REPUBLIC OF TURKEY
ADANA ALPARSLAN TÜRKEŞ SCIENCE AND TECHNOLOGY UNIVERSITY

GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES
DEPARTMENT OF ELECTRICAL AND ELECTRONIC ENGINEERING

ANALYSIS OF THE BACKPRESSURE-SURFACE AREA IN SMOKE TUBE HEAT EXCHANGERS

ALPER ÜNLÜ
MASTER OF SCIENCE

ADANA 2022

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Asst. Prof. Dr. FIRAT EKİNCİ

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I hereby declare that all information in this thesis has been obtained and presented in accordance with academic rules and ethical conduct. I also declare that, as required by these rules and conduct, I have fully cited and referenced all information that is not original to this work.

[Signature]

Alper ÜNLÜ

ABSTRACT

ANALYSIS OF THE BACKPRESSURE-SURFACE AREA IN SMOKE TUBE HEAT EXCHANGERS

Alper ÜNLÜ

Department of Electric and Electrical Engineering

Supervisor: Assoc. Prof. Dr. Fırat EKİNCİ

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The design of heat exchangers is crucial to achieving the highest efficiency. Gas engine usage increases in the industry to cover electrical power requirements. The gas engine systems are equipped with heat exchangers to increase efficiency. However, there is a lack of study on backpressure and heat transfer coefficient. In this study, the effect of backpressure on heat exchanger calculations is shown and explained why they are crucial in heat exchanger selection. For this purpose, a shell and smoke-tube heat exchanger configuration for a gas engine is considered and thermal calculations are carried out. The dimensions of the heat exchanger are determined and according to the analysis, capacity values are shown. It is shown the increase in back pressure decreases the required surface area of the heat exchanger and the overall heat transfer coefficient of the system increases.

Keywords: heat exchanger, pressure drop, heat transfer, surface area,

ÖZET

DUMAN BORULU ISI DEĞİŞTİRİCİLERİNDE KARŞI BASINÇ-YÜZEY ALANININ ANALİZİ

Alper ÜNLÜ

Elektrik ve Elektronik Mühendisliği Anabilim Dalı

Danışman: Doç. Dr. Üyesi Fırat EKİNCİ

Ağustos 2022, 73 sayfa

Isı eşanjörlerinin tasarımı, sistem verimini artırmakta önem arz etmektedir. Bunun ile beraber her geçen gün elektrik gücü gereksinimlerini karşılamak için endüstride gaz motoru kullanımı artmaktadır. Endüstride kullanılan gaz motoru sistemleri, verimliliği artırmak için ısı eşanjörleri ile donatılmıştır. Ancak, literatürde karşı basınç ve ısı transfer katsayısı ile ilgili yeterince çalışma bulunmamaktadır. Bu çalışmada, karşı basıncın eşanjör hesaplamalarına etkisi gösterilmiş ve eşanjör seçiminde bu ilişkinin neden önemli olduğu açıklanmıştır. Bu amaçla, bir gaz motoru için duman borulu ısı eşanjörü konfigürasyonu tasarlanmış ve termal hesaplamalar yapılmıştır. Eşanjör boyutları belirlenmiş ve analize göre kapasite değerleri gösterilmiştir. Karşı basınçtaki artışın ısı eşanjörünün gerekli yüzey alanını azalttığı ve sistemin toplam ısı transfer katsayısının arttığı gösterilmiştir.

Anahtar Kelimeler: ısı eşanjörü, basınç düşüşü, ısı transferi, yüzey alanı,

Dedication

I would like to dedicate my thesis to my family.

ACKNOWLEDGEMENTS

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NOMENCLATURE

W	: Watt
MW	: Megawatt
ORC	: Organic rankine cycle
STHX	: Segmental section
AHP	: Absorption heat pump
OAHP	: Open-cycle absorption heat pump
VPB	: Vapor pump boiler
THX	: Total heat exchanger
CHX	: Condensing heat exchanger
CI	: Compression ignition
VGT	: Variable Geometry Turbine
WG	: Waste-gate
PLC	: Programmable flow controllers
rpm	: Rotation per minute
Hg	: Mercury
NOx	: Nitrogen oxide
C	: Carbon
CO	: Carbon monoxide
CO_2	: Carbon dioxide
S	: Sulfur
SO_2	: Sulfur dioxide
H_2O	: Water
N_2	: Dinitrogen
O	: Oxygen
$T_{ex,I}$	: Exhaust inlet temperature

$T_{ex,o}$	: Exhaust outlet temperature
m_w	: Water flow
cp_w	: Specific heat of water
$T_{w,o}$	: Water outlet temperature
$T_{w,i}$	: Water inlet temperature
Re_i	: Reynolds number of the internal flow
Re_e	: Reynold value of the external fluid
V_i	: Velocity of the fluid in the tube
V_e	: Velocity of the external fluid
d_i	: Internal diameter of the tube
d_e	: Outer diameter of the tube
μ_i	: Dynamic viscosity of inner fluid
μ_e	: Dynamic viscosity of external fluid
ρ_i	: Density of the fluid of inner fluid
ρ_e	: Density of the fluid of external fluid
Nu_i	: Nusselt number of the fluid in the pipe
Nu_e	: Nusselt number of the external fluid
Pr	: Prandtl number
Pr_s	: Prandtl value of the two fluids at the average temperature
h_i	: Heat transfer coefficient of the inner fluid
h_i	: Heat transfer coefficient of the external fluid
k_i	: Thermal conductivity coefficient of the inner fluid
k_e	: Thermal conductivity coefficient of the external fluid
V_{max}	: Velocity that the external fluid reaches between the pipe bundle
S_T	: Axis distance of the pipes perpendicular to the external fluid
S_L	: Axis distance between two rows of pipes
R_d	: Radius of the tube
V	: Velocity
ε	: Epsilon value

f	: Friction factor
L	: Pipe length
R_i	: Radius of the inner tube

1. INTRODUCTION

Energy is an important physical foundation of socio-economic advancement. Throughout the history of humankind, energy have always been an important part of life. The energy resources which have started with firewood have evolved into fossil fuels, oil, coals, and natural gas. With the advances in technology, new energy resources have flourished such as wind power, solar power, and hydropower. Energy will keep on playing a key role for humankind. As a motive power for modernization, energy is closely related to national prosperity as well as people's livelihood and human well-being. Countries all over the world now share a strategic goal to hasten the development of a safe, reliable, economical, efficient, clean, and environment-friendly modern energy supply system by capitalizing on a new round of energy evolution.

The importance of energy highlights the need to use resources efficiently. There are different ways to increase the efficiency of energy generation systems, and recovering the waste heat from the exhaust is one of them. For instance, natural gas engines work with an efficiency ratio of 40-45%, and these engines are hugely utilized in industry. The exhaust from these engines has high temperatures and high mass flow. If the exhaust flow is not utilized, it is released into the atmosphere with a pressure of 30 to 50 mbar. This energy can be recovered with the usage of a heat exchanger.

The waste heat from exhaust can be recovered with smoke-tube heat exchangers. In the selection of the heat exchanger, backpressure and surface area are the main parameters. The backpressure is important to select the most suitable heat exchanger as too much backpressure can shut-off the engine. On other hand, surface area determines the capacity of the heat exchanger and the amount of heat recovered from the waste heat. The calculations for determining backpressure and surface area require a thermal analysis.

Thermodynamics deals with the equilibrium states and how a state in equilibrium changes to another. However, heat transfer analysis which is done for heat exchanger calculation deals with systems in a nonequilibrium phenomenon. Thus, the analysis of heat transfer can't be solely based on thermodynamics. Nevertheless, the foundation for the science of heat transfer lies in the laws of thermodynamics. The first law of thermodynamics states that energy cannot be created nor destroyed, in other words, the rate of energy loss from one system is equal to the rate of energy gain of the other system. The second law of thermodynamics states that heat flow requires to be from a warmer body to a colder body. The main requirement for heat transfer is the temperature difference and there cannot be heat transfer between two bodies that are in thermal equilibrium.

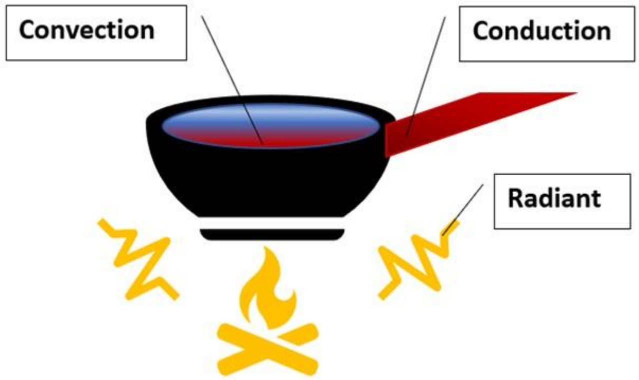

Figure 1-1. The diagrammatic representation of a ways of heat dissipation

Heat can be transferred in three different modes: conduction, convection, and radiation. All modes of heat transfer require the existence of a temperature difference, and all modes are from the high-temperature medium to a lower-temperature one.

Conduction heat transfer is the transfer of energy from the system with more energetic particles of a substance to a system with less energetic ones as a result of interactions between the particles. Heat transfer with conduction can take place in gases, liquids, or solids. Conduction in gases and liquids is caused by collisions and diffusions of the particles during random motion. Conduction in solids is caused by vibrations of the particles and the energy transfer from free electrons.

Conduction heat transfer rate through a system depends on the thickness, geometry, and material of the medium with the temperature difference across the system. Temperature is a physical quantity to measure the kinetic energy of the molecules or atoms of a substance. Kinetic energy for a gas or liquid is the random translational, vibrational, and rotational motions of the molecules. When two molecules with different kinetic energy collide, a portion of the kinetic energy of the molecule with higher energy is transferred to the molecule with less energy.

The cross-section of the material where heat transfer takes place and the length of the heat travel path plays an important role in heat transfer. The larger the size and length of a material, the higher the amount of energy required to heat that material. Also, the larger the surface area exposed to heat, the greater the heat loss. Therefore, small objects with small cross-sections have minimal heat loss.

The physical properties of the materials in the environment where heat transfer takes place determine which of the materials transfers heat better. Depending on the type of material and specifically, the thermal conductivity coefficient reveals that a conductive material will conduct heat better than an insulating material. As can be seen in the equation below, the heat transfer coefficient can be calculated:

$$Q = [U \times A \times (T_{outlet} - T_{inlet})]/d \tag{1}$$

Even if a fluid such as liquid or air heated by the heat source is removed from the heat source after being heated, it carries the thermal energy it is loaded with. The type of heat transfer that takes place with this charged energy is called convection. The fluid on a hot surface expands and becomes less dense and rises.

When the matter exposed to heat is examined at the molecular level, it is observed that the molecules expand as they are exposed to thermal energy. As the temperature of the heated fluid mass rises, the volume of the fluid increases in direct proportion to it. Due to this molecular effect on the fluid, there is displacement within the substance. The air whose temperature rises rapidly suppresses the cold air, which has less volume and higher density. As a result of these molecular events, it can be observed how convection currents are formed. The amount of heat transfer obtained per unit time by convection heat conduction can be calculated with the following equation:

$$Q = h_c \times A \times (T_{surface} - T_{fluid}) \tag{2}$$

Heat transfer by radiation, also known as radiance, occurs by the emission of electromagnetic waves emitted by the heat sources. These waves carry the energy away from the object from which it is emitted. Radiation occurs through a vacuum or any light-transmissible transparent medium (solid or liquid). Thermal radiation is the direct result of random movements of atoms and molecules in a matter.

All materials emit thermal energy according to their temperature. The hotter the inert material is, the more thermal energy it emits. The sun is an example of heat radiation that radiates heat throughout the solar system. At normal room temperature, objects propagate as infrared waves. The temperatures of objects also have an effect on the emitted wavelength and frequency. As the temperature increases, the wavelengths of the emitted radiation decrease so that shorter wavelengths with higher frequencies are emitted. The amount of heat transferred by thermal radiation can be calculated by the Stefan-Boltzman law:

$$P = e \times \sigma \times A \times (T_r^4 - T_c^4) \tag{3}$$

In this study, firstly, the theoretical modeling of–heat exchanger has been calculated then experimental studies are obtained. The relation between backpressure and heat transfer surface area is required to be investigated. For calculation purposes, a heat exchanger is designed and heat transfer analysis is carried out. Radiation was not taken into account in the calculations in this study as the heat exchanger is not exposed to flame. In the designed heat exchanger, only conduction heat transfer is considered. The system chosen as the subject of study is related to the recovery of waste heat. In such heat exchangers there is no active combustion, and therefore there will be no radiation heat transfer in it. For this reason, thermodynamic calculations of multiple hypothetical model heat exchangers, whose values are calculated for certain and same conditions, have been analyzed in computer environment in order to get fast results. Then, the data obtained from the field for the selected model were compared with the theoretical results.

2. LITERATURE REVIEW

The relationship between back pressure and surface area has been examined within the thesis's scope. During the thesis, a literature search and a review were made about heat exchangers, heat transfer, design, and auxiliary computer-aided programs from different articles and papers.

The aim of the studied study is to increase the heat transfer rate and to provide more evaluable concentration of the heat transfer path that occurs when a fluid flows in a serpentine tube in a tube heat exchanger. In the study, also observed different types of fluid flow, such as parallel flow through counter flow and heat exchange through the transition from laminar flow to turbulent flow. Numerical simulation has been carried out for nested tube coil heat exchanger and straight tube shell type heat exchanger, subject to different boundary conditions. Nusselt number, required pumping pressure, log average temperature difference, pressure drop variation according to Reynolds number due to different l/c ratios were investigated. As a result of the experimental and numerical study examined, it was observed that the Nusselt number of the inner tube increased with the increase of the Reynolds number. However, with the increase in the flow rate, the increase in turbulence between the fluid element, which will increase the mixing of the fluid, and finally the increase in the heat transfer rate with the Nusselt number has been investigated (Gayakwad, Nitnaware and Tapobrata, 2016).

Within the scope of the study, the development studies for the design of the fire tube hot water boiler with a capacity of 3 MW for different fuels (crude oil, diesel fuel and natural gas) were examined. During the review it was observed that the existing normative methods of heat calculation of boilers were created mainly for water-tube boilers with high thermal capacity, so it was discovered that the creation of a computer model should solve many problems. In the content of the study examined, the Russian program OPTSIM-K was used for the boiler design program and it was observed that a computer model of a fire tube hot water boiler was developed on its basis. With this program, it has been observed that the design of a new boiler with a thermal capacity of 3 MW has been successfully completed (Roslyakov et al., 2018).

Within the scope of this study, it has been observed that the thermo-hydraulic performance of segmental section (STHX) shell and tube heat exchangers is numerically examined for different section sections. In the study content, heat transfer coefficient and pressure drop were considered as key parameters. These two parameters depend on the fluid flow path, which is affected by the chamber characteristics (type, spacing, and chamber cut). It has been observed that the baffles are used to improve heat transfer on the shell side, but it has been noticed to increase the pressure drop, thus requiring a higher pumping power. In the content of the study, it was observed that a 3-dimensional numerical study of a shell and tube heat exchanger was made for different partition sections using the fluid analysis program Ansys-Fluent. As a result of these observations, it has been seen that the heat transfer coefficient and pressure drop are sensitive to the selection of partitioned section. It has been learned that the optimum performance can be achieved with a 25% partition cut (Ihssane et al. 2018).

In the study that was inspected, a small-sized liquid-to-liquid heat exchanger was designed and manufactured to be used for cooling purposes for servers in a data center. The designed heat exchanger was found to be designed under given conditions, including space constraints between racks, heat exchange capacity requirement, and energy efficiency and effectiveness requirement. It has been observed that the designed and manufactured heat exchanger performs 548W heat transfer under the specified conditions and meets the heat dissipation capability requirements (500W) of the developed cooling system. Meanwhile, the developed heat exchanger has been observed to show high efficiency and high energy efficiency of 0.9 ratios. In addition, it has been observed that the results obtained by the simulations are well matched with the measurement results showing that they are valid for the designed and manufactured liquid-liquid heat exchanger development (Gongyue et al. 2018).

The article that read, presents a waste heat recovery system specifically designed as a dipper for ICE in different power ranges for marine applications, using a simple Rankine cycle or an additional bottom Rankine cycle called ORC. This particular application under study shows several advantages for the installation of a waste heat recovery system, and in particular, the essentially infinite availability of the cooling medium represented by the sea of water has eliminated any problem in condenser design. It has been observed that a steady-state model of the system has been developed by means of the process simulator CAMEL-Pro™ to determine the most suitable configuration from the thermodynamic point of view of the model under consideration. According to the calculations made in the reviewed article, it has been seen that regeneration is a suitable method to increase the ORC efficiency, while at least the heat is recovered from a low-temperature source. Although it does not bring advantages in terms of power output for the waste heat recovery model analyzed in steady and stable state (Asfaw et al. 2015).

The article reviewed within the scope of the literature review is about the simulation model of the heat exchanger and its comparison with the real equipment. In the reviewed article, the mathematical model of the heat exchanger created with the balancing equations represents the simulation model created in Matlab Simulink using the s-function. Real data were obtained with prototype equipment in the laboratory environment and it was observed that they were compared with the simulation results. The main function of the model under consideration is to rearrange the outputs of the heat exchanger with the suggested inputs with as high probability as possible. The heat exchanger model, which is discussed in the article, mainly describes the change in ambient temperature during the heat transfer from one medium to another and during the transition from the heat exchanger. As a result of the examined article, it was observed that after the real temperature was compared with the model data, the simulation model of the heat exchanger showed the differences between the theoretical and the data obtained by the model. In the real-time process, there are heat transfer coefficients and heat capacity function of temperature. It has been seen that the theoretical temperature can reach the steady state of the actual temperature after allowing the specified correction of the model. Tubular exchangers are the most widely used heat exchangers in the industrial sector. It is essentially divided into two most common categories: shell and tube heat exchangers and coaxial exchangers. It is also possible to consider a shell and tube exchangers as a sum of simple coaxial tubes, which greatly facilitates the understanding and modeling of shell and tube exchangers. Currently, the optimization of heat transfer in exchanger is a recurring problem for thermal engineers. Therefore, several numerical and experimental studies have been carried out. Mounaam, A. and his colleagues work on the experimental and numerical study of heat transfer and dynamic behavior analysis of tubular heat exchanger is carried out in order to optimize the energetic performance of the exchanger. In this study, a simplified shell and tube exchanger with coaxial tubes is used. The heat exchanger disposes tubes, one in the other. The internal tube contains the recent water circuit and the alternative tube contains the cold-water circuit. Heat waft is exchanged among the 2 elements via way of

means of a median warmth switch area. Mounaam, A. et al. are studied on configuration: parallel flow and counter-contemporary flow. (Mounaam et al. 2015a).
Mounaam, A. et al. target to analyze the thermal and fluid flow conducted inside a tubular heat exchanger. In their article, an experimental and theoretical observation changed carried out for constant and dynamic states of coaxial tubes HE. The experimental observation changed into execution for 2 exchanger configurations: co-current and counter-current. Two examples were evolved and simulated in MATLAB/Simulink. In steady cases, the examples are estimated hot and cold outlet temperatures. In addition, the dynamic examples estimate the time regular and time rise. The example has been proven over an extensive variety of inlet temperatures and goes with the flow rates. This research gives a correct and quick approach for modeling and simulating a tubular heat exchanger. This approach lets in the prediction and optimization of the heat exchanger conduct. (Mounaam et al. 2015b).

For medium-low temperature flue gas, the technology for energy recovery from exhaust gas with waste heat recovery systems has been studied in detail over the last decade. In this studied paper, the technology of heat recovery systems, their working mechanism, and differences were reviewed at length. Boiler efficiency can be increased for a condensing boiler in instances where the waste heat energy of exhaust is used to heat the feedback water. Though it should be noted, the temperature difference in heat exchangers plays a huge role in the effectiveness of the heat transfer, and in most systems, the feedback water may not be cold enough to effectively recover the heat from the exhaust, which in return, decreases the efficiency of heat recovery exchangers in such systems. To overcome this problem, complex waste heat recovery systems were studied and proposed, which are reviewed in this paper. The general principle for these systems is to control the temperatures of the exhaust gas and the feedwater to increase the temperature differences of both fluids. These reviewed systems are categorized with respect to their approach to the problem: (1) decreasing the feedwater temperature; (II) decreasing the feedwater’s dew point temperature; (III) increasing the exhaust gas’s dew point temperature. These different systems are studied for their efficiency, entransy dissipation, pollutant emissions, and for their economic benefits (Men et al. 2021).

For systems using the first principle, decreasing the feedwater temperature, two heat pumps with different purposes and different specifications can be introduced to heat recovery systems. One of these pumps is the absorption pump. The absorption pumps can be driven with different sources which are hot water, flue gas, steam or by natural gas. The second pump that is introduced to the system is compressed heat pump which is driven by electricity. The compressed heat pump system includes electric compressor, condenser, evaporator and finally expansion valve as shown in Figure 2.1. The working fluid of the compressed heat pump system, circulates between these equipment in a closed cycle. The main principle for this system is to heat the working fluid with the energy of waste heat from exhaust and further increase the temperature and pressure of the working fluid with the compressor. The working fluid that reaches high temperature is introduced to feedback water in the condenser and heats the feedback water. It should be noted that the compressor uses electric motor and requires an additional electricity consumption in order to drive the motor. In 2014, Hebenstreit et al. studies compressed heat pump systems with different boiler systems and shown that the efficiency increases from 3% to 21% (Hebenstreit et al., 2014). Compared to conventional heat recovery boiler, the compressed pump arrangement can cool the flue gas to lower temperatures. The second pump option is absorption heat pump (AHP) and it is more popular compared to compressed heat pumps as it can be driven by steam, natural gas, flue gas or hot working fluid with high temperatures. AHP systems includes similarly a condenser, an evaporator which can be seen in compressed heat pump systems but also includes a generator and an absorber as shown in Figure 2.2. In AHP systems, the evaporator is also used to heat the working fluid using the waste heat from the exhaust gas. The condenser and absorber is used to heat feedback water using the heat recovered from exhaust gas that is carried with the working fluid. The AHP system and efficient heat transfer equipment together can decrease the temperature of the exhaust to 27 °C (Lin et al., 2003).

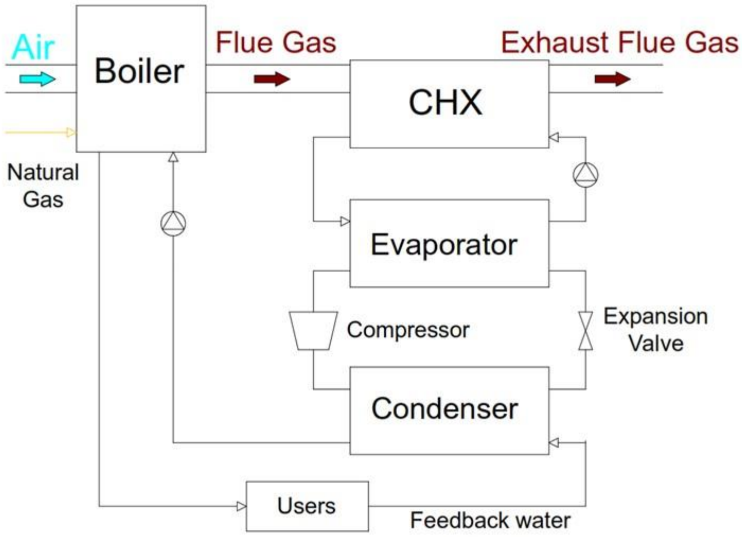

Figure 2-1. The heat pump system schematic diagram.

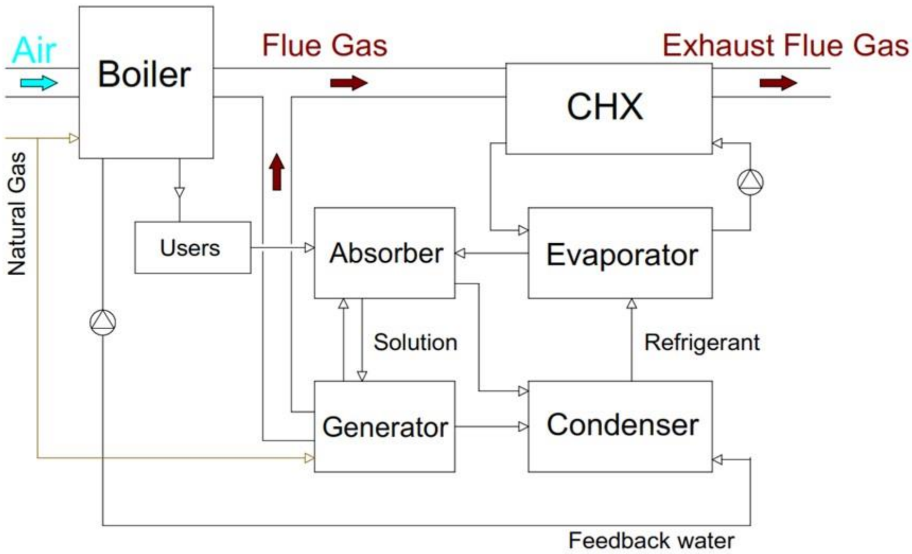

Figure 2-2. The absorption heat pump system schematic diagram.

For systems using the second principle mentioned above, the feedwater's dew point temperature is decreased. The decrease in the dew point temperature increases the driving force between two fluids, generally, in heat recovery systems these fluids are exhaust and feedwater, which increases the amount of waste heat recovered from the boiler exhaust, thus increasing the efficiency of the system. This principle can be achieved using an open-cycle absorption heat pump (OAHP) (Lazzarin et al. 1992). In OAHP systems, working fluid is in a closed-cycle and mainly consists of an absorber, a condenser, and a generator to achieve the desired cycle. The working fluid is a fluid with a low vapor pressure compared to water's vapor pressure at the same temperature (Riffat et al. 2006). Dehumidification takes place in OAHP on the exhaust side with the contact of the exhaust and working fluid. The water vapor of the exhaust is transferred to the working fluid, this process takes place in the absorber. The working fluid combined with the water vapor of the exhaust is needed to be heated and this is done in the generator. Similar to the APH systems, the OAHP systems can be driven with different heat resources such as hot water and steam. The main advantage of the OAHP system is that compared to conventional waste heat recovery boilers, it is capable of recovering more waste heat from the flue gas as is less limited to the effect of the dew point temperature of the flue gas. OAHP systems can achieve high efficiency, especially in high humidity ratio systems due to the dehumidification characteristics of the system (Wang et al. 2016).

Another approach to increase the heat recovery efficiency is to increase the dew point temperature of the exhaust. Increasing the dew point temperature of the flue gas increases the driving force between the flue gas and the cold source. This principle can be achieved by introducing a vapor pump boiler (VPB) to the system. There are two different heat exchangers in a VPB system, a total heat exchanger (THX) and a condensing heat exchanger (CHX). The THX provides the heat recovery from the exhaust gas by heating up air and CHX provides required condensation. In VPB systems, THX will heat the the inlet air of boiler and also helps with humidification of the air, this additional humidity of the air also reaches and effects the humidity of the exhaust gas. This increases exhaust gas dew point temperature and heat recovery by feedback water increases. This system is first demonstrated in theory (Kuck et al. 1996), in demonstration it was shown that VPB system decreases the temperature of exhaust gas to 29.9°C and boiler achieve and increased efficiency of 8.7 % compared to a heat exchanger system only with condensing boiler.

To investigate the performance and emissions of compression ignition (CI) engine under different speed and load conditions for the effect of engine backpressure, a study was done using a direct injection diesel engine investigation with an engine speed of 600, 950, and 1200 rpm with the backpressure of 0,40,60 and 80mm of Hg. In this study, two parameters were measured during the operation, the first being the engine performance and the other the exhaust emission (NOx, CO, and odor). With Hg (mercury) manometer, engine backpressure produced by the flow regulating valve in the exhaust line was measured. The exhaust emission was measured by a flue gas analyzer. From this experimental investigation, the following conclusions were presented. Firstly, the engine speed has no significant effect on engine performance for low engine speeds, however, at medium and high speeds, brake- specific fuel consumption is a little higher for all load conditions. The second conclusion was that CO is higher for low engine speed in all load conditions, however, with increased backpressure for higher engine speed and loads, CO decreased. Thirdly, with increased backpressure for various load conditions, NOx emission always decreased at low, medium, and high engine speeds. Lastly, in terms of odor magnitude, backpressure has no significant effect up to 40 mm of Hg pressure. However, above the backpressure of 40 mm Hg, odor is increased. Higher backpressure and engine load always increased the exhaust gas temperature (Roy et al. 2010).

The combustion from the solid fuels and natural gas results on a flue gas composition that varies with the fuel composition. It is crucial to examine both the fuel composition and the resultant exhaust composition as it effects the efficiency of combustion. The combustion requires air to for a full combustion and the amount of air to be introduced to burner becomes important and the required air is depended on the fuel composition. In addition, determining and monitoring the exhaust gas composition is important for CO emission reduction and environmental pollution reduction. The ideal combustion where full fuel combustion is achieved with minimum quantity of air is called theoretical combustion or stoichiometric. When oxygen is insufficient, incomplete combustion occurs. With respect to the composition of the fuel used in combustion, combustion gasses may contain CO, CO2, SO2, H2O, N2 as shown in Figure 2.3. To get better insight on flue gas, the composition of flue gas should be determined accurately, this is crucial to use correct flue gas properties for waste heat recovery boilers. By the mass conversation law, in combustion reactions, mass of the reactants and the mass of resulting products must be equal. On top of that, during reaction, total mass of elements do not differ and is preserved in resulting products. While the exact needed combustion air can be calculated for a fuel, as a common practice in real life applications due to burners imperfection, more air is introduced to the combustion process. This is done to achieve complete combustion. In this paper studied, a web application is presented to analyze the fuel composition and resultant exhaust gas composition for different excess air coefficients and variable fuel flows, while presenting the advantage of avoiding any errors that can be done during laborious calculations and quicker calculation (Paraschiv et al, 2020).

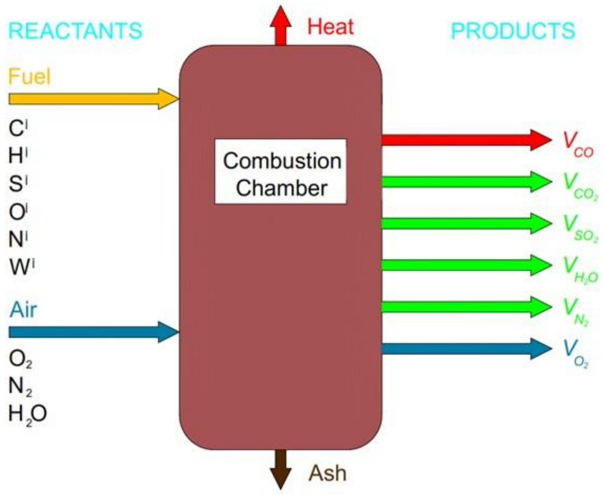

Figure 2-3. The solid fuel combustion.

To increase the rate of heat transfer in tubed applications, it is possible to employ turbulence promoters such as helically coiled wires or twisted tapes fitted inside tubes. This may be useful either to reduce the size of the proposed equipment or to increase the rate of heat transfer of current equipment. In this reviewed study, the usage of coiled wire turbulence promoters was observed to increase the heat transfer up to %280, though in terms of heat transfer per unit frictional power loss, coiled wire turbulence promoters are less effective than an empty tube. The increased frictional power loss was observed to be much greater than the increased heat transfer. Thus, where pumping power is not of great importance and the dominating factors are to achieve a reduction in size and weight of the equipment, usage of helically coiled wire turbulence promoters can be effectively employed (Kumar et al. 1970).

In power generation sectors, the usage of heavy fuel oil diesel engines attracts growing interest with waste heat recovery technologies achieving higher efficiency and decrease in fuel consumption. In that regard, Michos et al. investigated Organic Rankine Cycle (ORC) evaporator for the backpressure effect with different strategies. The ORC evaporator investigated in this study was an ORC evaporator on the exhaust line of a turbocharged, V12 heavy-duty diesel engine for typical power generation applications with the usage of commercial software Ricardo WAVE for the analysis of the backpressure effect. In order to counterbalance the increased pumping losses of the engine with the boiler installation, three different turbocharging strategies were evaluated: Variable Geometry Turbine (VGT), Waste-Gate (WG), and fixed turbine. The thermodynamic process analysis and optimization of the ORC systems and engine were presented with the purpose of utilizing the best configuration to increase the overall powertrain full efficiency. In the case of a fixed turbocharging system, the engine has to repress bigger pumping losses, and more fuel is required to be consumed while holding at a constant brake load, causing an increase in engine brake-specific fuel consumption, on the other hand, compressor pressure ratios are lessened, wherefore reduces the air mass flow in the engine. The trapped Air-Fuel-Ratio is decreased, worsening the quality of combustion. In addition, the increased exhaust temperatures in result increase thermal loading which causes the thermal failure of the cylinder heads, valves, and pistons. An option for a fixed turbocharger with the aim of lowering exhaust temperatures and preserving the trapped Air-Fuel-Ratio is the increase the boost pressure with a smaller turbine. This small turbine is equipped with a WG valve in order to bypass a portion of the exhaust gas when the requirements for increased boost pressure are lowered. This alternative, in contrast to the fixed turbocharger case, increases the pumping losses, which increases fuel consumption to preserve the steady brake load, thus causing furthermore in increased brake-specific fuel consumption. While maintaining the base case Air-Fuel-Ratio, to lessen the risen brake-specific fuel consumption with the usage of a smaller turbine of WG turbocharger, a VGT turbocharger is used. VGT turbocharger turbine nozzle area is modified between the two severe cases in the intermediate back pressures. In terms of engine efficiency, the VGT turbocharger has higher performance compared to WG for the studied ORC backpressure effect. Thus, in terms of engine operation, it is expressed that the VGT turbocharger was the most favorable solution. In the study, the largest fuel economy benefits were obtained with a VGT turbocharger on the engine side with recuperated cycle operation on acetone. Recuperated ORC arrangements also show efficiency advantages over their simple cycle counterparts, causing higher exhaust gas temperatures at the outlet for further heat recovery configurations (Michos et al. 2017).

Current gas cogeneration systems contain many issues regarding efficiency, with exhaust temperature of flue gasses generally above 90 °C. In order to lower the exhaust temperatures and increase efficiency of cogeneration system, Li et al. introduced technique that is based on absorption heat-exchange for flue gas waste heat recovery in systems with gas cogeneration.

With aim of lowering the temperature of water in the heat network to about 20 °C, absorption heat-exchange units were set up. This system heated return water orderly by gas-water heat exchanger, peak load heater and absorption heat pump. In the paper, this was compared with traditional systems and it was reported that systems with newly introduced technique the delivery capacity of the heat network increases greatly as the system runs with increased circuit temperature drop causing the improved system to have increased heating efficiency of 38.3% above against its conventional system. With additions of absorption heat-exchange, initial cost of gas cogeneration system increases, however with the increased efficiency, it is reported that static investment-increment payback period is around 4 years, thus giving an economical solution. With increased efficiency, the CO_2 emissions has also been reduced drastically. As a result, absorption heat-exchange gives a valid solution to consider with its advantages on economics, environmental protection and energy savings (Li et al. 2016).

Wu et al. studied the exergy transfer and the effectiveness of the transfer for the performance description of heat exchangers operating above/below the temperature of the surrounding with or without finite pressure drop. Heat transfer units' number, flow patterns on exergy transfer, and the ratio of working cold fluid heat capacity to that of hot fluids effects were discussed systemically. For a heat transfer process such as a tubular heat exchanger, the variation of net exergy consists of two parts: lost exergy due to friction of fluid flow and the thermal exergy changes due to heat transfer across a finitely varied temperature difference. It was concluded that the exergy transfer effectiveness for the heat exchangers was not only available for the systems that are operating above the surrounding temperature, but also for the systems that are operating below the surrounding temperature. In addition, pressure drop effects on heat exchange are stated to be safely negligible when the discussed medium is incompressible liquid and the heat exchanger operating temperature is above/below the surrounding temperature, but this is not the case when discussed medium is an ideal gas. Parallel flow heat exchanger affected greatest from the pressure drop, with the second being cross flow and the least being counter flow (Wu et al. 2007).

The heat transfer characteristics differ from heat exchanger construction and for a tube-in-tube helical heat exchanger, the heat transfer characteristics and the hydrodynamics were studied in the paper of Kumar et al. in 2006. They have carried out experiments with cold fluid in the outer tube area and hot fluid in the inner tube side in counter flow operations, where outer tubes were welded to plates to support the inner tube which also provided high turbulence in the outer tube area. The friction coefficient and the heat transfer data for fully developed are from numerical predictions and experimental presented work were compared to the available experimental works in the literature. With the experiment, overall heat transfer for different flow rates in the inner-coiled tube and the annulus, region obtained were reported. With gathered data, it was deduced that the heat transfer coefficient increases with higher inner tube Dean numbers for a constant flow rate observed in the annulus region. Similar to this observation, for a constant flow rate in the inner-coiled tube, variation of overall heat transfer coefficient was observed in the annulus region for different flow rates (Kumar et al. 2006).

A study done by Mahulikar et al. highlights certain principles for fluid friction physics with their study on fluid friction in compressible laminar convection. It is widely believed that when the Stanton number increases with increasing skin friction coefficient, Reynolds' analogy between the Stanton number and the skin friction coefficient holds for simple situations. In the investigation, the validity of Reynolds' analogy for skin friction coefficient between Stanton's number of liquids in different variations of fluid properties is re-examined. This led to finding Sieder-Tate's property-ratio methodology for obtaining Nusselt number corrections which are in theory based on the validity of Reynolds' analogy. The basis of validity for Reynold's analogy, in convective flows with various fluid properties, is the inverse dependence of skin friction coefficient and Reynolds number. This leads to a different outcome where Reynolds' analogy now results in Stanton number increasing with decreasing skin friction coefficient. Also in this study, the Theorem of Minimum Entropy Production applicability was linked to the validity of Reynolds' analogy. The region of invalidity of Reynold's analogy was connected to the hydrodynamic development of the flow phenomenon (Mahulikar et al. 2008).

San et al. investigated the heat transfer and fluid friction correlations for circular tubes that has inserted coiled-wire through experiment. In this experiment, the wire diameter-to-tube inner diameter ratio ranges were selected as 0.00725 to 0.134, and the coil pitch-to-tube inner diameter ratio ranges were selected as 1.304 to 2.319. In this experiment it was found that the Nusselt number increases with the wire diameter-to-tube inner diameter ratio value, whereas it increases with a decrease of the coil pitch-to-tube inner diameter ratio value. In experiment with the air as working fluid, it was reported that the dependence of the heat transfer enhancement of using the wire coil inserts on circular tubes on the Reynolds number was minor, whereas with water as working fluid, increase of the Reynolds number value decreases the heat transfer enhancement. With this experiment, wire diameter-to-tube inner diameter ratio and coil pitch-to-tube inner diameter ratio were proposed as 0.101 and 2.319, respectively, to effectively and efficiently enhance the heat transfer where the working fluid is air; for the water, the proposed values were 0.101 and 1.739, respectively (San et al. 2015).

Generalized equations used to describe the dynamics of mass flow and exchange of thermal energy are complicated for forced convection heat transfer by being nonlinear. These mathematical difficulties caused theoretical investigations for the fluid flow around and heat transfer from cylinders to be mainly focused upon asymptotic solutions. While these solutions are well documented in the open literature for very large and small Reynolds number ranges, there are no theoretical investigations to determine average heat transfer and drag coefficients from cylinders for low to moderate Reynolds numbers, including for large Prandtl numbers. The solution for this range of Reynolds numbers mostly relies on experimental or numerical methods. However, these approaches are hugely time-consuming, and expensive, and achieved results are applicable for a fixed range of conditions. In a study by Khan et al. a circular cylinder is considered in cross flow to analyze the heat transfer and fluid flow from a cylinder for a wide range of Reynolds and Prandtl numbers. With the integral approach employed, closed form solutions were developed for the heat transfer coefficient and the drag in terms of Reynolds and Prandtl numbers. For both isothermal and isoflux boundary conditions, heat transfer correlations were developed. Finally, in this paper, it was shown the equations presented were in close agreement with the experimental results for the studied range of Reynolds number in the absence of any blockage and free stream turbulence (Khan et al. 2005).

Size factor in industrial solutions may be critical where solutions need to be in small spaces. This being relevant for heat exchangers, it might be required to apply improvements in design in order to achieve the same heat transfer capacity with a smaller heat exchanger. One of these methods for the tubed heat exchanger is to use corrugated, fluted, finned, or ribbed tubes. However, these deformations on the tube increase pressure drop as it increases heat transfer efficiency as well as an increase the cost of manufacturing, thus it is required accurately study a selection of best approaches for the modified tubes. In these regards, Pethkool et al. investigated helically corrugated tubes to find the maximum thermal performance factor for the pitch ratio and rib-height ratio. Experimenting with over a wide range of turbulent flow with Reynolds number of 5500 to 60000 using water as working fluid, pitch ratio, and rib-height ratio configurations were studied. It was found that the Nusselt number, thermal performance factor, and friction factor were increased with the increase of pitch ratio and the rib-height ratio. In this study, it was reported that the maximum thermal performance factor of 2.33 was obtained for the tube with a rib-height ratio of 0.06 and pitch ratio of 0.27 for low Reynolds number where Nusselt number and friction factor were 2.14 times above the smooth tube (Pethkool et al. 2011).

Flue gasses in the process industry contain waste heat energy that can be utilized in order to decrease the cost of energy and emission. Through industrial condensing boilers, it is possible to achieve higher efficiency levels than conventional boilers by recovering the latent heat of water vapor in the flue gas. In addition, with particle removal, condensing boilers can help decrease emission as the boiler can also be optimized for emission abatement. However, for condensing boilers applications, there are two technical barriers: corrosion and return water temperatures. Chen et al. investigated condensing boilers with a single pass shell-and-tube condensing heat exchanger that requires approximately 4900 m2 of total heat transfer area in the case of using stainless steel as a construction material. To decrease the cost of production, the same case was studied with the assumption of using carbon steel as a construction material and using polypropylene coating as the corrosion-resistant coating material outside the tubes. With the addition of polypropylene coating, the required heat transfer area increased to approximately 5800 m2. Comparing two configurations, using a carbon steel condenser was presented to ensure cash return in a relatively shorter period of time when it is compared to a stainless-steel condenser (Chen et al. 2012).

Major source of global energy losses is represented by waste heat and Firth et al. reported a method to quantify future global waste heat emissions from the industry, transport, power generation and buildings sectors in their research paper and investigated their environmental effects. By simulating four projected energy landscapes to assess the amount of waste heat produced for given sub-sectors in the year 2030. In this study, quantity of waste heat was found to be significant in all years and scenarios, accounting for up to 53% of global input energy, which leads to a global theoretical waste hear recovery potential of up to 12% and economic potential of 6-9%. It is reported that the Transport sector was found to be largest sector for recoverable waste heat, with 44% of the global theoretical potential, as shown in Figure 2.4, with second largest being Industrial sector with 27%, followed by Buildings with 10% and lastly Power Generation sector with %10 (Firth et al. 2019).

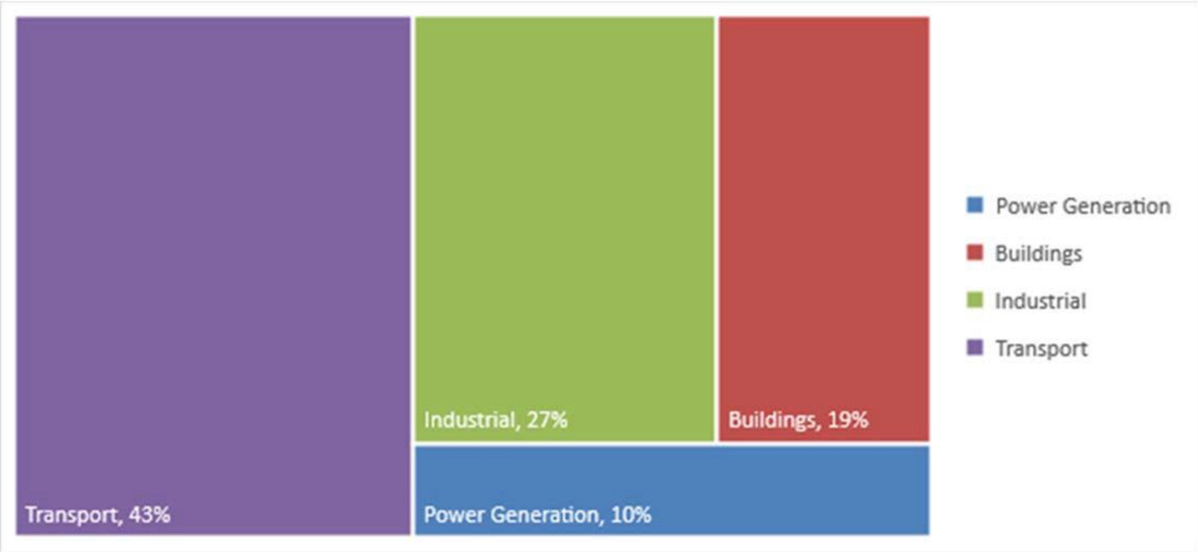

Figure 2-4. Global theoretical potential recoverable waste heat for sectors

The temperature of the waste heat plays a key role in heat recovery. The heat recovery for high temperature waste heat is always utilized, however medium-low temperature waste heat recovery is sometimes overlooked. Compared to high-temperature, it is harder to recover heat from medium-low temperatures. That being said, a huge portion of waste heat comes from medium-low waste heat sources. This increases the importance of heat recovery from medium-low temperature waste heat sources but it also brings more challenges. Figure 2.5 shows that the discharged waste heat is hugely portioned for low temperatures with 66% of the total waste heat is below 473.15 K (Haddad et al., 2014). Main source of combustion in industrial boilers are steam boilers which are gas boilers. The gas boilers use natural gas with high methane ratio, which increases the water vapor of the resultant exhaust air up to 15% of the volume and the temperature range of exhaust is around 150-200 °C. This recoverable heat, both the heat from the temperature and from the condensation heat, is discharged to the environment. The recovery of this waste heat plays a key role in gas boilers efficiency, which makes the waste heat recovery an important factor on the improvement of the energy utilization and energy conservation (Osakabe, 2000).

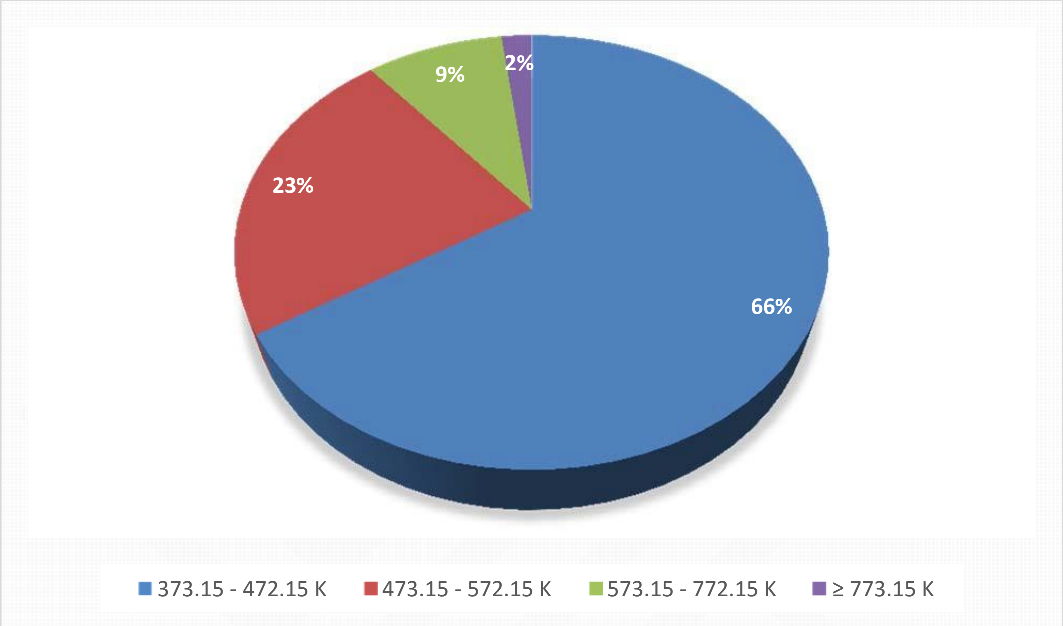

Figure 2-5. The waste heat temperature distribution in industry (60TWh)

Increasing energy production to meet the increasing energy demand with developing technology has caused environmental destruction for the world, and threatened human health and environmental health, and the ecological balance of the world with the developing industrial revolution. In order to use the energy produced more efficiently to meet the required energy needs, cogeneration systems, which are called the combined heat-power system, are used. Abdurrahman et. al. has analyzed cogeneration systems for small-scale capacities so that they can become more adaptable to our lives. Thus, the cogeneration system that will meet both electricity and heat needs for places such as homes, schools, hospitals, and shopping centers has been analyzed using Matlab-Simulink. According to Matlab simulation results, a model with high cogeneration efficiency that can be used in homes, hospitals, and schools has been examined. With the model examined, it has been observed that energy efficiency can be increased by installing the cogeneration system in places such as homes, universities, and hospitals, and carbon emissions will be reduced thanks to this micro-cogeneration system (Abdurrahman et al. 2019).

The study in which the smoke tube heat exchanger is used for cooling and the water in the biogas is separated by condensation has been examined. Alper analyzed the biogas condenser, which increases the fuel quality, in order for the internal combustion engines to use biogas as fuel to have a longer life. Biogas rich in water enters the heat exchanger designed within the scope of the study and cold water is circulated on the body side. Thus, it has been observed that the biogas in the gas phase is condensed and water is removed. The point of attention in the design of the system is that it has been observed that a design that will be low in terms of back pressure is made so that the whole system does not lock up during operation. It has been observed that the surface area that gives the desired capacity with limited back pressure is calculated and it has been observed that the analyzes are made in Solid. As a result of the analysis, it has been seen that it is possible to extend the life of the engine by preventing the system from locking and increasing the fuel quality by condensing the fuel to be burned (Alper et al. 2019).

Within the scope of the study examined, it was seen that the researchers aimed to experimentally and numerically examine an adsorption cooler with a new combined condenser-adsorbent infrastructure. With this experimental study, it has been observed that the experimental cycle of the adsorption cooler has been successfully achieved. In the numerical study, the effects of desorption temperature and condenser pressure on the performance values of the adsorption cooler were investigated. The numerical results obtained show that increasing the desorption temperature and decreasing the condensing pressure is successful in increasing the performance indicators. In order to further elaborate this study, a simulation study was carried out in Simulink. Study 2 focuses on improving the operating and construction parameters that affect the cooling performance of the new adsorption chiller with a combined condenser-adsorbent bed in the Simulink environment. It was seen that the effect of the variables on the new adsorption chiller was taken into account in the Simulink environment. The chiller's cooling performance has been observed to be analyzed on Simulink using simulations of the central composite design and response surface methodology. The effects of the amount of cooling/heating water of the adsorbent, the amount of chilled water in the evaporator, the evaporator heat exchangers, and the adsorption chiller on the cooling performance of the building material were analyzed with a dynamic-based model developed using MATLAB/Simulink. (Gamze et al. 2021).

Systems engineers are often faced with many complex flowcharts in their industries. System tools are needed to evaluate and control the correct performance of flow loops. In this reviewed article, it is seen that the approach to identification procedures in monitoring control performance for PLC application and PID control. A second study, a study using the MATLAB program for performance monitoring at a higher level of DCS, was examined. These programs are mainly implemented on PCs. As a result of the two studies examined, the opportunity to examine the application of identification procedures and control performance monitoring functions in programmable flow controllers (PLC). (Stepan et al. 2012).

2.1. Heat Exchangers

2.1.1. General Concepts

Systems designed to transfer heat energy from one environment to another are called heat exchangers. Heat exchangers in which two fluids come into direct contact and mix with each other are generally called direct heat exchangers. In these, the necessary heat transfer surface, edge planes of liquid droplets or liquid films, etc. is provided. Heat exchangers in which the two fluids are separated from each other by a partition wall that provides heat transfer are completed as indirect heat exchangers. The intermediate wall is called the heating or cooling surface.

Type of heat exchanger: according to its functions; It can be a heater, cooler, evaporator, condenser, superheater, economizer, hiter, and recuperator. According to the working mode of the heat exchanger, if the fluids that give off and receive heat flow through the same flow channel one after the other, it is called a regenerator. In order to meet the requirements of the process in a plant, special demands are applied in the design of the heat exchanger. Therefore, many types of heat exchangers have been developed.

2.2. Usage Areas

There are many types of heat exchangers, from automobile radiators to cooling towers, which have a wide range of applications in practice. Therefore, it plays a major role in applications in the industry. It has a very important place in heat treatment technique, which is the common application area of especially heating-cooling plants, power machines, thermal power plants, chemical plants, oil refineries and mechanical and chemical engineering.

Hot water is needed for comfort in residences and industry. With the heat exchanger, hot water can be produced centrally or individually. Compared to the old systems, with this system, which is more hygienic, more efficient, longer-lasting, more economical and more compact, the system can reach its old performance with minor revisions instead of completely replacing the system in problems such as calcification and excessive chlorine-induced deformation.

A region, a district, or even a whole province can be heated by using hot water from sources such as regional heating centers, geothermal resources, and electricity generation facilities. The area to be used is divided into sub-zones according to the need, and heat exchangers specially designed according to the type of source are placed under the buildings to produce hot water suitable for each building's own needs.

In underfloor heating systems, which are increasingly used as an effective, homogeneous and decorative heating system, tube heat exchangers used to prevent the heating source from being affected by corrosion act as a protective wall between the heated area and the heating source.

In multi-story and high-rise buildings, serious pressures occur due to the height of the system. Sending this pressure caused by the system to the heating-cooling system at the bottom, as it is, without reducing the pressure, causes the system to be overloaded and worn out. In addition, the initial investment cost is very high since it is necessary to install the installation with high pressure resistant armatures. In these systems, high pressure resistant heat exchangers to be placed between the boiler room or the cooling group and the installation meet the pressure coming from the system within itself and ensure that the boiler - cooling system in the primary circuit works comfortably at low pressures.

The use of central heating systems is encouraged within the framework of the new laws enacted in our country, and in some cases, it is compulsory. The reason for this is that central systems are more efficient and consume less energy compared to individual uses. With the heat exchangers, hot or hot water from a central source can be used to produce the hot water required for heating the houses, and at the same time, domestic hot water can be supplied.

2.2.1. Classification of Heat Exchangers

Heat exchangers can be classified as follows.

1. Classification According to the Type of Heat Exchange
2. Classification According to the Ratio of Heat Transfer Surface to Heat Transfer Volume
3. Classification According to the Number of Different Fluids
4. Classification According to Heat Transfer Mechanism
5. Classification by Current
6. Classification According to Construction Properties

2.2.1.1. Classification According to the Type of Heat Exchanger

2.2.1.1.1. Heat exchangers with direct between fluids

In this type of heat exchangers, the heat is transmitted between cold and hot fluids in direct contact, or a fluid and solid materials are transmitted by contacting each other. Cooling towers like Figure 2.6, spray and plate condensers, which are widely used in practice to remove the heat generated by industrial processes, are good examples of this type of heat exchanger.

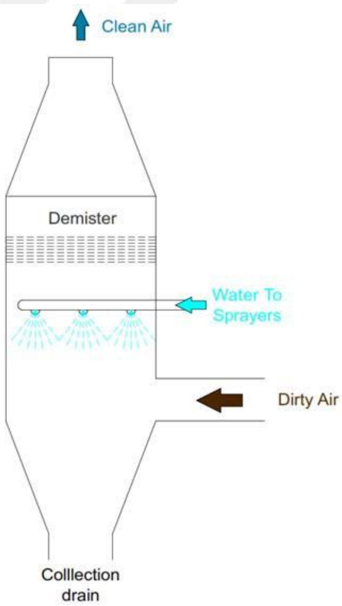

Figure 2-6. Heat Exchanger Flow Configurations

2.2.1.1.2. Heat exchangers with indirect contact between fluids

In these types, the heat first passes to a surface or a mass that separates it from the hot fluid into two fluids. This heat is then transferred from this surface or mass to the cold fluid. It can be examined in three groups as surface, filled and fluidized bed. Examples of tube heat exchangers, one of the important types of direct heat transfer heat exchangers used in practice, are given in Figure 2.7.

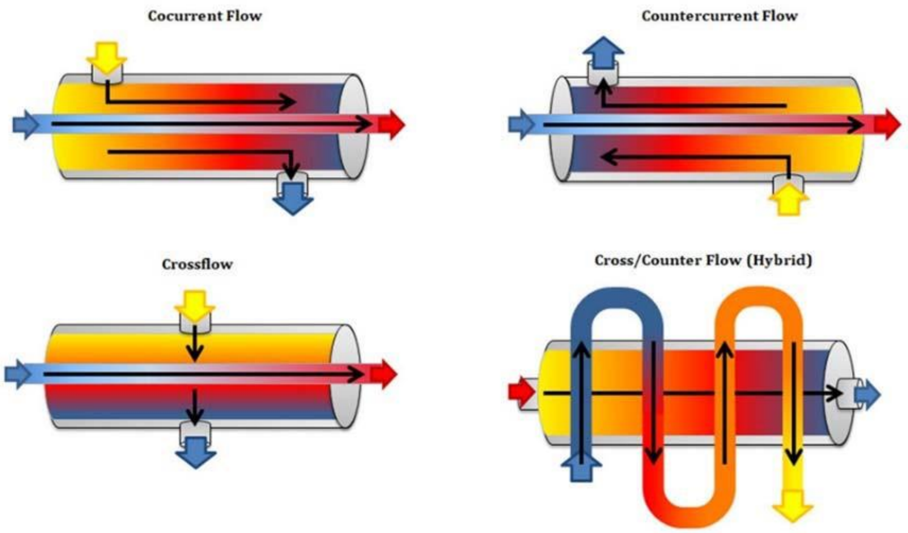

Figure 2-7. Heat Exchanger Flow Configurations

2.2.1.2. Classification According to the Ratio of Heat Transfer Surface to Heat Transfer Volume

The case of a very large heat transfer surface area per unit volume (>700 m2/m3) determines another important and special class of heat exchangers. Such exchangers, called compact heat exchangers, consist of multi-finned tubes or plates and are generally used where the heat transfer coefficient is small and at least one fluid is gas. As seen in Figure 2.8, the tubes used can be of flat or circular cross-sections, respectively. Parallel plate heat exchangers can be straight or wavy fins and can be used as single-pass or multi-pass. Flow cross-sections in compact heat exchangers are very small (Dh < 5 mm) and the flow in them is mostly laminar.

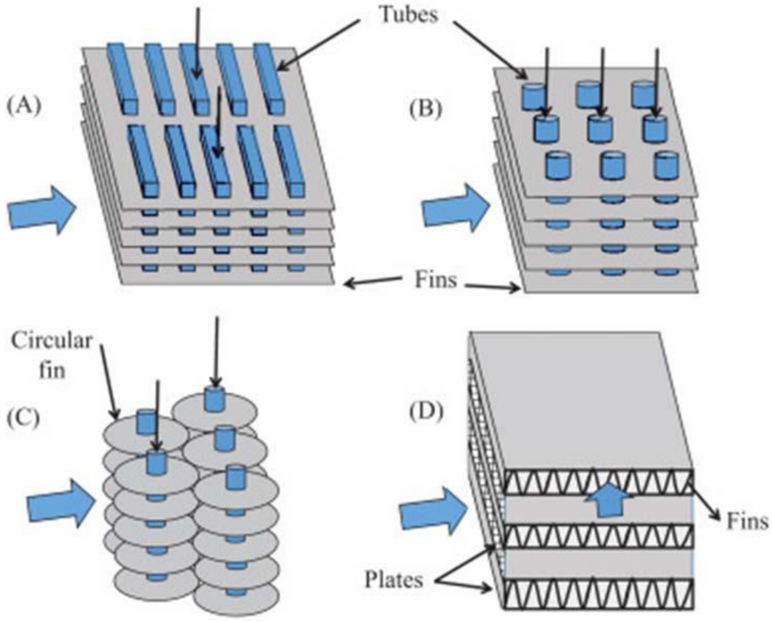

Figure 2-8. Compact heat exchangers: (A, B, C) fin-type heat exchanger and (D) plate-fin heat exchanger. (Green Chemistry and Engineering, 2007)

Compact heat exchangers are preferred because they save weight and volume and allow a more flexible project. On the other hand, the main drawbacks of this type of heat exchanger are that at least one of the fluids is gas, that the surface polluting and corrosive fluids cannot be used, and that additional fan or pump power is needed to overcome the overload losses that occur during flow.

2.2.1.3. Classification According to the Number of Different Fluids

In many practical applications, heat transfer between two fluids is generally considered in heat exchangers. However, in different situations, three-fluid heat exchangers can be encountered in chemical processes, cooling requirements, separation of air, purification, and liquefaction of the hydrogen. An example of a system using three-fluid heat exchangers is a pumpless cooling machine that works with heat energy obtained by using a third neutral gas, such as H2, in small-capacity, (NH3 + water) molten absorption plants in domestic and vehicle vehicles. Theoretical analyzes of this type of heat exchanger are quite complex and their design is difficult.

2.2.1.4. Classification According to Heat Transfer Mechanism

2.2.1.4.1. Single phase flow on both sides

Heat convection in single-phase flows on either side of the heat exchangers can be forced or naturally caused by a density difference, driven by a pump or a fan. Room heaters, steam boiler economizers and air heaters, vehicle radiators and air-cooled heat exchangers are important applications.

2.2.1.4.2. Single phase flow on one side and dual phase flow on the other

These heat exchangers have forced or single-phase flow on one side, and two-phase flow boiling or condensing on the other side. Examples of these are the condensers of thermal power plants, the condenser or evaporator of cooling systems, and steam boilers.

2.2.1.4.3. Dual phase flow on both sides

They are heat exchangers with evaporation on one side of the system and condensation on the other, with phase changes on both sides. They are used in the distillation of hydrocarbons to obtain low-pressure steam using high-pressure steam.

2.2.1.4.4. Heat transfer with convection and radiation

Especially in heat exchangers with high-temperature gas on one side, convection and radiation heat transfer are seen together. High-temperature filler regenerators, fossil fuel-fired heaters, steam boilers, and their superheaters, and pyrolysis furnaces are examples of this type of heat exchanger.

2.2.1.5. Classification by Current

2.2.1.5.1. Single pass heat exchangers

These are the types where two fluids meet only once in the heat exchanger relative to each other. These heat exchangers can be examined in three groups parallel, reverse and cross flow.

1) Parallel flow heat exchangers: In this arrangement, two fluids in the heat exchanger enter from the same end of the exchanger, flow parallel to each other and exit from the other end of the exchanger. The variation of fluid temperature across the heat transfer surface area of the heat exchanger equipment is one-dimensional. Since the wall temperature, which is the heat

transfer of the heat exchanger, does not change much, it is preferred where thermal stresses are not desired.

2) Counter-current heat exchangers: In this type, fluids flow in the heat exchanger in axially parallel but opposite directions with respect to each other. In counter-current heat exchangers, the average temperature difference and efficiency in the exchanger are greater than in any other flow arrangement. Due to this superiority, this type of heat exchanger is preferred in practice. However, this arrangement may not be preferred due to the fact that the temperature of the material with heat transfer varies greatly throughout the exchanger, resulting in increased thermal stresses and construction difficulties in manufacturing.

3) Cross-flow heat exchangers: In this arrangement, the fluids in the heat exchanger flow perpendicular to each other. According to the construction, with the help of fins or baffle plates, the fluids may or may not meet by themselves while they are moving through the exchanger. If the fluid flows in the pipes in the exchanger and does not mix with the fluid in the adjacent channel, this fluid is called immiscible. Otherwise, it is called a mixed fluid.

2.2.1.5.2. Multi-pass heat exchangers

1) Cross-reverse and cross-parallel flow arrangements: These arrangements are generally preferred in finned surface heat exchangers. Two or more crossovers are connected in series with the reverse or parallel current in succession. In applications at high temperatures, in terms of thermal stress materials in heat exchangers, manufacturing costs can be reduced by using expensive materials resistant to heat in regions where the temperature is high, and cheap materials in other regions.

2) Multi-pass shell-and-tube heat exchangers: Parallel-reverse, split-flow, and split-flow arrangements in which the shell fluid is mixed are the most commonly used types in practice. When the number of TEMA (Tubular Exchanger Manufactures Association) pipes, changes with these regulations, the flow direction increases, and the efficiency of the system approaches the cross-flow heat exchanger in which the two fluids are mixed. Although the efficiency of the odd number of pipe passage arrangements in a body is slightly better than the even number of arrangements, it is not preferred much in practice due to manufacturing difficulties and thermal stresses.

3) n Parallel plate transition arrangements: In plate type heat exchangers, multi-pass currents can be obtained by arranging the plates in various ways. These types are usually made of thin metal plates. The plates can be straight, sequential, corrugated or smooth. They cannot be used at very high temperature and pressure differences. Fluids are separated from each other by flat plates. Heat transfer occurs through these plates. They are used in any gas, liquid composition or two-phase flow.

2.2.1.6. Classification According to Construction Properties

Under this heading, the relevant heat exchangers are generally characterized according to their construction properties.

2.2.1.6.1. Tubular Heat Exchangers

In this type of heat exchangers, pipes with elliptical, rectangular and generally circular cross-sections are used. It provides great convenience in projecting as pipe diameter, length and arrangement can be changed easily. In addition, this type of heat exchanger can easily be used at high pressures, since pipes with circular cross-sections can withstand high pressures compared to other geometric shapes.

1) Straight Tube Heat Exchangers: In practice, besides the double tube ones, varieties made of tube bundles are also encountered. Double-pipe designs are the simplest heat exchangers to manufacture and design. While one of the hot or cold fluids flows through the inner pipe, the other fluid flows from the outside pipe according to the working style. Flow directions of fluids can be parallel or countercurrent. In order to increase the thermal capacity and heat transfer surface, they can be assembled in series.

2) Spiral Tube Heat Exchangers: To optimize the heat transfer efficiency and space, the coiled tube bundle placed in the body is called a spiral tube heat exchanger. The gaps between the coils of the spiral tube bundle become the body side flow path when the tube bundle is placed in the shell. The spiral shape of the flow for the tube-side and shell-side fluids creates centrifugal force. Thus, secondary circulating flow enhances the heat transfer on both sides in a proper counterflow arrangement. With centrifugal force, improvements can be made on the tube-side of the heat exchanger without the associated potential for clogging on both the

casing and pipe side. Heat transfer performance can be optimized since there are no partitions or dead spots in the heat exchanger that will reduce the velocity of the fluids.

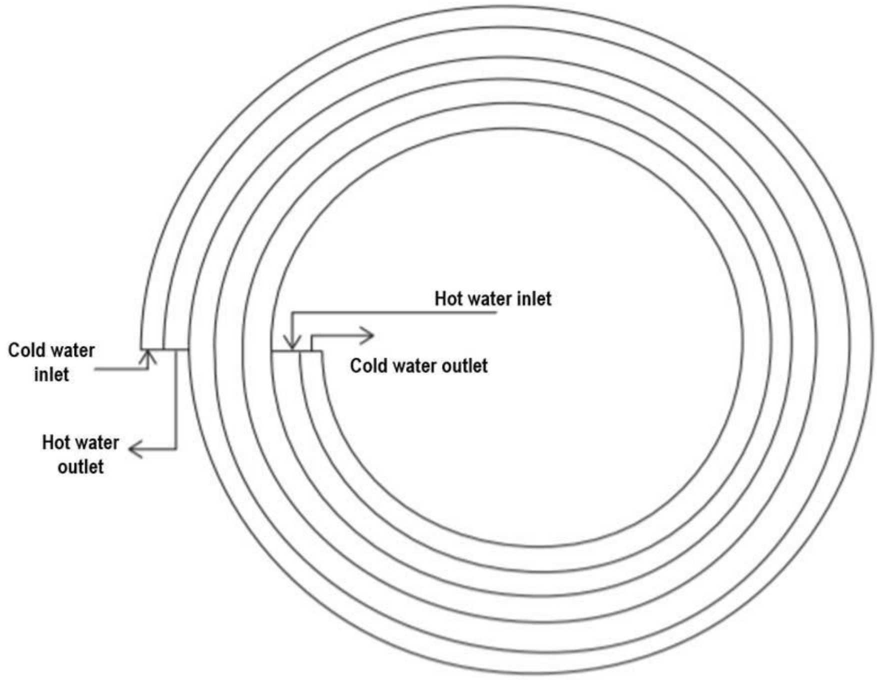

Figure 2-9. Spiral Heat Exchanger.

3) Shell and Tube Heat Exchangers: These exchangers are used by putting the tube bundle in a cylindrical body. One of the heating or cooling fluids moves inside the tube, and the other fluid moves outside the tube or inside the body as shown in Figure 2-9. The main elements of these heat exchangers are the tube bundle, the body, the two heads to which the tubes are fixed, and the baffles. The baffles are the plates that support the tubes and divert the body-side flow. These heat exchangers have applications in the petrochemical industry and many facilities where heat needs to be transported somewhere. TEMA, known as the tubular heat exchanger manufacturers association in the industry, has standardized the construction of these heat exchangers. In these arrangements, fixed tube bundle heat exchangers must be constructed in such a way as to accommodate the elongation that may occur due to pressure and temperature difference. The recommended pressure range for the fluids used in the shell and tubes of this type of heat exchanger is generally from 2 bar to 40 bar.

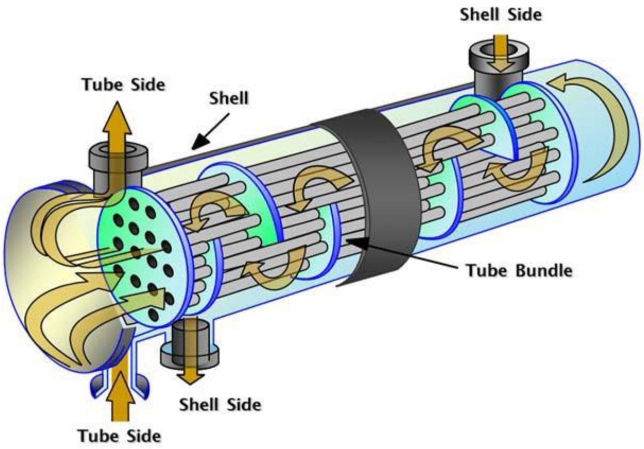

Figure 2-10. Shell and Tube Heat Exchanger. (Arveng, 2017)

4) Special Shell and Tube Heat Exchangers: Although these heat exchangers are similar to conventional shell and tube heat exchangers in terms of construction, they are manufactured for special uses. The protected body-tube heat exchanger is used where it is desired to prevent leakages. Graphite body and tube heat exchangers, on the other hand, are used especially in corrosive environments, as graphite is resistant to chemical substances and has a high heat transmission coefficient.

2.2.1.6.2. Plate Heat Exchangers

In plate heat exchangers, the surfaces where the heat transfer takes place are generally made of thin metal sheets. This metal can be straight or in a wavy form. The plates contain the inlet and outlet connections of the fluids that will exchange heat. The heat exchanger passes two different fluids through the channels formed by the plates arranged side by side. These plates are clamped between the two main plates with the help of studs. Between each plate, there is a gasket that ensures the impermeability and the flow of two fluids without mixing. The package and dimensions of the plates are determined depending on the flow rate of the fluid passing through, its physical properties, the current pressure drop and the temperature program. The different patterned structure on the plates helps the fluids to flow in a turbulent manner and helps to increase the thermal efficiency of the system by creating multiple contact points.

The plates are connected to the two main plates by means of two support bars at the top and bottom. In most of the applications the heat exchanger is manufactured as single pass only, but in some special cases it can be manufactured as multiple pass. Plate heat exchangers have much higher heat transfer coefficient values than other heat exchanger models. Thanks to the patterns on the plates that change the flow direction, the direction and speed of the fluid constantly changes during the flow, and thus, high turbulence value is easily reached even at low flow rates. In addition, these patterns do not leave an area where the fluid is bypassed, and the life of the heat exchanger is extended by preventing contamination and blunting. Compared to the tube heat exchangers that will do the same job, they need a working volume between 1/5 and 1/3. In order to prevent the mixing of fluids, a special seal design is used against seal failure.

1) Gasketed Plate Heat Exchangers: The heat exchanger mentioned in this section is used by packaging many thin metal sheets. For the fluids to be used for the system to pass through and to create heat transfer during this transition, appropriate gaskets are used while the metal sheets with holes are packaged. The capacity of this heat exchanger is obtained with the required number of plates according to the required thermal capacity. The design is shown in Figure 2-10.

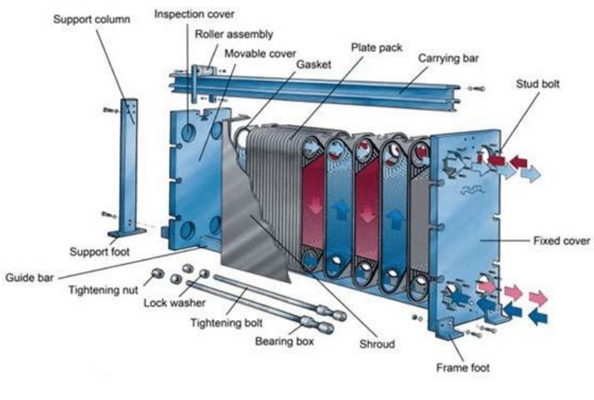

Figure 2-11. Gasketed Plate Heat Exchanger. (Alfa Laval, 2017)

2) Spiral Plate Heat Exchangers: It is a circular heat exchanger consisting of two tightly closed spiral channels to ensure that the fluids do not mix with each other. Since the flow arrangement in the heat exchanger is in the form of counter-flow, excellent results are obtained in applications where the fluid temperatures are very close to each other. The

structure of the spiral heat exchanger makes it possible to work with a fluid containing particles and fibers in one of the circuits and with a standard fluid in the other. It is very easy to reach the heat transfer surfaces of spiral heat exchangers. Both closed spiral channels can be accessed by opening the side covers of the heat exchanger without using any special apparatus. By opening one of the side covers, full access to the heat transfer surface of one of the circuits can be achieved and necessary cleaning can be done there without any contact with the other fluid circuit. The side cover is designed as hinged and can be opened without using lifting equipment.

c) Lamellar Heat Exchangers: This heat exchanger has a bundle of flattened tubes inside a shell. These heat exchangers find application in the paper, food, and chemical industries.

d) Thin Film Heat Exchangers: These heat exchangers have important application areas in heating and cooling of very high viscosity and temperature-sensitive materials. In practice, they are often used as evaporators because of the short residence time of the temperature-sensitive substances in the exchanger and their large heat transfer coefficient.

2.2.1.6.3. Finned Surface Heat Exchangers

Finned-tube heat exchangers consist of a large number of fins and a regularly arranged tube bundle in order to increase the heat transfer area. The reason why these heat exchangers contain a large number of fins is that the outer fluid is gas (usually air). If the external fluid is gas, the heat transfer coefficient value will be low, so more space will be needed for the desired amount of heat transfer. Having a large number of fins increases the heat transfer area and allows the heat transfer to be at the desired level. When the desired surface area is obtained by using the fin, the volume occupied by the equipment is reduced. In case the equipment gets smaller, attention should be paid to the back pressure value of the system. For this reason, the most suitable wing profiles in terms of construction should be used. These heat exchangers, which are divided into plate finned heat exchangers and tube finned heat exchangers, are named according to the use of two different fin profiles. Finned tube can be seen in Figure 2.11.

Figure 2-12. Finned Tube for Heat Exchanger.

2.2.1.6.4. Regenerative Heat Exchangers

Heat exchangers through which hot fluid and cold fluid pass alternately are called regenerators. The internal environment of the heat exchanger is heated by the hot fluid for a certain period of time. In this time period, called the hot period, the heat energy is transferred to the channel wall and stored there. The channel and wall shown in Figure 2.12. The compactness of the regenerators can reach very high values. Compared to other heat exchangers, the initial investment cost is low. The system has a self-cleaning feature. However, these heat exchangers also have a few disadvantages. Regenerators can only be used with gaseous fluids. Therefore, hot and cold fluids mix to some extent. In addition, this type of heat exchangers should never be used if the fluids are incompatible with each other. In practice, there are three types of regenerators: rotating and fixed filler and packed bed.

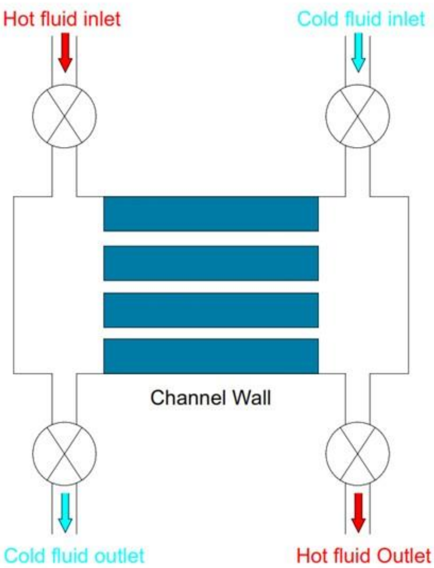

Figure 2-13. Regenerative Heat Exchanger.

Fixed channel regenerators heat air with the help of gases at high temperatures, such as in blast or glass furnaces. The most common technique to ensure the continuous operation is to use two or more regenerators. Thus, one regenerator supplies the heated fluid while the other regenerator stores the heat the hot fluid provides. Heat wheels usually consist of slightly separated plates made of high thermal capacity material fixed to the frame, which is connected to a slowly rotating shaft. The disc axis is positioned to center the volume formed by two adjacent channels. Some of these plates are arranged in such a way as to allow the passage of the air to be heated. The plates heat up as the gas stream slowly passes through them. Later, when the same plates enter the airflow, they cool down by giving the heat they have to the air they will heat. With the rotational movement that continues in this way, the plates carry the heat from the hot gases to the cold air. Packed bed regenerators are another example of continuously operating heat exchangers. Even though the construction of the bed regenerator is simple, the back pressure of the system is high.

3. MATERIALS AND METHODS

A number of formulas are needed in order to make analyzes within the scope of the thesis. Necessary entries of the configurations to be selected for the back pressure, which is the subject of the research, should be entered. Configuration selections have been made according to energy calculations. In the configurations selected according to the energy formulas given under the title, the total thermal conductivity of the system, in-pipe velocity, and pressure drop have been observed. Configuration selections have been chosen by assuming that the accepted 8.500 kg/h, 430 °C exhaust flow has decreased to 120 °C by heating the 70-90 °C water cycle. The total heat transfer coefficient and pressure drops of each configuration to be selected have been calculated.

While the exhaust gas is cooled in the cross-flow heat exchanger consisting of tube bundles, hot water is obtained with the heat obtained from here. The following formula is used when calculating the resulting hot water flow.

$$m_{ex}\, c_{pex}\, (T_{ex,i} - T_{ex,o}) = m_w\, c_{pw}\, (T_{w,o} - T_{w,i}) \tag{3.1}$$

In the formula given above, mex exhaust flow, cpex specific heat of exhaust temperature at average temperature, Tex,i exhaust inlet temperature, Tex,o exhaust outlet temperature, mw heated water flow, cpw specific heat of water at average temperature, Tw,o is the water outlet temperature, Tw,i is the water inlet temperature.

The research subject is about smoke tube, that’s why there is exhaust gas inside the tube, the fluid outside of the tube is water. The formula given below is used when calculating the Reynolds number of the internal fluid.

$$Re_i = \frac{V_i\, d_i}{\mu_i / \rho_i} \tag{3.2}$$

Here, Re_i is the Reynolds number of the internal flow, V_i is velocity of the fluid in the tube, d_i is the internal diameter of the tube, μ_i is the dynamic viscosity of the fluid, and ρ_i is the density of the fluid. The Nusselt number of the inner fluid is found by the formula given below.

$$Nu_i = 0{,}023\, Re_i^{0,8}\, Pr^{0,3} \tag{3.3}$$

Here, Nu_i is the Nusselt number of the fluid in the pipe, Re_i is the previously calculated Reynolds number, and Pr is the Prandtl number of the fluid at the average temperature.

The following formula is used when calculating the in-tube convection heat transfer coefficient.

$$h_i = \frac{Nu_i\, k_i}{d_i} \tag{3.4}$$

Here, h_i is the in-tube heat transfer coefficient, Nu_i is the previously calculated Nusselt number, k_i is the thermal conductivity coefficient of the fluid.

After the internal calculation of the system, the velocity of the water circulating around the pipes must be calculated for the external calculation. External fluid calculations are made assuming that the tubes are staggered and the pitch is 1.3 times the tube diameter. The maximum velocity of the water moving in the opposite direction to the exhaust gas is found with the help of the following formula.

$$V_{max} = \frac{S_T}{S_T - R_d}\, V \tag{3.5}$$

Here, "V_{max}" is the velocity that the external fluid reaches between the pipe bundle, "S_T" is the axis distance of the pipes perpendicular to the external fluid, "S_L" is the axis distance between two rows of pipes, "R_d" is the radius of the tube used, and "V" is the initial velocity of the fluid.

The Reynolds number of water, which is the outer fluid, is calculated by the formula 3.6.

$$Re_e = \frac{V_e\, d_e}{\mu_e / \rho_e} \tag{3.6}$$

Here, "Re_e" is the Reynold value of the external fluid, "V_e" is the velocity of the external fluid, "d_e" is the outer diameter of the pipe used, "u_e" is the dynamic viscosity of the external fluid, and "p_e" is the density of the external fluid.

The following formula was used to calculate the Nusselt number of the external fluid.

$$Nu_e = 1{,}04\, Re_e^{0{,}4}\, Pr^{0{,}36} \left(Pr / Prs \right)^{0{,}25} \tag{3.7}$$

Here, "Nu_e" is the calculated Nusselt value of the external fluid, "Re_e" is the Reynolds value of the previously calculated external fluid, "Pr" is the Prandtl value of the external fluid, and "Pr_s" is the Prandtl value of the two fluids at the average temperature.

The following formula is used when calculating the convection heat transfer coefficient of the external fluid.

$$h_e = \frac{Nu_e\, k_e}{d_e} \tag{3.8}$$

Here, h_e is the heat transfer coefficient of the external fluid, Nu_e is the previously calculated Nusselt number, k_e is the thermal conductivity coefficient of the external fluid.

The following formula is used when calculating the total heat transfer coefficient.

$$U = \frac{1}{\frac{1}{h_i}\frac{A_d}{A_i} + \frac{A_d}{2\,\pi\, l\, k}\ln\left(\frac{r_d}{r_i}\right) + \frac{1}{h_d}} \cong \frac{1}{\frac{1}{h_i} + \frac{1}{h_d}} \tag{3.9}$$

The correction factor required for the corrected logarithmic temperature difference calculation, which is used to find the area required for the realization of the heat transfer depending on the calculated total heat transfer coefficient, is calculated with the formula below. The correction factor changes according to the heat exchanger type.

$$P = \frac{T_{w,o} - T_{w,i}}{T_{ex,o} - T_{ex,i}} \tag{3.10}$$

$$R = \frac{T_{ex,i} - T_{ex,o}}{T_{w,o} - T_{w,i}} \tag{3.11}$$

$$X = \frac{1 - \left(\frac{R\,P - 1}{P - 1}\right)^{\frac{1}{N}}}{R - \left(\frac{R\,P - 1}{P - 1}\right)^{\frac{1}{N}}} \tag{3.12}$$

$$F = \frac{\frac{\sqrt{R^2 + 1}}{R - 1}\ln\left(\frac{1 - X}{1 - R\,X}\right)}{\ln\left(\frac{\frac{2}{X} - 1 - R + \sqrt{R^2 + 1}}{\frac{2}{X} - 1 - R - \sqrt{R^2 + 1}}\right)} \tag{3.13}$$

The corrected logarithmic temperature difference correction factor according to the waste heat boiler design is calculated with the help of the formula below.

$$\Delta T_{lm} = \frac{(T_{ex,i} - T_{w,o}) - (T_{ex,o} - T_{w,i})}{\ln\left(\frac{T_{ex,i} - T_{w,o}}{T_{ex,o} - T_{w,i}}\right)}$$

F

(3.14)

The friction factor to be used to calculate the pressure drop inside the pipe is calculated with the following formula.(Incopera et.al, 1996)

$$\frac{1}{\sqrt{f}} = -1{,}8 \log \left[\frac{6{,}9}{Re_i} + \left(\frac{\varepsilon / D_i}{3{,}7} \right)^{1{,}11} \right] \tag{3.15}$$

Here, "ε" is the epsilon value and "f' is the friction factor. The formula used to calculate the pressure drop inside the pipe is as follows.

$$\Delta P_K = f \frac{L}{R_i} \frac{\rho_i V_i^2}{2} \tag{3.16}$$

Here, "L" is the pipe length, "ρi" is the density of the inner fluid, and "Ri" is the radius of the tube through which the inner fluid passes.

3.1. The Design of The Heat Exchanger for the Study

To carry out thermal analysis, a shell and tube heat exchanger is designed. The calculations have been done for the designed heat exchanger. To select the capacity of the boiler, an exhaust source is determined. In industry, the usage of 1.6 MW is common for small and medium-sized enterprises. The exhaust source of the designed heat exchanger has been a natural-gas powered generator. The specification of exhaust for the selected generator is given in Table 3.1.

Table 3-1. The exhaust values for the natural gas engine.

Specification	**Amount**
Exhaust temperature	430 °C
Exhaust gas flow for 0°C and 101.3 kPa	13.903 Nm^3/h
Exhaust gas mass flow	8500 kg/h

The designed shell and tube heat exchanger has received exhaust given in Table 3.1. The exhaust has been hot working fluid for the heat exchanger where it has gone through the tubes

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inside the heat exchanger. The cold working fluid that is water, goes through the body as explained in 2.2.1.6.1. Tubular Heat Exchangers.

The flow of the fluids will be in the opposite direction of each other. In opposite directions of flow, the hot working fluid can be decreased to lower temperatures, ensuring better heat regeneration from the exhaust.

The hot working fluid, exhaust, has unique thermal characteristics depending on the natural gas used during combustion. Depending on natural gas specifications, the air is provided for combustion. The exactly needed air for combustion can be calculated but in practice, to ensure full combustion, additional air is provided, generally the additional value is 5%-15% extra of the needed air for the combustion.

The natural gas used for the combustion can vary in time even for the same supplier or source. For the sake of analysis, a natural gas chemical content is assumed for this thesis, which is given in Table 3.2.

Table 3-2. The chemical content of natural gas

Chemical Content of Gas	
Gas	**Content Ratio**
Methane	**90,03%**
Ethane	**3,50%**
Propane	**1,14%**
n-Butane	**0,28%**
Isobutane	**0,20%**
n-Pentane	**0,03%**
Isopentane	**0,04%**
Neopentane	**0,00%**
n-Hexane	**0,04%**
Nitrogen (atm)	**4,19%**
Carbon monoxide	**0,00%**
Carbon dioxide	**0,56%**
	100,00%

The molecular weight of the components that reach with oxygen are taken into consideration and these values can be found in a combustion constants table, in this study, the table used is from Ganapathy (2002) as well as other combustion related tables. Chemical components of the natural gas determine the calorific values of the fuel, combustion air requirements, fuel density, and fuel molecular weight.

Combustion consists of reactions involving different components of the fuel. The important ones in combustion which are C, S, H2, CO, O, CmHn is given below:

$$\begin{aligned} &C + O_2 = CO_2 \\ &S + O_2 = SO_2 \\ &2H_2 + O_2 = 2H_2O \\ &2CO + O_2 = 2CO_2 \\ &CH_4 + 2O_2 = CO_2 + 2H_2O \\ &C_mH_n + \left(m + \frac{n}{4}\right) O_2 = mCO_2 + \frac{n}{2} H_2O \end{aligned} \tag{3.17}$$

For the combustion calculations, chemical reactions and combustion constants are found and based on 100 mol of fuel. The theoretical dry air required is found, which is increased by an excess of 15%.

The moles of chemical components CO2, H2O, N2, and O2 resulting in the reaction are found. The resolution of chemical components and ratios to each other gives the characteristics of the exhaust gas. For our considered natural gas given in Table 3-2, the results of calculations are given in Appendix A-1 in datasheet format.

4. RESULTS AND DISCUSSIONS

In order to compare the exhaust heat transfer coefficient hi and backpressure, a shell and tube heat exchanger with different sizes, thicknesses, lengths, and amounts for each configuration is considered and thermal analysis has been done. The heat exchanger's capacity is kept constant, the exhaust flow rate is 8500 kg/h and the temperature is 430 °C for each configuration. The exhaust exits the heat exchanger at 120 °C, so the capacity of the heat exchanger is 818 kW. Backpressure analysis is performed for 8 different design combinations. In each combination, the required heat transfer surface area and pipe side pressure drop depending on the heat transfer coefficient of the system were calculated. The heat transfer analyses and back pressure results for the configuration with Ø26, 9- 2500 mm - 3,6 wall thickness are shown in Table 4-1. The calculated backpressure is 3,06 mbar and the exhaust heat transfer coefficient is 49,83. The capacity of 818 kW/h is reached with 772 tubes and 119,56 m2 of heat transfer area.

Table 4-1. The heat transfer analyzes and back pressure results for the configuration according to Ø26,9- 2500 mm - 3,6 wall thickness.

Accepted U (kcal/hm²C)	Calculated U (kcal/hm²C)	Heat Transfer Area(m²)	Needed Tube (pcs.)	Exhaust Velocity (m/s) V_i	Exhaust Reynolds Number Re_i	Exhaust Nusselt Number Nu_i	Exhaust Heat transfer coefficient (kcal/m²C) h_i	Back pressure (mbar)
50	46,58	85,99	555,74	21,47	9962,84	32,05	64,87	5,54
46,58	44,06	92,30	596,52	20,00	9281,80	30,28	61,30	4,88
44,06	42,18	97,57	630,61	18,92	8780,04	28,97	58,63	4,41
42,18	40,76	101,92	658,75	18,11	8404,94	27,97	56,62	4,08
40,76	39,67	105,48	681,75	17,50	8121,35	27,21	55,08	3,84
39,67	38,84	108,37	700,41	17,04	7905,01	26,63	53,91	3,65
38,84	38,19	110,70	715,45	16,68	7738,84	26,18	53,00	3,52
38,19	37,69	112,56	727,51	16,40	7610,50	25,84	52,29	3,41
37,69	37,31	114,06	737,16	16,19	7510,95	25,57	51,75	3,33
37,31	37,00	115,24	744,84	16,02	7433,47	25,35	51,32	3,27
37,00	36,77	116,19	750,95	15,89	7373,02	25,19	50,98	3,22
36,77	36,58	116,94	755,79	15,79	7325,74	25,06	50,72	3,19
36,58	36,43	117,53	759,63	15,71	7288,71	24,96	50,52	3,16
36,43	36,32	118,00	762,67	15,65	7259,67	24,88	50,36	3,13
36,32	36,23	118,37	765,07	15,60	7236,87	24,82	50,23	3,12
36,23	36,16	118,67	766,97	15,56	7218,96	24,77	50,13	3,10
36,16	36,10	118,90	768,47	15,53	7204,88	24,73	50,05	3,09
36,10	36,06	119,08	769,65	15,50	7193,80	24,70	49,99	3,08
36,06	36,02	119,23	770,59	15,48	7185,09	24,67	49,94	3,08
36,02	36,00	119,34	771,33	15,47	7178,23	24,66	49,90	3,07
36,00	35,98	119,43	771,91	15,46	7172,83	24,64	49,87	3,07
35,98	35,96	119,50	772,36	15,45	7168,57	24,63	49,85	3,06
35,96	35,95	119,56	772,72	15,44	7165,22	24,62	49,83	3,06

The heat transfer analyzes and back pressure results for the configuration with Ø26,9- 3000 mm - 3,6 wall thickness are shown in Table 4-2. The calculated backpressure is 16,59 mbar and exhaust heat transfer coefficient is 97,37. The capacity of 818 kW/h is reached with 334 tubes and 62,10 m^2 of heat transfer area. The results are hugely different from Ø26,9- 2500 mm - 3,6 wall thickness tube configuration given in Table 4-1. The increase of 500mm increased the backpressure as the exhaust is required to travel for longer length. The exhaust heat transfer coefficient increased 1,95 times while backpressure increased 5,42 times. The required tube amount for achieving desired heat transfer capacity is achieved in less than half of what was needed in for configuration given in Table 4-1.

Table 4-2. The heat transfer analyzes and back pressure results for the configuration according to Ø26,9- 3000 mm - 3,6 wall thickness.

Accepted U (kcal/hm²C)	Calculated U (kcal/hm²C)	Heat Transfer Area(m^2)	Needed Tube (pcs.)	Exhaust Velocity (m/s) V_i	Exhaust Reynolds Number Re_i	Exhaust Nusselt Number Nu_i	Exhaust Heat transfer coefficient (kcal/m²C) h_i	Back pressure (mbar)
50	53,73	85,99	463,12	25,77	11955,41	37,08	75,05	9,23
53,73	56,84	80,01	430,94	27,69	12847,99	39,28	79,50	10,51
56,84	59,40	75,63	407,36	29,29	13591,74	41,09	83,16	11,63
59,40	61,47	72,38	389,85	30,61	14202,12	42,56	86,14	12,59
61,47	63,13	69,95	376,72	31,67	14697,07	43,74	88,53	13,39
63,13	64,45	68,10	366,80	32,53	15094,67	44,68	90,44	14,05
64,45	65,51	66,70	359,26	33,21	15411,72	45,43	91,96	14,58
65,51	66,34	65,63	353,49	33,76	15663,09	46,03	93,16	15,02
66,34	66,99	64,81	349,07	34,18	15861,48	46,49	94,10	15,36
66,99	67,50	64,18	345,67	34,52	16017,52	46,86	94,84	15,63
67,50	67,90	63,69	343,05	34,78	16139,91	47,14	95,42	15,85
67,90	68,21	63,32	341,02	34,99	16235,70	47,37	95,87	16,02
68,21	68,46	63,03	339,46	35,15	16310,56	47,54	96,23	16,15
68,46	68,65	62,80	338,25	35,28	16368,97	47,68	96,50	16,26
68,65	68,80	62,63	337,31	35,38	16414,51	47,78	96,72	16,34
68,80	68,91	62,49	336,58	35,45	16449,98	47,87	96,88	16,40
68,91	69,00	62,39	336,02	35,51	16477,60	47,93	97,01	16,45
69,00	69,07	62,31	335,58	35,56	16499,09	47,98	97,12	16,49
69,07	69,13	62,24	335,24	35,59	16515,80	48,02	97,19	16,52
69,13	69,17	62,19	334,98	35,62	16528,80	48,05	97,26	16,54
69,17	69,20	62,16	334,77	35,64	16538,91	48,07	97,30	16,56
69,20	69,23	62,13	334,61	35,66	16546,76	48,09	97,34	16,57
69,23	69,25	62,10	334,49	35,67	16552,87	48,11	97,37	16,59

The heat transfer analyzes and back pressure results for the configuration with Ø26,9- 3500 mm - 3,6 wall thickness are shown in Table 4-3. The calculated backpressure is 64,45 mbar and exhaust heat transfer coefficient is 166,21. The capacity of 818 kW/h is reached with 171 tubes and 37,13 m^2 of heat transfer area. The results show a similar difference compared to Table 4-2 as it was for difference in configurations Table 4-2 to Table 4-1. The exhaust heat transfer coefficient and backpressure increased greatly. However, the increase in backpressure is near critical levels at this point. While some systems may permit 64,45 mbar backpressure, it is not advised to use high backpressure heat exchangers in risk of decreasing the efficiency of gas engine. And if system does not allow 64,45 mbar backpressure, this heat exchanger can't be used as it will prevent gas engine to work properly.

Table 4-3. The heat transfer analyzes and back pressure results for the configuration according to Ø26,9- 3500 mm - 3,6 wall thickness.

Accepted U (kcal/hm²C)	Calculated U (kcal/hm²C)	Heat Transfer Area(m²)	Needed Tube (pcs.)	Exhaust Velocity (m/s) V_i	Exhaust Reynolds Number Re_i	Exhaust Nusselt Number Nu_i	Exhaust Heat transfer coefficient (kcal/m²C) h_i	Back pressure (mbar)
50	60,61	85,99	396,96	30,06	13947,98	41,95	84,90	14,22
60,61	70,39	70,94	327,48	36,44	16906,90	48,93	99,03	20,10
70,39	79,06	61,07	281,95	42,32	19637,35	55,15	111,63	26,32
79,06	86,47	54,38	251,06	47,53	22053,55	60,52	122,49	32,43
86,47	92,65	49,72	229,54	51,98	24121,49	65,02	131,60	38,11
92,65	97,69	46,40	214,23	55,70	25845,31	68,71	139,07	43,15
97,69	101,75	44,01	203,16	58,73	27252,85	71,69	145,09	47,47
101,75	104,97	42,25	195,07	61,17	28383,75	74,06	149,89	51,08
104,97	107,49	40,96	189,09	63,10	29281,08	75,92	153,67	54,02
107,49	109,46	40,00	184,64	64,62	29986,23	77,38	156,62	56,38
109,46	110,99	39,28	181,32	65,81	30536,24	78,52	158,92	58,26
110,99	112,17	38,73	178,82	66,73	30962,79	79,39	160,69	59,73
112,17	113,08	38,33	176,94	67,44	31292,15	80,07	162,06	60,88
113,08	113,78	38,02	175,52	67,98	31545,61	80,58	163,11	61,77
113,78	114,31	37,79	174,44	68,40	31740,16	80,98	163,91	62,46
114,31	114,72	37,61	173,62	68,73	31889,21	81,29	164,53	62,99
114,72	115,04	37,48	173,01	68,97	32003,22	81,52	165,00	63,39
115,04	115,27	37,37	172,54	69,16	32090,33	81,70	165,36	63,71
115,27	115,46	37,30	172,18	69,30	32156,82	81,83	165,63	63,94
115,46	115,59	37,24	171,91	69,41	32207,56	81,93	165,84	64,12
115,59	115,70	37,19	171,70	69,49	32246,24	82,01	166,00	64,26
115,70	115,78	37,16	171,55	69,56	32275,72	82,07	166,12	64,37
115,78	115,84	37,13	171,43	69,61	32298,19	82,12	166,21	64,45

The heat transfer analyzes and back pressure results for the configuration with Ø26,9- 4000 mm - 3,6 wall thickness are shown in Table 4-4. The calculated backpressure is 194,39 mbar and exhaust heat transfer coefficient is 255,84. The capacity of 818 kW/h is reached with 100 tubes and 24,75 m^2 of heat transfer area. The increase in backpressure is above critical levels at this point. This heat exchanger can't be used in natural gas engines found in the market as the engine won't be able to push the exhaust out of the system in desired speed and system will clog. This may be solved by adding a fan to increase pressure of exhaust; however, this will increase the cost of the system and complexity. The required heat transfer area decreased greatly, which means the cost of heat exchanger will also be cheap compared to low backpressure heat exchangers. In the other hand, if the heat exchanger is not suitable to the system and will not perform properly, being cheap is not an advantage.

Table 4-4. The heat transfer analyzes and back pressure results for the configuration according to Ø26,9- 4000 mm - 3,6 wall thickness.

Accepted U (kcal/hm²C)	Calculated U (kcal/hm²C)	Heat Transfer Area(m²)	Needed Tube (pcs.)	Exhaust Velocity (m/s) V_i	Exhaust Reynolds Number Re_i	Exhaust Nusselt Number Nu_i	Exhaust Heat transfer coefficient (kcal/m²C) h_i	Back pressure (mbar)
50	67,25	85,99	347,34	34,35	15940,55	46,68	94,48	20,66
67,25	84,61	63,93	258,25	46,20	21439,43	59,17	119,75	35,23
84,61	100,95	50,82	205,27	58,13	26973,29	71,10	143,90	53,26
100,95	115,53	42,59	172,04	69,36	32183,32	81,89	165,74	73,19
115,53	127,99	37,21	150,33	79,38	36831,73	91,22	184,63	93,30
127,99	138,29	33,59	135,69	87,94	40805,28	99,01	200,40	112,20
138,29	146,59	31,09	125,58	95,02	44089,57	105,33	213,20	128,97
146,59	153,13	29,33	118,47	100,72	46734,09	110,36	223,37	143,23
153,13	158,22	28,08	113,41	105,22	48821,06	114,28	231,32	154,95
158,22	162,13	27,17	109,76	108,71	50442,94	117,31	237,44	164,34
162,13	165,11	26,52	107,12	111,40	51688,78	119,62	242,12	171,71
165,11	167,36	26,04	105,19	113,44	52637,41	121,38	245,67	177,43
167,36	169,05	25,69	103,77	114,99	53354,98	122,70	248,35	181,81
169,05	170,32	25,43	102,73	116,15	53895,10	123,69	250,36	185,13
170,32	171,27	25,24	101,97	117,02	54300,14	124,44	251,86	187,64
171,27	171,98	25,10	101,40	117,68	54603,07	124,99	252,98	189,53
171,98	172,51	25,00	100,98	118,16	54829,15	125,40	253,82	190,95
172,51	172,90	24,92	100,67	118,53	54997,63	125,71	254,45	192,01
172,90	173,19	24,87	100,44	118,80	55123,03	125,94	254,91	192,79
173,19	173,41	24,82	100,27	119,00	55216,30	126,11	255,25	193,38
173,41	173,57	24,79	100,15	119,15	55285,63	126,24	255,51	193,82
173,57	173,69	24,77	100,05	119,26	55337,13	126,33	255,70	194,14
173,69	173,78	24,75	99,99	119,34	55375,37	126,40	255,84	194,39

The heat transfer analyzes and back pressure results for the configuration with Ø33,7- 2500 mm - 3,6 wall thickness are shown in Table 4-5. The calculated backpressure is 0,18 mbar and exhaust heat transfer coefficient is 15,75. The capacity of 818 kW/h is reached with 1674 tubes and 348,36 m^2 of heat transfer area. The results differ greatly in contrast to configuration given in Table 4-1. The lengths and thickness are same however diameter of tubes is different. This difference increases the area which the exhaust flows through the tubes which decreases backpressure greatly. In the result of backpressure decreasing, exhaust heat transfer coefficient also decreases. The required tube amount is 1674 which is highly impractical for a heat exchanger construction unless the system can permit critically low backpressure from the heat exchanger. In Table 4-6, tubes lengths are increased for 500mm. The backpressure increases to 1,07 mbar. This configuration is better compared to Table 4-5 however unless specifically required, 1 mbar is still too low for most systems.

Table 4-5. The heat transfer analyzes and back pressure results for the configuration according to Ø33,7- 2500 mm - 3,6 wall thickness.

Accepted U (kcal/hm²C)	Calculated U (kcal/hm²C)	Heat Transfer Area(m²)	Needed Tube (pcs.)	Exhaust Velocity (m/s) V_i	Exhaust Reynolds Number Re_i	Exhaust Nusselt Number Nu_i	Exhaust Heat transfer coefficient (kcal/m²C) h_i	Back pressure (mbar)
50	37,33	85,99	413,13	15,96	9962,84	32,05	48,22	2,28
37,33	29,65	115,16	553,31	11,92	7438,85	25,37	38,17	1,35
29,65	24,71	145,01	696,74	9,46	5907,54	21,10	31,74	0,89
24,71	21,38	174,01	836,08	7,89	4922,96	18,23	27,44	0,64
21,38	19,07	201,06	966,03	6,83	4260,75	16,24	24,44	0,49
19,07	17,41	225,47	1083,33	6,09	3799,39	14,82	22,30	0,40
17,41	16,20	246,95	1186,53	5,56	3468,94	13,78	20,73	0,34
16,20	15,29	265,47	1275,48	5,17	3227,03	13,01	19,57	0,30
15,29	14,61	281,16	1350,90	4,88	3046,85	12,42	18,69	0,27
14,61	14,09	294,30	1414,04	4,66	2910,81	11,98	18,02	0,25
14,09	13,69	305,19	1466,35	4,50	2806,98	11,63	17,50	0,23
13,69	13,38	314,14	1509,33	4,37	2727,04	11,37	17,10	0,22
13,38	13,13	321,44	1544,42	4,27	2665,07	11,16	16,79	0,21
13,13	12,94	327,37	1572,92	4,19	2616,78	11,00	16,55	0,21
12,94	12,79	332,17	1595,98	4,13	2578,98	10,87	16,36	0,20
12,79	12,68	336,04	1614,56	4,08	2549,29	10,77	16,21	0,20
12,68	12,58	339,15	1629,51	4,05	2525,91	10,69	16,09	0,19
12,58	12,51	341,65	1641,50	4,02	2507,46	10,63	15,99	0,19
12,51	12,45	343,65	1651,11	3,99	2492,87	10,58	15,92	0,19
12,45	12,41	345,25	1658,79	3,98	2481,32	10,54	15,86	0,19
12,41	12,37	346,52	1664,94	3,96	2472,17	10,51	15,81	0,19
12,37	12,34	347,54	1669,84	3,95	2464,90	10,48	15,78	0,18
12,34	12,32	348,36	1673,75	3,94	2459,14	10,46	15,75	0,18

Table 4-6. The heat transfer analyzes and back pressure results for the configuration according to Ø33,7- 3000 mm - 3,6 wall thickness.

Accepted U (kcal/hm²C)	Calculated U (kcal/hm²C)	Heat Transfer Area(m²)	Needed Tube (pcs.)	Exhaust Velocity (m/s) V_i	Exhaust Reynolds Number Re_i	Exhaust Nusselt Number Nu_i	Exhaust Heat transfer coefficient (kcal/m²C) h_i	Back pressure (mbar)
50	43,09	85,99	344,28	19,15	11955,41	37,08	55,79	3,79
43,09	38,33	99,77	399,48	16,51	10303,35	32,92	49,54	2,90
38,33	34,96	112,15	449,05	14,69	9166,01	29,98	45,11	2,35
34,96	32,51	122,98	492,40	13,39	8359,08	27,85	41,91	1,99
32,51	30,70	132,25	529,53	12,45	7773,00	26,28	39,54	1,75
30,70	29,33	140,06	560,81	11,76	7339,44	25,10	37,76	1,58
29,33	28,30	146,56	586,82	11,24	7014,07	24,20	36,42	1,45
28,30	27,51	151,91	608,23	10,84	6767,13	23,52	35,39	1,36
27,51	26,90	156,28	625,72	10,54	6578,03	22,99	34,60	1,29
26,90	26,43	159,82	639,90	10,31	6432,20	22,58	33,98	1,24
26,43	26,06	162,68	651,36	10,12	6319,12	22,27	33,50	1,20
26,06	25,77	164,98	660,56	9,98	6231,06	22,02	33,13	1,17
25,77	25,55	166,82	667,94	9,87	6162,23	21,82	32,83	1,15
25,55	25,37	168,30	673,84	9,79	6108,29	21,67	32,60	1,13
25,37	25,23	169,47	678,54	9,72	6065,93	21,55	32,42	1,12
25,23	25,12	170,41	682,29	9,66	6032,61	21,45	32,28	1,11
25,12	25,03	171,15	685,27	9,62	6006,36	21,38	32,17	1,10
25,03	24,96	171,74	687,64	9,59	5985,66	21,32	32,08	1,09
24,96	24,91	172,21	689,53	9,56	5969,32	21,27	32,01	1,09
24,91	24,87	172,59	691,02	9,54	5956,41	21,24	31,95	1,08
24,87	24,83	172,88	692,20	9,53	5946,22	21,21	31,91	1,08
24,83	24,81	173,12	693,14	9,51	5938,16	21,18	31,88	1,08
24,81	24,79	173,30	693,89	9,50	5931,78	21,17	31,85	1,07

The heat transfer analyzes and back pressure results for the configuration with Ø33,7- 3500 mm - 3,6 wall thickness are shown in Table 4-7. The calculated backpressure is 4,64 mbar and exhaust heat transfer coefficient are 56,97. The capacity of 818 kW/h is reached with 355 tubes and 97,74 m^2 of heat transfer area. This configuration can be compared to configuration given in Table 4-1 as their backpressures are similar. However, the advantage of using configuration in Table 4-7 is that the needed tube amount is 335 and overall size of heat exchanger will be much smaller to Table 4-1 configuration which have 772 tube requirements.

Table 4-7. The heat transfer analyzes and back pressure results for the configuration according to Ø33,7- 3500 mm - 3,6 wall thickness.

Accepted U (kcal/hm²C)	Calculated U (kcal/hm²C)	Heat Transfer Area(m²)	Needed Tube (pcs.)	Exhaust Velocity (m/s) V_i	Exhaust Reynolds Number Re_i	Exhaust Nusselt Number Nu_i	Exhaust Heat transfer coefficient (kcal/m²C) h_i	Back pressure (mbar)
50	48,63	85,99	295,10	22,35	13947,98	41,95	63,12	5,84
48,63	47,59	88,40	303,40	21,74	13566,46	41,03	61,73	5,56
47,59	46,78	90,35	310,06	21,27	13274,63	40,32	60,67	5,34
46,78	46,16	91,90	315,40	20,91	13050,14	39,77	59,85	5,18
46,16	45,68	93,14	319,65	20,63	12876,66	39,35	59,21	5,06
45,68	45,30	94,12	323,02	20,41	12742,14	39,02	58,71	4,96
45,30	45,01	94,90	325,70	20,25	12637,53	38,76	58,33	4,89
45,01	44,78	95,52	327,81	20,12	12556,01	38,56	58,03	4,83
44,78	44,60	96,00	329,48	20,01	12492,38	38,41	57,79	4,79
44,60	44,46	96,39	330,80	19,93	12442,64	38,29	57,61	4,76
44,46	44,35	96,69	331,84	19,87	12403,73	38,19	57,46	4,73
44,35	44,27	96,93	332,65	19,82	12373,25	38,11	57,35	4,71
44,27	44,20	97,12	333,30	19,79	12349,37	38,06	57,26	4,69
44,20	44,15	97,26	333,80	19,76	12330,65	38,01	57,19	4,68
44,15	44,11	97,38	334,20	19,73	12315,97	37,97	57,14	4,67
44,11	44,08	97,47	334,51	19,71	12304,45	37,94	57,09	4,66
44,08	44,05	97,54	334,76	19,70	12295,41	37,92	57,06	4,65
44,05	44,03	97,60	334,95	19,69	12288,31	37,90	57,03	4,65
44,03	44,01	97,64	335,10	19,68	12282,74	37,89	57,01	4,65
44,01	44,00	97,68	335,22	19,67	12278,37	37,88	57,00	4,64
44,00	43,99	97,71	335,32	19,67	12274,94	37,87	56,98	4,64
43,99	43,99	97,73	335,39	19,66	12272,24	37,87	56,97	4,64
43,99	43,98	97,74	335,45	19,66	12270,12	37,86	56,97	4,64

The heat transfer analyses and back pressure results for the configuration with Ø33,7- 4000 mm - 3,6 wall thickness are shown in Table 4-8. The calculated backpressure is 15,85 mbar and the exhaust heat transfer coefficient is 92,70. The capacity of 818 kW/h is reached with 183 tubes and 60,78 m^2 of heat transfer area. This configuration is a balanced design construction for a heat exchanger if the system permits backpressures of 16 mbar. In most natural gas engines, this backpressure is acceptable.

Table 4-8. The heat transfer analyzes and back pressure results for the configuration according to Ø33,7- 4000 mm - 3,6 wall thickness.

Accepted U (kcal/hm²C)	Calculated U (kcal/hm²C)	Heat Transfer Area(m²)	Needed Tube (pcs.)	Exhaust Velocity (m/s) V_i	Exhaust Reynolds Number Re_i	Exhaust Nusselt Number Nu_i	Exhaust Heat transfer coefficient (kcal/m²C) h_i	Back pressure (mbar)
50	53,99	85,99	258,21	25,54	15940,55	46,68	70,23	8,49
53,99	57,33	79,63	239,12	27,58	17213,39	49,64	74,69	9,75
57,33	60,08	74,99	225,18	29,28	18278,54	52,08	78,36	10,86
60,08	62,32	71,55	214,87	30,69	19155,54	54,07	81,35	11,82
62,32	64,12	68,99	207,16	31,83	19868,44	55,67	83,77	12,62
64,12	65,56	67,05	201,35	32,75	20442,15	56,95	85,70	13,28
65,56	66,70	65,58	196,94	33,48	20900,24	57,97	87,23	13,82
66,70	67,60	64,46	193,57	34,07	21263,76	58,78	88,44	14,26
67,60	68,31	63,60	190,99	34,53	21550,85	59,41	89,39	14,61
68,31	68,86	62,94	189,01	34,89	21776,74	59,91	90,14	14,88
68,86	69,30	62,43	187,48	35,17	21953,94	60,30	90,73	15,10
69,30	69,64	62,04	186,31	35,39	22092,64	60,60	91,19	15,28
69,64	69,90	61,74	185,40	35,57	22201,01	60,84	91,55	15,41
69,90	70,11	61,50	184,69	35,70	22285,57	61,03	91,82	15,52
70,11	70,27	61,32	184,15	35,81	22351,48	61,17	92,04	15,60
70,27	70,40	61,18	183,73	35,89	22402,80	61,28	92,21	15,66
70,40	70,49	61,07	183,40	35,96	22442,75	61,37	92,34	15,71
70,49	70,57	60,99	183,15	36,01	22473,82	61,44	92,44	15,75
70,57	70,63	60,92	182,95	36,04	22497,99	61,49	92,52	15,78
70,63	70,67	60,87	182,80	36,07	22516,77	61,53	92,59	15,81
70,67	70,71	60,83	182,68	36,10	22531,36	61,56	92,63	15,83
70,71	70,74	60,80	182,59	36,12	22542,71	61,59	92,67	15,84
70,74	70,76	60,78	182,52	36,13	22551,52	61,61	92,70	15,85

The results of different configurations are summed in Table 4-9. As it can be observed, for the same exhaust and capacity, different designs can be utilized. It is required to select most suitable heat exchanger for the system where the heat exchanger will be introduced to. For this reason, the backpressure is a great tool. Before designing heat exchanger, systems backpressure availability should be studied thoroughly and a heat exchanger should be designed in respect to system.

Table 4-9. All configurations result

Tube Diameter	Tube Length	Capacity (kW/h)	Exhaust Flow (kg/h)	Heat Transfer Area(m²)	Needed Tube	Exhaust Heat transfer coefficient hi	Calculated U	Backpressure (mbar)
Ø26,9	2500	818	8500	119,6	772,7	49,8	35,9	3,1
Ø26,9	3000	818	8500	62,1	334,5	97,4	69,2	16,6
Ø26,9	3500	818	8500	37,1	171,4	166,2	115,8	64,5
Ø26,9	4000	818	8500	24,8	100,0	255,8	173,8	194,4
Ø33,7	2500	818	8500	348,4	1673,8	15,7	12,3	0,2
Ø33,7	3000	818	8500	173,3	693,9	31,8	24,8	1,1
Ø33,7	3500	818	8500	97,7	335,4	57,0	44,0	4,6
Ø33,7	4000	818	8500	60,8	182,5	92,7	70,8	15,9

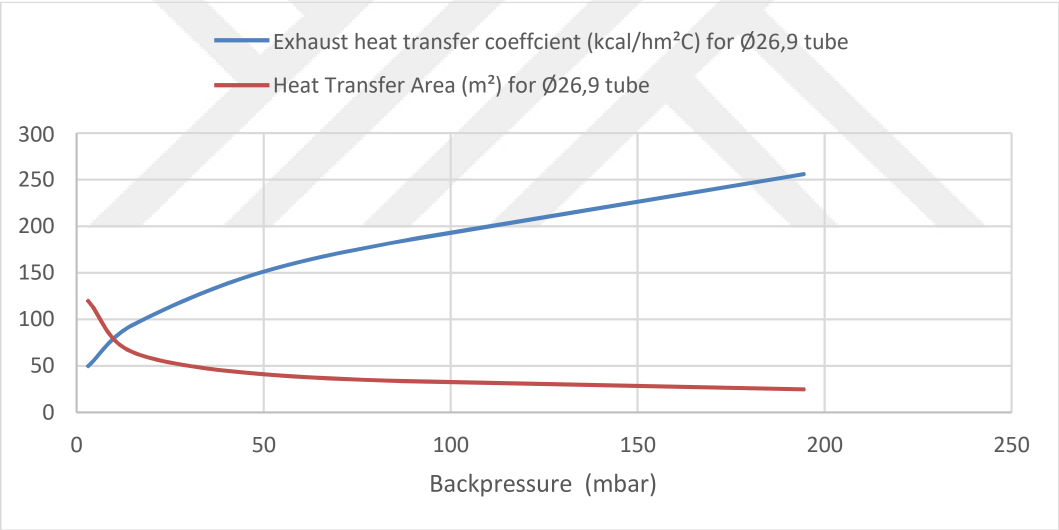

Figure 4-1. The change in heat transfer area and exhaust heat transfer coefficient with the change of backpressure for Ø26,9 tube.

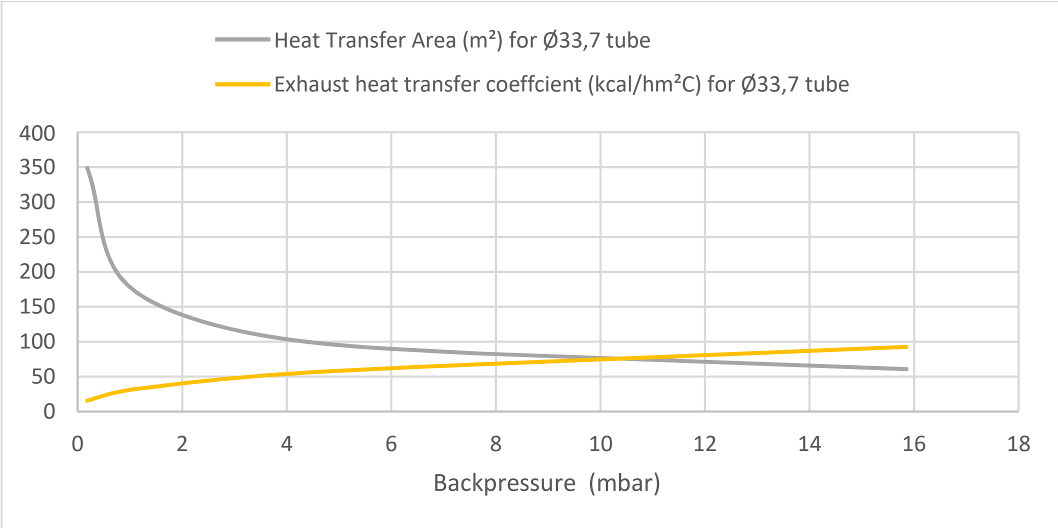

Figure 4-2. The change in heat transfer area and exhaust heat transfer coefficient with the change of backpressure for Ø33,7 tube

In both graph for Figure 4-1 and Figure 4-2, curve fitting studies is done and power trendline correlation between backpressures to heat transfer area and exhaust heat transfer coefficients are found. The equation for backpressure and heat transfer area is given in equation (4.1) and the equation for backpressure and exhaust heat transfer coefficient is given in equation (4.2).

$$y = 180{,}295 * x^{-0{,}386} \tag{4.1}$$

$$z = 31{,}518 * x^{0{,}3959} \tag{4.2}$$

In equations 4.1 and 4.2, x refers to backpressure. In equation 4.1, y is the heat transfer area and in 4.2, z is the exhaust transfer coefficient.

The calculations were done in two ways, Excel and Simulink. The Simulink designs are given in Figure 4-3 to Figure 4-6. In the Figure 4-3, the Cp calculation for exhaust side is done. The water side Cp calculation Is given in Figure 4-4. The calculated flow of water is shown in Figure 4-5. Theoretical simulations were done in Simulink.

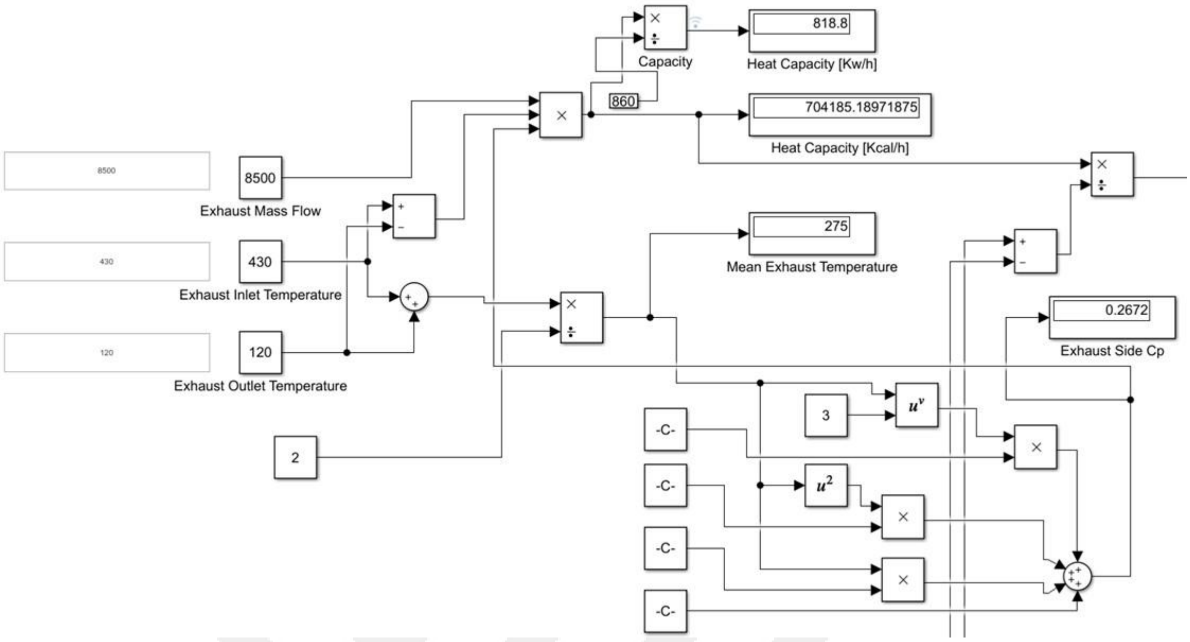

Figure 4-3. Exhaust Side Cp calculation on Simulink.

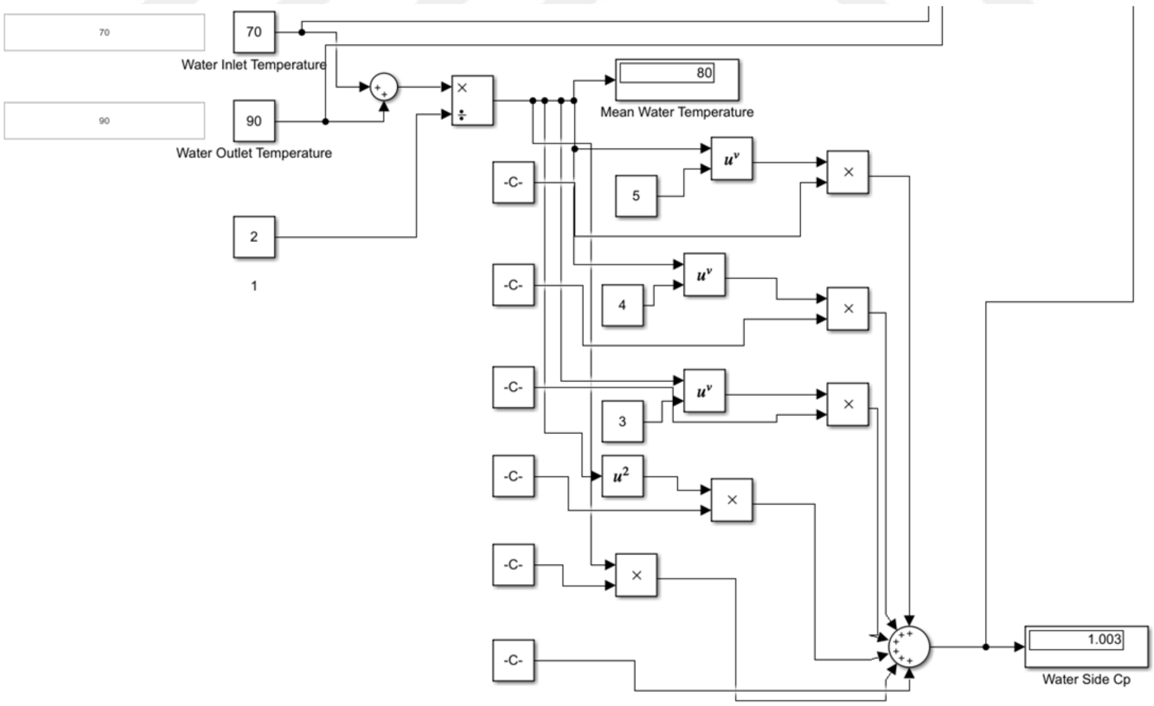

Figure 4-4. Water Side Cp calculation on Simulink.

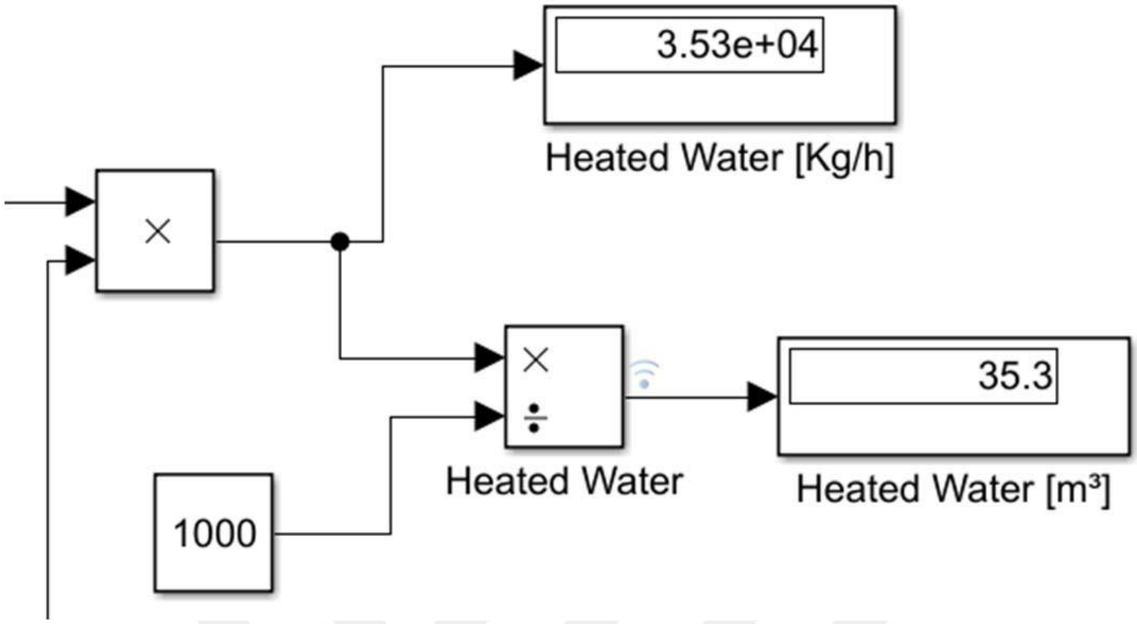

Figure 4-5. Calculated flow for water side on Simulink.

The interface for the heat exchanger calculations is done in Simulink. In this interface, there are five variables for calculation are shown. These variables are exhaust inlet and outlet temperature, exhaust mass flow, and water inlet and outlet temperature. Simulink calculates the amount of heated water mass flow and shows the heat exchanger's heat transfer capacity. The interface and data inlet on Simulink to visualize results are shown in Figure 4-6.

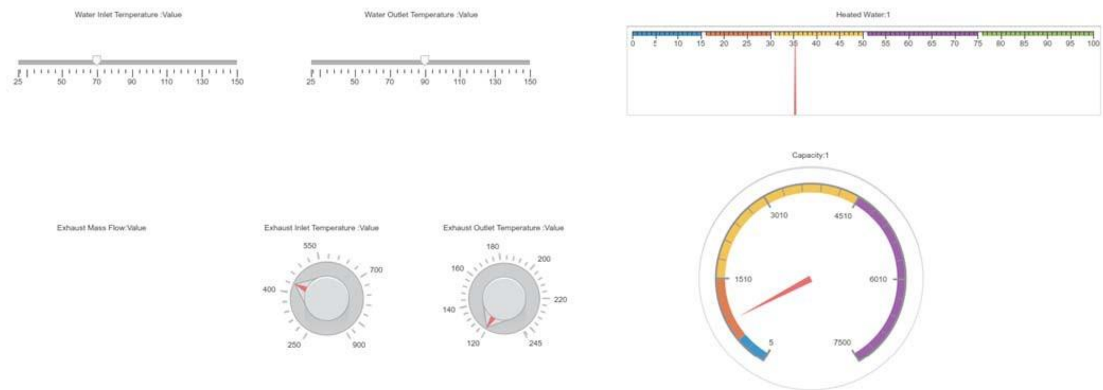

Figure 4-6. Data inlet on Simulink.

Figure 4-7. Manufacturing of heat exchanger.

Figure 4-8. Manufacturing of heat exchanger.

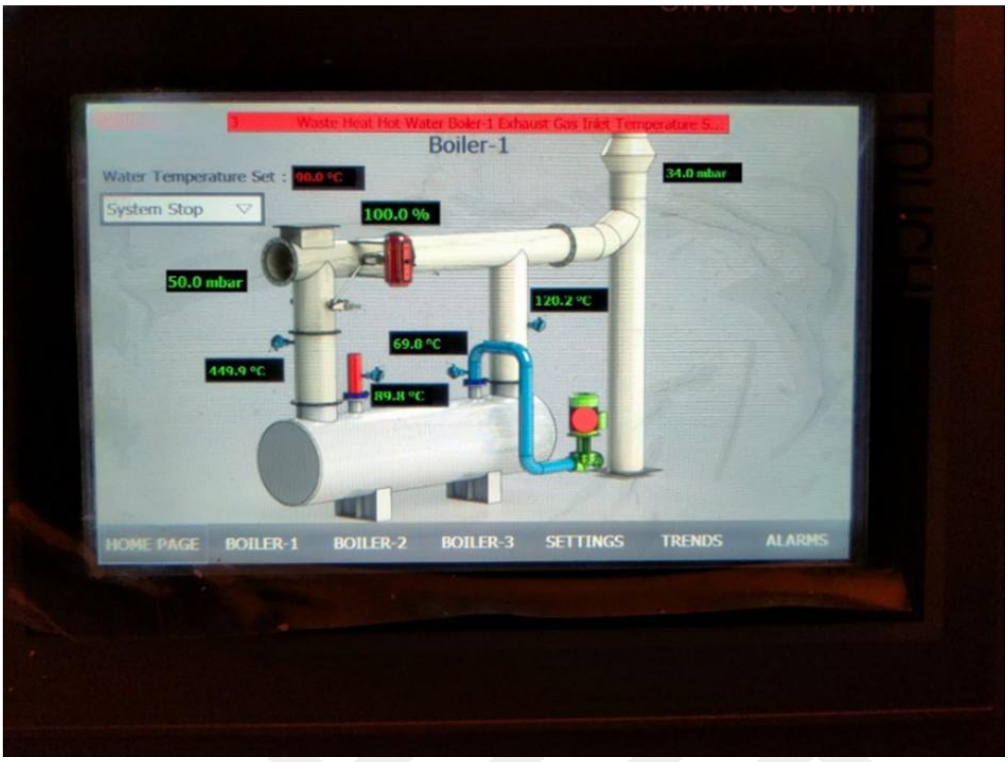

Figure 4-9. Control Panel

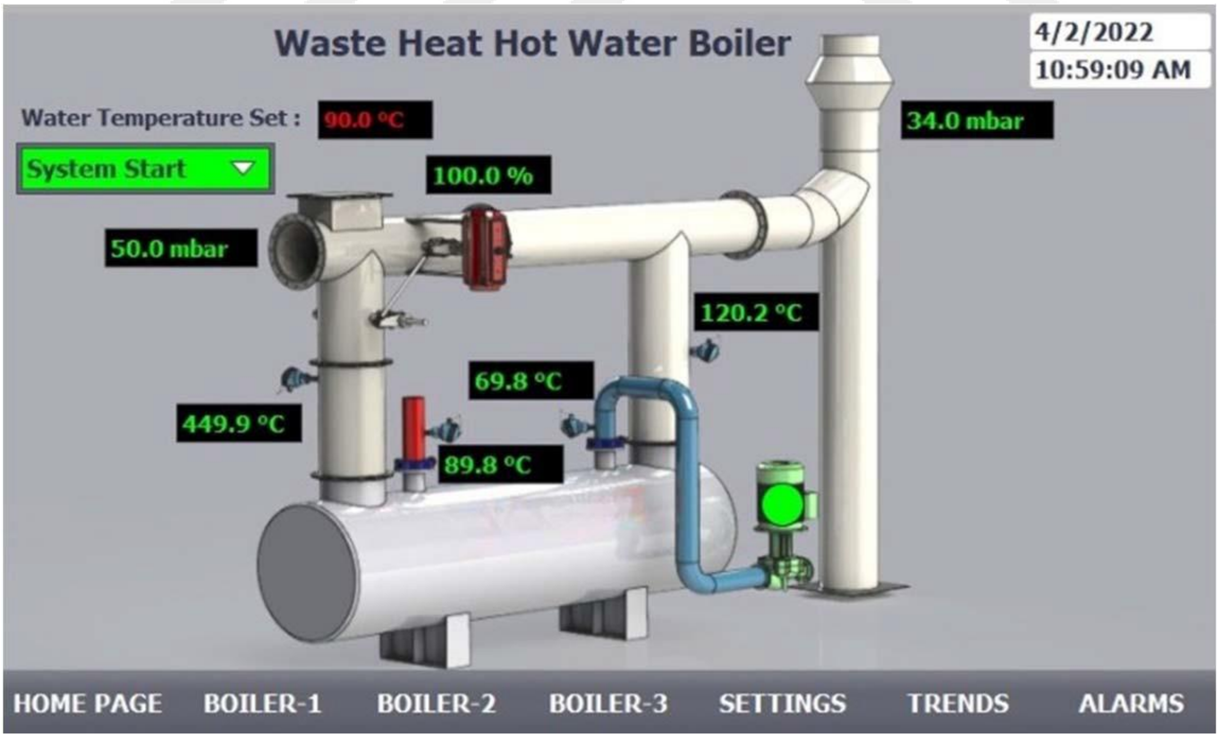

Figure 4-10. Control Panel on Scada System

Field measurements and a control panel view of the designed heat exchanger are given in Figures 4-9 and 4-10. The heat exchanger applied in the field is the heat exchanger with a back pressure of 16 mbar. The configuration of the manufactured heat exchanger is taken from Table 4-2. 335 tubes with Ø26.9 x 3000 x 3.6 mm specifications were used.

The reason for manufacturing the design given in Table 4-2 is clearly seen in the cost table below at Table 4-10. The most cost-effective belongs to the design in table 4-2. Designs marked with red cannot be selected because their mbar values are too high. While calculating the cost, tube price research was carried out in the market. It has been learned that the price of Ø26.9 x 3.2 pipe is $5.22 per meter. Ø33.7 x 3.2 pipe costs $7,105 per meter

Table 4-10. Manufacturing Costs

Designs	Tube Diameter (mm)	Tube Length (mm)	Heat Transfer Area(m^2)	Needed Tube (pcs.)	Backpressure (mbar)	Manufacturing Cost ($)
Table 4-1	Ø26,9	2500	119,6	772,7	3,1	10083,7
Table 4-2	Ø26,9	3000	62,1	334,5	16,6	5238,3
Table 4-3	Ø26,9	3500	37,1	171,4	64,5	3131,5
Table 4-4	Ø26,9	4000	24,8	100	194,4	2088,0
Table 4-5	Ø33,7	2500	348,4	1673,8	0,2	32806,5
Table 4-6	Ø33,7	3000	173,3	693,9	1,1	16320,5
Table 4-7	Ø33,7	3500	97,7	335,4	4,6	9203,4
Table 4-7	Ø33,7	4000	60,8	182,5	15,9	5723,2

For the designs, it is desired to make a selection by calculating the efficiency. But since the capacities of the selected designs are the same, their efficiency will not differ. For this reason, the selection was made by cost analysis.

Figure 4-11. Field Application

5. CONCLUSIONS

The energy needs are increasing, as well as the need to use this energy efficiently. The heat exchanger systems increase overall efficiency of systems with exhaust gas. The systems with higher efficiency can run more cheaply with amount of fuel saved compared to low efficiency systems, which have increased of importance for using heat exchanger systems in industry, factories and even in residential areas. Heat exchangers are in most cases unique to the systems they are to be assembled. The capacity, thus size and manufacture cost, depends on the system requirement. Thus, before a heat exchanger is to be introduced to the system, it must be calculated and designed with input values taken from system.

There are many input values in designing a heat exchanger, namely a few of them are: Input and output temperatures of the working fluids, flow amount, densities, specific heat values of working fluids, and working environment. However, the importance of backpressure in some instances is overlooked. The backpressure affects the heat transfer coefficient and is required to be calculated in detail to ensure heat transfer coefficient. Unless the effect of backpressure on the heat transfer coefficient is calculated, the capacity of the system may decrease or increase in value, preventing the heat exchanger to be the perfect fit for the system.

In addition, in industry, heat exchanger capacities are associated with the heating surface area. This can easily mislead engineers in comparing different heat exchangers in their search to find the most suitable heat exchanger for their system requirements. Two heat exchangers with the same heating surface area but different backpressure have different capacities. In this study, the effect of backpressure on heat exchanger calculations is shown and explained why they are crucial in heat exchanger selection.

The backpressure calculations are explained in detail through the thesis, and the formulas and equations used for deriving the calculations are given. A representative heat exchanger is designed for calculation purposes. The designed heat exchanger is a shell and tube heat exchanger with 335 pieces of 3-meter pipes with an outer diameter of Ø26.9 made of carbon steel. The studied exhaust values are taken from a 1.6 MW gas engine. The exhaust gas specifications are depended on the used natural gas chemical content; thus, natural gas is introduced and combustion calculations are done.

The results of calculations are given and systems with different backpressure are compared. The comparisons are given in graph format for easier observation. From the graph, it is observed that the increase of backpressure increases heat transfer coefficients and heat transfer area requirements decrease for same heat capacity.

With the design, 818 kW of thermal energy to be thrown into the atmosphere was recovered. Today, humanity is making a great effort to prevent environmental conditions from getting worse. With the recovery of this waste heat, 85 Sm^3/h natural gas was used effectively. Thanks to the effective use of natural gas, 0.402 tons of annual carbon emissions have been reduced.

6. RECOMMENDATIONS

The need for thermal energy is essential in many industries. For this reason, the installation of systems that will generate thermal energy falls to us engineers. Establishing the system that will meet the thermal need in the most efficient way should be considered as a future investment. Although the equipment that increases the efficiency seems to increase the initial investment price, it will be seen that these systems pay for themselves in 3-4 months if the depreciation calculation is made.

Equipment that is not selected correctly can cause unpredictable damage to the system and this will return as maintenance cost. Choosing equipment with low mbar values to avoid strain on the system will unnecessarily increase the equipment price, as can be seen by research, and the effort spent on labor in equipment production will increase. At the same time, choosing equipment with high mbar values so that the initial investment price is appropriate will cause damage to auxiliary equipment such as fans and pumps.

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Appendix

A-1. Natural Gas Combustion Analysis Datasheet

Alper Ünlü	Chemical Combustion Analysis	18.06.2022

Chemical Content of Gas

Gas	Content Ratio
Methane	90,03%
Ethane	3,50%
Propane	1,14%
n-Butane	0,28%
Isobutane	0,20%
n-Pentane	0,03%
Isopentane	0,04%
Neopentane	0,00%
n-Hexane	0,04%
Ethylene	0,00%
Propylene	0,00%
n-Butene (butylene)	0,00%
Isobutene	0,00%
n-Pentene	0,00%
Benzene	0,00%
Toluene	0,00%
Xylene	0,00%
Carbon	0,00%
Hydrogen	0,00%
Oxygen	0,00%
Carbon monoxide	0,00%
Carbon dioxide	0,56%
Acetylene	0,00%
Naphthalene	0,00%
Methyl alcohol	0,00%
Ethyl alcohol	0,00%
Ammonia	0,00%
Sulfur	0,00%

Analysis

Fuel Values

Lower Calorific Value	8.242	kcal/Sm³	Upper Calorific Value	9.126	kcal/Sm³
	8.694	kcal/Nm³		9.627	kcal/Nm³
	8.101	kcal/m³		8.970	kcal/m³
	10.922	kcal/kg		12.094	kcal/kg
	45.698	kj/kg		50.601	kj/kg
	34.484	kj/Sm³		38.183	kj/Sm³
	36.378	kj/Nm³		40.280	kj/Nm³
Theoretical Air Need				9,62	mol/ fuel mol
Theoretical O_2 Need				2,02	mol/ fuel mol

Additional %15 Combustion Needs for Air

Combustion Air Requirement (Dry)	11,06	mol/ fuel mol
Combustion Air Requirement (Wet)	11,27	mol/ fuel mol
Fuel Density	0,7546	kg/Sm³
	0,7960	kg/Nm³
	0,7417	kg/m³
Fuel Molecular Weight	17,77	g/mol
Amount of Fuel Burned	452,76	kg/h
	600,00	Sm³/h
Combustion Air Requirement (Wet)	6.636	Sm³/h
	6.751	m³/h

Exhaust Values

In Weight	Wet Air / Fuel Ratio	18,06	
	Wet Exhaust / Fuel Ratio	19,06	
	Wet Exhaust Weight	8.500	kg/h
	Wet Exhaust Volume	13.093	Nm³/h

Hydrogen

Exhaust sulfide	0,00%	**Density**		**1,18**	**kg/Sm³**	
				1,24	**kg/Nm³**	
Sulfur dioxide	0,00%	**Combustion Enthalpy Value**		**573**	**kcal/kg**	
Water vapor	0,00%	**Combustion Temperature**		**1.841**	**°C**	
Air	0,00%	**Exhaust Ratios by Weight**		**Exhaust Ratios by Volume**		
	100,00 %	**Carbon dioxide**	**13,44%**	**Carbon dioxide**		**8,48%**

System/Environment Information	
Ambient temperature	20 °C (273 K)
Atmospheric pressure	1,01325 bar
Gauge Pressure	0 bar
Air Humidity	40,00%

Water vapor	11,08%	Water vapor	17,08%
Nitrogen (atm)	72,61%	Nitrogen (atm)	71,96%
Oxygen	2,86%	Oxygen	2,48%

Example Cp Values

725°C for 0,310 kcal/ kg K	420°C for 0,288 kcal/ kg K
15°C for 0,263 kcal/ kg K	0°C for 0,262 kcal/ kg K