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Research Article

Investigation of Outdoor Air Temperature Effects on Heat Recovery in Cross Flow Heat Exchanger

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Abstract

A heat recovery device was designed and manufactured to provide air-to-air heat transfer by using a cross-flow heat exchanger on the pipe bundles. For this purpose, different cold air inlet temperature values and the temperature values of high temperature hot exhaust gases in the tube bundle heat exchanger were determined as operating parameters. According to the temperature increase of the cold air entering the heat exchanger; The kinematic viscosity values of the air increased and the maximum Reynolds number in the flow state of the air passing through the test heat exchanger was decreased. The average Nusselt numbers and average air convection coefficients in the flow of air decreased according to the temperature increase of the cold air. According to the temperature increase of the cold air entering the heat exchanger, both the analytical and experimental outlet temperature values of the cold air leaving the heat exchanger increase very close to each other. With this increase, the amount of heat transferred from the fluid to the fluid in the heat exchanger decreases. While the temperature value of the cold air entering the heat exchanger increases, the temperature difference value of the advancing air from the beginning to the outlet ($\Delta T = T_s - T_i$) decreases. Due to this decrease, a decrease was observed in the amount of heat transfer.

Keywords: cross-flow heat exchanger, heat recovery, industrial waste heat, pipe bundles

Çapraz Akışlı Isı Değiştiricisindeki Isı Geri Kazanımında Dış Hava Sıcaklık Etkilerinin İncelenmesi

Öz

Boru demetleri üzerinde çapraz akışlı ısı değiştiricisi kullanılarak havadan havaya ısı geçişini sağlayacak bir ısı geri kazanım cihazı tasarlanmış ve imal edilmiştir. Bu amaçla boru demetli ısı değiştiricisinde farklı soğuk hava giriş sıcaklık değerleri ile yüksek sıcaklıktaki sıcak egzoz gazlarının sıcaklık değerleri çalışma parametreleri olarak belirlenmiştir. Isı değiştiricisine giren soğuk havanın sıcaklık artışına göre; havanın kinematik viskozite değerleri artmakta ve deney ısı değiştiricisinden geçen havanın akış halindeki maksimum Reynolds sayısında azalma görülmüştür. Havanın akış halindeki ortalama Nusselt sayıları ve ortalama hava taşınım katsayıları soğuk havanın sıcaklık artışına göre azalma elde edilmiştir. Isı değiştiricisine giren soğuk havanın sıcaklık artışına göre çıkan soğuk havanın hem analitik hemde deneysel çıkış sıcaklık değerleri bir birlerine çok yakın değerlerde artmaktadır. Bu artış ile ısı değiştiricisinde akışkandan akışkana transfer edilen ısı miktarı azalmaktadır. Isı değiştiricisine giren soğuk havanın sıcaklık değeri artarken başlangıçtan itibaren çıkışa kadarki ilerleyen havanın sıcaklık farkı değeri olan ($\Delta T = T_s - T_i$) azalmaktadır. Bu azalış sebebi ile ısı geçiş miktarında da azalma görülmüştür.

Anahtar Kelimeler: çapraz akışlı ısı eşanjörü, ısı geri kazanımı, endüstriyel atık ısı, boru demetleri

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1. Introduction

Since our country is a foreign-dependent country in terms of energy resources, it makes it necessary to use energy efficiently in all stages from production to consumption. For this reason, energy saving is very important. Today, heat exchangers are widely used in the industry, especially in thermal power plants, ceramic, cement, iron-steel and textile industries in order to utilize waste heat. As in developed countries, they are preferred in many industrial applications in our country, especially for heat recovery from low-temperature waste heat[1].

If we pay attention to the developed countries, the places where energy is used the most are industrial enterprises. Approximately 30-40% of the existing energy is used in industrial enterprises. Most of the energy consumed in the industrial manifests itself as process heat. Process heat is the heat energy used directly in the stages from the entry of the raw material product to the exit of the finished product. The inactive heat energy has a value between 70-80% of the total energy consumed. As it can be understood from these amounts, it has an important potential in terms of energy saving, since the energy lost is insignificant[1,2].

38% of the energy consumption in our country is consumed by industrial enterprises. 75% of this consumed energy is spent for process heat. According to the usage areas, the thermal energy used in the industry has different temperatures according to the processes. While it is around 1200 °C in the ceramic, cement, iron-steel industry; It is used in the textile, chemical and food industries at 150-200 °C. While more than 60% of the energy used in the industry is used below the temperature value of 300 °C, around 20-30% is used below the temperature value of 150 °C. One of the sectors where waste heat is generated to a large extent is the textile sector. Hot water is needed in the temperature range of 70-200 °C for heating-drying in the textile industry. Due to the excess of energy inputs in the industry, the chance to compete in the world market has become one of the important policies of the enterprises today to be able to produce high quality and cheaper by making the waste heat useful heat. In the industry, how much electricity and how much heat energy is spent as input for profit per unit product is carefully calculated. One of the most important moves that our country's developing industrialization sector can make for the energy bottleneck is waste heat recovery[1].

We can list the benefits of waste heat recovery for countries as follows;

•For the health; It reduces air-environmental pollution with the reduction of fuel consumption and the scarcity of exhaust gases released into the atmosphere,

• In terms of plants and other living things; It prevents thermal (thermal) pollution of the environment by releasing waste heat with reduced temperature value to the environment thanks to the recovery of waste heat at high temperature,

• In terms of security and peace; It ensures that aerobic treatment is carried out in a safe manner, as the temperature of the fluid with waste heat recovery enters the treatment plant at the lowest temperature value it should be,

• In terms of operating profitability; It provides profitability in operation by reducing fuel consumption,

• In terms of process-process time; By shortening the process time in the industry, the time per unit product is shortened and it ensures the production of more products,

• In terms of earnings; It provides profit by ensuring the economical use of our own resources and workforce,

• In terms of foreign exchange loss; With the economical use of existing energy resources, it prevents the loss of foreign currency,

• In terms of competition; It provides an increase in competition due to the reduction of energy inputs on the product and the reduction of its costs,

• In terms of economic vitality; With the increase of profitable companies in the industry, it provides economic vitality by encouraging the investor[1].

Waste heat direct utilization systems are both cheaper and easier to implement, despite other waste heat systems. However, this system has important drawbacks that limit its use in many places. For example, considering the system of directly drying process raw materials with flue gases, flue gases generally contain sulfur and moisture. If the flue gas temperature drops below the dew point temperature during the drying process, the acid that will form will both adversely affect the quality of the products and cause corrosion on the chimney and similar surfaces. As long as the production is not affected, this method is preferred to other systems economically only in cases where there is a corrosion problem[3,4].

The main method of recovery of waste heat in industrial factories is the use of heat exchangers. In the determination of an industrial heat exchanger, the heat transfer capacity, the temperatures of the fluids, the pressure drops that can be allowed in each fluid circuit, the properties of the fluids entering the heat exchanger and the volumetric flow rates should be known. These values are the design parameters of the heat exchanger.

The final design will be achieved by a compromise of the triad of pressure drop, heat exchanger efficiency and cost. It is the comparison of the maintenance and operating costs of the whole system against the fixed costs that guide the decisions in the final design. Thus, the total costs can be minimized.

The basic parameters in the selection of the most suitable waste heat device can be listed as follows:

- Temperature of waste heat fluid, - Flow rate of waste heat fluid, - The lowest allowable temperature for the waste heat fluid, - Chemical composition of the heated fluid, - The highest allowable temperature of the heated fluid, - If control is required, control temperature[4].

It is possible to recover waste heat at temperatures between 200-500 °C with heat exchangers and other heat equipment[4].

The recovery of waste energy is achieved by using the waste gas of the combustion installation so that heat is needed for later use and regains the existing latent heat. While

installing these systems, economic conditions should be taken into consideration and the return time of waste heat and life-cost calculations should be considered while applying. Researchers have used it in incinerators, glass furnaces, cement factories, etc. investigated the methods of energy recovery from waste heat. Economizers only recover sensible heat, and waste heat boilers work optimally at temperatures above 300 ºC. The use of waste heat from the cooling water used to cool the electric generators used to heat the greenhouses is an example of a heat recovery system. Apart from that, the gas-gas heat exchanger heats the gas inside with the waste gas thrown out, and energy is recovered. As a result of the use of these burned flue gases, there is a 6% reduction in fuel consumption. Heat exchangers such as heat wheels, heat pipes and recuperators are used in gas-to-gas heat recovery systems. Some researchers recommend the use of a liquid-to-liquid heat exchanger for heat recovery in high pressure gas compressors. Thus, in order to increase the thermal efficiency of the gas turbine, this result is achieved by pre-cooling the incoming gas and intercooling in the compressor stages[4].
It is possible to recover the energy by using the waste gas thrown out in the heat recovery systems under the following conditions;

1. The waste gas discharged must be as clean as possible. Because contamination in the heat exchanger affects the heat transfer negatively, 2. The dew point temperature should not be dropped below. Because condensation causes corrosion, 3. Thermal stress should not be caused. Otherwise, the material gets tired[4].

Failure to meet these conditions described above reduces the life and efficiency of the heat recovery system.

Depending on the lowest temperature value of the exhaust gas, waste heat recovery is possible. Because these gases can be cooled down to the desired temperature value. The polluted gases contain Sulfuroxide and nitrogenoxide. If these are cooled below 150 ºC, they condense to form sulfuric and nitric acids. This dangerous situation should be avoided. Otherwise, acid solutions will destroy the heat recovery equipment[4].

An improved methodology is presented to determine the amounts that can be recovered from the dryer section of a typical pulp and paper mill. A series of mathematical models have been created that closely represent the various stages of the drying process, based on the laws of thermo-fluid dynamics and heat transfer. In calculating the recoverable amounts of the two sources, a set of equations was developed along with the measured real operating parameters and used[5].

To achieve waste heat recovery, heat pumps include suitable switching or reversing valves to provide cooling, thereby recovering the expected final heat product. In this study, an overview of reverse heat pump systems used for waste heat recovery is presented. Both direct and indirect benefits have been identified regarding the benefits of waste heat recovery systems. Direct benefits include improving the efficiency of a system, while indirect benefits include reducing pollution, reducing equipment sizes and reducing auxiliary energy consumption. These findings have created a comprehensive knowledge base on reverse heat pumps for waste heat recovery[6].

Most researchers focus on how to efficiently recover waste exhaust heat using different heat exchangers, but few consider the relationship between heat factors and the heating load among these factors, including the effect of outside air on heat transfer parameters. During the actual heating time, the heating load and efficiency changes daily. Therefore, the temperature value of the outside air has a very important effect on the efficiency and economy of the system. Therefore, based on the heating load characteristics, this article came up with the design idea of a heat recovery device that will provide cross-flow air-to-air heat transfer over a bundle of pipes to recover the waste exhaust heat.

In cases where direct use of waste heat is not possible, various heat exchanger systems are used that provide heat transfer. For this purpose, in this study, a heat recovery device was designed and manufactured to provide cross-flow air-to-air heat transfer on pipe bundles made of copper material. While the high temperature air in the form of waste heat passes between the pipe bundles, the low temperature air taken from the outside environment is passed through the pipe bundles and the temperature of the air rises due to many heat transfer factors. One of these factors is the outside temperature. In our study, the maximum Reynolds numbers, average Nusselt numbers and average air convection coefficients were examined and interpreted among the heat transfer parameters of the outdoor air temperature.

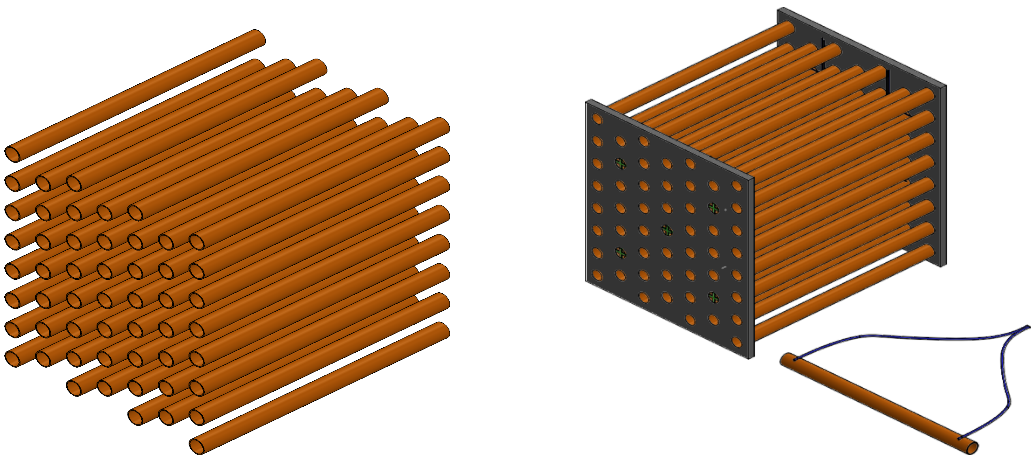
2. Material and Methods

Air to air heat recovery is the process of recovering the energy contained in a high temperature moist air stream into another low temperature moist or slightly humid air stream. This process keeps the indoor environment quality at an acceptable level and comfort. Heat recovery ventilation systems continuously transmit fresh hot air into the environment with the heat they transfer from the exhaust air. Energy can be recovered from either sensible heat (temperature only) or latent heat (humidity) or both from mixed and multiple sources[7].

Heat recovery systems from air to air can be grouped into three classes according to their applications: from process to process, from process to comfort and from comfort to comfort. These; In dryers and furnaces, there is a process-to-process application. Heat is taken from the exhaust air and transferred to the supply air. There is an application from process to comfort in enterprises where heated air is exhausted, such as foundries, pulp and paper mills. In winter, the heat from the process is captured and transferred to the supply air used for heating the environment. While a full energy recovery is desired in process-to-process applications, it may be necessary to reduce the recovery rate, especially in warm weather, in order not to overheat the supply air in process-to-comfort applications. There is an application from comfort to comfort in businesses where the ventilation of residences, swimming pools, operating rooms, animal shelters and plant production greenhouses are made. It is aimed to reduce the enthalpy in summer and increase it in winter by performing heat exchange between the supply air used in ventilation and the exhaust air streams[7].

The basis of the methods of re-utilizing the heat energy lost during ventilation is to recover the heat energy in the waste polluted air by using heat exchangers from air to air. The heat energy existing in the exhaust air, which is seen as a heat carrier in many systems, is thrown to the outside atmosphere from the indoor environment by means of conventional ventilators. However, in the winter months, the heat energy in the exhaust air thrown into the cold air entering the interior is transferred from the air to the air by using heat exchangers and the heat is recovered. Heat recovery from air to air was carried out using a cross-flow heat exchanger on pipe bundles manufactured and designed for this study.

The schematic views of the cross-flow heat exchanger on the pipe bundles manufactured and designed for this study are shown below. In Figure 2.1, 320 mm long, 16 mm outer diameter, 1 mm wall thickness, 53 pieces of copper material shifted in-line pipe bundle is designed.



(a) (b) (c)

Figure 2.1 a- The view of the shifted row pipe bundle, b-both sides of the pipes connected with tube-sheets, c- 5 pipes connected with mutual temperature gauges.

While the hot fluid containing waste heat moves over and through the pipes, the cold fluid that needs to be freshly heated is passed through the pipe bundles in such a way that it cross-flows with respect to this hot fluid. In order to calculate the heat transfer values of the heat exchanger consisting of a slid tube bundle, copper with a high thermal conductivity value was chosen as the material of the pipes. In addition, J-type precision temperature measuring thermocouples are connected to the opposite ends of 5 of them in the shifted-row tube bundle seen in Figure 2.1.

The dimensions of the displaced row tube bundles and the tube-sheets connecting the ends of these tube bundles are shown in Figure 2.2 below. The dimensions of the designed shifted in-line cross-flow heat exchanger are considered in such a way that the diameter of a tube in the tube bundle is proportional to 16 mm.

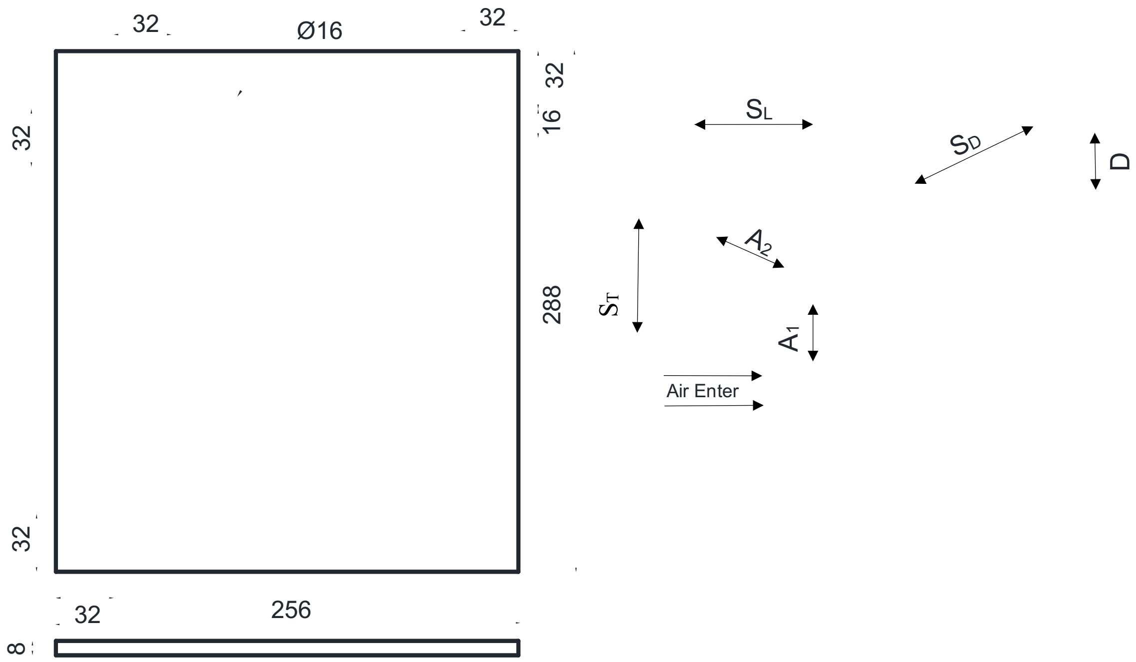


Figure 2.2 Arrangement of the displaced row pipe bundle and view of the dimensions of the tube-sheets.

A total of 10 thermocouples, 5 of which are close to the left tube-sheet side and 5 to the right side of the tube-sheet, which can precisely measure temperature values, are shown in Figure 2.3.

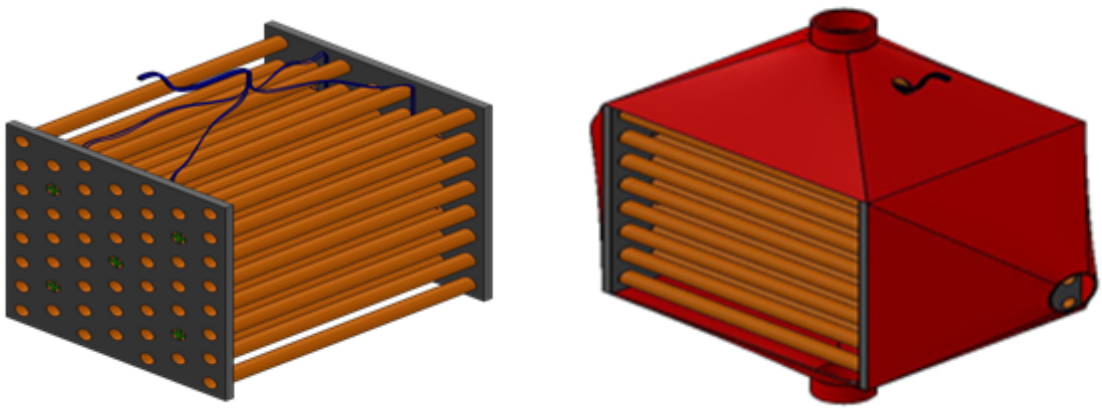


Figure 2.3. a-Appearance of 10 thermocouples whose temperature values can be measured precisely (on the left), b- View of the designed heat exchanger and 10 thermocouples attached to the pipes inside (on the right).

The heat exchanger, to which the inner design and thermocouples are connected, is covered with a metal sheet plate with a wall thickness of 0.5 mm, as seen in Figure 2.3. In order to reduce the heat loss from the outer wall of this coating to the environment, the surface of the heat exchanger was completely insulated and experiments were carried out in the insulated state.

J-type thermocouples have been installed in the designed heat exchanger so that the temperature values in the inlet and outlet regions of the hot air and cold air passing through the heat exchanger can be read and recorded with precision.

A total of 15 thermocouples, together with the thermocouple that reads the temperature value of the environment where the experiment was conducted, were connected to the temperature measuring device. This temperature measuring device was measured with the OMB-TEMPSCAN-1100 temperature measuring device, which has 32 channels and each channel can collect, read and save 100 data per second.

The inlet velocities of hot air and cold air entering the heat exchanger were measured and recorded moment by moment with Tenmars TM-4001 air velocity and temperature gauge thermal anemometer. Thus, it was ensured that the air velocities entering the heat exchanger remained constant at the desired value during the experiment. Hot and cold air entering the heat exchanger is provided separately with two Black & Decker KX2001K-XX air guns.

The experiments were carried out in a special room in the Hydromechanics Laboratory of the Mechanical Engineering Department. The experiment was carried out in a time period of exactly 1 year, without consuming energy, in a way that the temperature of the private room and its surroundings remained constant.

Figure 2.4 The view of the air guns used in the experiments and the anemometer reading the air velocity values.

2.1. Mathematical Studies and Formulations

While one of the fluids with different temperatures passes through the pipes, the other one passes through the pipes and the heat transfer by convection in cross flow over the pipe bundles from a pipe bundle arranged (triangle) in the direction of the flow velocity of the fluid V. Experimental design has been carried out in industrial applications. The arrangement of the displaced pipe bundle is determined by the pipe diameter (D), the distance between the pipe axes perpendicular to the flow (S_T), the distance parallel to the flow (S_L), and the distance (S_D) on the triangular diagonal plane between the displaced (triangular) pipes. It is seen in Fig.2.2.

The heat transfer coefficient of the pipes in the first row of the pipe bundle at the entrance point of the fluid to the heat exchanger is equivalent to the heat transfer coefficient of a single pipe in cross flow, while the heat transfer coefficient of the pipes perpendicular to the flow in the inner rows is greater than that of the pipes in the front row. It can be thought that the heat transfer coefficient value in the pipes after the first few rows of pipes increases due to the fact that the hot fluid flowing over the pipes perpendicular to the flow, which comes after the first few rows of pipes perpendicular to the flow, acts as a "Turbulence Grid". While the heat transfer coefficient value in the pipes increases after the first few rows of pipes, very little heat increase occurs after 5-6 rows perpendicular to the flow. While the heat transfer coefficient changes in the pipes along the pipe bundle perpendicular to the flow from the entry point to the heat exchanger, it is generally desired to calculate the "Average Heat Transfer Coefficient ($\bar{h}$)" for the entire pipe bundle of the heat exchanger in terms of engineering applications [8].

At the point of entry of the fluid to the heat exchanger, the row of tube bundles perpendicular to the flow over the surface of the tube bundles consisting of 10 or more rows (N_L > 10) the dimensionless average "Nusselt number $\overline{(\mathrm{Nu_D})}$" equivalent to the surface temperature change for cross-air flow [9] It was obtained in the form of the following relation.

$$\overline{Nu_D} = C_1\ Re_{D,max}^{m} \quad \overrightarrow{\substack{Valıdıty\ Lımıts \\ of\ the\ Equatıon}} \quad \begin{bmatrix} N_L \geq 10 \\ 2000 < Re_{D\,Max} < 4000 \\ Pr = 0{,}7 \end{bmatrix} \tag{1}$$

In the equation;

$Re_{D,\,max}$: Reynolds number corresponding to the maximum velocity of the air passing between the tube bundles in the air heat exchanger,

Pr: Prandtl number, which determines the thermal ratio of the viscous forces formed by the physical properties of the air,

N_L: Number of successive alignments of planes perpendicular to the direction the fluid flows (N_L=7 in the heat exchanger)

The (C_1) and (m) values are taken from the table below according to the $\underbrace{(S_T/D)}_{=2}$ and $\underbrace{(S_L/D)}_{=2}$ values in the dimensions of the heat exchanger designed in our study.

Table 1. Numerical values of constants in the expression (Nu_D) in airflow over a bundle of pipes consisting of 10 or more rows [7].

	$(S_T/D) = 2$	
	C_1	m
$(S_L/D) = 2$	0,482	0,556

What is the importance of the friction coefficient created by the fluid passing between the pipe bundles in the velocity boundary layer; The importance of the Nusselt number in the thermal boundary layer is the same. The mean Nusselt number $\overline{Nu_D}$, depends on the functions of the Reynolds number passing between the tube bundles of the fluid and the dimensionless Prandtl number, which consists of the physical properties of the fluid passing between the tube bundles. That's why; It has been proved by the researchers that the above Average Nusselt Number value is both dependent on the dimensionless Prandtl number and so that different fluids can be used, by multiplying the equation(1) with a numerical value such as 1.13 $Pr^{1/3}$, a solution convergent to the values in the applications is obtained. The above statement of Average Nusselt number $\overline{Nu_D}$, is valid for applications and for different fluids;

$$\overline{Nu_D} = (1{,}13)C_1 \; Re_{D,max}^{m} \; Pr^{1/3} \quad \xrightarrow[of\ the\ Equation]{Validity\ Limits} \begin{bmatrix} N_L \geq 10 \\ 2000 < Re_{D\,Max} < 40000 \\ Pr = 0{,}7 \end{bmatrix} \tag{2}$$

is seen.

The maximum Reynolds number ($Re_{D\,Max}$) in Equation(2) is the value of the fluid advancing in the heat exchanger at the maximum velocity as it passes over the pipes. V_{max} is the resulting value. The V_{max} value depends on the tube bundle arrangement of the designed heat exchanger.

$$\underbrace{2(S_D - D)}_{(S_D)unknown} < \underbrace{(S_T - D)}_{0{,}016} \tag{3}$$

The place where V_{max} occurs is determined by mathematically examining whether it occurs between the pipes on the plane perpendicular to the flow (A1) or on the diagonal plane between the shifted pipes (A2) in the pipe bundle arrays, the inequality of which can be seen in Figure 2.2. The factor of 2 in the left term of the inequality equation [3]; It is the expression of the separation of the fluid flowing from the (A1) plane to the (A2) plane into 2 separate branches. According to these statements; Let's try to find the diagonal plane distance (S_D) value between the displaced pipes developed by Grimison, 1937 and V_{max} expressions according to this value.

$$S_D = \underbrace{\left[S_L^2 + \left[\frac{S_T}{2}\right]^2\right]^{\frac{1}{2}}}_{0,0358} < \underbrace{\frac{S_T+D}{2}}_{0,0240} \quad (4)$$

Since the inequality does not occur, that is, 0.0358 < 0.0240; V_{max} value does not occur in the diagonal range of (A2) pipes. The V_{max} value of the fluid moving inside the designed heat exchanger is realized in the range (A1) (between the pipes on the plane perpendicular to the flow). Thus, the value of the maximum velocity at the flow velocity of the fluid flowing in the (A1) range is obtained according to the equation below.

$$V_{max} = \frac{S_T}{S_D - D} V \quad (5)$$

The most recent test results of the mean Nusselt number according to the V_{max} value close to the true values are given by Whitaker, 1972[10] and Zhukauskas, 1972[11].

$$\overline{Nu_D} = C\, Re_{D,max}^m\, Pr^{0,36} \left[\frac{Pr}{Pr_s}\right]^{1/4} \xrightarrow{\text{Validity Limits of the Equation}} \left[\begin{array}{c} N_L \geq 20 \\ 1000 < Re_{D\,Max} < 2x10^6 \\ 0,7 < Pr < 500 \end{array}\right] \quad (6)$$

Prs : Prandtl number of the fluid at the tube bundle surface temperature (T_s), constants (C) and (m) seen in equation [6] are given in Table 3.2.

Table 2. Numerical values of the constants in the equation of the air flow (NuD) over a bundle of pipes consisting of 20 or more rows [11].

	$\underbrace{Re_{D\,max}} \cong 10^3 - 2x10^5$	
	$C = \underbrace{0,35(S_T/S_L)^{1/5}}$	m
$\underbrace{(S_T)/(S_L)} < 2$	0,35	0,60

Since the pipe row in the planes perpendicular to the direction of the flow of the fluid entering the test device we have prepared is 7< 20; Using the equation below, values close to the experimental results are obtained.

$$\overline{Nu_D}\,\Big|_{\cdot(N_L<20)} = C_2\,\overline{Nu_D}\Big|_{\cdot(N_L\geq 20)} \quad (7)$$

The average Nusselt number ($\overline{Nu_D}$) that we will calculate in equation(7) should be multiplied by the coefficient (C2) shown in Table 3.3 before the expression¯. According to this;

Table 3. Numerical values of the constant (C_2) in the expression (Nu_D) in the air flow over the tube bundle consisting of 20 or less ($N_L < 20$) rows [11].

	$N_L = 7$
C_2	0,95

The equation of the average Nusselt number ($\overline{Nu_D}$) that creates the surface temperature change of the fluid passing through the 7 rows of pipes along the plane surfaces perpendicular to the direction in which the fluid entering the prepared heat exchanger flows is;

$$\overline{Nu_D} = C_2 C\, Re_{D,max}^{m}\, Pr^{0,36} \left[\frac{Pr}{Pr_s}\right]^{1/4} \xrightarrow[\text{of the Equation}]{\text{Validity Limits}} \begin{bmatrix} N_L = 7 \\ 1000 < Re_{D\,Max} < 2x10^6 \\ 0,7 < Pr < 500 \end{bmatrix} \tag{8}$$

As a result, the final state of the Nusselt number ($\overline{Nu_D}$) to the average is shown in equation(8).

In the experimental study, the value of the inlet velocity of the hot air entering the heat exchanger was kept constant at V=5 m/s. As seen in Figure 2.1, a total of 10 temperature measuring thermocouples were mounted on the pipe surface, 5 pairs (2 on the left and right side of the outer surface of a pipe) out of the pipe bundle consisting of a total of 53 copper pipes. At the time of the experiment, these temperature values were collected 100 pieces of data per second and their averages were recorded. The temperature averages on the surface of 10 copper pipes were tried to be kept constant at 59 °C. Experiments and analytical calculations were carried out by keeping the surface temperature of the pipe bundle (T_S= 59 °C) constant.

3. Results and Discussion

The thermal performance (T_{ci}) of all heat exchangers depends on the fresh air inlet temperature, the exhaust air velocity (V) and the surface temperature (T_s) of the exhaust air in the pipe bundle. As mentioned in the Introduction part of the study, the variation of the outdoor temperature (T_{ci}) value in applications realized with waste heat recovery was investigated in this study.

The variation of the maximum Reynolds number obtained according to the inlet temperature values of the cold air entering the heat exchanger in experiments and analytical studies is shown in Figure 4.1. The variation of the maximum Reynolds number formed by the maximum velocity value of the fluid passing through the gap (A1) shown in Figure 2.2 between the pipes on the plane surface perpendicular to the flow, according to the changing outdoor temperature value. As the inlet temperature of the cold air entering the heat exchanger increases, the average temperature value of the air advancing in the heat exchanger also increases. With the increase in the average temperature of the air, the kinematic viscosity values, which are the physical properties of the fluid, also increase. It is seen that the maximum Reynolds number in the flow state of the fluid passing through the test device decreases with the increase in the kinematic viscosity of the air.

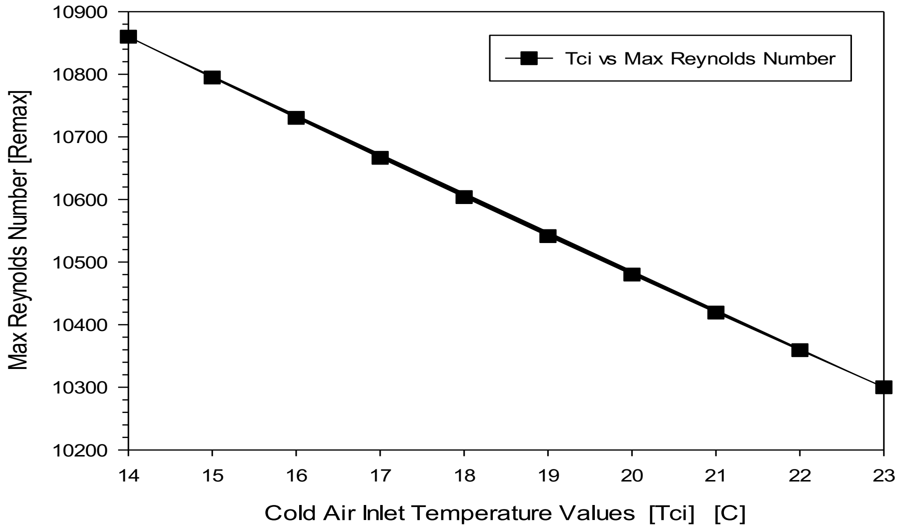


Figure 4.1 The variation of the maximum Reynolds number according to the inlet temperature of the cold air entering the heat exchanger.

The variation of the average Nusselt number, which includes the heat transfer coefficient created by the hot air passing between the pipes in the pipe bundle consisting of the shifted pipes, on the surface of the pipes is shown in Figure 4.2.

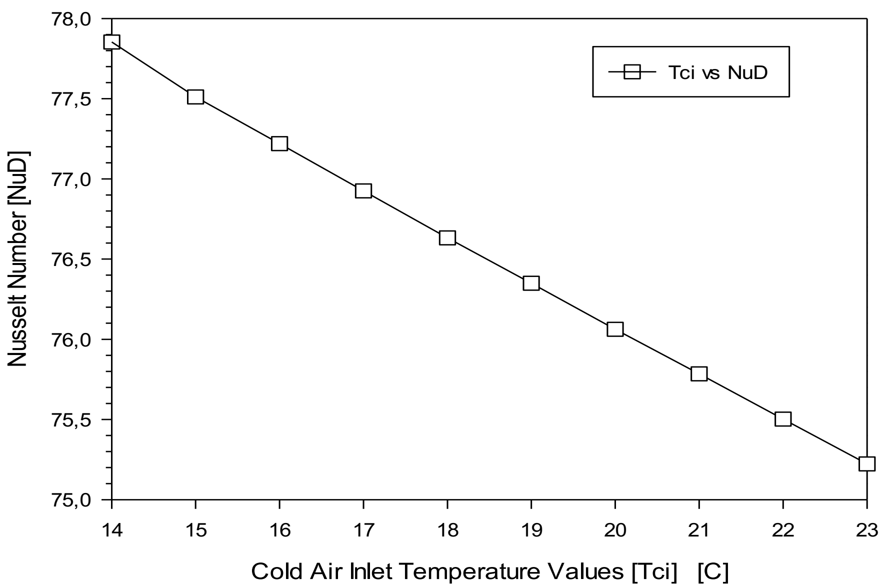


Figure 4.2 Change of the average Nusselt number according to the inlet temperature values of the cold air entering the heat exchanger.

Due to the increase in the inlet temperature of the cold fluid, it will cause an increase in the average temperature of the fluid. This temperature increase will increase the kinematic viscosity of the air and the Prandtl number will increase. Since the increase in the aforementioned Prandtl number cannot compensate for the decrease in the maximum Reynolds number as the fluid passes between the tube bundles, a decrease in the calculated average Nusselt number is observed.

The variation of the values found from the average Nusselt number, whose average convection coefficient ($\bar{h}$) value is calculated on all surfaces of the pipes in the pipe bundle and whose graph is shown in Figure 4.2, according to the inlet temperature values of the cold air to the heat exchanger is given in Figure 4.3.

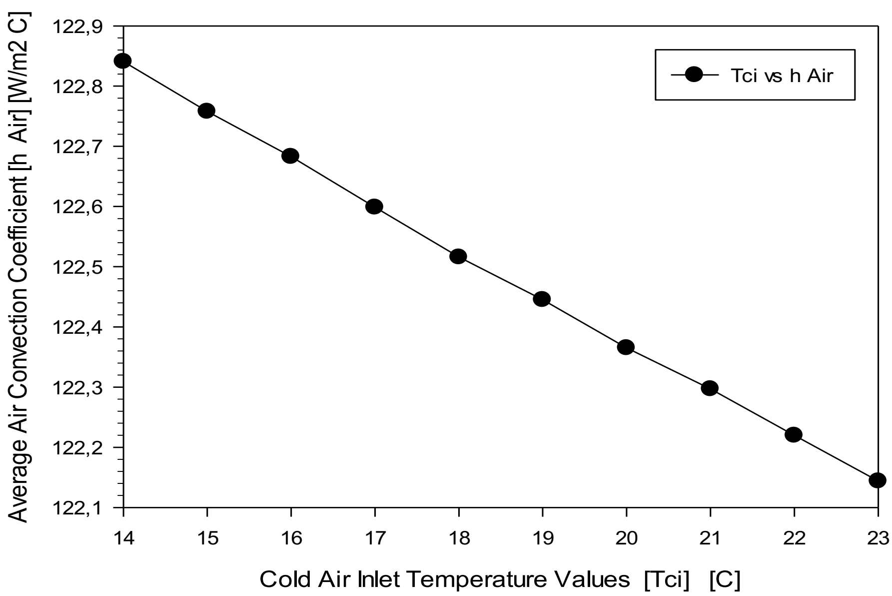


Figure 4.3 Change of the Average Convection Coefficient according to the inlet temperature values of the cold air entering the heat exchanger.

In the experimental and analytical study, it is seen that the calculated average transport coefficient decreases in direct proportion with the calculated average Nusselt number. In addition, as mentioned in the paragraphs above; It can be thought that the heat transfer coefficient value in the pipes after the first few rows of pipes increases due to the fact that the hot fluid flowing over the pipes perpendicular to the flow, which comes after the first few rows of pipes perpendicular to the flow, acts as a "Turbulence Grid". After the first few rows of pipes, the increase in the heat transfer coefficient value in the pipes occurred, while after 5-6 rows perpendicular to the flow, very little temperature increase was observed, even in a linear state.

Some of the design parameters of the designed heat exchanger and their dimensions, the flow velocity value of the fluid, the number of tube bundles, the average convection heat transmission coefficient, the inlet temperature of the cold fluid and the surface temperature values of the pipes in the tube bundle are known, the cold fluid entering the heat exchanger, the exit temperature of the heat exchanger. (T_{co}) has made it computable. According to these data, the outlet temperature of the fluid to be heated is calculated by the equation [9] given below.

$$\frac{(T_s - T_o)}{(T_s - T_i)} = \exp\left(-\frac{\pi\, D\, N\, \bar{h}}{\rho\, V\, N_T\, S_T\, C_p}\right) \tag{9}$$

The outlet temperature of the cold air from the heat exchanger, calculated with experiments and analytically, is given in Figure 4.4.

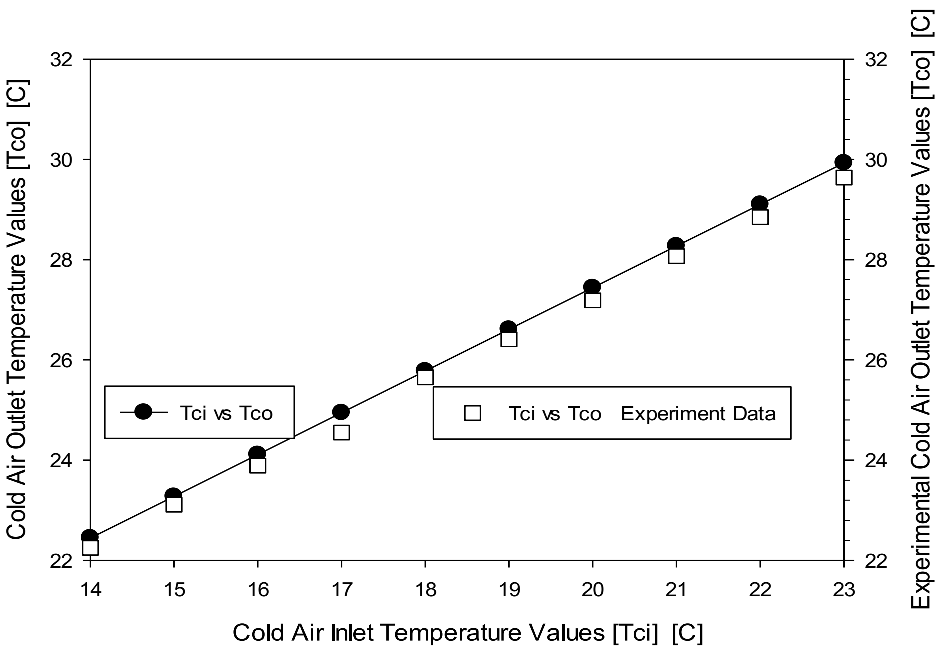


Figure 4.4 The variation of the exit temperature values found experimentally and analytically calculated according to the inlet temperature values of the cold air entering the heat exchanger.

It is seen that the exit temperature values of the cold fluid in the heated state are almost close according to both experimental measurements and analytical calculations. It is seen that the output temperature value increases linearly with the increase in the inlet temperature of the cold fluid entering the heat exchanger. Although the equation changes in the form of exponential function, it is seen that the results change linearly.

Initially, as the fluid continues to flow over the tube bundles, the expression ($\Delta T = T_s - T_i$) will undergo a large temperature difference. According to this temperature difference, the heat transfer found by using Newton's law of cooling may be high. Then, the fluid in question approaches the temperature (T_s), which is the surface temperature value of the pipes in the pipe bundle, with the heat it gains while flowing over the pipe bundles. Thus, the value of ($\Delta T = T_s - T_i$), which is an important parameter in heat transfer, decreases. Thus, the (ΔT) value is initially high, but gradually decreases as the fluid progresses, depending on the design of the heat exchanger. Since the function of the change (ΔT) of the fluid as it passes through the heat exchanger is not known exactly, the calculation is made according to the average logarithmic temperature difference value in order to calculate a convergent heat value of the heat amount passing from the fluid to the fluid in the heat exchanger. Accordingly, the average logarithmic temperature difference value is given in Figure 4.5.

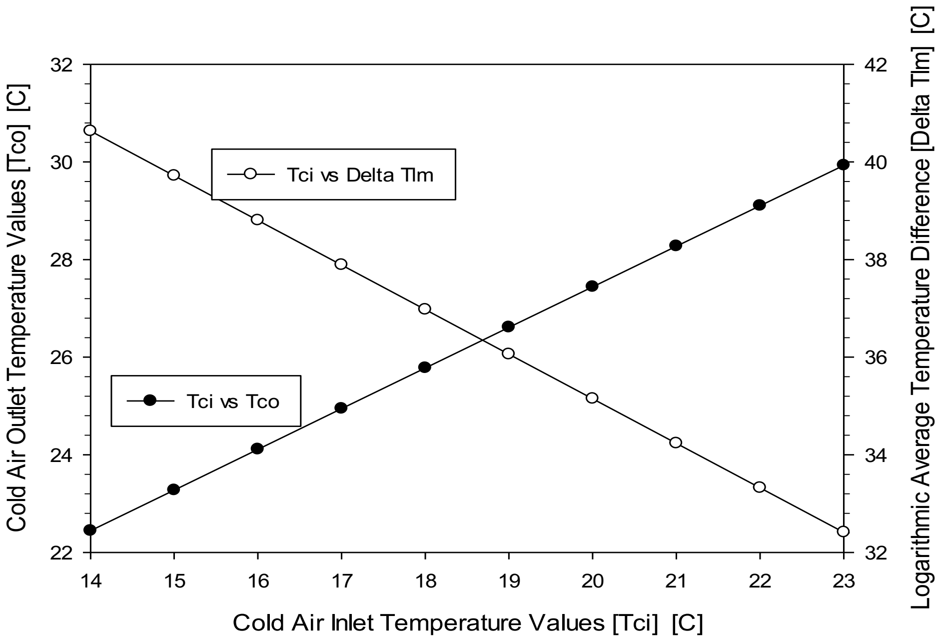


Figure 4.5 Variation of both the analytical outlet temperature values and the average logarithmic temperature difference value according to the inlet temperature values of the cold air entering the heat exchanger.

The temperature ($\Delta T = T_s - T_i$) that the fluid gains while flowing on the pipe bundles from the beginning to the outlet decreases. This decrease value is seen in Figure 4.5. The amount of heat transfer per unit length in the heat exchanger, which is formed according to the average logarithmic temperature difference value, is shown in Figure 4.6.

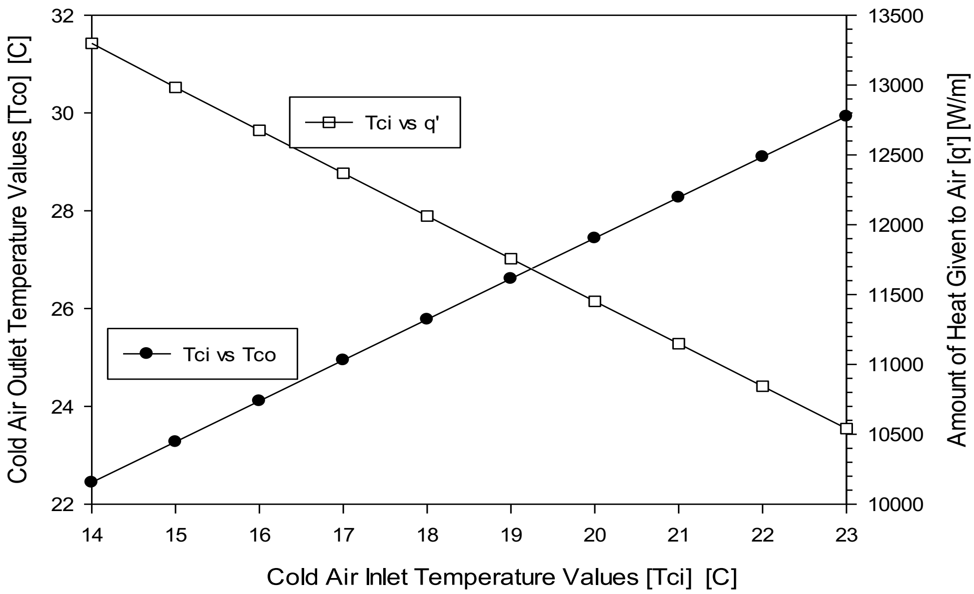


Figure 4.6 The variation of both the analytical outlet temperature values and the heat transfer value according to the inlet temperature values of the cold air entering the heat exchanger.

As seen in Figure 4.6, as the inlet temperature of the cold air entering the heat exchanger increases, the outlet temperature of the cold air leaving the heat exchanger increases. With this increase, the amount of heat transferred from the fluid to the fluid in the heat exchanger decreases. While the temperature value (T_{ci}) of the cold air entering the heat exchanger increases, the temperature difference value ($\Delta T = T_s - T_i$) of the advancing fluid from the beginning to the outlet decreases. Due to this decrease, there is a decrease in the amount of heat transfer in the heat exchanger.

In the heat exchanger where heat transfer from the fluid to the fluid is carried out, the pressure drop that the fluid is exposed to as it moves through the pipe bundle is as important a parameter as the heat transfer. Accordingly, the pressure drop of the fluid advancing in the designed heat exchanger is shown in the equation [9].

$$\Delta P = N_L \; X \left(\frac{\rho \, V_{max}^2}{2}\right) f \tag{10}$$

N_L: Number of successive alignments of planes perpendicular to the direction the fluid flows (N_L=7 in the heat exchanger).

X : Correction Factor (Found X =1.0 from Figure 7.13 on page 410 in Source [8].

f : Friction Multiplier (F=0.36 from Figure 7.13 on page 410 in Source [8].

The pressure drop per unit length created by the fluid moving between the pipes of the heat exchanger tube bundle is shown in Figure 7.

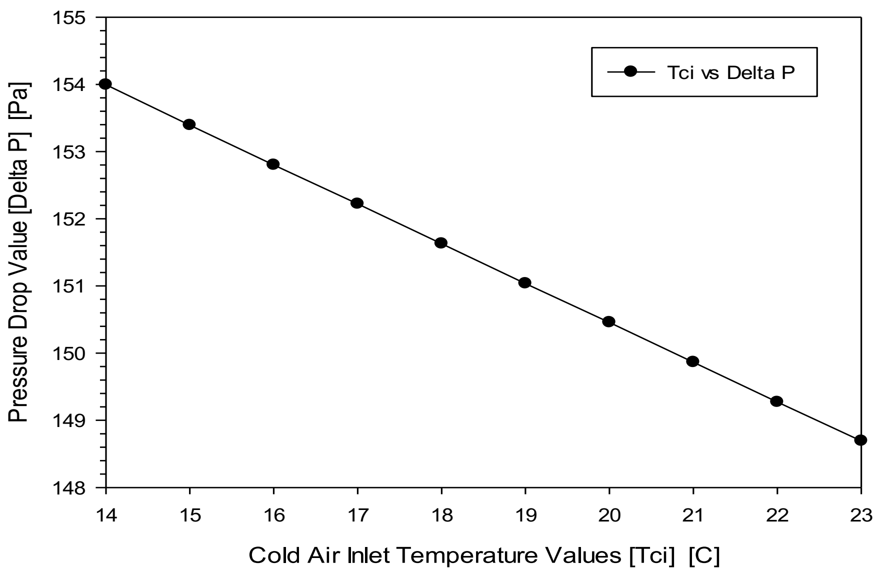


Figure 4.7 Pressure drop change according to the inlet temperature of the cold air entering the heat exchanger.

With the increase in the temperature of the cold air entering the heat exchanger, the increase in the average temperature of the air period period in the heat exchanger causes a decrease in the density, which is one of the physical properties of the air. Therefore, the pressure drop decreases due to the decrease in the density of the air as it passes through the heat exchanger. This decrease is shown in Figure 4.7.

4. Conclusion

Experimental and analytical results are discussed in this study to determine the thermal performance of the heat exchanger designed for waste heat recovery. As in all heat exchangers, it has been observed that the thermal performance (T_{ci}) of this type of cross-flow tube bundle heat exchanger depends on the fresh air inlet temperature, the exhaust air velocity (V) and the surface temperature (T_s) of the exhaust air in the tube bundle.

It was observed that while the inlet temperature values of the cold air entering the heat exchanger increased, the outlet temperature values of the cold air also increased. It has been observed that the exit temperatures of the cold air from the heat exchanger are very close to each other, both the values measured in the experiment and the analytically calculated values.

While the temperature value (T_{ci}) of the cold air entering the heat exchanger increases, the temperature difference value ($\Delta T = T_s - T_i$) of the advancing fluid from the beginning to the outlet decreases. Due to this decrease, the amount of heat transfer from the fluid to the fluid in the heat exchanger also decreased.

As the heat transfer from the fluid to the fluid passes between the pipe bundles, the pressure drop value of the air decreases due to the decrease in the density of the air as the inlet temperature of the cold air increases.

Ethics in Publishing

There are no ethical issues regarding the publication of this study.

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