frac{m^3}{h} \times 1.247\,\frac{kg}{m^3} \times 0.67 \times 1.005\,\frac{kJ}{kg\,°K} \times \left(\frac{(19-(-4))°K \times 7600\,\dfrac{h}{yr}}{2} \right) \times 2.778 \times 10^{-7}\,\frac{MWh}{kJ}$$

Q_{max} = 206.315 MWh/yr

$$Q_{R\max} = 206.315\,\frac{MWh}{yr} - 13.219\,\frac{MWh}{yr}$$

$Q_{R\max}$ = 193.096 MWh/yr

Table 4.2 highlights the entire results calculated by means of graphical method.

Table 4.2: Results of graphical method solution for both facilities

	Q_V (MWh/yr)	Q (MWh/yr)	Q_R (MWh/yr)
Facility 1	188.337	2939.402	2751.065
Facility 2 (max.)	7.932	123.789	115.858
Facility 2 (min.)	13.219	206.315	193.096

4.3 Single frequency method

Single frequency method is based on the same data in DIN 4710. Heating requirement for each hour, during which the working space temperature has to be attained are calculated independent on each other. In other words, frequency of the temperatures below t_{fin}=19 °C and related heating requirement are taken into consideration separately. That's why, single frequency method points out more accurate results in comparison with graphical method.

4.3.1 Facility 1

As seen in Appendix A1 supply air temperatures at the entrance vary from -18 to 36 °C. However, no heat recovery will be needed at temperatures more than 18 °C, which means more than 18.9 °C according to DIN 4710. Each ΔQ and ΔQ_R should be calculated for each temperature in this range via following equations [10]:

$$\Delta Q = V_A \rho \, z \, c_L \, (t_{fin}\text{-}t_{21}) \tag{4.5}$$

$$\Delta Q_R = V_A \rho \, z \, c_L \, \phi_A \, (t_{21}\text{-}t_{11}) \tag{4.6}$$

ΔQ: Heating requirement to attain the air conditioned space temperature within each temperature interval without heat recovery equipment
ΔQ_R: Waste heat recovery within each temperature interval
V_A: Volumetric flow rate
ρ: Density
z: Frequency
c_L: Specific heat capacity
t_{fin}: Working space temperature
t_{21}: Supply air temperature at the entrance
t_{11}: Exhaust air temperature at the entrance

Possible heat recovery will be illustrated here not for the whole range but from t_{21}=3 °C to t_{21}=5 °C. ϕ_A is again assumed to be 0.6. Moreover, t_{11} (=28.6°C), t_{fin} (=19 °C), c_L= 1.005 kJ/kg°K, conversion factor 2.778×10^{-7} MWh/kJ remains constant.

For t_{21}=3 °C:

$$\Delta Q = 144181 \frac{m3}{h} \times 1.279 \frac{kg}{m3} \times 341.5 \frac{h}{yr} \times 1.005 \frac{kJ}{kg°K} \times (19-3)°K$$

$\Delta Q = 281.312$ MWh/yr

$$\Delta Q_R = 144181 \frac{m3}{h} \times 1.279 \frac{kg}{m3} \times 341.5 \frac{h}{yr} \times 1.005 \frac{kJ}{kg°K} \times 0.6 \times (28.6-3)°K$$

$\Delta Q_R = 270.059$ MWh/yr

At t_{21}=3 °C, recoverable amount of energy is less than needed. Additional heating has to be supplied to the working space. In other words waste heat recovery does not enhance the working space temperature up to t_{fin}=19 °C. As the supply air temperature at the entrance increases, lower amount of additional heating is needed.

For t_{21}=4 °C:

$$\Delta Q = 144181 \frac{m3}{h} \times 1.275 \frac{kg}{m3} \times 352.1 \frac{h}{yr} \times 1.005 \frac{kJ}{kg°K} \times (19-4)°K$$

$\Delta Q = 271.065$ MWh/yr

$$\Delta Q_R = 144181 \frac{m3}{h} \times 1.275 \frac{kg}{m3} \times 352.1 \frac{h}{yr} \times 1.005 \frac{kJ}{kg°K} \times 0.6 \times (28.6-4)°K$$

$\Delta Q_R = 266.728$ MWh/yr

For both t_{21}=3 °C and t_{21}=4 °C, values of ΔQ are always more than ΔQ_R. On the other hand at t_{21}=5 °C, recoverable amount of energy is calculated to be more than required heating to achieve t_{fin}=19 °C. Therefore, overheating must be prevented. ΔQ_R is so assumed that, it is regulated to be equal to ΔQ practically.

For t_{21}=5 °C:

$$\Delta Q = 144181 \frac{m3}{h} \times 1.27 \frac{kg}{m3} \times 380.3 \frac{h}{yr} \times 1.005 \frac{kJ}{kg°K} \times (19-5)°K$$

$\Delta Q = 272.185$ MWh/yr

$$\Delta Q_R = 144181 \frac{m3}{h} \times 1.27 \frac{kg}{m3} \times 380.3 \frac{h}{yr} \times 1.005 \frac{kJ}{kg°K} \times 0.6 \times (28.6-5)°K$$

$\Delta Q_R = 275.295$ MWh/yr

Table 4.3 illustrates all of the calculated results. As the outside temperature goes up, additional heating diminishes. If it is sufficiently high, like $t_{21}=5$ °C, then even surplus of heat exists, which may be avoided through process control equipments.

Table 4.3: Application of single frequency method

ρ (kg/m³)	z (h/yr)	t_{A1} (°C)	ΔQ (MJ/yr)	ΔQ (MWh/yr)	Q (MWh/yr)	Q_R (MWh/yr)	Q_V (MWh/yr)
1.279	341.5	3	1012641	281.312	188.479	180.940	7.539
1.275	352.1	4	975757	271.065	182.614	178.708	2.906
1.27	380.3	5	979788	272.185	182.364	182.364	-

Through adding heating requirement and heat recovery for each interval total values are computed. Besides, not only weekend restrictiveness factor but also operating time factor f_B has to be considered. It depends on the duration and daytime on daily basis. Since both facilities are operating for 24 hours per day, f_B is equal to 1 [10]. Rest of the operating time factors are tabulated in Table A2 at Appendix A2.

$$Q = f_B \times f_A \times \sum_{t_{A1}=-18}^{t_{A1}=18} \Delta Q \tag{4.7}$$

$$Q_R = f_B \times f_A \times \sum_{t_{A1}=-18}^{t_{A1}=18} \Delta Q_R \tag{4.8}$$

$$Q_V = Q - Q_R = f_B \times f_A \times (\sum_{t_{A1}=-18}^{t_{A1}=18} \Delta Q - \sum_{t_{A1}=-18}^{t_{A1}=18} \Delta Q_R) \tag{4.9}$$

Q: Annual heating requirement to attain the air conditioned space temperature without heat recovery equipment
Q_R: Annual waste heat recovery
Q_V: Additional heating requirement to achieve the air conditioned space temperature
f_B: Operating time factor
f_A: Weekend restrictiveness factor

The following calculations are valid for the same example:

$$Q = 1 \times 0.67 \times \sum_{t_{A1}=-18}^{t_{A1}=18} (0.276 + 0.803 + 1.167 + ... + 33.785 + 20.645 + 9.343)$$

$Q = 2893.302$ MWh/yr

$$Q_R = 1 \times 0.67 \times \sum_{t_{A1}=-18}^{t_{A1}=18} (0.209 + 0.611 + 0.892... + 33.785 + 20.645 + 9.343)$$

$Q = 2748.231$ MWh/yr

$Q_V = 2893.302 - 2748.231$

$Q_V = 145.071$ MWh/yr

To be able to get used to differing recovered heat coefficients quicker, Q, Q_R and Q_V are calculated for a range of recovered heat coefficients. The range starts from $\phi=0.5$. The upper boundary is selected, where total heating requirement is covered by waste heat recovery equipment, in other words where no heating in addition to waste heat recovery is necessary. Table A.3 in Appendix A3 is constituted by calculated values for the whole range in detail.

Figure 4.4 visualizes the trend of the series of data within the reasonable range. As illustrated, recovered heat coefficients higher than 0.8 is rationally not necessary, where lower values may also be acceptable due to almost flat run of the curve.
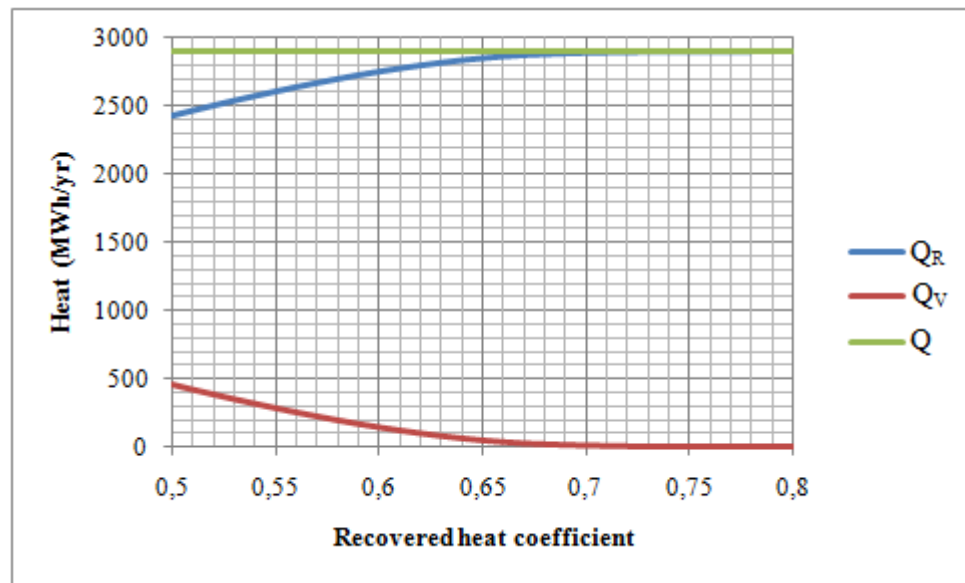
Figure 4.4: Waste heat recovery potential in Facility 1

4.3.2 Facility 2

Same calculations are realized for Facility 2. Like in the graphical method results are computed both for minimum and maximum values of volumetric flow rates. Detailed table is shown in Appendix A4, which will be necessary for the comparative economic analysis of selected equipments.

Visualization of those data on Figure 4.5 concludes to a very similar graph to Figure 4.4 which belongs to Facility 1. Waste heat recovery equipments with recovered heat

coefficients around 0.65 seems sufficient, if prices change dramatically, since all four curves of Q_R and Q_V become flat for higher coefficients.
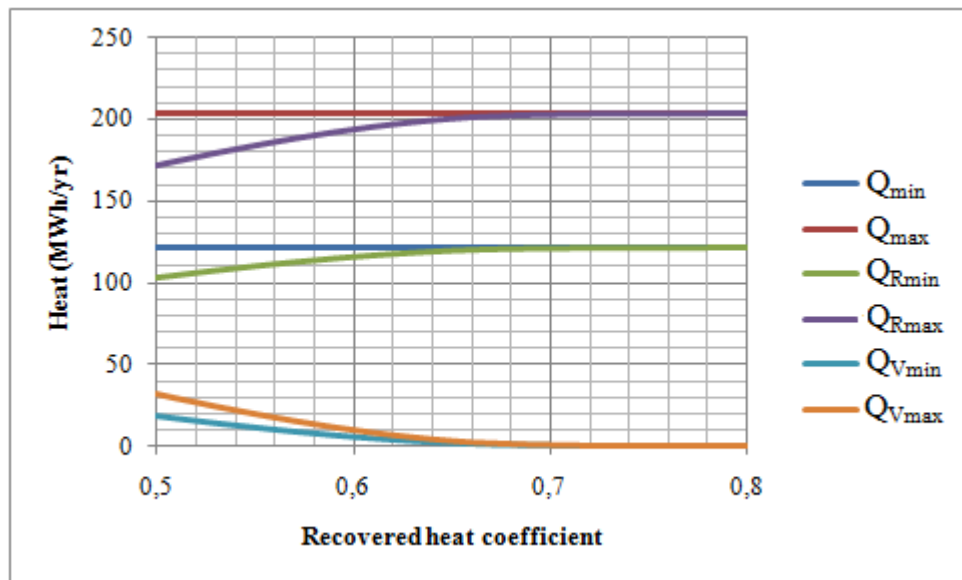
Figure 4.5: Waste heat recovery potential in Facility 2

5. ECONOMIC ANALYSIS

In this section comparative economic analysis of selected equipments will be proceeded and annual profit of each option will be evaluated.

5.1 General information and assumptions

Previously, theoretical information is summarized and recoverable amount of waste heat is calculated regarding to various recovered heat coefficients.

Prior to theoretical information, the number of options for waste heat recovery is decreased to three subject to cost effectiveness: rotary heat exchanger, plate heat exchanger and run around coil. Rotary heat exchangers seem to be the most cost effective choice among both recuperators and regenerators. On the other hand, plate heat exchangers may offer the optimal solution where exhaust and supply air should not be mixed. What's more, run around coil may also be necessary, if streams are located far away from each other and fed to waste heat recovery equipment separately. On the whole, rotary heat exchanger, plate heat exchanger and run around coil will be assessed in economic analysis. To achieve the closest results to reality, more than 50 manufacturers in Germany, Switzerland and Liechtenstein are contacted via e-mail and telephone without naming the company. Results are highlighted in Tables 5.1 and 5.2. All of the prices include 19% value added tax.

Table 5.1: Properties of equipments intended for Facility 1

	Plate heat exchanger	Rotary heat exchanger		Run around coil
Company	P1	P2	P3	P4
Number of equipment	2	3	3	2
Recovered heat coefficient (%)	75,2	74,8	76,3	48
Pressure drop (Pa)	348,5	83	143	71
Total annual power saving (kW)	675	681,9	692,7	220
Total price (€)	48208	52961	37589	94486

Table 5.2: Properties of equipments intended for Facility 2

	Plate heat exchanger	Rotary heat exchanger		Run around coil	
Company	P1	P2	P3	P4	P5
Number of equipment	1	1	1	1	1
Recovered heat coefficient (%)	72,8	76,3	76,1	45	57,6
Pressure drop (Pa)	222,5	76	145	138	120
Total annual power saving(kW)	46,1	49,1	48,7	28,65	36,5
Total price (€)	2529	4959	3451	7497	8687

Furthermore installation costs have to be regarded as a part of initial investment costs. Both plants should be examined by engineering offices, which will be available in the following phases of this project. As a result of restricted amount of practical data like in the equipment costs, heuristics belonging to state of Baden Württemberg and Handbook of Heating and Air Conditioning 2009 are considered at this phase, which are expressed in Table 5.3.

Table 5.3: Heuristics of investment costs [25,26]

	Recovered heat coefficient (without condensation)	Total costs including installation costs	
		Handbook of Heating and Air Conditioning 09/10 in € pro m^3/h	Ministry of Economics-Baden Württemberg in € pro m^3/h
Plate heat exchanger	45-65 %	0,35-0,65	0,30-0,60
Run around coil	40-70 %	0,70-1,40	0,60-1,30
Rotary heat exchanger	65-80 %	0,50-0,80	0,50-0,70

As summarized in Table 5.3, heuristics define the investment cost which is sum of equipment and installation costs with regard to volumetric flow rate. Run around coils achieve slightly higher recovered heat coefficients than plate heat exchangers but they offer the highest total cost per volumetric flow rate. Despite reaching the lowest unit investment costs, recovered heat coefficients of plate heat exchangers are lower than the others, where rotary heat exchangers attain the highest recovered heat coefficient. All in all, the data belonging to the handbook is preferred since it is a more recent source providing more accurate information intended particularly for this field. Maximum total costs per unit volumetric flow rate in the range are

considered. At this point, previously collected data from companies made it possible to acquire a general perspective about installation costs of each equipment.

Properties like pressure drop and motor power are also acquired. Moreover, comparison of the recovered heat coefficients and related recoverable amount of energy are taken into account to understand how realistic they are. Calculations are proceeded subject to those comparisons. By way of illustration, 75% of recovered heat coefficient is selected, which is close to maximum value, since the data from manufacturer correspond to the theoretical data. Same opinion is valid for run around coil and 55% is regarded whilst calculations. However, recovered heat coefficient encountered by the manufacturers seemed high and alleviated down to 60%. To sum up, generally either only recovered heat coefficients or both recovered heat coefficients and related recoverable amount of energy are evaluated optimistic and recoverable energy computed in the previous section is considered.

5.2 Sample calculation: Rotary heat exchanger for Facility 1

Economic analysis is proceeded according to VDI 2067, which distinguishes the annuity costs step by step into 4 subgroups: Annuity of the outgoing capital-related, requirement (consumption)-related, operation-related and other costs. Revenues are computed as incoming payments [27]. By way of illustration the economic analysis of the rotating heat exchanger at the first facility is introduced.

5.2.1 Annuity of capital-related costs

Volumetric flow rate in Facility 1 is measured to be 144181 m^3/h. Regarding to Table 5.3 investment cost including installation cost is 0.80 €.

$$A_0 = V \times UCI \tag{5.1}$$

$$A_0 = 144181 \frac{m^3}{h} \times 0.80 \frac{€}{m^3}$$

$$A_0 = 115344,8 \text{ €}$$

A_0: Investment premium
V: Volumetric flow rate
UCI: Unit cost of investment subject to volumetric flow rate

Investment premium of heat wheel is calculated to be 115344,8 €. Moreover, observation period is selected to be 20 years. The reason for this is different calculated service lives of a heat wheel, a run around coil and a plate heat exchanger. According to Table A3 of VDI 2067 service lives of a run around coil and a plate heat exchanger last for 20 years, where the calculated service life of a heat wheel is 15 years. Consequently, the observation period is chosen to be 20 years, which also leads to one time replacement of the heat wheel, although no replacement is needed for plate heat exchanger and run around coil. To compute the investment costs, price change factor for capital related costs and interest rate must be considered, which are respectively 1.03 and 1.07 in the guideline illustrated example. Cost of the first replacement is given by the equation [27]:

$$A_1 = A_0 \times \frac{r^{(1 \times T_N)}}{q^{(1 \times T_N)}}$$

(5.2)

$$A_1 = 115344.8 \times \frac{1.03^{(1 \times 15)}}{1.07^{(1 \times 15)}}$$

$A_1 = 65132.8$ €

A_1: Cash value of the first replacement
A_0: Investment premium
r: Price change factor
q: Interest factor
T_N: Service life of installation components in years

After computing the cash value of the first replacement as 65132.8 €, net book value should be considered. The net book value is determined by straight line depreciation of the investment premium until the end of the observation period and discounted at the beginning of the observation period [27]

$$Rw = A_0 \times r^{(n \times T_N)} \times \frac{(n+1) \times T_N - T}{T_N} \times \frac{1}{q^T}$$

(5.3)

R_W: Net book value
A_0: Investment premium
r: Price change factor
q: Interest factor
n: Number of replacements within the observation period
T: Observation period in years
T_N: Service life of installation components in years

$$R_W = 115344.8 \times 1.03^{(1 \times 15)} \times \frac{(1+1) \times 15 - 20}{15} \times \frac{1}{1.07^{20}}$$

$R_W = 30959{,}19 \ €$

The annuity factor is calculated as follows [27]:

$$a = \frac{q^T \times (q-1)}{q^T - 1} \tag{5.4}$$

a: Annuity factor
q: Interest factor
T: Observation period in years

$$a = \frac{1.07^{20} \times (1.07 - 1)}{1.07^{20} - 1}$$

$a = 0.094393$

Another factor is f_K which corresponds to factors for repair of the investment premium per year may be taken from Table A3 of VDI 2067 as 3%.

$f_K = 3 \ \%$

Cash value factor b is defined according to different subgroups of annuity costs in the guideline. However, they do not differ, unless the interest factor q, price change factor r and observation period T remain same. Since these variables are same for all sorts of waste heat recovery equipment, the one and only difference are the indexes subject to the bundle of costs. For instance, cash value factor b is symbolized as b_{IN} in terms of capital related costs. Moreover, it refers to maintenance and repair payments. Generally all types of cash value factor may either be taken from VDI 2067 or calculated with the following equation [27]:

$$b = \frac{1 - \left(\dfrac{r}{q}\right)^T}{q - r} \tag{5.5}$$

b: Cash value factor
q: Interest factor
r: Price change factor
T: Observation period in years

$$b = \frac{1 - \left(\dfrac{1.03}{1.07}\right)^{20}}{1.07 - 1.03}$$

$b = 13.332$

From VDI 2067 b is read also as 13.332 which is equal to the calculated value. In order to consider price changes of regular payments for repair cash value factor, b_{IN} is multiplied by annuity factor α [27].

$$ba_{IN} = b_{IN} \times a \tag{5.6}$$

ba_{IN} = 13.332 × 0.094

ba_{IN} = 1.258415

ba_{IN}: Price dynamic annuity factor for repair payments
b_{IN}: Cash value factor for repair payments
a: Annuity factor

The annuity of payments linked to capital in €/year is given by the equation [27]:

$$A_{N,K} = (A_0 + A_1 + A_2 + ... + A_N - R_w) \times a + \frac{f_K}{100} \times ba_{IN} \tag{5.7}$$

$$A_{N,K} = (115344.8 + 65132.8 - 30959.19) \times 0.094 + \frac{3}{100} \times 115344.8 \times 1.258$$

$A_{N,K}$ = 18468 €/yr

$A_{N,K}$: Annuity of payments linked to capital
A_0: Investment premium
$A_{1, 2, ..., n}$: Cash value of 1st, 2nd, nth replacement
R_W: Net book value
a: Annuity factor
f_K: Factor for repairs in % of the investment
ba_{IN}: Price-dynamic annuity factor for repair payments

Consequently, previously calculated variables are inserted and annuity of payments linked to capital are computed to be 18468 €/yr.

5.2.2 Annuity of requirement (consumption) - related costs

Requirement (consumption) related costs include energy requirement for compression of supply and exhaust air and for the motor to rotate the heat wheel.

5.2.2.1 Energy requirement for the compression of supply and exhaust air

Energy requirement for the compression of supply and exhaust air is given by the following calculation [28]:

$$E_C = \frac{\Delta P \times (V_E + V_S) \times \sum z \times f_A \times f_B}{\eta}$$ (5.8)

E_C: Energy requirement for the compression of supply and exhaust air
ΔP: Pressure drop
V_E: Volumetric flow rate of the exhaust air
V_S: Volumetric flow rate of the supply air
z: Number of hours during which temperature is below working space temperature
t = 19 °C
f_A: Weekend restrictiveness factor
f_B: Operating time factor
η: Ventilator efficiency

There exist two alternatives for heat wheel from the manufacturers intended for the Facility 1. Pressure drops belonging to two different systems are 143 and 83 Pa, respectively. Volumetric flow rate for both streams:

$$144181 m^3 / h \times 2 = 288362 m^3 / h$$

Total number of hours during which temperature is below working space temperature T = 19 °C is 7467.4 hours/yr according to DIN 4710. Weekend restrictiveness factor f_A and operating time factor f_B which were mentioned in the previous section are 0,67 and 1, respectively. Ventilator efficiency to compress the streams is assumed to be 0.75 subject to manufacturers' reports. Conversion factors 10^6 and 3600 are applied to convert W into kW and m^3/h to m^3/s. Finally, energy requirement to compress both streams are computed as follows:

$$E_C = \frac{113 Pa \times 288362 m^3 / h \times 7467.4 h / a \times 0.67 \times 1}{0.75 \times 1000 W / kW \times 3600 s / h}$$

E_C = 60380.53 kWh/year

5.2.2.2 Energy requirement for the motor to rotate the heat wheel

Motor power from manufacturers' spreadsheets are equal and 2.25 to rotate the heat wheel.

P = 2.25 kW

Prior to the total number of hours during which waste heat will be recovered [28]:

$$E_M = P \times \sum z \times f_A \times f_B \tag{5.9}$$

$$E_M = 2.25kW \times \sum 7467.4 \times 0.67 \times 1$$

E_M = 11257.11 kWH/year

E_M: Energy requirement to rotate the heat wheel
P: Motor power
z: Frequency
f_A: Weekend restrictiveness factor
f_B: Operating time factor

5.2.2.3 Total energy requirement

Total energy requirement is the sum of the energy requirement for the compression of both air streams and rotation of the heat wheel.

$$E_T = E_C + E_M \tag{5.10}$$

E_T = 60380.53 + 11257.11

E_T = 71637.64 kWh/year

E_T: Total energy requirement
E_C: Energy requirement for the compression of supply and exhaust air
E_M: Energy requirement to rotate the heat wheel

Unit electricity and heating costs are considered as 0.15 € per kWh and 40 € per MWh [28]. The requirement (consumption) related costs in the first year:

$$A_{VI} = E_T \times UCE \tag{5.11}$$

$A_{VI} = 71637.64\ kWh \times 0.15\ €/kWh$

$A_{VI} = 10745.65\ €$

A_{V1}: The requirement (consumption) related costs in the first year
E_T: Total energy requirement
UCE: Unit cost of electricity

Requirement (consumption) related costs in the first year are computed as 10745.65 €. Price dynamic annuity factor for capital related costs was already calculated as 13.332. Cash flow rate remains same since interest factor q, price change factor r and observation period T do not change [27].

$$ba_V = b_V \times a \tag{5.12}$$

$$ba_V = 13.332 \times 0.094393$$

$$ba_V = 1.258446$$

ba_V :Price dynamic annuity factor for consumption related costs
b_{IN}: Cash value factor for requirement (consumption) related costs
a: annuity factor

Annuity of requirement (consumption) related costs are given by the equation [27]:

$$A_{N,V} = A_{V1} \times ba_V \tag{5.13}$$

$A_{N,V}$ = 10745.65 × 1.258446

$A_{N,V}$ = 34199.58 €/year

$A_{N,V}$: Annuity of requirement (consumption) related costs
A_{V1}: Requirement (consumption) related costs in the first year
ba_V :Price dynamic annuity factor for consumption related costs

Annuity of requirement (consumption) related costs is calculated to be 34199.58 €/yr. Next parameter is annuity of operation related costs.

5.2.3 Annuity of operation related costs

Operation related costs imply cleaning and servicing which are indicated to be the 10% of the investment costs at VDI 2067 [27].

$$A_{B1} = (A_0 + A_1) \times \frac{10}{100} \tag{5.14}$$

$$A_{B1} = (115344.8 + 65132.8) \times \frac{10}{100}$$

A_{B1} = 18047.76 €

A_{B1}: Operation related costs in the first year
A_0: Investment premium
$A_{1, 2, ..., n}$: cash value of 1st, 2nd, nth replacement

As in capital and requirement (consumption) related costs cash value factor b_B is equal to 13.332. Subsequently, price dynamic annuity factor is given by the equation [27]:

$$ba_B = b_B \times a \tag{5.15}$$

$$ba_B = 13.332 \times 0.094393$$

$$ba_B = 1.258446$$

b_{AB}: Price dynamic annuity factor for operation related costs
b_B: Cash value factor
a: Annuity factor

Analogous to the other annuities, the annuity of operation related costs is the product of cash value factor and operation related costs in the first year [27]:

$$A_{N,B} = A_{B1} \times ba_B \tag{5.16}$$

$$A_{N,B} = 18047.76 \times 1.258446$$

$A_{N,B}$= 22712.14 €/year

$A_{N,B}$: Annuity of operation related costs
A_{B1}: Operation related costs in the first year
b_{AB}: Price dynamic annuity factor for operation related costs

Finally, annuity of operation related costs is computed as 22712 €/yr.

5.2.4 Annuity of other costs

Since there do not exist any other subgroup of costs, the annuity of other costs is assumed to be 0.

5.2.5 Annuity of revenue

Recovered heat coefficients received from the manufacturers range from 64.6% and 80.7%. Since the coefficients are mostly scattered around 75%, and prior to the coefficients in the literature, recovered heat coefficient representing the general is assumed as 0.75.

Table 5.4: Comparison of graphical and single frequency method

	Graphical method (MWh/yr)	Single frequency method (MWh/yr)
Q_V	1.547	0.729
Q	2978.079	2893.302
Q_R	2976.532	2892.573

Table 5.4 summarizes the outcome of graphical and single frequency methods. The recoverable energy is calculated to be 2892,573 MWh/year via single frequency method and 2976.532 MWh/year via graphical method. As shown in Figure 5.1 there exists a very little amount of difference between both results. Since single frequency method concludes to more accurate results, it is considered during the calculations.
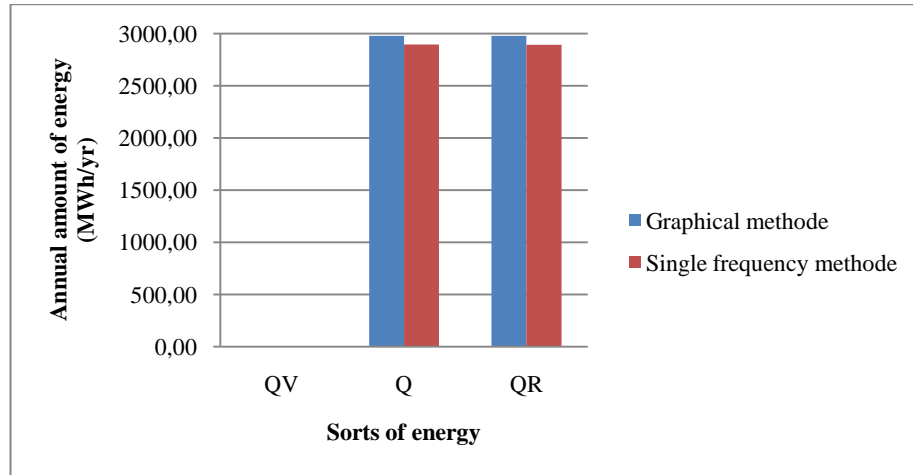
Figure 5.1: Comparison of graphical method and single frequency method

The product of annual recoverable energy and unit cost of heating is the revenue in the first year.

$$E_1 = Q_R \times \text{UCH}$$ (5.17)

$E_1 = 2892,573 \; MWh \, / \, yr \times 40 \, € \, / \, MWh$

$E_1 = 115702.9 \, €$

E_1: Revenue of waste heat recovery in the first year
Q_R: Annual recoverable amount of energy
UCH: Unit cost of heating

The rest of the calculation is analogous to the previous ones [27]:

$$ba_E = b_E \times a$$ (5.18)

ba_E: Price dynamic annuity factor for revenues
b_E: Cash value factor for revenues
a: Annuity factor

$ba_E = 13.332 \times 0.094393$

$ba_E = 1.258446$

65

$$A_{N,E} = A_{E1} \times ba_E \qquad \text{(5.19)}$$

$$A_{N,E} = 115702.9 \times 1.258446$$

$$A_{N,E} = 145605.9$$

$A_{N,E}$: Annuity of revenues
A_{E1}: Revenue in the first year
b_{AE}: Price dynamic annuity factor for revenues

Consequently, annuity of revenues is calculated to be 145605.9 €/yr.

5.2.6 Total annuity

Total annuity is given by the equation [27]:

$$A_N = A_{N,E} - (A_{N,K} + A_{N,V} + A_{N,B} + A_{N,S}) \qquad \text{(5.20)}$$

$A_{N,K}$: Annuity of the outgoing capital-related costs (€)
$A_{N,V}$: Annuity of requirement-consumption related costs (€)
$A_{N,B}$: Annuity of operation-related costs (€)
$A_{N,S}$: Annuity of other costs (€)
$A_{N,E}$: Annuity of revenue (€)
A_N: Total annuity

Inserting all the variables into the equation leads to the result

$$A_N = 145605.9 - (18468.03 + 13522.82 + 22712.14 + 0)$$

A_N = 90902.95 €/year

Inserting all the annuities of costs and revenues concludes to the total annuity in other words total annual profit of 90902.95 € /yr.

5.3 Additional assumptions for run around coil and plate heat exchanger:

Several differences whilst calculations of economic analysis of run around coil and plate heat exchangers may be listed respectively: (1) Observation period and calculated service life of run around coil and plate heat exchanger are equal and 20 years. Hence, no replacement takes place (2) Since there is no moving part in both plate heat exchangers and run around coils, annuity of requirement (consumption) related costs does not include annual energy required for the motor (3) Recovered heat coefficient and related recoverable amount of energy are selected to be 60% for

plate heat exchangers and 55% for run around, the reason for which is put into detail above.

5.4 Results and evaluation of economic analysis for Facility 1

Comparison of rotary heat exchanger, plate heat exchanger and run around coil enlightened in Table 5.5:

Table 5.5: Comparison of waste heat recovery equipment for Facility 1.

		Rotary heat exchanger	Plate heat exchanger	Run around coil
A_0	Initial costs (€)	115345	93718	201853
A_1	Cost of the first replacement (€)	65133	-	-
$A_{N,K}$	Annuity of outgoing capital-related costs (€/yr)	18468	11205	24134
$A_{N,V}$	Annuity of requirement-consumption related costs (€/yr)	13523	35152	14323
$A_{N,B}$	Annuity of operation-related costs (€/yr)	2271	9627	20736
$A_{N,S}$	Annuity of other costs (€/yr)	0	0	0
$A_{N,E}$	Annuity of revenue (€/yr)	145606	138340	131198
A_N	Total annuity (€/yr)	90903	82356	72005
T	Payback period (yr)	1.99	1.14	2.8

Figures 5.2-5.6 visualize the data tabulated in Table 5.5 and make familiar which subgroup affects the total annuity how. Figure 5.2 demonstrates initial and investment costs and Figure 5.3 shows how the annuity of capital related costs alter.
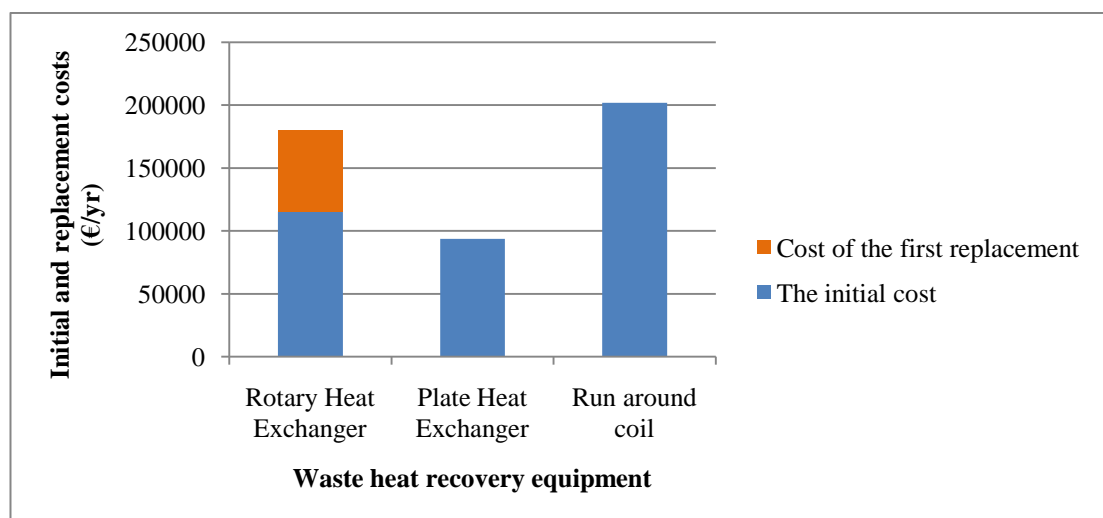
Figure 5.2: Comparison of annuities of initial and replacement costs

Even though investment cost per volumetric flow rate of a plate and rotary heat exchanger do not fluctuate dramatically, substitution of rotary heat exchanger increases the annuity of capital related costs. Figure 5.3 shows another outcome of high capital related costs, which is the enhancement of operation related costs due to its dependence upon capital related costs. In other words, operation related costs vary correlated to capital related costs.
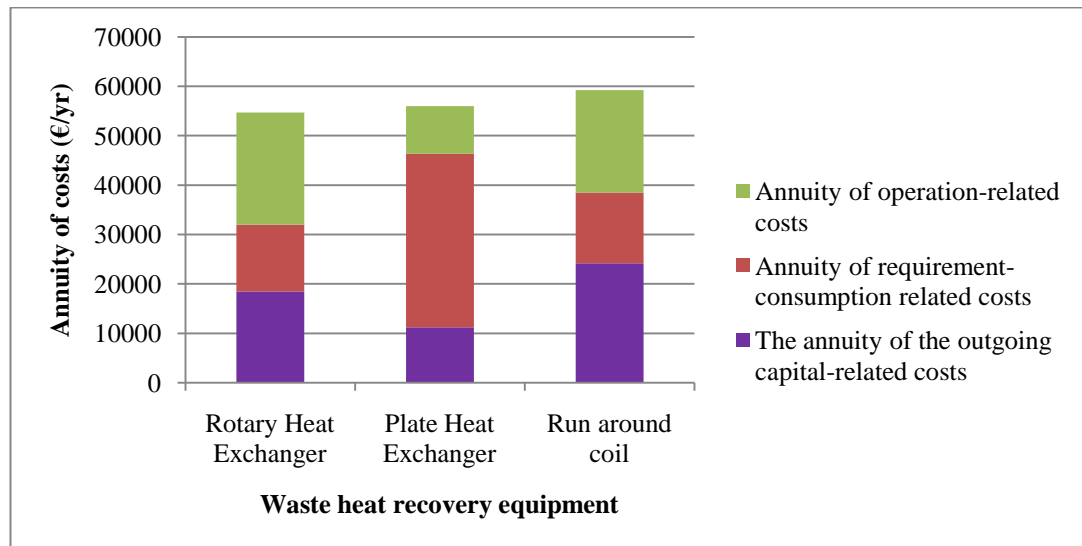
Figure 5.3: Comparison of annuities of costs

As expected, the highest annuity of outgoing capital related costs belongs to run around coil. Final significant point to pay attention in Figure 5.3 is the affect of pressure drop to operation related costs. In particular, it leads to a remarkable rise at the annuity of operation related costs of plate heat exchanger.

Figure 5.4 enlightens the annuity of revenues. As a result of higher recovered heat coefficient application of a rotary heat exchanger yields higher savings. On the other hand, there does not exist a significant difference between rotary and plate heat exchanger, since the curve of recovered heat coefficient becomes flatter in this range of the scale as illustrated in previous section.
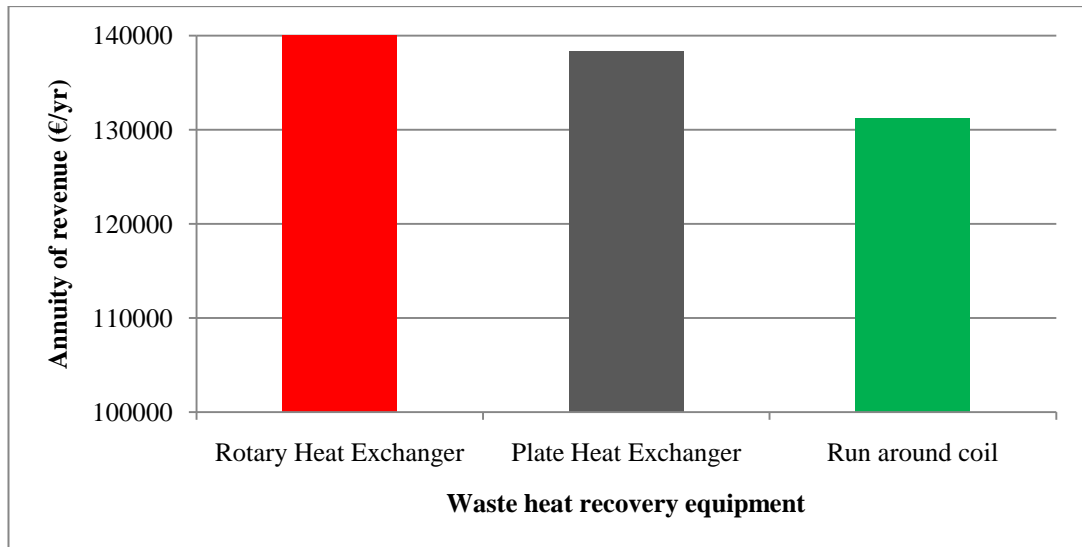
Figure 5.4: Comparison of annuities of revenues

Taking both annuity of costs and revenues, total annuity of each equipment is shown in Figure 5.5. Since the values are positive, it would not be wrong to call the total annuity as annual profit. Clearly indicated by the graph, rotary heat exchanger provides the maximum profit.
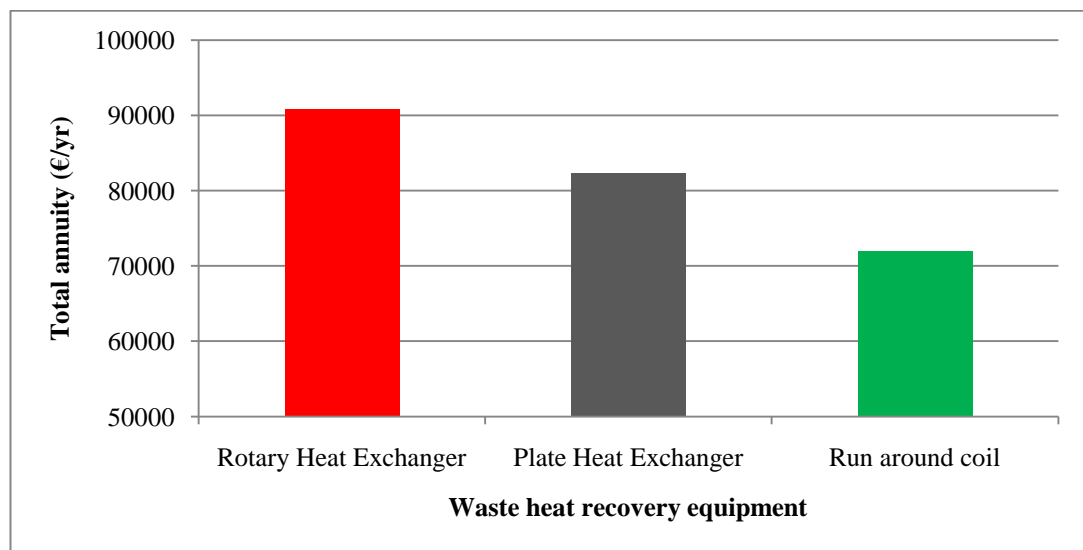
Figure 5.5: Comparison of total annuities

Since interest rates, price change factors and other relevant parameters are taken into consideration during calculation of each subgroup, payback period is the division of initial investment costs by total revenue. Therefore, plate heat exchanger takes the advantage of low initial and replacement costs, which overcomes the negative effect of lower total annuity as shown in Figure 5.6.
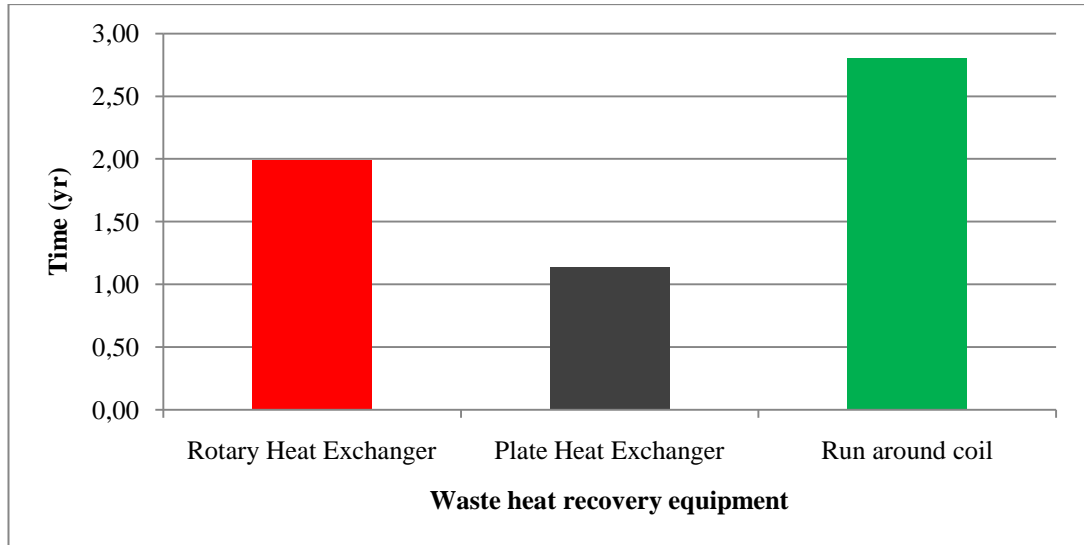
Figure 5.6: Comparison of payback periods

5.5 Results and evaluation of economic analysis for Facility 2

Whilst proceeding the economic analysis for Facility 2, tabulated in Table 5.6, an additional assumption is considered. Volumetric flow rate in Facility 2 ranges from 6072 to 10120 m^3/h. Heuristics define investment costs on the basis of volumetric flow rates, which conclude to different investment costs for the same application.

Table 5.6: Comparison of waste heat recovery equipment for Facility 2

		Rotary heat exchanger	Plate heat exchanger	Run around coil
A_0	Initial costs (€)	8096	6578	14168
A_1	Cost of the first replacement (€)	4572	-	-
$A_{N,K}$	Annuity of the outgoing capital-related costs (€/yr)	1296	786	1694
$A_{N,V}$	Annuity of requirement-consumption related costs (€/yr)	668	1269	731
$A_{N,B}$	Annuity of operation-related costs (€/yr)	1594	676	1455
$A_{N,S}$	Annuity of other costs (€/yr)	0	0	0
$A_{N,E}$	Annuity of revenue (€/yr)	10220	9728	9233
A_N	Total annuity (€/yr)	6662	7006	5353
T	Payback period	1.9	0.94	2.65

According to one manufacturer's prices for both volumetric flow rates, equipment costs do not differ. For instance, plate and rotary heat exchanger cost respectively 1264.67 € and 3451.21 €, which are valid for both volumetric flow rates. Hence, maximum volumetric flow rate is considered to calculate capital and relevant

operation related costs. In other words, variation of volumetric flow rate does not affect capital and operation related costs. On the other hand, average volumetric flow rate is considered to compute the requirement consumption related costs. Outcome of the economic analysis for Facility 2 is visualized by Figures 5.7-5.11. Similar to initial and replacement costs at Facility 1, a possible application of run around coil concludes to the highest and a possible application of plate heat exchanger to the lowest costs as demonstrated in Figure 5.7.
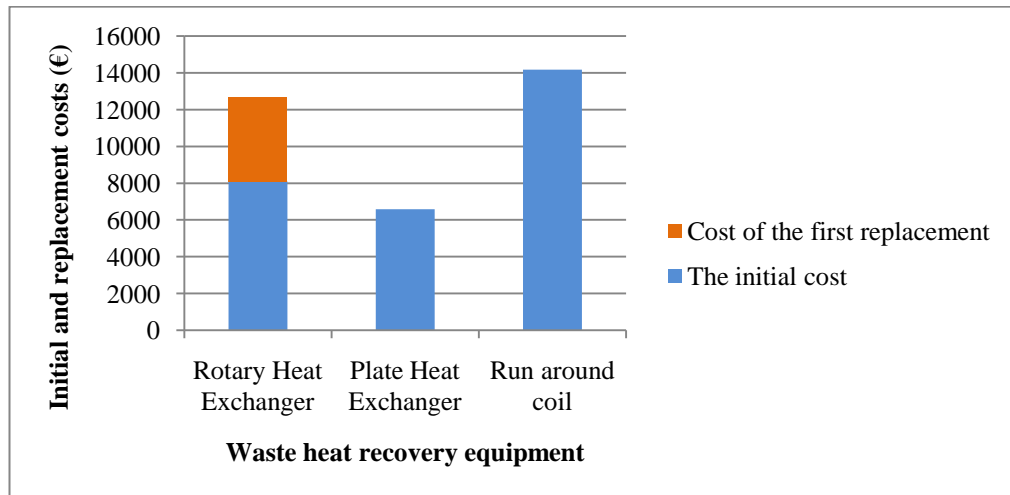
Figure 5.7: Comparison of annuities of initial and replacement costs

Three annuities of cost are illustrated in Figure 5.8. The most remarkable difference between possible applications of Facility 1 and Facility 2 is that decreasing volumetric flow rate alleviates in particular the pressure drop of plate heat exchanger significantly, even though it remains still a bit higher than the others.
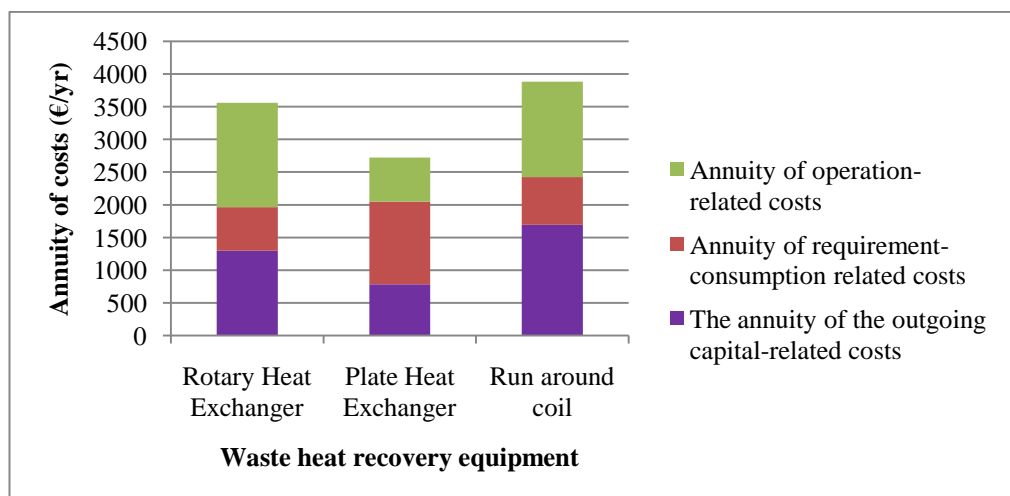
Figure 5.8: Comparison of annuities of costs

Figure 5.9 proves that, the highest annuity of revenue belongs to rotary heat exchanger like at a possible application in Facility 1. Since there exists a scale down in comparison with Facility 1, annuity of revenues and the relevant difference of annuities decrease.
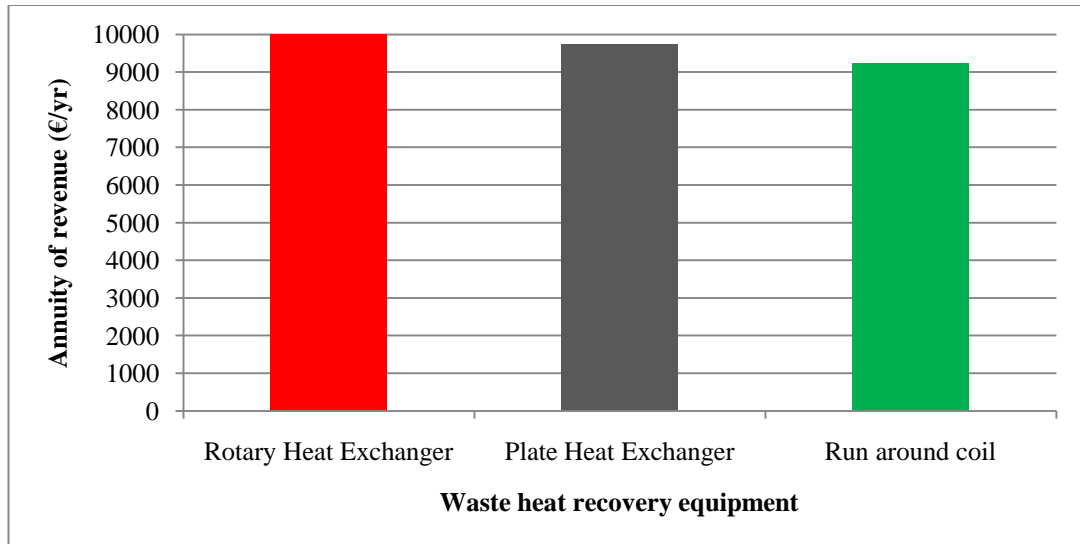
Figure 5.9: Comparison of annuities of revenues

On the contrary to rotary heat exchanger's slightly higher revenue, plate heat exchanger's pressure drop sinks enough to diminish the costs as illustrated in Figure 5.9. Hence, it becomes the most cost effective option for a potential waste heat recovery in Facility 2 as shown in Figure 5.10.
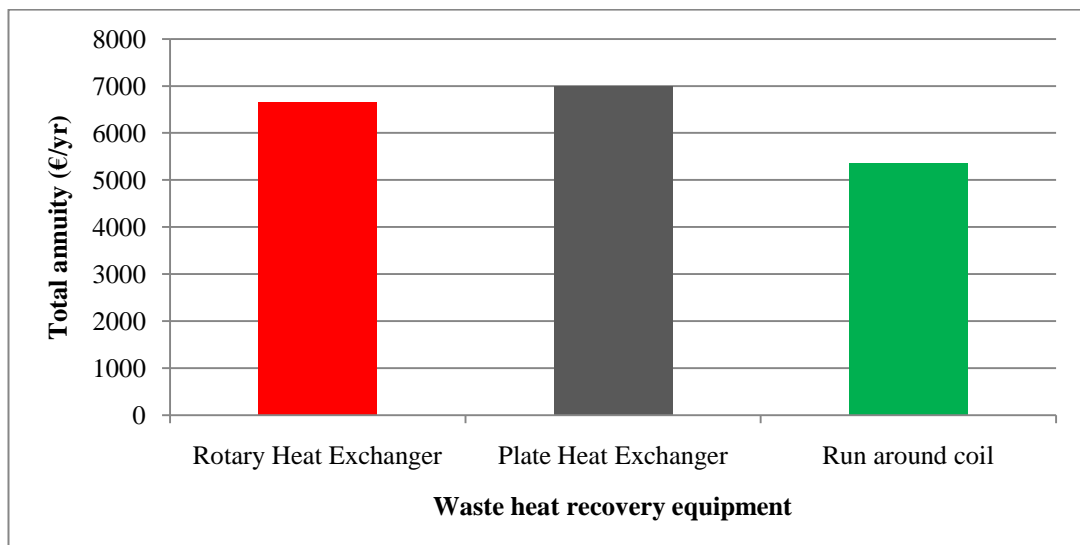
Figure 5.10: Comparison of total annuities

Figure 5.11 demonstrates that, a possible application of plate heat exchanger leads to the shortest payback period as in the previous facility.
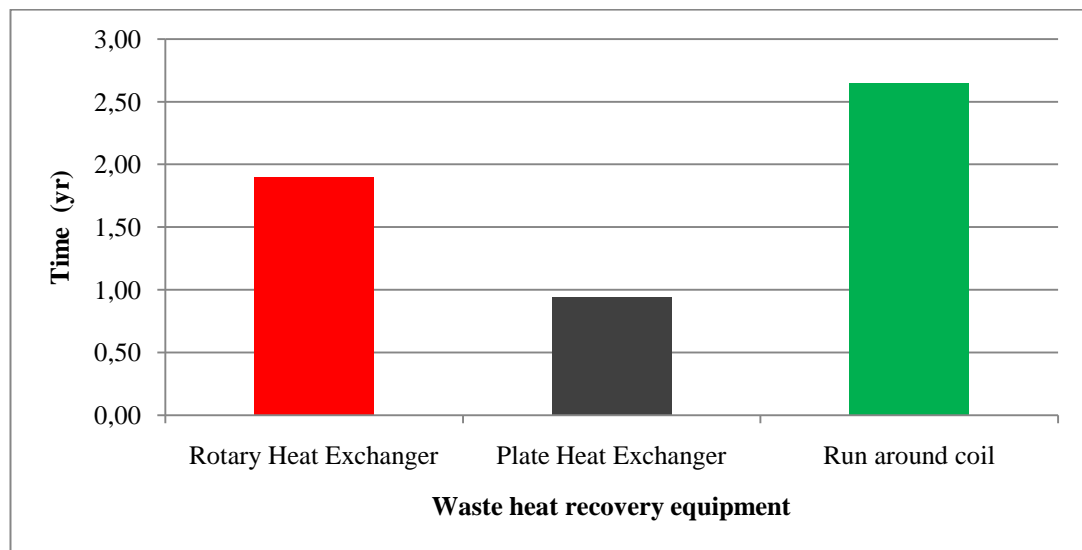
Figure 5.11: Comparison of payback periods

6 CONCLUSION AND RECOMMENDATIONS

The major purpose of this research is to help the decision makers to meet the requirements of two Daimler AG facilities with respect to the assumptions. Particularly, it is realized for selecting the optimum choice among low temperature waste heat recovery options. Subtopics of the theme are handled step by step from theory to practice. Moreover, they are narrowed down not only in the light of current literature but also real data received from manufacturers to achieve the most accurate results. Consequently, economic potential of such an investment is computed according to several guidelines.

6.1 Recommendation for Facility 1: Rotary heat exchanger

For Facility 1 an application of rotary heat exchanger seems to possess the highest total annuity of 90903 €/yr. Utilization of plate heat exchanger and a run around coil conclude to 82356 €/yr and 72005 €/yr total annuity. Hence, it is confirmed that rotary heat exchanger is the most cost effective option if heat exchange takes place between large masses of streams. Even though the least unit cost per volumetric flow rate belongs to plate heat exchanger, high volumetric flow rate increases pressure drop of plate heat exchanger. In other words, high requirement-consumption related costs in particular costs for compressing supply and exhaust air streams at a plate heat exchanger leads to a disadvantage. High investment costs and relatively low recovered heat coefficient of run around coil diminish revenue and hence total annuity significantly. As a result of the same reason, run around coil has the longest payback period equal to 2.8 years. It is also consistent to theoretical information which states that run around coil should only be employed, if sources and load locate not far away from each other. On the contrary, plate heat exchanger possesses the shortest payback period of 1.14 year. The reason originates mainly from its least unit cost per volumetric flow rate which decreases investment costs. Moreover, payback period of rotary heat exchanger is placed in the middle of the classification with 1.99

years. Although rotary heat exchanger has the highest total annuity, high investment costs hinder shorter payback periods.

6.2 Recommendation for Facility 2: Plate heat exchanger

In Facility 2, an application of plate heat exchanger is calculated to be the most profitable waste heat recovery option. Total annuity of such an application is 7006 €/yr. Relatively low volumetric flow rate ranging from 6072 m^3/h up to 10120 m^3/h made it possible to decrease pressure drop, which was the main problem of plate heat exchanger in the previous case. In fact, current pressure drop leads again to highest requirement consumption related costs. Nevertheless, costs for compression of exhaust and supply air at a possible plate heat exchanger application is so alleviated that rotary heat exchanger cannot compensate by means of its higher annuity of revenue. Therefore, total annuity is kept high for plate heat exchanger and competition with rotary heat exchanger becomes possible. Furthermore, it is correlated with the theoretical data which expresses that plate heat exchanger is the most cost effective solution for volumetric rate up to 20000 m^3/h [3]. Scale down of the waste heat recovery in Facility 2 does not allow high investment and relevant capital related costs of rotary heat exchanger to be compensated. Thus, the total annuity of rotary heat exchanger is 6662 €/yr. The lowest total annuity of 5353 €/yr and longest payback period of 2.65 years belong again to run around coil as a consequence of its high investment cost and low recovered heat coefficient. Also in Facility 2 plate heat exchanger possesses the shortest payback period of 0.94 year as a result of previously introduced causes. Payback period of rotary heat exchanger stands again in the middle with 1.9 years.

On the whole, both investment of waste heat recovery are foreseen to be profitable and to reduce energy costs significantly. Moreover, all three equipments discussed here acquire very short payback periods in comparison with their lifetime. Both findings of analysis underline that, waste heat recovery is remarkable to decrease industrial end-use energy. In particular process waste heat, which is the main theme of this work, plays the major role in reduction of industrial end-use energy. Last but not the least benefit of waste heat recovery is to alleviate CO_2 emissions. Substituting energy costs with recovered heat will also decrease CO_2 emissions, which will provide competitive advantage to organizations for future global emissions trading.

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APPENDICES

APPENDIX A: Tables related with annual heat recovery

APPENDIX A.1

Table A.1: Annual average temperature data for Stuttgart (DIN 4710)

t_{A1} (°C)	Frequency z (h/yr)	t_{A1} (°C)	Frequency z (h/yr)
-18	0.2	9	4307.3
-17	0.8	10	4671.4
-16	1.7	11	5031.4
-15	3.2	12	5408.1
-14	6.3	13	5779.4
-13	11.1	14	6152.9
-12	18.7	15	6525.6
-11	28.8	16	6867.3
-10	43.5	17	7181.8
-9	60.7	18	7467.4
-8	83.1	19	7707.5
-7	112.4	20	7912.5
-6	153.9	21	8083.7
-5	208.7	22	8230
-4	276.7	23	8351.9
-3	366	24	8453.5
-2	486.9	25	8537.8
-1	656.8	26	8601.8
0	831.4	27	8653.4
0	1111.8	28	8689.2
1	1411.5	29	8718.5
2	1736.6	30	8734.4
3	2078.1	31	8751.5
4	2430.2	32	8758.7
5	2810.5	33	8762.5
6	3188.5	34	8764.5
7	3562	35	8765.2
8	3940.5	36	8765.3

APPENDIX A.2

Table A.2: Operating time factor

													To											
	Hour	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
	0	0,05	0,1	0,14	0,19	0,24	0,29	0,34	0,38	0,42	0,46	0,5	0,54	0,57	0,6	0,64	0,67	0,7	0,74	0,78	0,82	0,86	0,91	0,95	1
	1	1	0,05	0,1	0,15	0,2	0,25	0,3	0,33	0,37	0,42	0,46	0,49	0,52	0,56	0,59	0,62	0,66	0,69	0,73	0,77	0,82	0,86	0,9	0,95
	2	0,95	1	0,05	0,1	0,15	0,2	0,25	0,28	0,33	0,37	0,41	0,44	0,47	0,51	0,54	0,57	0,61	0,65	0,69	0,72	0,77	0,81	0,86	0,9
	3	0,9	0,95	1	0,05	0,1	0,15	0,2	0,23	0,28	0,32	0,35	0,39	0,42	0,46	0,49	0,52	0,56	0,6	0,64	0,67	0,72	0,76	0,81	0,86
	4	0,85	0,9	0,95	1	0,05	0,1	0,15	0,18	0,23	0,27	0,3	0,34	0,37	0,41	0,44	0,47	0,51	0,55	0,59	0,62	0,67	0,72	0,76	0,81
	5	0,8	0,85	0,9	0,95	1	0,05	0,1	0,14	0,18	0,22	0,26	0,29	0,32	0,36	0,39	0,42	0,46	0,5	0,54	0,57	0,62	0,67	0,71	0,76
	6	0,75	0,8	0,85	0,9	0,95	1	0,05	0,09	0,13	0,17	0,21	0,24	0,27	0,31	0,34	0,38	0,41	0,45	0,49	0,53	0,57	0,62	0,66	0,71
	7	0,7	0,75	0,8	0,85	0,9	0,95	1	0,04	0,09	0,13	0,16	0,19	0,23	0,26	0,3	0,33	0,37	0,4	0,44	0,48	0,52	0,57	0,61	0,66
	8	0,67	0,72	0,77	0,82	0,86	0,91	0,96	1	0,04	0,08	0,12	0,15	0,18	0,21	0,25	0,29	0,32	0,36	0,4	0,44	0,48	0,52	0,57	0,62
	9	0,63	0,67	0,72	0,77	0,82	0,87	0,91	0,96	1	0,04	0,08	0,11	0,14	0,17	0,2	0,25	0,28	0,32	0,35	0,4	0,44	0,48	0,53	0,58
	10	0,58	0,63	0,68	0,73	0,78	0,83	0,87	0,92	0,96	1	0,04	0,07	0,1	0,13	0,17	0,21	0,24	0,28	0,31	0,36	0,4	0,44	0,49	0,54
From	11	0,54	0,59	0,65	0,7	0,74	0,79	0,84	0,88	0,92	0,96	1	0,03	0,07	0,1	0,13	0,17	0,2	0,24	0,27	0,32	0,36	0,4	0,45	0,5
	12	0,51	0,56	0,61	0,66	0,71	0,76	0,81	0,85	0,89	0,93	0,97	1	0,03	0,07	0,1	0,13	0,16	0,2	0,24	0,28	0,33	0,37	0,41	0,46
	13	0,48	0,53	0,58	0,63	0,68	0,73	0,77	0,82	0,86	0,9	0,93	0,97	1	0,03	0,07	0,1	0,13	0,17	0,21	0,25	0,3	0,34	0,38	0,43
	14	0,44	0,49	0,54	0,59	0,64	0,69	0,74	0,79	0,83	0,87	0,9	0,93	0,97	1	0,03	0,07	0,1	0,14	0,17	0,22	0,27	0,3	0,35	0,4
	15	0,41	0,46	0,51	0,56	0,61	0,66	0,7	0,75	0,8	0,83	0,87	0,9	0,93	0,97	1	0,04	0,07	0,1	0,14	0,19	0,23	0,27	0,32	0,36
	16	0,38	0,43	0,48	0,53	0,58	0,62	0,67	0,71	0,75	0,79	0,83	0,87	0,9	0,93	0,96	1	0,03	0,07	0,11	0,16	0,2	0,24	0,29	0,33
	17	0,34	0,39	0,44	0,49	0,54	0,59	0,63	0,68	0,72	0,76	0,8	0,84	0,87	0,9	0,93	0,97	1	0,04	0,08	0,12	0,16	0,2	0,25	0,3
	18	0,31	0,35	0,4	0,45	0,5	0,55	0,6	0,64	0,68	0,72	0,76	0,8	0,83	0,86	0,9	0,93	0,96	1	0,04	0,08	0,12	0,17	0,21	0,26
	19	0,27	0,31	0,36	0,41	0,46	0,51	0,56	0,6	0,65	0,69	0,73	0,76	0,79	0,83	0,86	0,89	0,92	0,96	1	0,04	0,08	0,13	0,17	0,22
	20	0,23	0,28	0,33	0,38	0,43	0,47	0,52	0,56	0,6	0,64	0,68	0,72	0,75	0,78	0,81	0,84	0,88	0,92	0,96	1	0,04	0,09	0,13	0,18
	21	0,18	0,23	0,28	0,33	0,38	0,43	0,48	0,52	0,56	0,6	0,64	0,67	0,7	0,73	0,77	0,8	0,84	0,88	0,92	0,96	1	0,04	0,09	0,14
	22	0,14	0,19	0,24	0,28	0,33	0,38	0,43	0,48	0,52	0,56	0,6	0,63	0,66	0,7	0,73	0,76	0,8	0,83	0,87	0,91	0,96	1	0,05	0,09
	23	0,1	0,14	0,19	0,24	0,29	0,34	0,39	0,43	0,47	0,51	0,55	0,59	0,62	0,65	0,68	0,71	0,75	0,79	0,83	0,87	0,91	0,95	1	0,05
	24	0,05	0,1	0,14	0,19	0,24	0,29	0,34	0,38	0,42	0,46	0,5	0,54	0,57	0,6	0,64	0,67	0,7	0,74	0,78	0,82	0,86	0,91	0,95	1

APPENDIX A.3

Table A.3: Results of single frequency method regarding to ϕ for Facility 1

ϕ	Q_R	Q_V
0.50	2431.503	461.799
0.51	2469.249	424.053
0.52	2504.617	388.685
0.53	2539.936	353.366
0.54	2573.665	319.637
0.55	2606.343	286.959
0.56	2637.801	255.501
0.57	2667.735	225.567
0.58	2696.276	197.026
0.59	2723.295	170.007
0.60	2748.231	145.071
0.61	2772.104	121.198
0.62	2793.070	100.232
0.63	2812.530	80.772
0.64	2830.211	63.091
0.65	2845.167	35.427
0.66	2857.875	25.915
0.67	2867.387	19.131
0.68	2874.171	13.831
0.69	2879.471	9.724
0.70	2883.578	6.586
0.71	2886.716	4.255
0.72	2889.047	2.586
0.73	2890.716	1.476
0.74	2891.826	0.729
0.75	2892.573	0.305
0.76	2892.997	0.097
0.77	2893.205	0.021
0.78	2893.281	0.021
0.79	2893.301	0.001
0.80	2893.302	0.000

Q = 2893.302 MWh/yr

APPENDIX A.4

Table A.4: Results of single frequency method regarding to ϕ for Facility 2

ϕ	Q_{Rmin}	Q_{Rmax}	Q_{Vmin}	Q_{Vmax}
0.5	102.712	171.187	19.135	31.893
0.51	104.284	173.806	17.564	29.274
0.52	105.777	176.295	16.071	26.785
0.53	107.270	178.783	14.578	24.297
0.54	108.667	181.112	13.181	21.968
0.55	110.048	183.414	11.799	19.666
0.56	111.348	185.581	10.499	17.499
0.57	112.614	187.690	9.234	15.390
0.58	113.789	189.648	8.059	13.432
0.59	114.940	191.551	6.908	11.529
0.6	115.952	193.254	5.896	9.826
0.61	116.930	194.883	4.918	8.197
0.62	117.816	196.360	4.032	6.720
0.63	118.607	197.679	3.240	5.401
0.64	119.324	198.874	2.524	4.206
0.65	119.954	200.814	1.894	3.157
0.66	120.464	201.370	1.383	2.266
0.67	120.822	201.819	1.026	1.710
0.68	121.092	202.172	0.756	1.261
0.69	121.303	202.446	0.545	0.908
0.7	121.467	202.655	0.380	0.634
0.71	121.593	202.809	0.255	0.425
0.72	121.685	202.914	0.162	0.271
0.73	121.749	202.987	0.099	0.166
0.74	121.792	203.036	0.055	0.093
0.75	121.822	203.036	0.026	0.044
0.76	121.837	203.062	0.011	0.018
0.77	121.844	203.074	0.003	0.006
0.78	121.847	203.078	0.001	0.002
0.79	121.848	203.080	0.000	0.000
0.8	121.848	203.080	0.000	0.000

Q_{min} = 121.848 MWh/yr

Q_{max} = 203.080 MWh/yr