ÇUKUROVA UNIVERSITY
INSTITUTE OF NATURAL AND APPLIED SCIENCES

WASTE HEAT RECOVERY USING REGENERATIVE HEAT EXCHANGER

BAHADIR ATALAY

MSc THESIS

DEPARTMENT OF MECHANICAL ENGINEERING

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ABSTRACT

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Rotary-type heat exchanger is one of the most important regenerative heat exchangers. This type of heat exchangers are mostly used for waste heat recovery in industry. Rotary-type heat exchangers have various application areas such as gas turbines, thermal power stations, schools, business centers, food industry, drying systems, and nowadays in air-conditioning systems.

In this thesis, a rotary-type heat exchanger used for waste heat recovery is considered. Effectiveness of this heat exchanger was investigated experimentally, analytically and numerically for various values of rotational speed and flow rate.

Key words: Rotary-type heat exchanger, waste heat recovery, effectiveness.

ÖZ

YÜKSEK LİSANS TEZİ

REJENERATİF ISI EŞANJÖRÜ KULLANARAK ATIK ISI GERİ KAZANIMI

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: Prof.Dr. Aziz ERTUNÇ

Rejeneratif ısı eşanjörlerinin en önemlilerinden birisi döner tip ısı eşanjörleridir. Bu tip ısı eşanjörleri endüstride özellikle atık ısı geri kazanımı sağlamak için kullanılırlar. Döner tip ısı eşanjörlerinin gaz türbinleri, güç santralleri, okullar, iş merkezleri, gıda sanayi, kurutma sistemleri ve son yıllarda iklimlendirme sistemlerinde çeşitli uygulama alanları bulunmaktadır.

Bu çalışmada, atık ısı geri kazanımı sağlamak amacıyla kullanılan döner tip bir ısı eşanjörü ele alınmıştır. Bu ısı eşanjörünün etkinliği dönme hızı ve hava debisi parametrelerine bağlı olarak deneysel, analitik ve sayısal olarak incelenmiştir.

Anahtar Kelimeler: Döner tip ısı eşanjörü, atık ısı geri kazanımı, etkinlik.

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LIST OF SYMBOLS

Symbols

a	A length of equilateral triangle	(m)
A	Area	(m^2)
L	Width of the rotary heat exchanger	(m)
C	Heat capacity	(J/kgK)
Cp	Specific heat capacity	(J/kgK)
d	Diameter	(m)
d^*	Dimensionless diameter	-
F	Surface area for heat transfer	(m^2)
f	Fresh air	-
Fo	Fourier number	-
k	Total heat transfer coefficient	(W/m^2K)
M_s	Mass of the solid body of the rotary heat exchanger	(kg)
n	Shape of the channel	-
N	Rotational speed	(rev/min)
Ntu	Number of transfer unit	-
Ntu_∞	Number of transfer unit for $Z^* \rightarrow \infty$	-
P	Periphery	(m)
Pr	Prandtl number	-
Q	Heat transfer	(W)
R	Mass rate	-
Re	Reynolds number	-
T	Temperature	(K)
t	Time	(s)
$\dot{V}$	Volume flow rate	(m^3/s)
w	Waste air	-
x	Coordinate through fluid flow	-
Z^*	Dimensionless coordinate	-

ε	Effectiveness	-
ε_o	Effectiveness of cross-flow heat exchanger	-
ϕ	Heat transfer factor	-
φ_c	Correction factor for cleaning region	-
φ_r	Correction factor for rotational speed	-
ψ	Shape factor for developed flow	-
Φ	Heat transfer factor	-
α	Heat transfer coefficient	(W/m^{2}K)
Δ	Differential	-

Subscript and Superscript Symbols

*	Ratio
a	Analytical
e	Experimental
eq	Equivalent
fl	Fluid
flS	Fluid storage
i	Inlet
max	Maximum
min	Minimum
n	Numerical
o	Outlet
r	Rotary heat exchanger
wl	Wall
wlS	Wall storage
∞	Infinitive

1. INTRODUCTION

Nowadays, energy requirement is mostly related to industrial applications and in houses. Energy cost just cannot be ignored. Sharp increase in energy cost, decreasing energy resources and environmental anxiety cause using energy much more efficiently. Reducing energy consumption has to be achieved by energy conservation. Heat is one of the energy forms. Energy requirements in the form of heat can be conserved with better insulation, improved conversion technologies, partial recovery of heat from waste exhaust gasses and liquids, greater use of heat pumps, co-generation of heat and electricity.

Energy conservation can be achieved using waste heat recovery systems known as heat exchanger. There are two types of heat exchangers commonly used for waste heat recovery, namely, recuperators and regenerators. Regenerator type heat exchangers are divided into two groups, these are valved-type and rotary-type.

Rotary-type heat exchangers have a porous matrix and transfer heat between two gas streams. These types have various application areas and mostly used for waste heat recovery. The porous matrix is arranged in the form of a drum which rotates about an axis, so that the body of the porous matrix passes periodically through the hot stream and then through the cold stream. The heat which is stored in the porous matrix during its contact with the hot gas is transferred to the cold gas during its contact with the cold stream. Knowing the variation of effectiveness with the rotational speed of a rotary regenerator is important to get the maximum benefit from it and for capacity control and, therefore, automatic control.

In this study, rotary-type heat exchangers are considered. Experiments were performed using a laboratory made rotary-type heat exchanger in order to test the influence of rotational speed and flow rate on effectiveness. For the same regenerator, and for the same parameters, the effectiveness was also calculated with the help of analytical equations developed in a previous study. In addition, a numerical study was also carried out to compare the effectiveness found by the experiments.

2. PREVIOUS STUDIES

First analytical studies on regenerators were carried out by Nusselt (1927), Hausen (1929) and Schumann (1929).

Burns and Willmott (1978) solved the differential equations derived for regenerator analytically with some assumptions. They investigated the influence of various parameters on regenerator effectiveness.

Razelos and Benjamin (1978) solved nonlinear regenerator differential equations with finite difference method. In their study, the physical properties of fluid and wall were cared due to temperature variation. This solution method was used for all regenerator types.

Holmberg (1979) investigated rotary-type heat exchanger with dehumidification properties which was used in air-conditioning systems. The heat transfer on the regenerator wall through the flow direction was also taken into account. The differential equations written for regenerator were solved with finite difference method.

Applegate (1981) investigated air-to-air heat recovery systems, principle of operations, basic thermodynamics, approximate recovery rates, selection data required and construction considerations of these systems, and gave installation examples.

Li (1983) calculated effectiveness of rotary-type heat exchangers using numerical methods. The influence of the rotary regenerator effectiveness on regenerator wall through the flow direction was investigated. In his study, regenerator differential equations were solved using two assumptions: Fluid and wall thermal properties are constant and vary with temperature.

Kays et. al. (1984) developed a simple empirical formulation for the influence of ratio of the heat capacity of rotary heat exchanger (C_r) to the minimum heat capacity of rotary heat exchanger (C_{min}). The equation is valid for effectiveness of heat exchanger smaller than 90 %and it is:

$$\varepsilon \approx \varepsilon_0\left[1-\frac{1}{9(C_r / C_{min})^{1.93}}\right]$$

where, ε_0 is the effectiveness of a cross-flow heat exchanger.

Worsøe-Schmidt (1991) developed a simple empirical equation for the effectiveness of rotary heat exchanger as:

$$\varepsilon = \varepsilon_0\left(1-\frac{0.114\left[1-e^{(-Ntu)}\right]}{C^{*\,0.44}C_r^{*\,1.93}}\right)$$

where, ε_0 is the effectiveness of a cross-flow heat exchanger, Ntu is the number of transfer unit, C^* is the heat capacity ratio, C_r^* is the heat capacity ratio of the rotary heat exchanger.

Romie (1987) investigated how the fluid thermal capacity affects the regenerator effectiveness for counter-flow regenerators analytically using the Laplace transformations.

Romie and Baclic (1988) suggested an integral method for quick calculation of fluid and wall temperatures for counter-flow regenerators.

Hill and Willmott (1989) proposed a method for quick and true calculation of temperature distribution for regenerators. They used Richardson extrapolation method to solve the problem quickly.

Scaricabarozzi (1989) investigated the conditions of heat conduction through the flow direction or not on regenerator wall. Regenerator differential equations were solved with some assumptions using the discretization method and developed analytical solution for regenerator effectiveness.

Romie (1990a) developed a table which depends on four different parameters to found the cross-flow regenerator effectiveness. The effectiveness for the parameters which are not in the table can be found by interpolation.

Romie (1990b) investigated the effect of small variation of fluid flow rate on exit gas temperature for rotary-type regenerators.

Varol (1991) investigated how much energy recovery can be achieved in air-conditioning systems with rotary-type heat exchanger. He also investigated the effect of various dimensionless parameters (heat capacity ratio of fluids, heat capacity ratio of regenerator, corrected number of transfer units, etc.) on regenerator performance.

A research group in the Mechanical Engineering Department of Çukurova University has maintained an ongoing research activity in waste heat recovery and especially in rotary-type heat exchangers.

Yılmaz and Cihan (1993) designed the rotary-type heat exchangers and developed empirical equations for heat exchanger effectiveness. Pressure losses of cleaning region and heat capacity ratio were taken into consideration for the calculation of cross-flow heat exchanger effectiveness.

Candar (1995) designed and constructed a rotary-type heat exchanger in the Laboratories of Mechanical Engineering Department of Çukurova University. He tested effectiveness of the rotary-type heat exchanger manufactured in an experimental setup and obtained some preliminary results.

Later, this study was extended by Doğruyol (1997) (see also Büyükalaca and Doğruyol; 1998). Doğruyol investigated the influence of rotational speed in the range of 2 rev/min to 7 rev/min experimentally. She compared her experimental results with the results of an analytical study performed in the Mechanical Engineering Department of Çukurova University by Yılmaz et. al. (1996) and found an acceptable agreement between the two results. In the study of Yılmaz et. al. (1996) a calculation method for the effectiveness of rotary heat exchangers for various matrix geometries was given.

Apart from rotary-type heat exchangers, some studies on recuperative-type heat exchangers were also carried out in the Mechanical Engineering Department of Çukurova University. Küçük (1996) and Doba (1996) manufactured a recuperative heat exchanger. They tested this recuperator in the experimental setup constructed by Candar (1995) and compared their results with the results for regenerator obtained by Candar (1995). Doba (1996) also investigated optimization of regenerative-type heat exchangers from the view of the second law of thermodynamics.

Parallel to these experimental and analytical studies, some numerical studies were also performed in the Mechanical Engineering Department of Çukurova University. Ünal (1996) calculated rotary-type heat exchanger effectiveness numerically. First, regenerator differential equations for fluid and wall temperatures in the regenerator were derived using the first law of thermodynamics. Then, these coupled differential equations were solved using the finite difference method and temperatures were obtained. Regenerator effectiveness was determined from these obtained temperatures. Numerical results were compared with the results given in the literature. Afterwards, the influence of various parameters on regenerator effectiveness was investigated.

3. HEAT EXCHANGERS

Heat exchangers are devices that transfer heat between two or more fluids at different temperatures. Many types of heat exchangers have been developed for use at such varied levels of technological sophistication and sizes as steam power plants, chemical processing plants, building heating and air conditioning, household refrigerators, car radiators, radiators for space vehicles, etc. and classified below (Section 3.1). In the common types, such as shell-and-tube heat exchangers and car radiators, heat transfer is primarily by conduction and convection from a hot to a cold fluid, which are separated by a metal wall. In boilers and condensers, heat transfer by boiling and condensation is of primary importance. In certain types of heat exchangers, such as cooling towers, hot fluid (i.e. water) is cooled by direct mixing with the cold fluid (i.e. air); that is, the water sprayed or falling down into an induced air draft is cooled by both convection and vaporization. In radiators for space applications, the waste heat carried by the coolant fluid is transported by convection and conduction to the fin surface and from there by thermal radiation into the atmosphere-free space.

3.1. Classification of Heat Exchangers

Heat exchangers are made in so many sizes , types, configurations, and flow arrangements that some kind of classification, even though arbitrary, is necessary for the study of heat exchangers. Özışık (1988) broadly classified heat exchangers. In the following discussion we consider classifications based on the transfer process, compactness, construction type, flow arrangement, heat transfer mechanism and number of different fluids.

Heat exchanger classification is shown in Fig. 3.1. In this thesis, periodic-flow drum-type rotary heat exchanger was used in the experimental study.

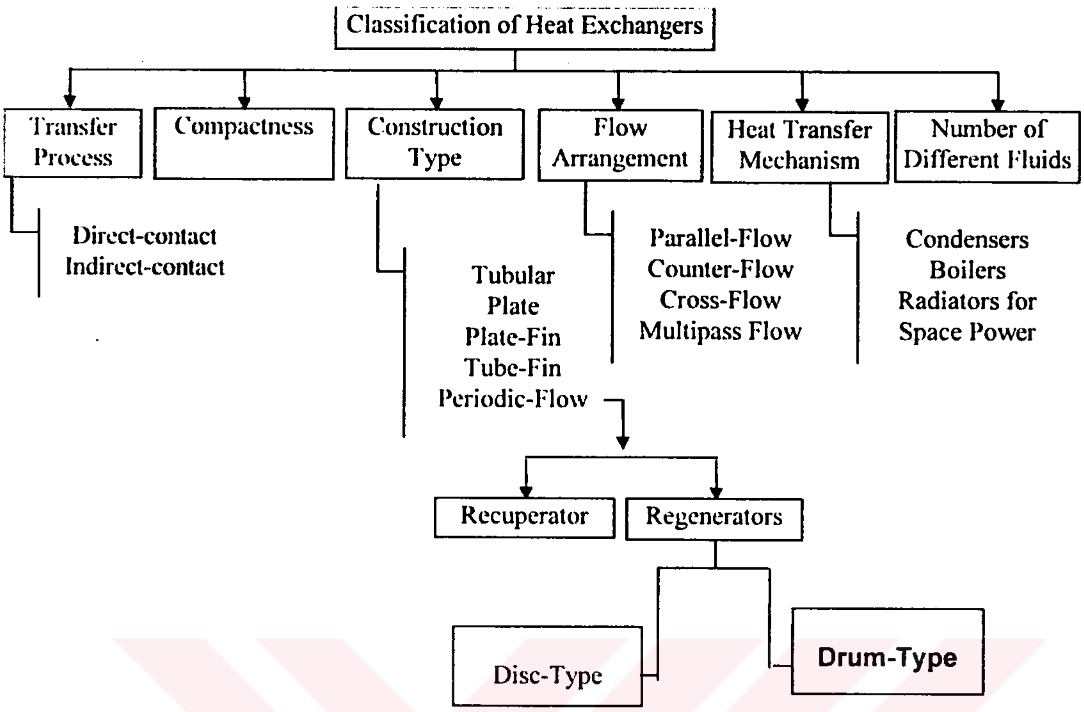


Figure 3.1. Classification of heat exchanger.

3.1.1. Classification by Transfer Process

Heat exchangers can be classified as direct contact and indirect contact. In the direct contact type, heat transfer takes place between two inmiscible fluids, such as a gas and a liquid, coming into direct contact. For example, cooling towers, jet condensers for water vapor, and other vapors utilizing water spray are typical examples of direct-contact exchangers.

In the indirect-contact type of heat exchangers such as automobile radiators, the hot and cold fluids are separated by an impervious surface, and they are referred to as surface heat exchangers. There is no mixing of the two fluids.

3.1.2. Classification by Compactness

The definition of compactness is quite an arbitrary matter. The ratio of heat transfer surface area on one side of the heat exchanger to the volume can be used as a measure of the compactness of heat exchangers. A heat exchanger having a surface area density on any one side greater than about 700 m^2/m^3 quite arbitrarily is referred to as a compact heat exchanger regardless of its structural design. For example, automobile radiators having an area density on the order of 1100 m^2/m^3 is a compact heat exchanger. The human lungs, with an area density of about 20000 m^2/m^3, are the most compact heat-and-mass exchanger. The very fine matrix regenerator for the Stirling engine has an area density approaching that of human lung.

On the other extreme of the compactness scale, plane tubular and shell-and-tube type exchangers, having an area density in the range of 70 to 500 m^2/m^3, are not considered compact.

An example of compact heat exchanger surface better than 1000 m^2/m^3, is shown in Fig. 3.2.

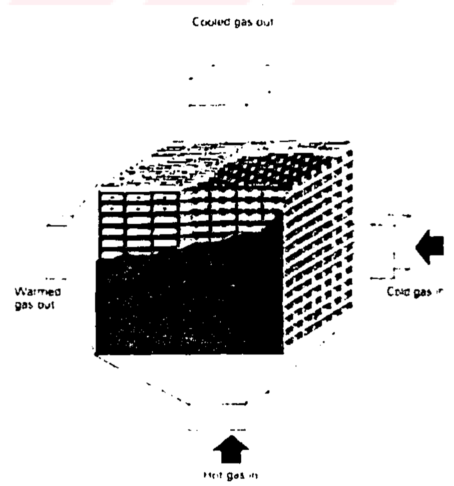


Figure 3.2. An example configuration for compact heat exchanger.

3.1.3. Classification by Construction Type

Heat exchangers can also be classified according to their construction features. For example, there are tubular, plate, plate-fin, tube-fin, and periodic exchangers.

Tubular Heat Exchangers: Tubular exchangers can accommodate a wide range of operating pressures and temperatures in many sizes, flow arrangements, and types. A commonly used design, called the shell-and-tube exchanger, consists of round tubes mounted on a cylindrical shell with their axes parallel to that of the shell. Fig. 3.3 illustrates the main features of a shell-and tube exchanger having one fluid flowing inside the tubes and the other flowing outside the tubes. The principal components of this type of heat exchanger are the tube bundle, shell, front and rear end headers, and baffles. The baffles are used to support the tubes, to direct the fluid flow approximately normal to the tubes, and to increase the turbulence of the shell fluid.

The character of the fluids may be liquid-to-liquid, liquid-to-gas, or gas-to-gas. Liquid-to-liquid exchangers have the most common applications. Both fluids are pumped through the exchanger; hence the heat transfer on both the tube side and the shell side is by forced convection. Since the heat transfer coefficient is high with the liquid flow, generally there is no need to use fins.

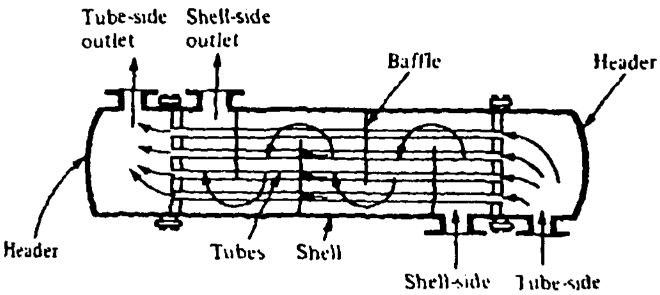


Figure 3.3. An example for shell-and-tube heat exchanger.

The liquid-to-gas arrangement is also commonly used; in such cases, the fins usually are added on the gas side of the tubes, where the heat transfer coefficient is low.

Gas-to-gas exchangers are used; in the exhaust-gas and air preheating recuperators for gas-turbine systems, cyrogenic gas-liquefaction systems, and steel furnaces. Internal and external fins generally are used in the tubes to enhance heat transfer.

Plate Heat Exchangers: As the name implies, plate heat exchangers usually are constructed of thin plates. The plates may be smooth or may have some form of corrugation. Since the plate geometry can not accommodate as high pressure and/or temperature differentials as a circular tube, they are generally designed for moderate temperature and/or pressure. The compactness factor for plate exchangers ranges from 120 to 230 m^2/m^3. Fig. 3.4 shows, an example of plate heat exchanger.

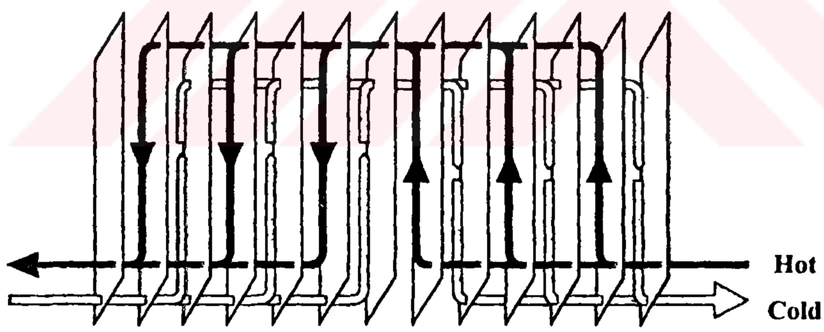


Figure 3.4. An example of plate heat exchanger.

Plate-Fin Exchangers: The compactness factor can be significantly improved (i.e. up to about 6000 m^2/m^3) by using the plate-fin type of heat exchanger. Fig. 3.5 illustrates typical plate-fin configurations. Louvered or corrugated fins are separated by flat plates. Cross-flow, counter-flow, or parallel-flow arrangements can be obtained readily by properly arranging the fins on each side of the plate. Plate-fin exchangers are generally used for low-pressure applications not exceeding about 10 atm. The maximum operating temperatures are limited to about 800°C. Plate-fin heat exchangers have also been used for cyrogenic applications.

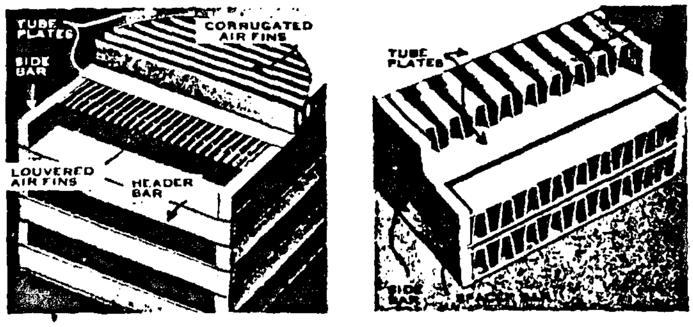
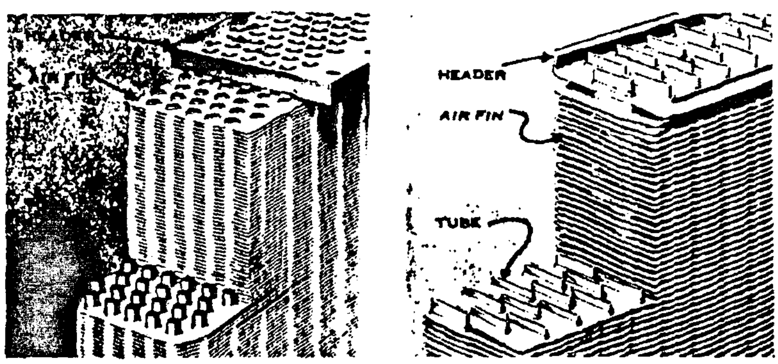


Figure 3.5. An example for plate-fin heat exchangers.

Tube-Fin Heat Exchanger: When a high operating pressure or an extended surface is needed on one side, tube-fin exchangers are used. Fig. 3.6 illustrates two typical configurations, one with round tubes and the other with flat tubes. Tube-fin exchangers can be used for a wide range of tube fluid operating pressures not exceeding about 30 atm and operating temperatures from low cyrogenic applications to about 870°C. The maximum compactness ratio of about 330 m^2/m^3 is less than that obtainable with plate-fin exchangers.



(a) Round tube and fin (b) Flat tube and fin

Figure 3.6. An example for tube-fin heat exchanger, (a) round tube and fin, (b) flat tube and fin.

The tube-fin heat exchangers are used in gas-turbine, nuclear, fuel cell, automobile, airplane, heat pump, refrigeration, electronics, cyrogenics, air conditioning, and many other applications.

Periodic-Flow Heat Exchangers: Periodic-flow heat exchangers can be either recuperative or regenerative. The recuperative type has no moving parts and consists of a porous mass (i.e. balls, pebbles, powders, etc.) through which hot and cold fluids pass alternately. A flow-switching device regulates the periodic flowing of the two fluids. During the flow of the hot fluid, the heat is transferred from the hot fluid to the matrix of the regenerative exchanger. Then the hot fluid flow is switched off, and the cold fluid flow is switched on. During the flow of the cold fluid, heat is transferred from the matrix to the cold fluid. Fig. 3.7 shows, an example for recuperative heat exchangers.

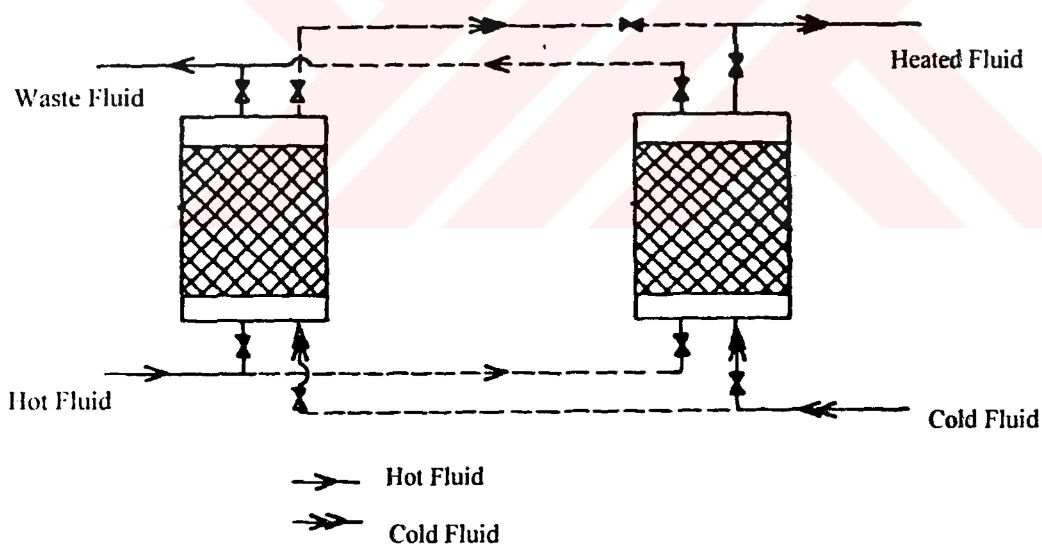


Figure 3.7. An example for recuperative heat exchangers.

Regenerative-type heat exchangers can be called as rotary heat exchangers. Rotary heat exchangers have two types which are disc-type and drum-type and they are schematically shown in Fig. 3.8. Rotary-type heat exchanger matrix is arranged in the form of a drum which rotates about an axis so that a given portion of the matrix passes periodically through the hot stream and then through the cold stream. The heat stored in the matrix during its contact with the hot gas is transferred to the cold gas during its contact with the cold stream. Rotary-type regenerators are suitable only for gas-to-gas heat exchange, because only for gases, the heat capacity of the porous matrix much greater than the heat capacity of the gas contained in the flow passages. It is not suitable for liquid-to-liquid heat exchange, because the heat capacity of the porous matrix is much less than that of the liquid.

Since the heat transfer matrix is rotating, the temperatures of the gases, and the wall depend on space and time; as a result, the heat transfer analysis of regenerators is involved, in that the periodic flow introduces several new variables. For conventional, stationary heat exchangers it is sufficient to define the inlet and outlet temperatures, the flow rates, the heat transfer coefficients for two fluids, and the surface areas of two sides of the heat exchangers. For the rotary heat exchanger, however, it is also necessary to relate the heat capacity of the rotor to that of the fluid streams, the fluid flow rates, and the rotation speed under consideration.

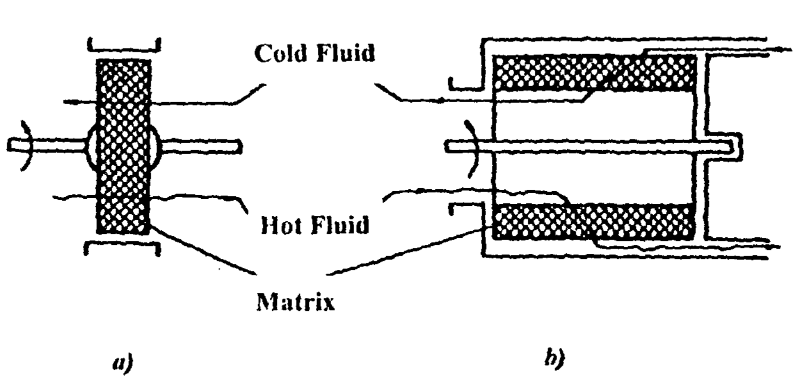


Figure 3.8. An example for rotary-type heat exchanger, (a) disc-type, (b) drum-type.

In this thesis, rotary-type (drum) heat exchanger was used in experimental, analytical and numerical studies.

3.1.4. Classification By Flow Arrangement

Numerous possibilities exist for flow arrangement in heat exchangers. We summarize here the principal ones.

Parallel-Flow: The hot and cold fluids enter at the same end of the heat exchanger, flow through in the same direction, and leave together at the other end, as shown in Fig. 3.9.

Counter-flow: The hot and cold fluids enter in the opposite ends of the heat exchanger and flow through in opposite directions, as shown in Fig. 3.10.

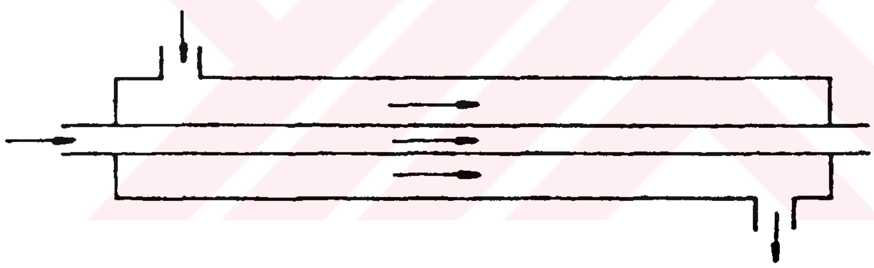


Figure 3.9. An example for parallel-flow heat exchanger.

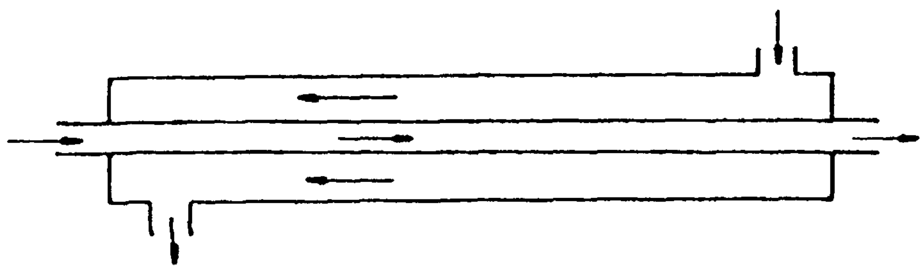


Figure 3.10. An example for counter-flow heat exchanger.

Cross-Flow: In the cross-flow exchanger, the two fluids usually flow at right angles to each other, as shown in Fig. 3.11. In the cross-flow arrangement, the flow may be called mixed or unmixed, depending on the design.

Fig. 3.11a shows an arrangement in which both hot and cold fluids flow through individual channels formed by corrugation; therefore the fluids are unmixed. Fig. 3.11b illustrates a typical temperature profile for the outlet temperatures when both fluids are unmixed, as shown in Fig. 3.11a. The inlet temperatures for both fluids are assumed to be uniform, but the outlet temperatures exhibit variation transverse to the flow.

In the flow arrangement shown in Fig. 3.11c, the cold fluids flows inside the tubes and so is not free to move in the transverse direction. Therefore, the cold fluid is said to be unmixed. However, the hot fluid flows over the tubes and is free mixed. The mixing tends to make the fluid temperature uniform in the transverse direction; therefore, the exit temperature of a mixed stream exhibits negligible variation in the crosswise direction.

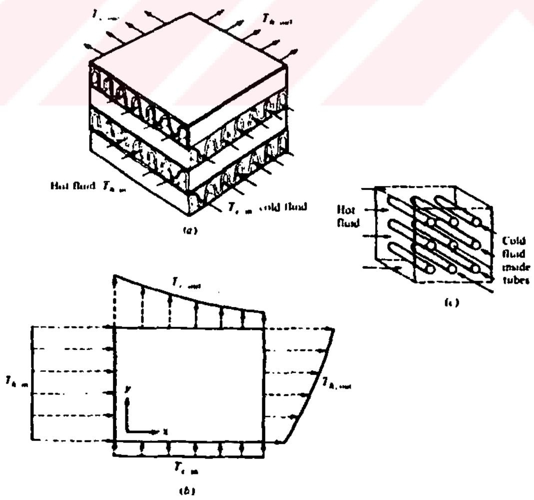


Figure 3.11. Examples for cross-flow heat exchanger, (a) both fluids unmixed, (b) temperature profile when both fluids are unmixed, (c) cold fluid unmixed, hot fluid mixed.

In general, in a cross-flow exchanger three idealized flow arrangements are possible: (1) Both fluids are unmixed; (2) one fluid is mixed, and the other is unmixed; and (3) both fluids are mixed. The last arrangement is not commonly used.

Multipass Flow: The multipass flow arrangements are frequently used in heat exchanger design, because multipassing increases the overall effectiveness over individual effectiveness . A wide variety multipass flow arrangements are possible. It is seen from Fig. 3.12 an example of multipass flow typical arrangements. The heat exchanger in Fig. 3.12a is a "one shell pass, two tube pass" arrangement, which is also referred to as a "one-two" heat exchanger. Fig. 3.12b shows a "two shell pass, for tube pass" arrangement, and Fig. 3.12c shows a "three shell pass, six tube pass" arrangement.

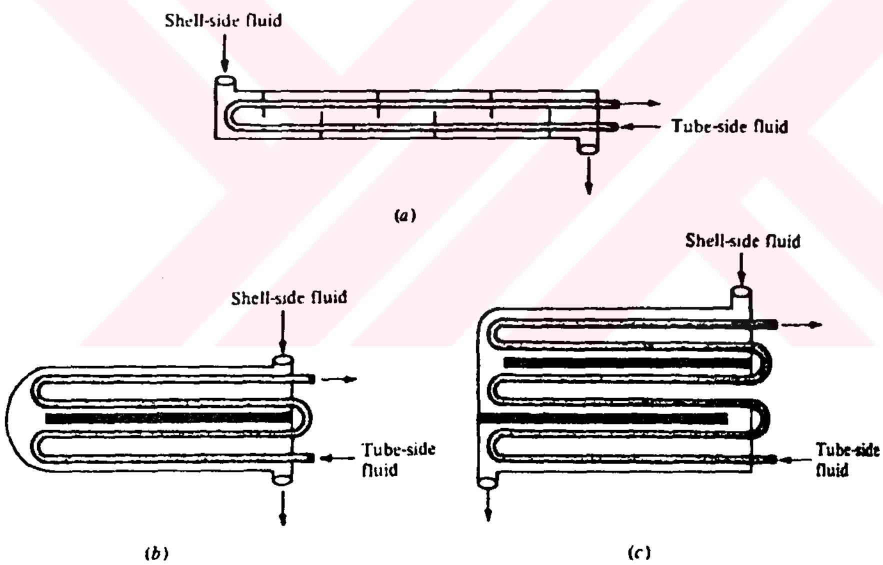


Figure 3.12. Examples for multipass flow heat exchanger, (a) one shell pass, two tube pass, (b) two shell pass, four tube pass, (c) three shell pass, six tube pass.

3.1.5. Classification by Heat Transfer Mechanism

The possibilities for the heat transfer mechanism include a combination of any two of the following; single-phase forced or free convection, phase change (boiling or condensation), radiation or combined convection and radiation.

For all the cases discussed earlier we considered single-phase forced convection on both sides of the heat exchanger. Condensers, boilers, and radiators for space power plants include the mechanisms of condensation, boiling, and radiation, respectively, on one of the surfaces of the heat exchanger.

Condensers: Condensers are used for such varied applications as steam power plants, chemical processing plants, and nuclear electric plants for space vehicles. The major types include the surface condensers, jet condensers, and evaporative condensers. The most common type is the surface condenser, which has the advantage that the condensate is returned to the boiler through the feedwater system.

Boilers: Steam boilers are one of the earliest applications of heat exchanger. The term steam generator is often applied to boilers in which the heat source is a hot fluid stream rather than the product of combustion. An enormous variety of boiler types exist, ranging from small units for house heating applications to huge, complex, expensive units for modern power stations.

Radiators for Space Power Plants: The rejection of waste heat from the condenser of a power plant intended to produce electricity for the propulsion, guidance, or the communication equipment of a space vehicle poses serious problems even for a power plant producing only a few kilowatts of electricity. The only way the waste heat can be dissipated from a space vehicle is by thermal radiation by taking advantage and the radiative-heat flux. Thus, in the operation of some power plants for space vehicles, the thermodynamic cycle is at such high temperatures that the radiator runs red-hot. Even so, it is difficult to keep the radiator size within a reasonable envelope for launch vehicles.

3.1.6. Classification by Number of Different Fluids

Heat exchangers generally transfers heat between two fluids. But sometimes, when the applications are chemical operation, refrigeration technology, air separation and hydrogen processes, it was encountered more than two fluids in heat exchangers.

4. MATERIAL AND METHOD

In this thesis, the effect of rotational speed and flow rate on effectiveness of a rotary-type heat exchanger was studied experimentally, analytically and numerically.

4.1. Experimental Study

The effectiveness of a rotary heat exchanger with various rotational speeds and flow rates was investigated experimentally. The details of the experimental study is given below:

4.1.1. Experimental Setup

The experimental setup used in the experiments is shown schematically in Fig. 4.1. As shown from the figure, fresh air is taken into the system with a radial fan. When it passes through one side of the rotary heat exchanger, it receives heat from the porous matrix. Then it passes from a damper which is used to adjust the flow rate of fresh air. Pre-heated fresh air comes to a coil which is heated by a central heating system. After the coil, heated air represents waste air. It was possible to change the temperature of the air leaving the coil (waste air) by adjusting the temperature of the water circulating within the coil. The waste air leaves part of its heat to the porous matrix when it passes through the other side of the rotary heat exchanger and leaves the system. Wire sieves are used to improve the flow in the channels at various sections. The heat exchanger was rotated with the help of an AC motor and a special speed varying system was used to change the rotational speed. Two different views of the experimental setup are given in Fig. 4.2 and Fig. 4.3.

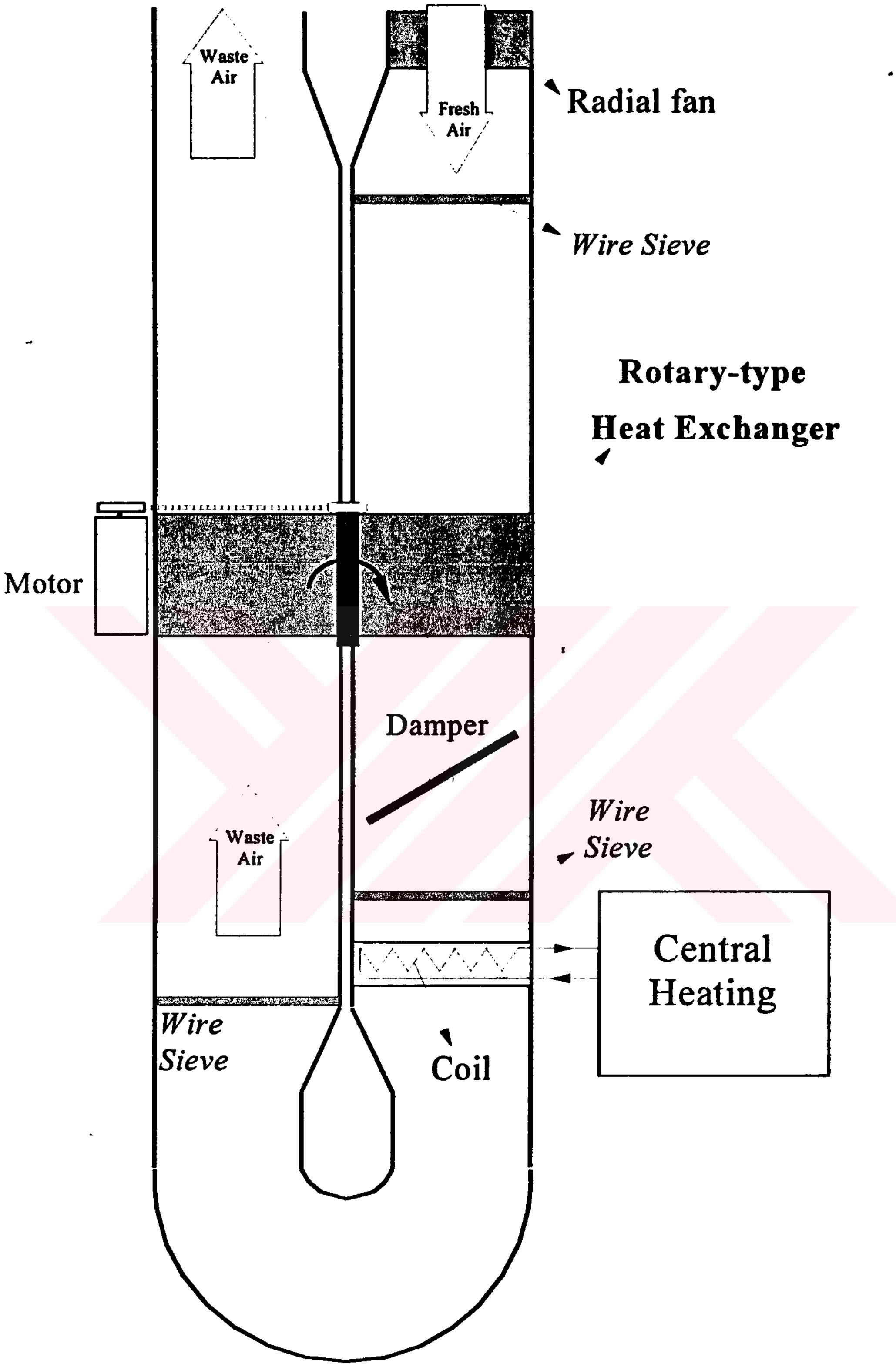


Figure 4.1. Schematical representation of the experimental setup.

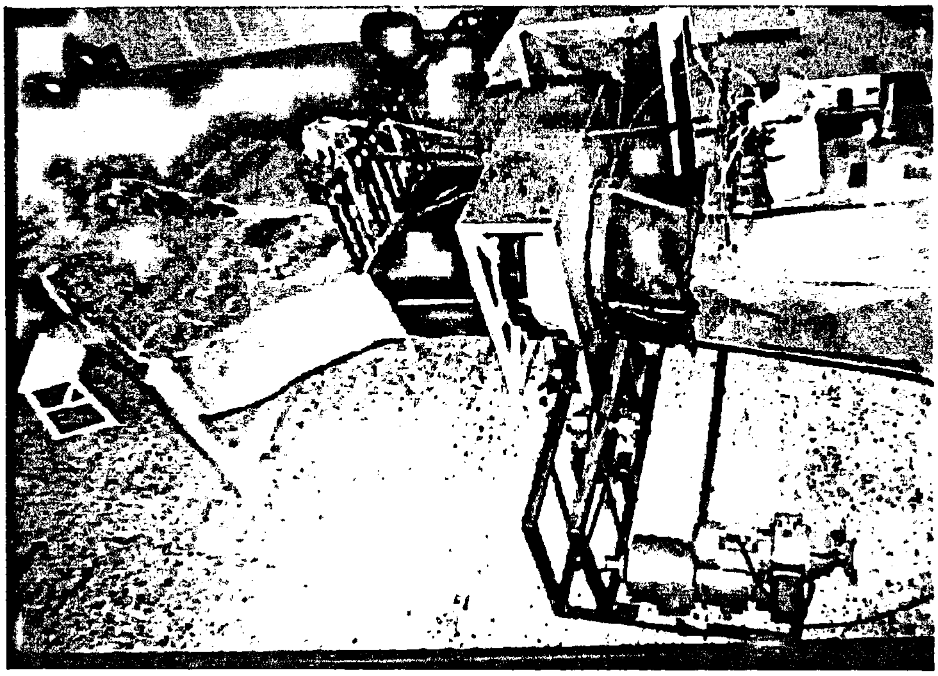


Figure 4.2. A picture of the experimental setup.

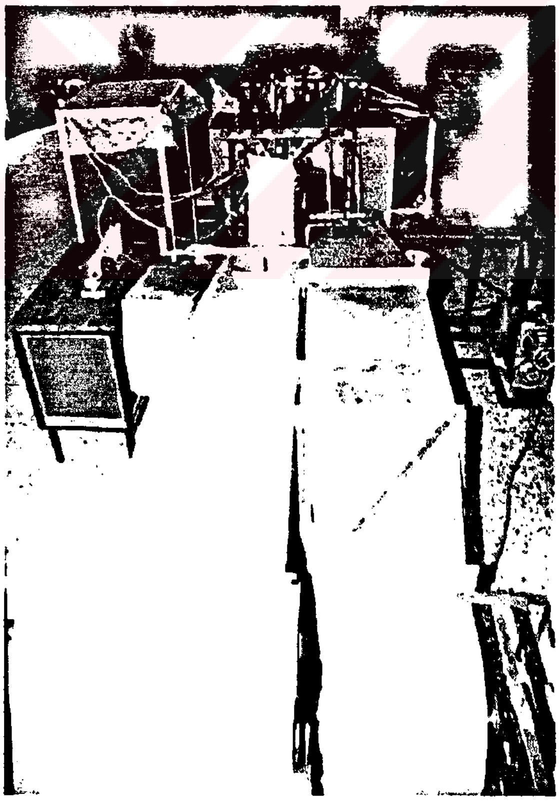


Figure 4.3. A picture of the experimental setup.

Rotary-type heat exchangers are not manufactured in Turkey. Although it is possible to import them from abroad, the prices are quite high. Thus, a rotary-type heat exchanger was manufactured by Candar (1995) in the Laboratories of Çukurova University Mechanical Engineering Department, and it is shown in Fig. 4.4. The rotary-type heat exchanger has a diameter of 690 mm, length of 200 mm and weight of 90 kg.

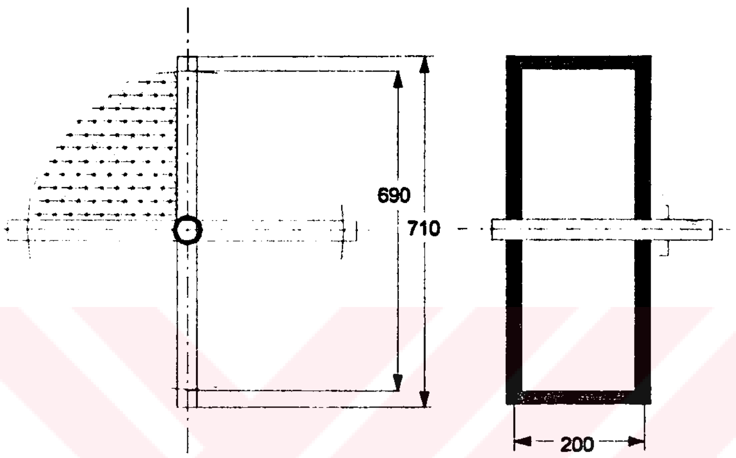


Figure 4.4. Representation of the rotary-type heat exchanger matrix.

The flow passage of the rotary heat exchanger is produced with the arrangement shown in Fig. 4.5. The rotary heat exchanger porous matrix is shown schematically in Fig. 4.6 and a picture of it is shown in Fig. 4.7. Flow passage geometry has an equilateral structure. The length of one side is approximately 3.44 mm. Matrix material is aluminum and the plate thickness is 0.35 mm.

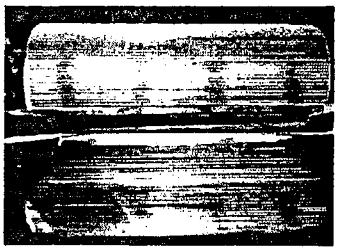


Figure 4.5. The device used for obtaining flow passage geometry of the rotary heat exchanger.

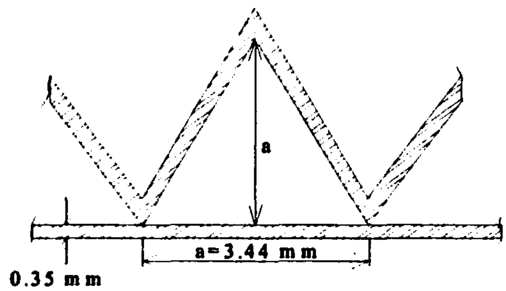


Figure 4.6. Flow passage geometry of the rotary heat exchanger used in the experiments.

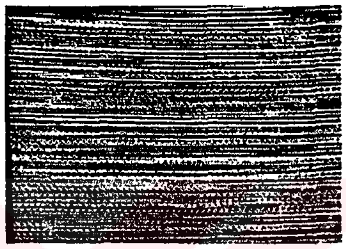


Figure 4.7. A picture of the porous matrix of the rotary heat exchanger.

Candar (1995) constructed originally the rotary heat exchanger and the experimental facility used in this study, and made some preliminary tests. Since it was not possible to change the rotational speed of the rotary regenerator in the test facility utilized, the tests were performed for only one rotational speed and different fluid flow rates.

Doğruyol (1997) made some improvements on the system which was constructed originally by Candar (1995). The rotational speed was made variable using an AC motor in the range of 2-7 rev/min. Fluid velocity measurements were performed at about seventy points at each cross-section of the channel and flow rates were calculated using the velocity values. Temperature measurements using K-type thermocouple were performed at seventy points at each cross-section of the channel and then the average temperatures calculated for each channel. Due to the limitations of data logger used that allows a maximum of eight input signals to be logged in and that each signal can be read in about one second, Doğruyol (1997) was not able to detect simultaneously the temperatures from the several points in any cross-section of the channel. Therefore, there was a considerable time difference between the readings. She was also not able to investigate the behavior of the rotary regenerator at low values of rotational speed (<2 rev/min). She investigated the effectiveness of this rotary heat exchanger for rotational speed between 2 rev/min and 7 rev/min, and it was found that, the effectiveness of the rotary heat exchanger was nearly constant in these rotational speed interval. However, due to the limitations of the data logging system and procedure, the accuracy of the results are questionable.

In this thesis, a frequency controller to vary the rotational speed of the rotary regenerator was thought to be best solution. However, due to financial limitations of the project, a special chain and chain-gear system was designed and constructed as shown in Fig. 4.8. The new system allowed us to vary rotational speed in the range of 0.05 rev/min and 7 rev/min. A picture of the new system is shown in Fig. 4.9.

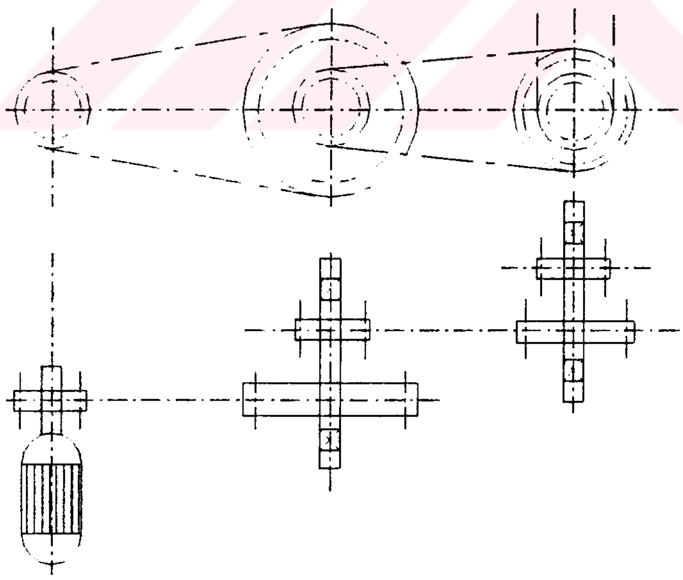


Figure 4.8. The chain-gear system used to vary the rotational speed of the rotary regenerator.

Figure 4.9. A picture of the chain-gear system used to vary the rotational speed of the rotary regenerator.

In addition, temperature measurements were improved by arranging thermocouples to give average temperature at any cross-section of the channel. This enabled us to collect simultaneously data from the several points at any cross-section and from several cross-sections.

4.1.2. Measurements on Experimental Setup

During the experiments fresh and waste air flow rates and temperatures, and the heat exchanger rotational speed were measured. Temperature measurements were performed with K-type thermocouples at the inlet and at the outlet of both sides of the rotary heat exchanger as shown in Fig. 4.12 (totally at four different cross-sections). Nine thermocouples were located into the channels at each cross-section and they were arranged to give average temperature at that section (Fig.4.10). A computer controlled data acquisition system was used to collect data as shown in Fig. 4.11 and a schematical representation of this system is shown in Fig. 4.12. A picture of the computer controlled data acquisition system used in the experiments is shown in Fig. 4.13.

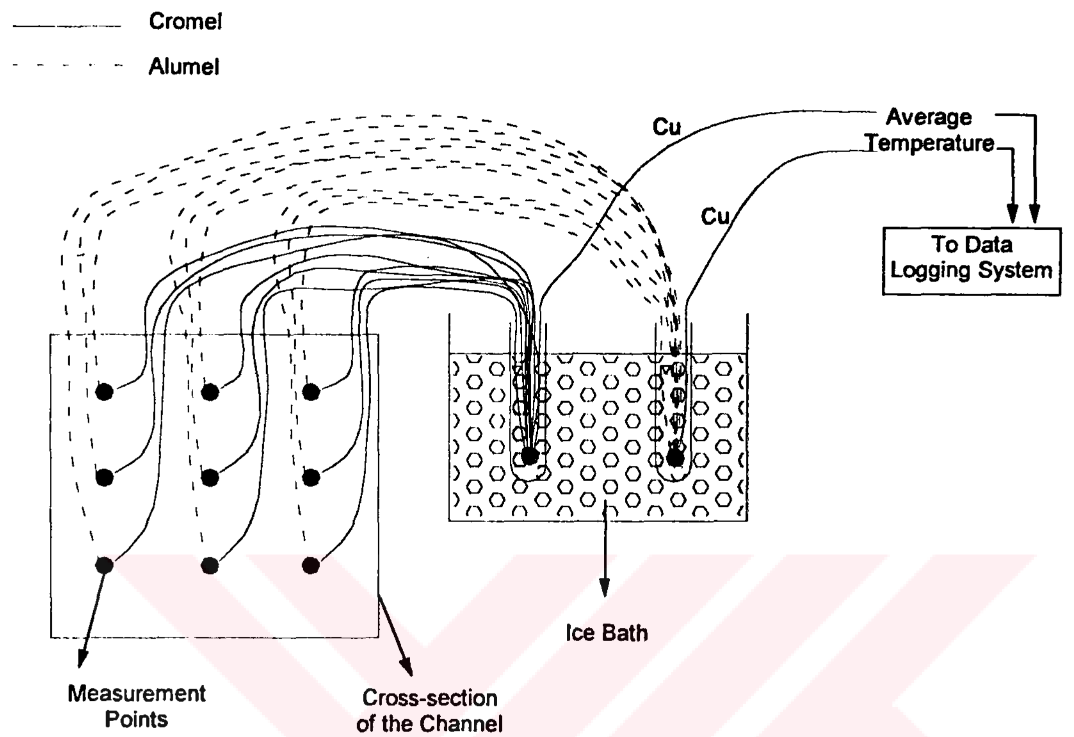


Figure 4.10. The arrangement of thermocouples to obtain average temperature at each cross-section of the channel.

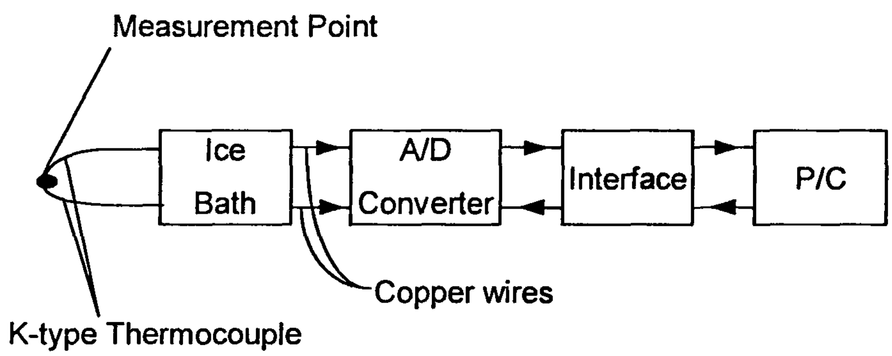


Figure 4.11. The computer controlled data acquisition system used in the experiments.

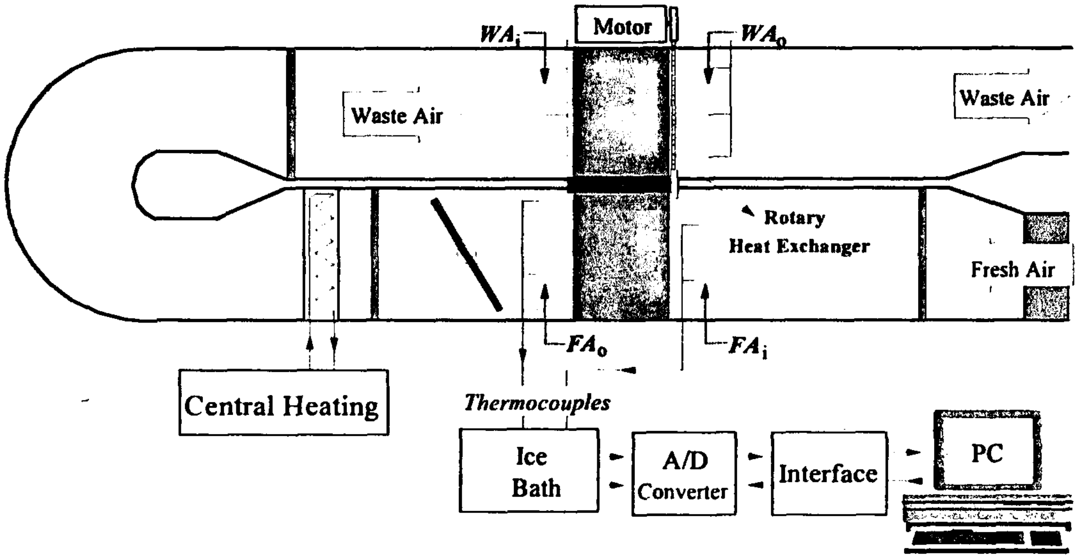


Figure 4.12. Schematical representation of computer controlled data acquisition system and measurement points on the experimental setup.

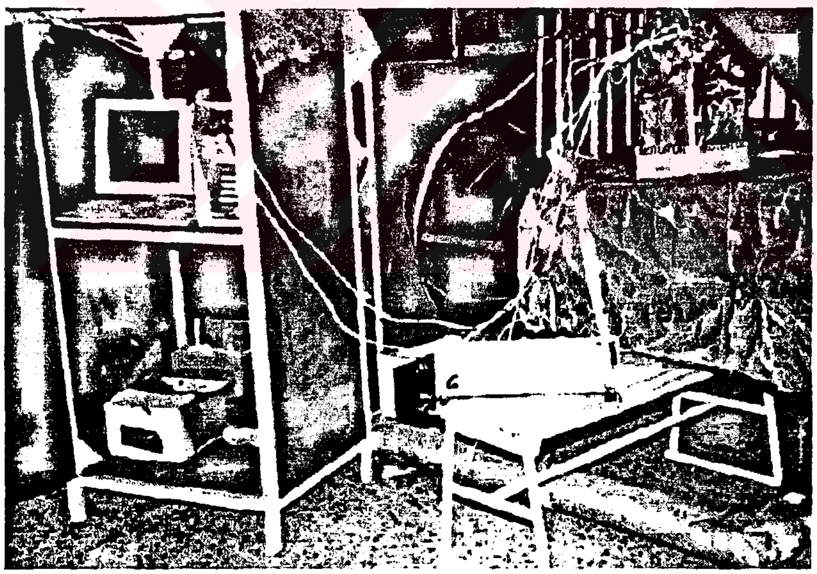


Figure 4.13. A picture of the computer controlled data acquisition system used in the experiments.

The measurement of thermocouple emf's as made by using a computer controlled data acquisition system which consisted of a personal computer, an

ADC-16 interface, and ADC-16 analog/digital converter and an ice bath (Fig. 4.11).

A/D converter has an eight differential input channels with 16-bit resolution. Input range can be adjustable to ± 5V or ± 3.2768V (jumper selectable, in the experiments ± 3.2768V was used) and input gains to 1,10 or 100 (software selectable, in the experiments input gain was selected as 100). Minimum data collection frequency was 1 data/sec.

In the experiments, four signals that give the average temperature at four different cross-sections were logged into the data acquisition system. Each temperature measurement section was scanned continuously 600 times. The time difference between the readings is about 1 sec. A typical data file collected is given in Appendix-A. The data stored in the file were in the form of mV, so the collected data were processed after the experiments to obtain the temperatures as ^{0}C using the calibration curve given in Appendix-B. Then average of 600 data for each channel is calculated to obtain the temperature (^{0}C) of that section.

Temperature measurements were made using the K-type thermocouples. All thermocouples were routed directly into ice/water cold junctions. A separate cold junction was employed for each temperature measurement section. Copper wires were then used to connect the cold junction to the A/D converter.

Careful consideration was given to the problem of minimizing pick-up of electrical noise by the thermocouple wires during transmission. All the thermocouples used were shielded separately from their measurement points to the A/D converter using the aluminum foil which was connected to the ground.

Flow rate of the air at four cross-sections of the channel (Fig. 4.12) was obtained from the velocity measurements at that cross-sections. At each cross-section, velocities from the seventy different points (Fig. 4.14) were measured using a hot-wire anemometer. The hot-wire anemometer used in the experiments was an Air-Flow AT-2 anemometer/thermometer having a measurement range of 0-15 m/sec. The hot-wire anemometer was calibrated using a pitot tube in an air-flow bench (Fig. 4.15). A typical calibration curve is given in Appendix-C.

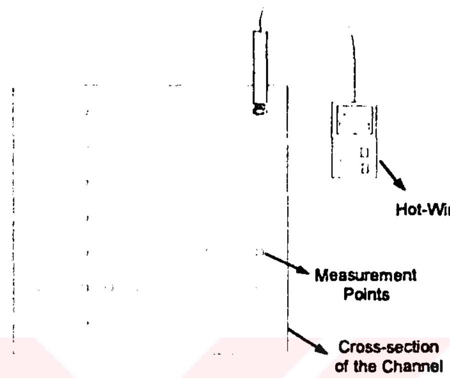


Figure 4.14. Schematical representation of velocity measurement for each channel.

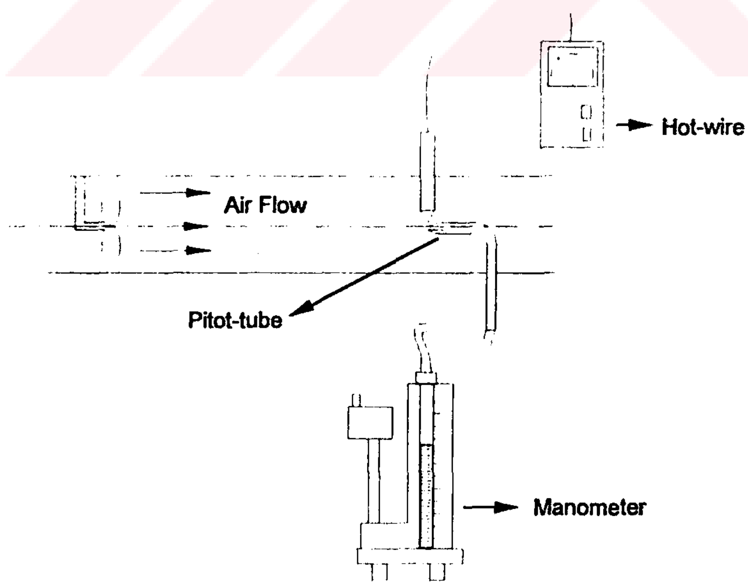


Figure 4.15. Air-flow bench with pitot tube used for hot-wire calibration.

A calculation procedure was developed for evaluating the average flow rate at each cross-section as follows: First, it was assumed that, each measurement point represents a region (Fig. 4.16). The average velocity was found for each cross-section using the equation given below. The flow rate at each cross-section can be found by knowing the average velocity and cross-sectional area of the section.

$$U_{average} = \frac{\sum(AU)}{\sum(A)} = \frac{(A_1U_1 + A_2U_2 + \ldots + A_nU_n)}{(A_1 + A_2 + \ldots + A_n)}$$

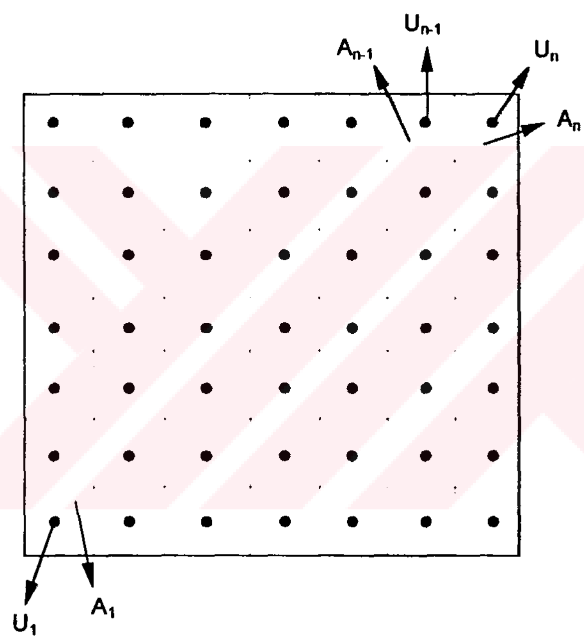


Figure 4.16. Calculation procedure for average velocity.

4.1.3. Experimental Parameters

Fresh air inlet temperature was constant at about 20 °C during the experiments, and waste air temperature was varied between 40 °C to 50 °C. Due to the experimental arrangement utilized in this study, the waste air flow rates are approximately equal to that of fresh air and they are 1.042 m^3/s, 0.874 m^3/s and

0.576 m^3/s. A series of experiments were performed at various rotational speeds of rotary regenerator between 0.05 rev/min and 7 rev/min. Table 4.1 shows the list of the experiments performed in this study.

Table 4.1. List of the experiments performed in this study.

Run No	Rotational Speed (rev/min)	Waste Air Temperature (^0C)		
		Air Flow Rate (1.042 m^3/s)	Air Flow Rate (0.874 m^3/s)	Air Flow Rate (0.576 m^3/s)
1	0.050	42.40	45.16	48.16
2	0.058	42.79	45.35	49.08
3	0.068	42.99	45.84	49.30
4	0.080	43.16	45.99	49.50
5	0.094	43.38	45.80	49.63
6	0.110	43.52	46.06	49.56
7	0.126	43.84	46.49	49.44
8	0.148	44.15	46.51	49.62
9	0.173	44.44	46.85	49.80
10	0.204	44.83	47.06	49.66
11	0.260	44.99	47.51	49.57
12	0.326	44.58	47.04	48.40
13	0.378	44.89	47.64	49.59
14	0.447	45.26	47.74	50.30
15	0.522	45.88	47.94	50.12
16	0.608	45.82	48.02	50.10
17	0.720	46.13	48.12	50.16
18	0.831	46.35	48.17	50.03
19	0.967	46.13	47.79	50.14
20	1.113	46.56	47.87	49.94
21	1.341	46.35	47.95	50.53
22	1.595	46.72	48.32	50.43
23	2.000	46.10	48.21	50.14
24	3.000	46.38	47.89	49.34
25	4.000	45.97	48.34	49.29
26	5.000	46.26	48.34	49.26
27	6.000	46.40	48.44	50.04
28	7.000	45.91	48.27	50.13

4.1.4. Error Analysis

For the calculation of the effectiveness, one needs to know the heat transfer, which requires the measurement of the velocity and temperature. In the experiments, the mean uncertainty values of the velocity and temperature were estimated to be %10 and ± 0.5 ^{0}C respectively. The uncertainty analyses of the overall experimental procedures were made using the method described by Moffat (1988). It was found that, uncertainty of effectiveness of the rotary heat exchanger varied between %23 - %16.9 for flow rate of 1.042 m^3/s, %20 - %16.4 for flow rate of 0.874 m^3/s and %19 - %16 for flow rate of 0.576 m^3/s.

4.1.5. Calculation of Effectiveness

Using the data obtained from the experiments ε_e was calculated from its definition:

$$\varepsilon_e = \frac{Q}{Q_{max}} \tag{4.1}$$

Here, Q is the actual heat transfer rate within the regenerator:

$$Q = C(T_{outlet} - T_{inlet}) \tag{4.2}$$

and Q_{max} is the maximum possible heat transfer rate from one stream to the other:

$$Q_{max} = C_{min}(T_{w,inlet} - T_{f,inlet}) \tag{4.3}$$

In these equations, C and C_{min} respectively denote the heat capacity of the air and minimum heat capacity of the air.

4.2. Analytical Study

In this study, analytical results for the effectiveness of the rotary regenerator (ε_a) was obtained using the procedure given by Yılmaz, Büyükalaca and Candar (1996):

$$\varepsilon_a = \varepsilon_o \varphi_r \varphi_c \tag{4.4}$$

in which, ε_o denotes the effectiveness of a cross-flow heat exchanger, φ_r is a correction factor for rotational speed and φ_c represents a correction factor for cleaning region (Kays and London 1984).

Effectiveness of a cross-flow heat exchanger can be calculated using Eqn. (4.5) suggested by Kays and London (1984):

$$\varepsilon_o = \frac{1 - e^{\left[-Ntu(1-C^*)\right]}}{1 - C^* e^{\left[-Ntu(1-C^*)\right]}} \tag{4.5}$$

As can be seen from this equation, ε_o depends on two parameters which are the heat capacity ratio (C^*) and the number of transfer units (Ntu). These parameters are defined as:

$$C^* = \frac{C_{min}}{C_{max}} \tag{4.6}$$

$$Ntu = \frac{kF}{C_{min}} \tag{4.7}$$

Heat capacity of a heat exchanger (kF) is calculated from the equation given below:

$$\frac{1}{kF}=\frac{1}{\alpha_f F_f}+\frac{1}{\alpha_w F_w} \tag{4.8}$$

where, F is the heat transfer surface area of the rotary regenerator and α is heat transfer coefficient and can be calculated knowing Nusselt number. In this study, Nusselt number was calculated using the equations developed by Yılmaz and Cihan (1993):

$$Nu = Nu_\infty\left[1+\frac{4.212\psi\Phi^3}{Z^* Nu_\infty^3}-0.8\left(\frac{\psi\Phi^3}{Z^* Nu_\infty^3}\right)^{2/3}\right]^{1/3} \tag{4.9}$$

in which, Nu_∞ is a Nusselt number for $Z^* \rightarrow \infty$ and can be found as follows:

$$Nu_\infty = 3.657\,\phi \tag{4.10}$$

where, ϕ is a heat transfer factor and is given below:

$$\phi = 1+\frac{\phi_\infty - 1}{1+1/(n-1)}+\Delta\phi \tag{4.11}$$

other parameters ϕ_∞ and $\Delta\phi$ can be calculated from equations given below:

$$\phi_\infty = 0.5155\frac{d^{*2}}{3-d^*} \tag{4.12}$$

$$\Delta\phi = \Delta\phi_{max}\frac{0.95(n-1)^{0.5}}{1+0.038(n-1)^3} \tag{4.13}$$

here, $\Delta\phi_{max}$ is defined as:

$$\Delta\phi_{max} = \frac{7.10^{-3} d^{*8}}{(1 + 10 d^{*28})(1 + 64.10^{-8} d^{*28})^{1/2}} \qquad \textbf{(4.14)}$$

In the last equation, d^* is a dimensionless diameter and n is a parameter which defines the solid area of the rotary heat exchanger. For the rotary heat exchanger used in the experimental studies, these parameters can be found as follows:

$$d^* = \frac{d_{eq}}{d_{max}} \qquad \textbf{(4.15)}$$

$$n = \frac{A}{A_{eq}} = \frac{P}{P_{eq}} \qquad \textbf{(4.16)}$$

here, A_{eq} and P_{eq} are, respectively denote the cross-sectional area and periphery of the equivalent diameter.

Equivalent diameter is defined as in Fig. 4.17 for equilateral triangles.

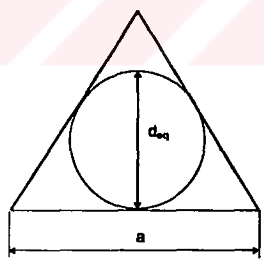


Figure 4.17. Definition of equivalent diameter.

$$d_{eq} = 4.A / P \qquad \textbf{(4.17)}$$

$$A = \frac{a^2\sqrt{3}}{4}; \qquad P = 3.a; \qquad \textbf{(4.18)}$$

so, equivalent diameter is found as:

$$d_{eq} = \frac{2.a}{\sqrt{3}} \tag{4.19}$$

Maximum diameter is equal to the equivalent diameter for equilateral triangles. From these equations, d^* and n can be found as follows for equilateral triangle channels of rotary heat exchanger:

$$d_{max} = \frac{2.a}{\sqrt{3}} \qquad d^* = 1 \qquad n = \frac{3\sqrt{3}}{\pi} \tag{4.20}$$

The other parameter for the calculation of Nusselt number is shape factor for developed flow (ψ) and is given below:

$$\psi = 1 + \left\{ \frac{\psi_\infty - 1}{\left[1 + 0.33 d^{*2.25} / (n-1)\right]} \right\} \tag{4.21}$$

where, ψ_∞ is defined as:

$$\psi_\infty = \frac{3}{8} d^{*2} (3 - d^*) \tag{4.22}$$

From Eqn. (4.21) and Eqn. (4.22), ψ is found as follows for the rotary heat exchanger which is used in the experimental study:

$$\psi = 0.834 \tag{4.23}$$

The parameters which are also used for the calculation of Nusselt number (Eqn. 4.9) are heat transfer factor (Φ) and dimensionless coordinate (Z^*), and given as follows:

$$\Phi = 1 + \frac{\left[3(d^* / 2)^{7/8} / (1 + d^*)\right] - 1}{1 + 0.25 / (n - 1)} \tag{4.24}$$

$$Z^* = L / (d_{eq} . Re . Pr) \tag{4.25}$$

Inserting the values of d^* and n given in Eqn. (4.20) into the related equations, we obtain:

$$Nu_{\infty} = 2.589 \quad \text{and} \quad \Phi = 0.868 \tag{4.26}$$

Using the calculated values for the rotary heat exchanger, Eqn. (4.9) has a simple formulation and can be rewritten as:

$$Nu = 2.589\left[1 + \frac{0.1324}{Z^*} - \frac{0.0797}{Z^{*2/3}}\right]^{1/3} \tag{4.27}$$

Kays and London (1984) suggested the equation given below for the calculation of the correction factor for rotational speed (φ_r):

$$\varphi_r = 1 - \frac{1}{9\, C_r^{*\,1.93}} \tag{4.28}$$

Here, C_r^* is heat capacity ratio of the rotary heat exchanger and is defined as below:

$$C_r^* = \frac{C_r}{C_{min}} \tag{4.29}$$

In which, C_r is the heat capacity of the rotary heat exchanger and it can be calculated as:

$$C_r = M_s Cp_r N \tag{4.30}$$

where, M_s and Cp_r are respectively, mass of the solid body of the rotary heat exchanger and specific heat capacity of the rotary heat exchanger. N is the rotational speed of the rotary heat exchanger.

The correction factor for rotational speed (φ_r) can also be calculated using the equation given by Worsøe-Schmidt (1991):

$$\varphi_r = 1 - \frac{0.114\left[1 - e^{(-Ntu)}\right]}{C^{*\,0.44} C_r^{*\,1.93}} \tag{4.31}$$

For the regenerator used in the experiments, there was no cleaning region, so the correction factor for cleaning region (φ_c) was taken as 1 in calculations.

The effectiveness of the rotary heat exchanger was obtained analytically using a computer code program and given in Appendix-D.

4.3. Numerical Study

In this study, the effectiveness of the rotary heat exchanger was also obtained numerically using a computer code (given in Appendix-E) written by Ünal (1996) (see also Yılmaz and Ünal, 1996).

Ünal (1996) obtained numerical results by writing a mathematical model for the fluid and for the wall of the rotary heat exchanger matrix as shown in Fig. 4.18.

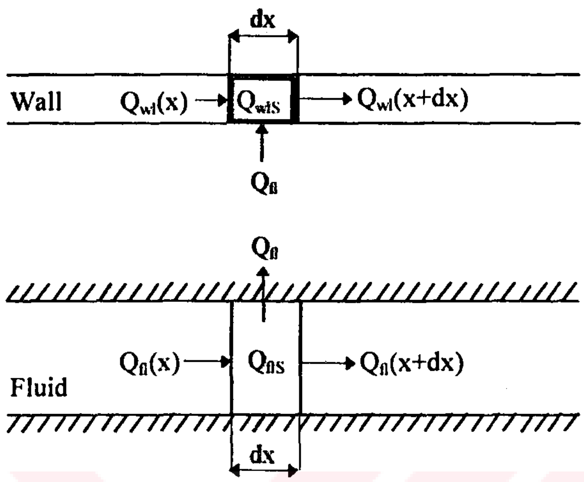


Figure 4.18. Energy balance for the unit element of rotary heat exchanger, and fluid.

Applying the first law of thermodynamics to the unit element of the wall and the fluid, Ünal (1996) wrote the differential equations for wall temperature and fluid temperature with some assumptions. Dimensionless form of the equations are given below:

$$\frac{1}{R}\frac{\partial T_{fl}^{*}}{\partial t^{*}} + \frac{\partial T_{fl}^{*}}{\partial x^{*}} - \frac{Fo_{fl}}{R}\frac{\partial^2 T_{fl}^{*}}{\partial x^{*2}} + Ntu\, T_{fl}^{*} = Ntu\, T_{wl}^{*} \qquad \text{(for fluid)} \qquad \textbf{(4.32)}$$

$$\frac{\partial T_{wl}^{*}}{\partial t^{*}} - Fo_{wl}\frac{\partial^2 T_{wl}^{*}}{\partial x^{*2}} + \frac{Ntu}{C_r^{*}} T_{wl}^{*} = \frac{Ntu}{C_r^{*}} T_{fl}^{*} \qquad \text{(for wall)} \qquad \textbf{(4.33)}$$

For these coupled differential equations and the boundary conditions (constant temperature of cold and hot fluid at the inlet and zero conductive heat transfer on the wall at the inlet and at the outlet of the regenerator) discretization equations using finite differences were obtained. At the beginning of the calculations ($t^* = 0$), a constant temperature in the regenerator (a mean temperature of hot and cold fluid) is assumed. Firstly, fluid temperature and then wall temperature are calculated. These calculations are carried out using a computer program written in Fortran programming language and continued until the temperature does not change in each period. From these temperatures, the effectiveness of the regenerator is calculated. Detailed information about the numerical method employed and about the influences of the different boundary conditions of the fluid at the end of the regenerator ($\partial T_{fl}^*/\partial x^* = 0$, $\partial^2 T_{fl}^*/\partial x^{*2} = 0$ and $\partial T_{fl}^*/\partial x^* = \partial^2 T_{fl}^*/\partial x^{*2} = 0$) are given in Ünal (1996).

5. RESULTS AND DISCUSSIONS

Effectiveness of the rotary-type heat exchanger was calculated using the measured values and the equations given in material and method section. The effect of rotational speed and flow rate was examined using the results obtained experimentally, analytically and numerically.

5.1. Experimental Results

In the experiments, various rotational speeds (in the range of 0.05 rev/min and 7 rev/min) and three different flow rates (1.042 m^3/s, 0.874 m^3/s and 0.576 m^3/s) were covered for the investigation of effectiveness of rotary-type heat exchanger.

The effectiveness of the rotary heat exchanger for various rotational speeds at the flow rate of 1.042 m^3/s is shown in Fig. 5.1. As can be seen from the figure, when the rotational speed is higher than 1.5 rev/min, the effectiveness is constant and has a value of about 0.56. For the lower values of rotational speed (below 1.5 rev/min), the effectiveness increases sharply with the increase of rotational speed.

When the flow rate is decreased to 0.874 m^3/s, the variation of the effectiveness with rotational speed is shown in Fig. 5.2. After the rotational speed of 2 rev/min, the effectiveness of the rotary heat exchanger is nearly 0.63. Below 2 rev/min, the effectiveness increases rapidly with the increase of rotational speed.

The results for the flow rate of 0.576 m^3/s is given in Fig. 5.3. Similar to the previous cases, higher than a certain value of rotational speed (2 rev/min), the effectiveness does not change with rotational speed and it is around 0.67. Firstly, the effectiveness increases sharply until 0.6 rev/min, then slowly between 0.6 rev/min and 2 rev/min.

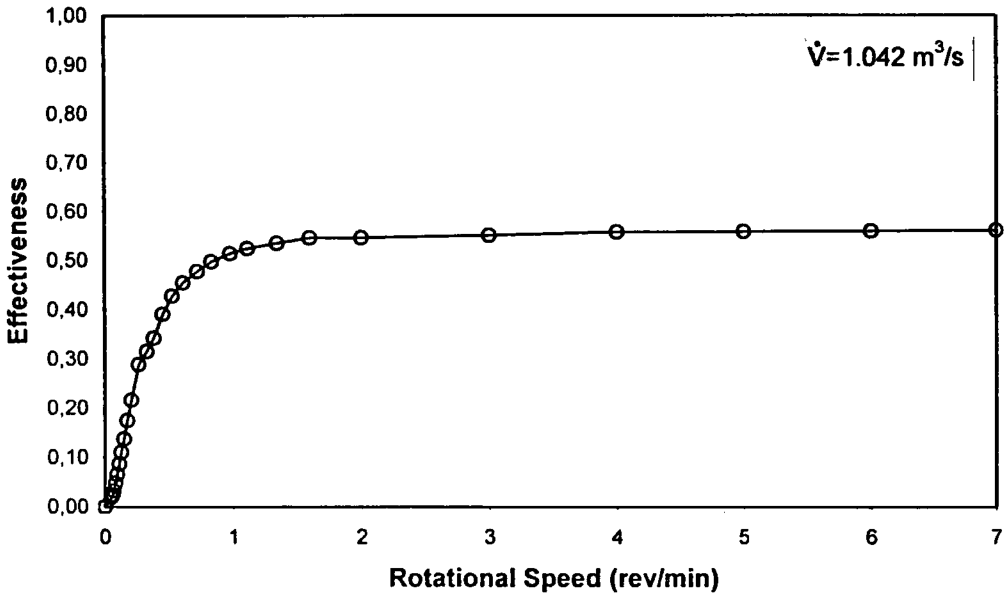


Figure 5.1. Variation with rotational speed of effectiveness obtained from the experiments for flow rate of 1.042 m^3/s.

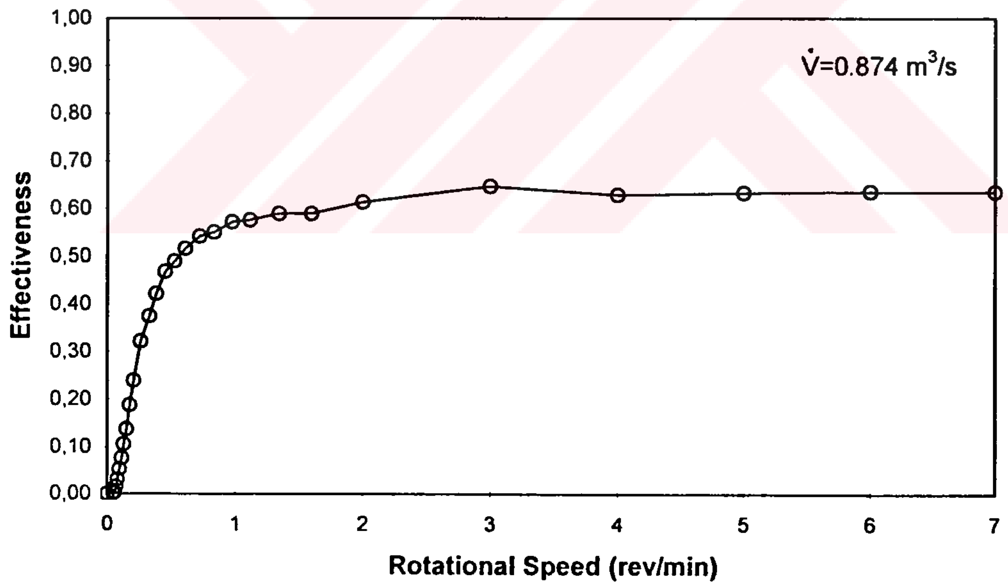


Figure 5.2. Variation with rotational speed of effectiveness obtained from the experiments for flow rate of 0.874 m^3/s.

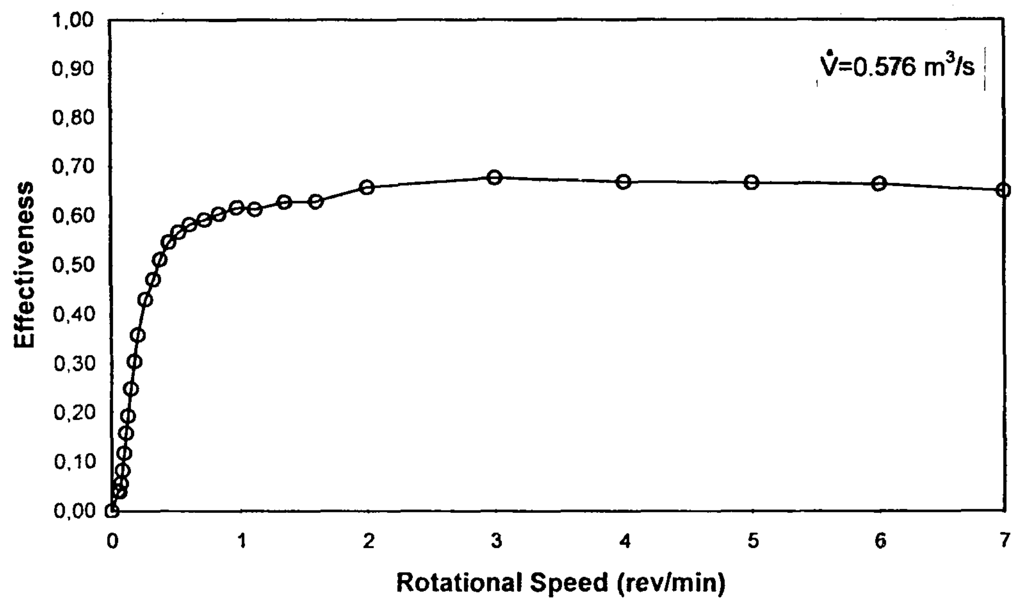


Figure 5.3. Variation with rotational speed of effectiveness obtained from the experiments for flow rate of 0.576 m^3/s.

5.2. Analytical Results

The effectiveness of the rotary heat exchanger was calculated analytically using the equations given in Section 4.2. The calculations were made for the same parameters covered in the experiments. The influence of the flow rate and the rotational speed on effectiveness of the rotary heat exchanger was obtained analytically. Both equations given by Kays and London (1984) (Eqn. 4.28) and Worsøe-Schmidt (1991) (Eqn. 4.31) for the effect of rotational speed were used in the calculations.

Analytically obtained effectiveness of the rotary heat exchanger for various rotational speeds for a flow rate of 1.042 m^3/s is shown in Fig. 5.4. Kays and London (1984) and Worsøe-Schmidt (1991) produced almost the same results for all range and the variation of effectiveness with rotational speed is very similar to the experimental ones. Here again, the effectiveness increases very rapidly with the increase of rotational speed up to a certain value (0.26 rev/min) and then it is almost constant at 0.58.

The results for lower flow rates (0.874 m^3/s and 0.576 m^3/s) are shown in Fig. 5.5 and Fig. 5.6. The pattern of behavior in the analytical results with lower flow rates continues to be similar to that seen in the high flow rate case. For the higher values of rotational speed, the flow rate does not change with rotational speeds. The critical values are 0.26 rev/min for the flow rate of 0.874 m^3/s and 0.173 rev/min for the flow rate of 0.576 m^3/s. Below these critical values, an increase in rotational speed is followed by the increase of effectiveness. Kays and London and Worsøe-Schmidt continues to be in good agreement with each other for these values of flow rate.

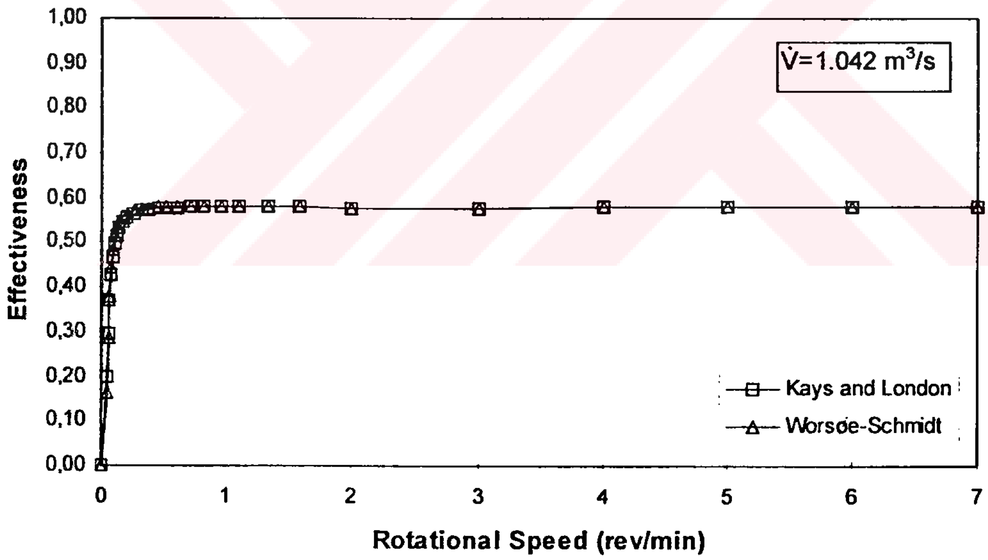


Figure 5.4. Variation with rotational speed of effectiveness obtained from the analytical study for flow rate of 1.042 m^3/s.

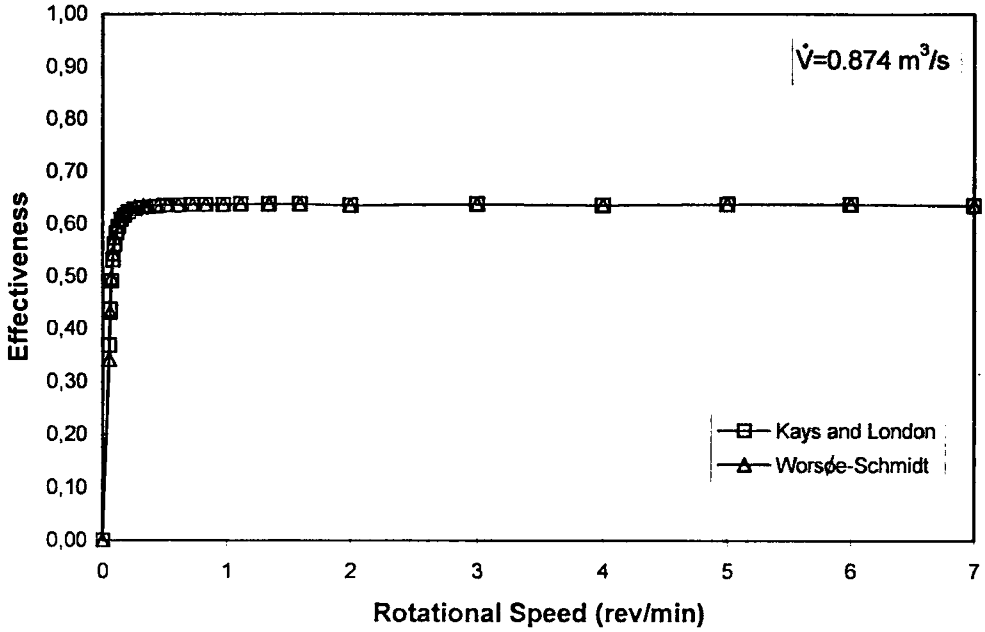


Figure 5.5. Variation with rotational speed of effectiveness obtained from the analytical study for flow rate of 0.874 m^3/s.

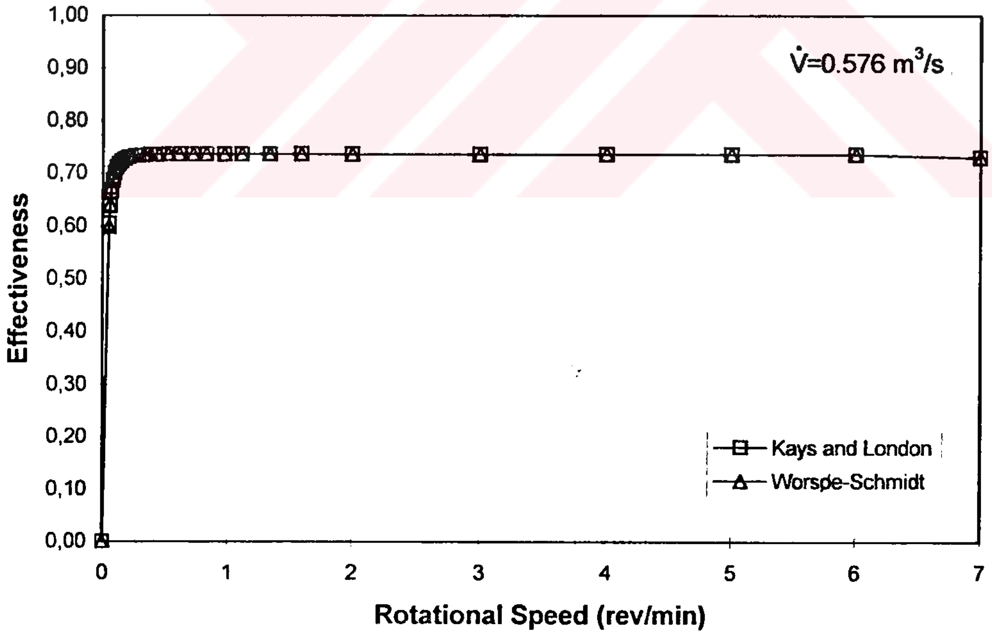


Figure 5.6. Variation with rotational speed of effectiveness obtained from the analytical study for flow rate of 0.576 m^3/s.

5.3. Numerical Results

The numerical results were obtained by solving the differential equations of rotary regenerator.

The numerically obtained effectiveness values show a similar trend seen in the experimental and analytical ones. Fig. 5.7 to Fig. 5.9 show the results for the flow rates values of 1.042 m^3/s, 0.874 m^3/s and 0.576 m^3/s respectively.

The critical rotational speed above which the effectiveness almost constant is 1.5 rev/min for the flow rate of 1.042 m^3/s, 2 rev/min for the flow rate of 0.874 m^3/s, and 1 rev/min for the flow rate of 0.576 m^3/s. The effectiveness values for the constant region are 0.56, 0.60 and 0.69 for the flow rates of 1.042 m^3/s, 0.874 m^3/s and 0.576 m^3/s respectively.

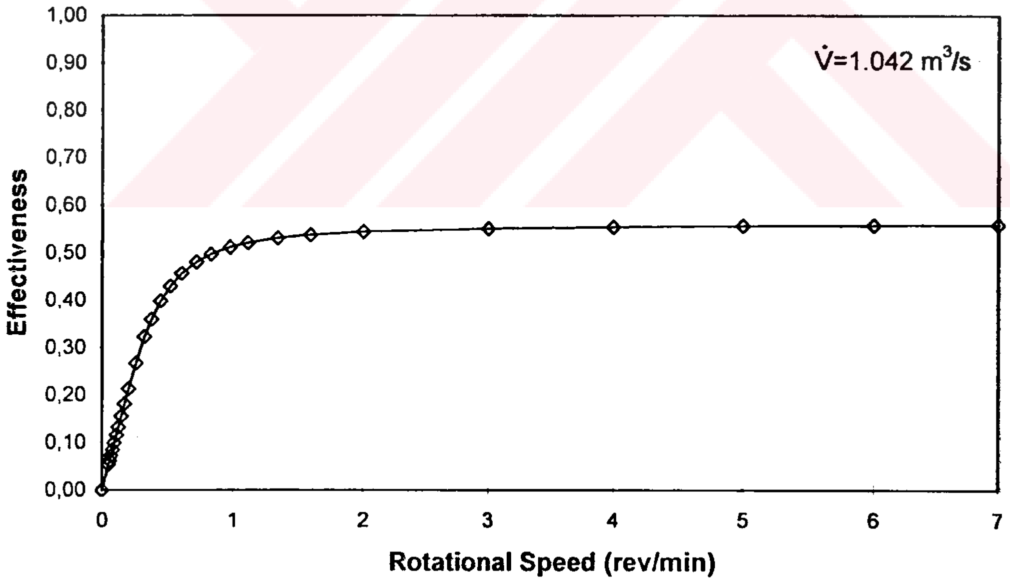


Figure 5.7. Variation with rotational speed of effectiveness obtained from the numerical study for flow rate of 1.042 m^3/s.

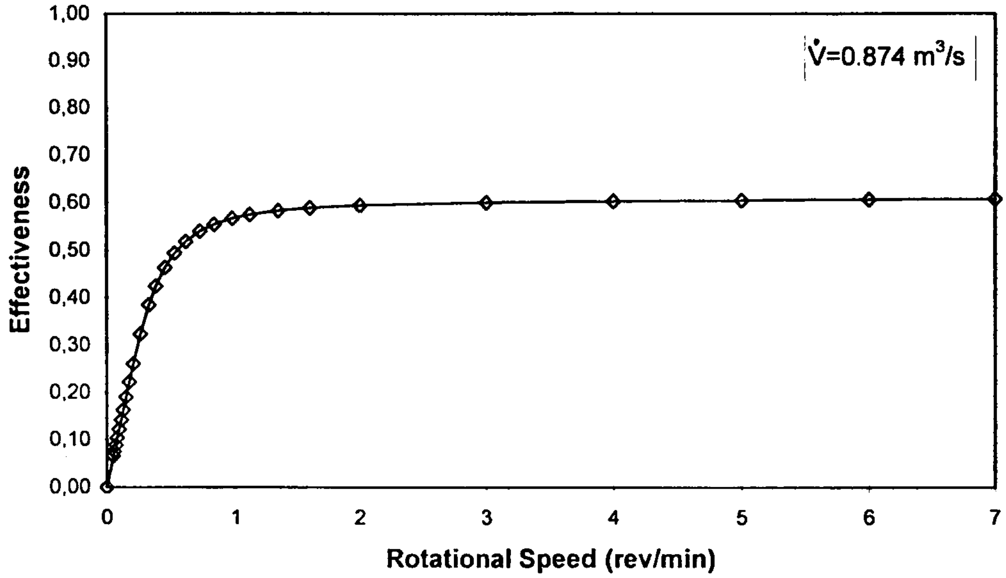


Figure 5.8. Variation with rotational speed of effectiveness obtained from the numerical study for flow rate of 0.874 m^3/s.

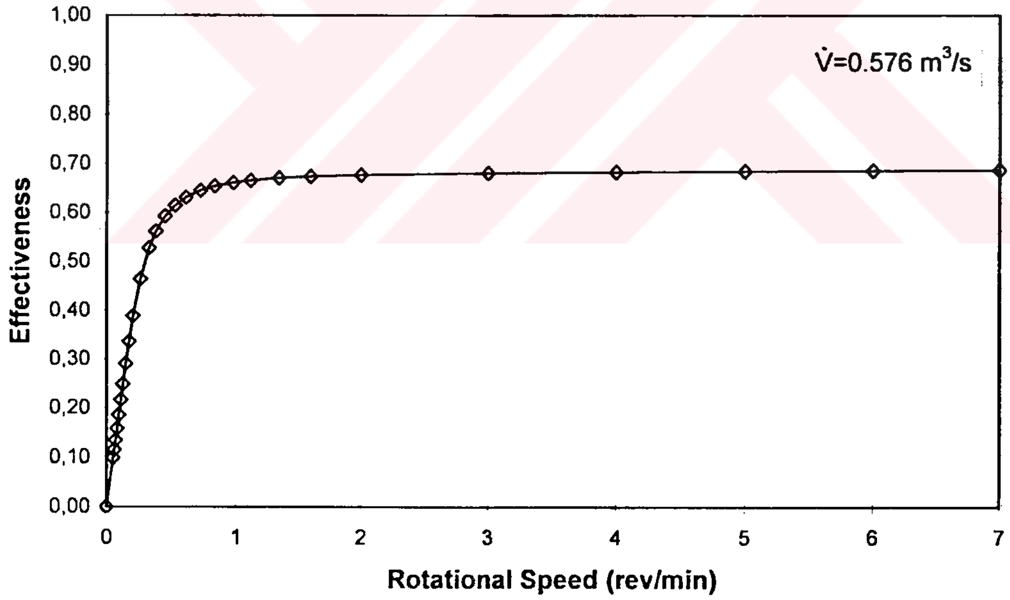


Figure 5.9. Variation with rotational speed of effectiveness obtained from the numerical study for flow rate of 0.576 m^3/s.

5.4. Comparison and Discussion of Results

The influence of rotational speed and of flow rate on the effectiveness of the rotary regenerator is considered firstly. Then, the experimental, the analytical and the numerical results are compared with each other for each flow rate tested.

5.4.1. Influence of Rotational Speed on the Effectiveness of the Rotary-Type Heat Exchanger

All the experimental results (Fig. 5.10) show that, the effectiveness increases with the increase of rotational speed up to a critical value. This critical value of rotational speed is around 2 rev/min. At zero rotational speed, there is no heat transport from hot side to cold side except heat conduction through the solid matrix of the rotary regenerator. With the increase of rotational speed, the rate of transport of heat by the solid matrix of the rotary regenerator is increased. As a result the effectiveness of the rotary regenerator gets better. However, increasing rotational speed above a certain value, does not enhance heat transport, hence the effectiveness. This is due to the fact that, the time in which the solid matrix of the regenerator is exposed to the hot or cold fluid (exposed time) decreases with the increase of rotational speed. This deteriorates the amount of heat transferred to or from the solid matrix.

The trends seen in the experimental results (Fig. 5.10) are also evident in the case of the analytical (Fig. 5.11) and the numerical ones (Fig. 5.12).

The analytical results shown in Fig. 5.11 were obtained using the equation (Eqn. 4.28) given by Kays and London (1984). The results of Worsøe-Schmidt equation (Eqn. 4.31) are not shown here, since they are very close to that of Kays and London equation and Eqn. (4.28) is simpler in form than Eqn. (4.31).

5.4.2. Influence of Flow Rate on the Effectiveness of the Rotary-Type Heat Exchanger

In Fig. 5.10, the effectiveness obtained from the experiments for three different flow rates are compared. It is evident from the figure that, decrease in flow rate, causes increase of effectiveness of the rotary regenerator. For example, a decrease of flow rate from 1.042 m^3/s to 0.576 m^3/s causes the increase of effectiveness 16.4% in the constant effectiveness region. However, the influence of flow rate on the effectiveness is not significant for low rotational speeds (≤0.5 rev/min). When the rotational speed approaches to zero, all the flow rates tested produce almost the same value.

When the flow rate decreases, the time required for travel of a fluid particle in the channel of rotary heat exchanger (residence time) increases. For example, residence time increases from 0.032 s to 0.060 s when the flow rate decreased from 1.042 m^3/s to 0.576 m^3/s. In turn, increase of residence time leads to an increase in the amount of heat transfer between the fluid and the wall and to a better effectiveness.

The pattern of behavior in the case of the analytical (Fig. 5.11) and the numerical (Fig. 5.12) results continues to be similar to that seen in the experiments, although differences in terms of the rate of increase of effectiveness are evident. For example, a decrease of flow rate from 1.042 m^3/s to 0.576 m^3/s causes the increase of analytical effectiveness 20.5% and the increase of numerical effectiveness 18.8% in the constant effectiveness region. The corresponding value was 16.4% in the experiments.

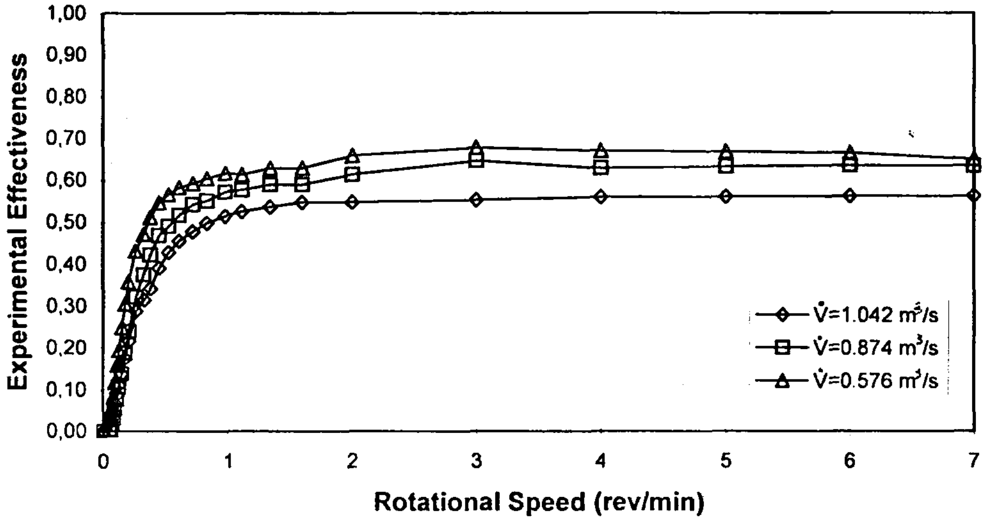


Figure 5.10. Comparison of experimental effectiveness for three different flow rates.

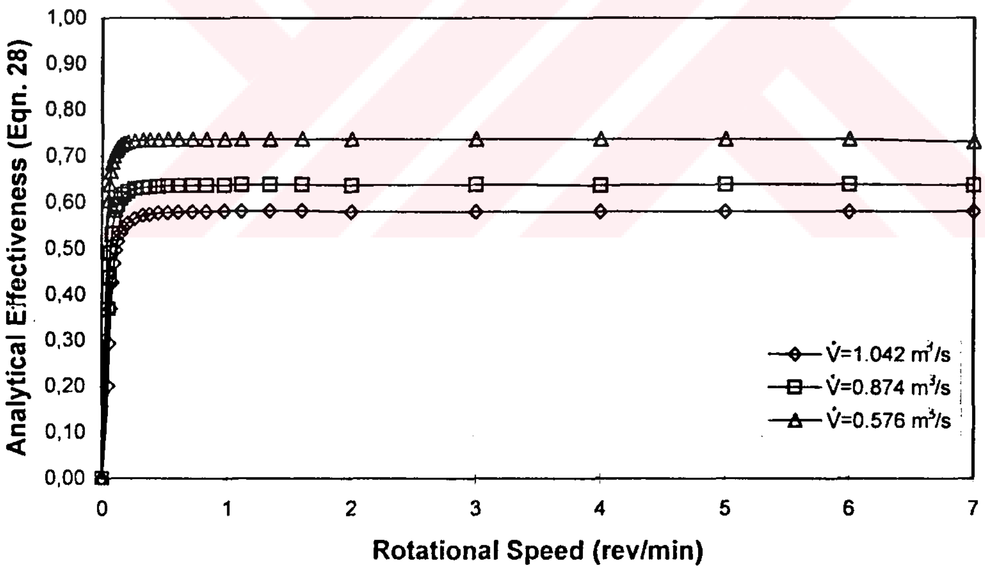


Figure 5.11. Comparison of analytical effectiveness for three different flow rates.

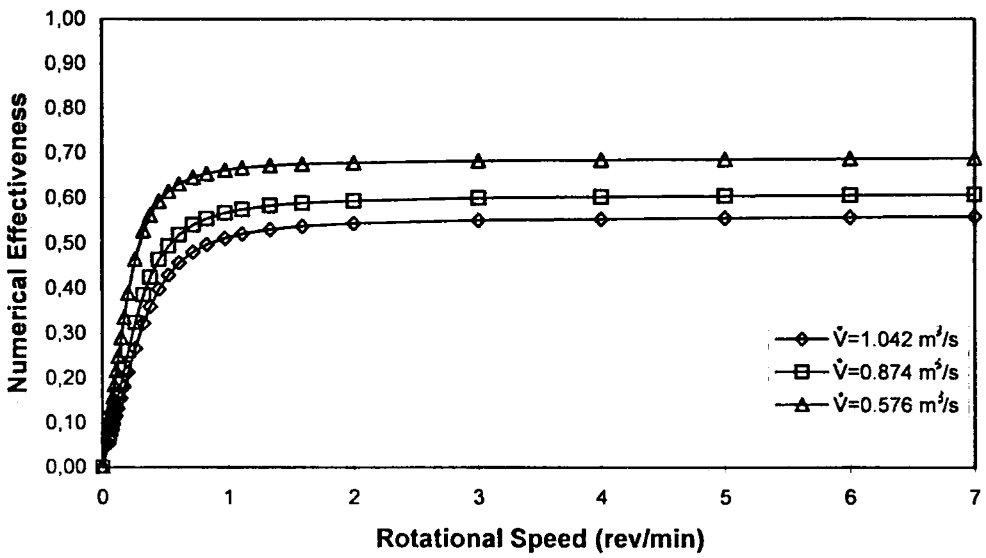


Figure 5.12. Comparison of numerical effectiveness for three different flow rates.

5.4.3. Comparison of Experimental, Analytical and Numerical Results

The effectiveness values of the rotary heat exchanger obtained experimentally, analytically and numerically for various rotational speeds were compared for each flow rate tested. As explained in proceeding section, the analytical effectiveness was obtained using the equation (Eqn. 4.28) given by Kays and London (1984), due to its simplicity in form.

Fig. 5.13 shows a comparison for the flow rate of 1.042 m^3/s. A very good agreement is seen between the experimental and the numerical results for all values of rotational speed. However, the analytical effectiveness is slightly higher than the other two for high rotational speeds (> 1.5 rev/min). The difference between the analytical results and the other two is significantly high for low rotational speeds (<1.5 rev/min). Similar results are obtained for the flow rate values of 0.874 m^3/s and 0.576 m^3/s (Fig. 5.14 and Fig. 5.15).

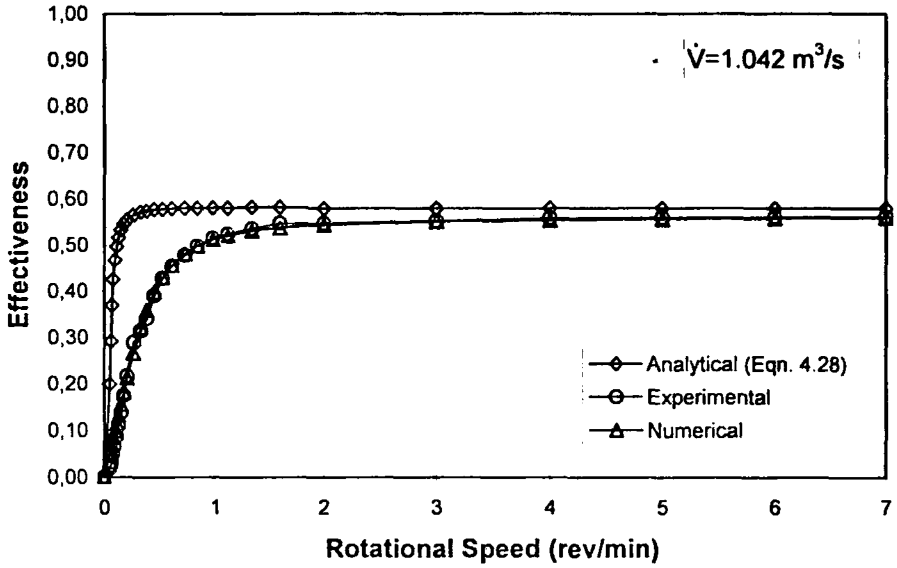


Figure 5.13. Comparison of the effectiveness values obtained from the experimental, the analytical and the numerical studies for flow rate of 1.042 m^3/s.

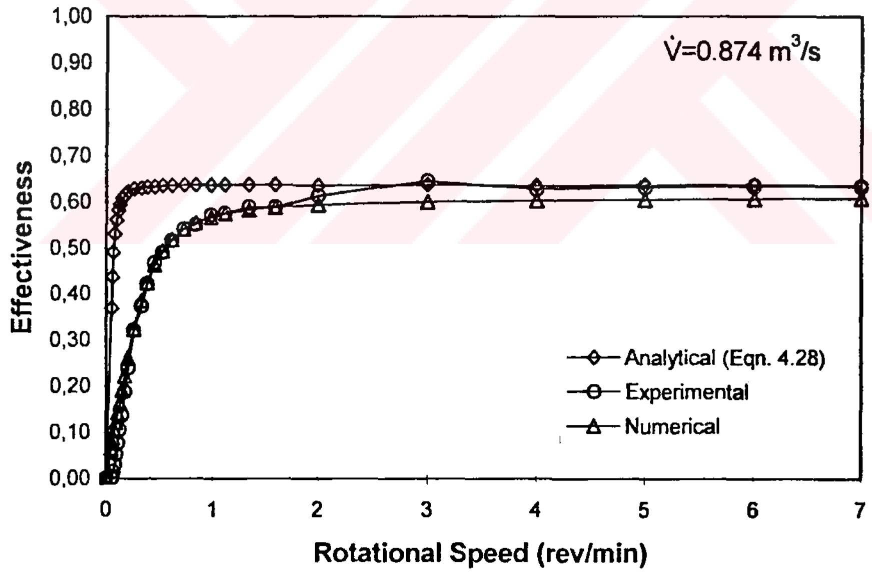


Figure 5.14. Comparison of the effectiveness values obtained from the experimental, the analytical and the numerical studies for flow rate of 0.874 m^3/s.

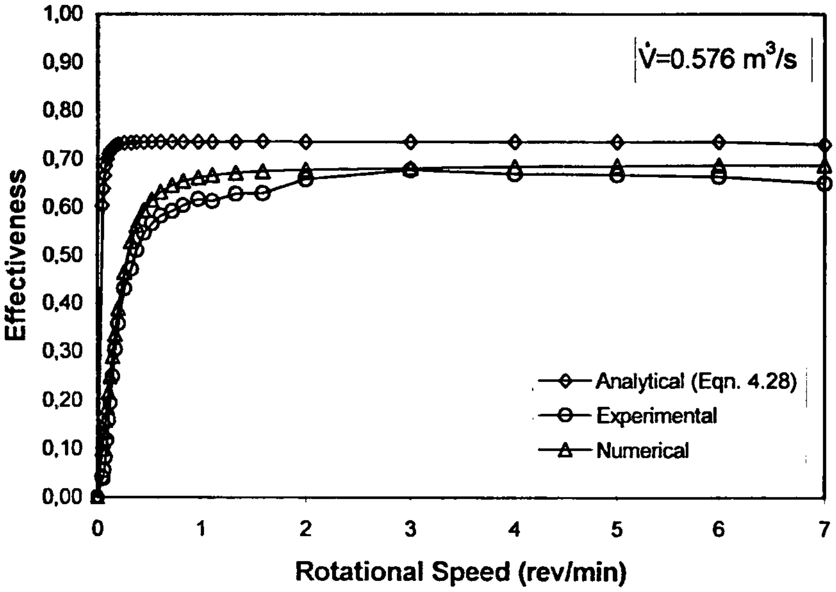


Figure 5.15. Comparison of the effectiveness values obtained from the experimental, the analytical and the numerical studies for flow rate of 0.576 m^3/s.

Investigation of figures from Fig. 5.13 to Fig. 5.15 reveals that Eqn. (4.28) (or Eqn. 4.31) does not reproduce correctly the influence of rotational speed on the effectiveness of the rotary regenerator for low values of rotational speed (generally less than 2 rev/min). Therefore, Eqn. (4.28) was modified to fit the experimental data over the range of rotational speed covered in this study (0.05 rev/min ~ 7 rev/min). The new correction factor for rotational speed (φ_r) is:

$$\varphi_r = \frac{0.24\, C_r^*}{\left[1 + 0.02\, C_r^{*1.82} + \left(0.24\; C_r^*\right)^3\right]^{1/3}} \qquad \textbf{(5.1)}$$

The analytical effectiveness values obtained using the Eqn. (5.1) are shown in Fig. 5.16 to Fig. 5.18 for three different values of flow rate. As shown from the figures, using the new correction factor for rotational speed, for high flow rates

(1.042 m^3/s and 0.874 m^3/s) the experimental, the analytical and the numerical effectiveness values were in a good agreement. When the flow rate was decreased to 0.576 m^3/s, the experimental and the numerical results were nearly close to each other, but the analytical result was not in a good agreement with the others for high rotational speeds.

The critical rotational speed above which the effectiveness almost constant is 2 rev/min for the flow rate of 1.042 m^3/s, 1.5 rev/min for the flow rate of 0.874 m^3/s and 1 rev/min for the flow rate of 0.576 m^3/s. The experimental, the analytical and the numerical effectiveness values are very close to each other for high flow rates, and in the constant effectiveness region, the effectiveness are 0.58, 0.63 for the flow rates of 1.042 m^3/s, 0.874 m^3/s respectively. However, flow rate of 0.576 m^3/s, the analytical effectiveness is slightly higher than the others. In the constant effectiveness region, the discrepancy is about 5.5%.

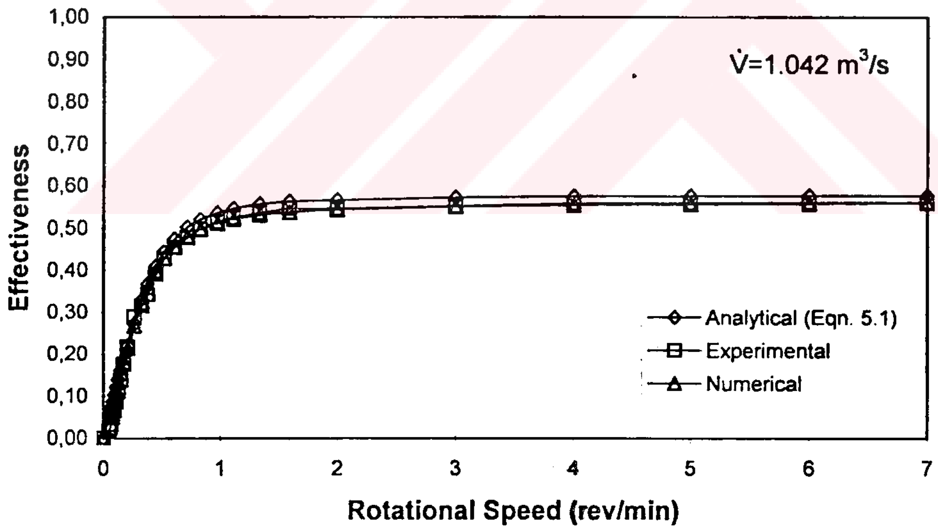


Figure 5.16. Comparison of the effectiveness values obtained from the experimental, the analytical and the numerical studies for flow rate of 1.042 m^3/s.

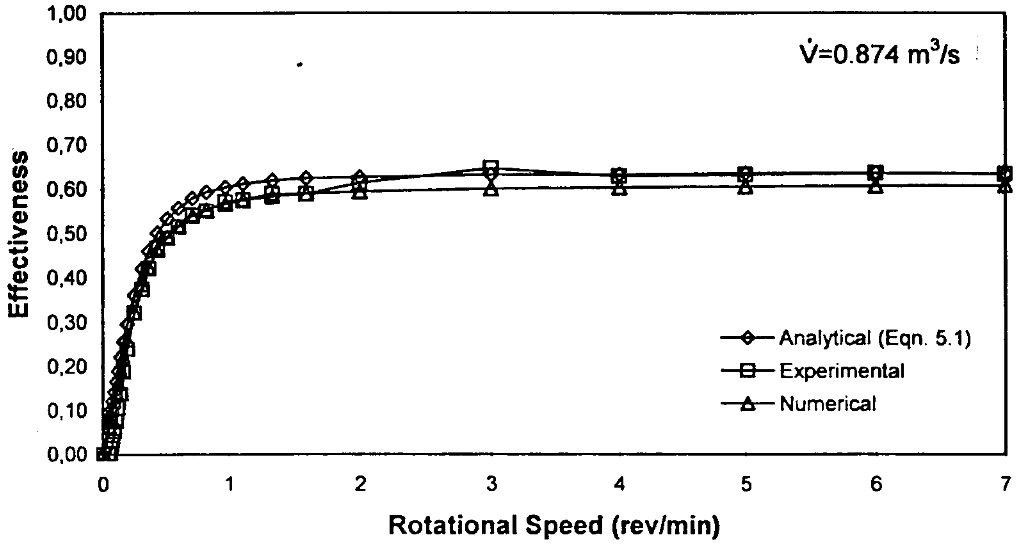


Figure 5.17. Comparison of the effectiveness values obtained from the experimental, the analytical and the numerical studies for flow rate of 0.874 m^3/s.

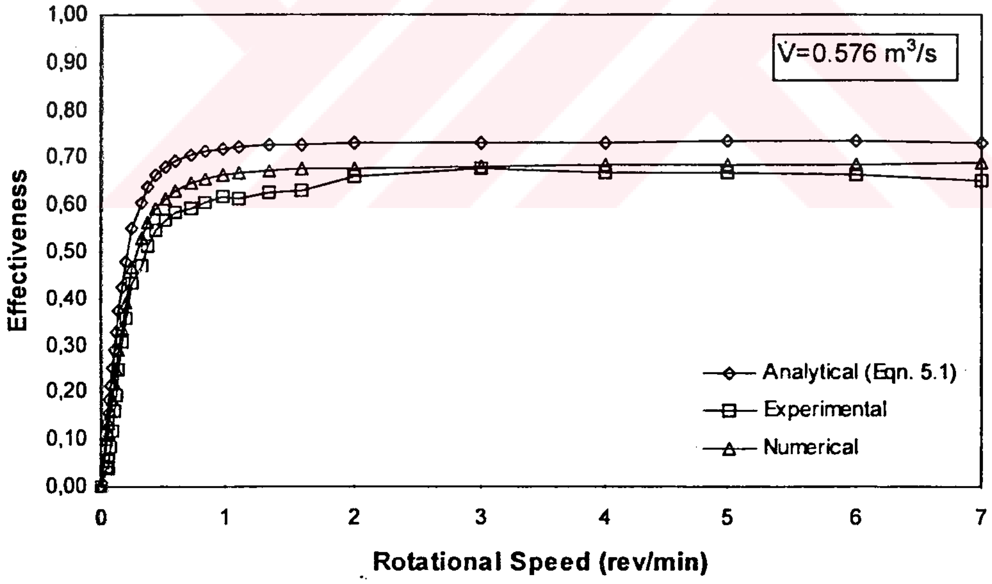


Figure 5.18. Comparison of the effectiveness values obtained from the experimental, the analytical and the numerical studies for flow rate of 0.576 m^3/s.

6. CONCLUSIONS

In this thesis, the influence of rotational speed and flow rate on effectiveness of a rotary-type heat exchanger was investigated experimentally, analytically and numerically.

The major conclusions from this study are as follows:

1) With increase of rotational speed up to a certain value, the rate of transport of heat from hot fluid to cold one by the solid matrix of the rotary regenerator is increased. This leads to the effectiveness of the rotary regenerator gets better. However, increasing rotational speed above a certain value does not enhance heat transport, hence the effectiveness. This is due to the fact that, the time in which the solid matrix of the regenerator is exposed to the hot or cold fluid decreases with the increase of rotational speed.

2) Decrease of flow rate causes an increase in the effectiveness of the rotary regenerator. This is due to the time required for travel of a fluid particle in the rotary heat exchanger (residence time) increases with the decrease of flow rate. Increase of residence of time leads to an increase in the amount of heat transfer between fluid and the wall and hence, the effectiveness increases.

3) For the high rotational speeds (2-7 rev/min) the experimental, the analytical and the numerical results were found to be in a good agreement. However, for the low rotational speeds (0.05-2 rev/min) the analytical effectiveness results obtained using the correction factor for rotational speed suggested by Kays and London (Eqn. 4.28) and by Worsøe-Schmidt (Eqn. 4.31) were higher than the others. A new correction factor for rotational speed (Eqn. 5.1) was obtained to describe the influence of rotational speed and it was seen that the analytical effectiveness

calculated using this new correction factor produced results that were in a good agreement with the experimental and the numerical ones.

7. RECOMMENDATIONS FOR FUTURE WORK

1) It was shown that from the temperature measurements, there was a temperature fluctuation at measurement sections as shown in Fig. 7.1. The reason for that is that central heating system works as on-off. This affects the temperature of the waste air first and then the others. To stabilize the temperature variation, heating system was modified as shown in Fig. 7.2. The new heating system can work in two modes. The system can either work described in this thesis or the other was can be used changing the valve position to stabilize the temperature variation in the coil.

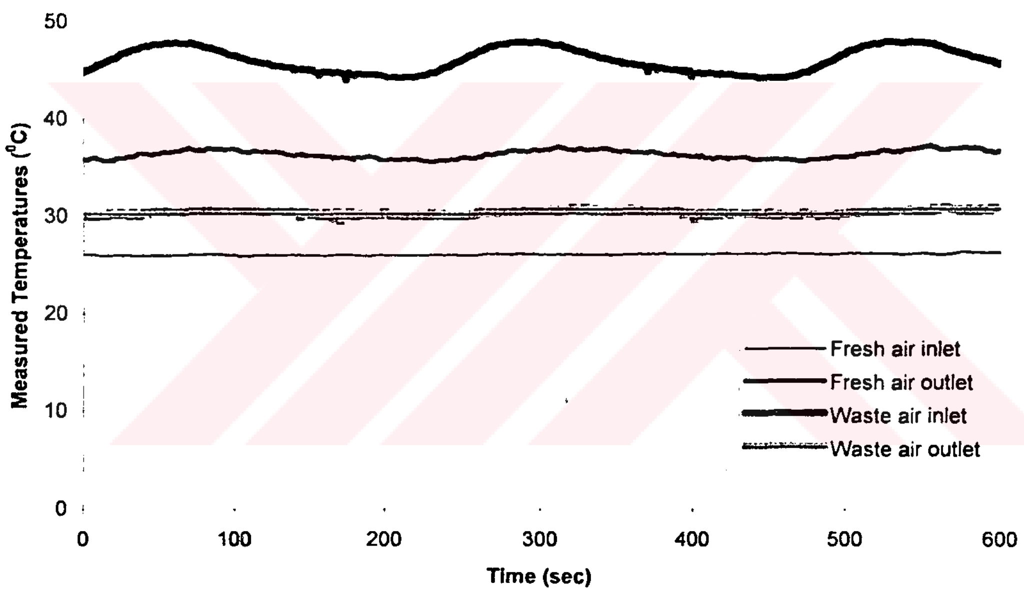


Figure 7.1. Temperature fluctuation at each section.

As shown in Fig. 7.2 that, there is a storage tank after the central heating system, then a small capacity heater was replaced after the tank to stabilize the temperature of the hot water which is fed to the coil in the experimental setup. The heater is controlled using a temperature controller to keep the temperature of the hot water at desired level. After the heater, a circulation pump is used to pump the hot water to the coil.

Although this system has been constructed, it is not tested yet. A series of experiments should be performed to see the influence of fluctuation of temperature on the effectiveness.

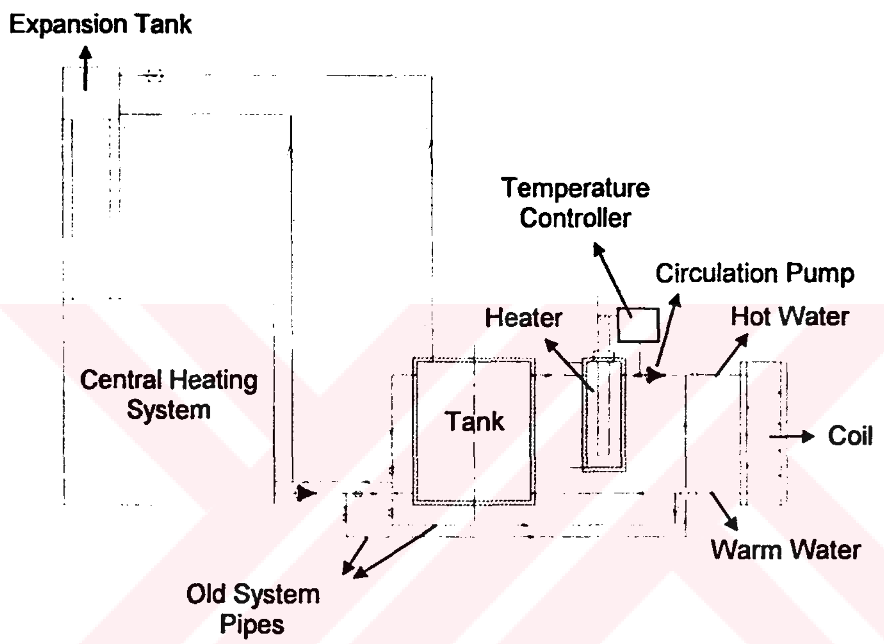


Figure 7.2. A new system for stabilize the temperature fluctuation.

2) Using the new correction factor for rotational speed, it was also concluded that, for high flow rates (1.042 m^3/s and 0.874 m^3/s) the experimental, the analytical and the numerical effectiveness were in good agreement. When the flow rate was decreased to 0.576 m^3/s, the experimental and the numerical results continue to be in good agreement. However, the analytical values departs from the two for high rotational speeds. Further experiments should be carried out with lower flow rates, to see whether this departure is due to experimental errors or due to actual phenomena. If the latter is the case, the correction factor for rotational speed should be modified.

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CURRICULUM VITAE

Bahadır ATALAY was born in Kırıkkale, 1974. He finished his first and high school education in Mersin. He gained his Bachelor of Science in Çukurova University Mechanical Engineering Department in 1996. On completion of his degree, he started his Master of Science studies in Çukurova University, Institute of Natural and Applied Sciences, Mechanical Engineering Division. He still works as a Research Assistant in this division.

APPENDIXES

Appendix-A

A typical data file collected by the data acquisition system used in the experimental study.

```
* Product: ADC+16
* Date:      06/16/97
* Time:      14:06:25
*
* Channel Sequence:     00 + 03
* Units:     Millivolts
*
Start Data
00001 14:06:26 +01.046 +01.444 +01.818 +01.203
00002 14:06:28 +01.046 +01.446 +01.822 +01.210
00003 14:06:29 +01.047 +01.445 +01.824 +01.207
00004 14:06:30 +01.045 +01.448 +01.826 +01.205
00005 14:06:31 +01.043 +01.450 +01.829 +01.204
00006 14:06:32 +01.045 +01.450 +01.831 +01.203
00007 14:06:33 +01.045 +01.452 +01.836 +01.202
00008 14:06:34 +01.044 +01.454 +01.840 +01.202
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00010 14:06:35 +01.042 +01.449 +01.847 +01.217
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Appendix-B

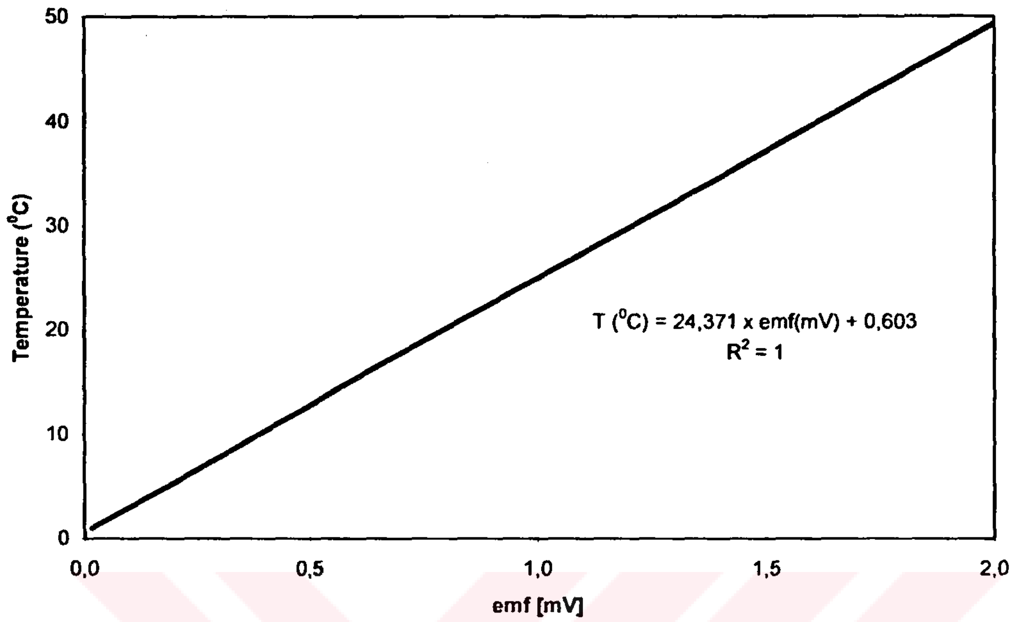

Figure A. Calibration curve for K-type thermocouples used in the experimental study.

The calibration curve was taken from Omega Temperature Measurement and Encyclopedia (Vol.28) for K-type thermocouples.

Appendix-C

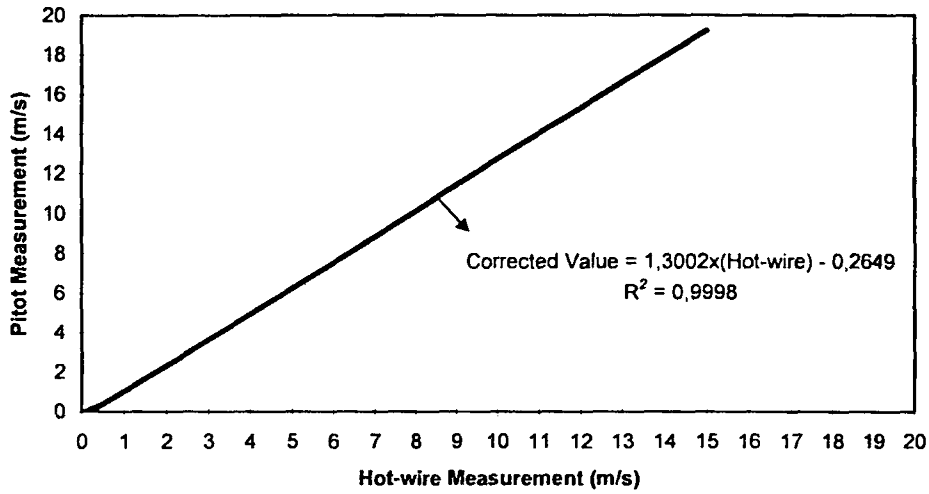

Figure B. Calibration curve for the hot-wire used in the experimental study.

Hot-wire was calibrated using a pitot-tube in an air flow bench system. Hot-wire and pitot-tube measurements were taken at the same point in the bench for various air velocities. Then the hot-wire readings were calibrated with respect to pitot-tube values. So, from the calibration curve, corrected value for hot-wire was found.

Appendix-D

A Pascal Computer Program to calculate the experimental and the analytical effectiveness of the rotary-type heat exchanger using the experimental data:

```
Program Effectiveness;
Uses Crt;
Var
  TG1,TG2,TC1,TC2,VG1,VG2,V1,V2,U1,U2,T1,T2,A1,A2:REAL;
  RO1,RO2,CP1,CP2,PR1,PR2,LAM1,LAM2,VIS1,VIS2:Real;
  AF,C,DEQ,DMAX,D1,N:Real;
  FIS,DFIM,DFI,FI,NUS,KAS,KA,ITA,KIS,KI:Real;
  RE1,RE2,X1,X2,DELTP1,DELTP2,PE1,PE2,HEADLOSS1,HEADLOSS2:Real;
  C1,C2,CMIN,CMAX,CY,Z1,Z2,NU1,NU2,ALFA1,ALFA2,K,NTU:Real;
  EPSO,FIR,CR,CRY,N1,DS,ANALYEFF,EXPEFF,EXPEFF1,EXPEFF2:REAL;
  AC1,FKA,X,ATF:REAL;
  FL,KF:REAL;
  Q1,Q2,Q:REAL;
  SEC,TUS:CHAR;
  NAME1,NAME2:STRING[15];
  CAL1,CAL2:TEXT;

Const
  PI=3.14; S=0.35E-3; L=0.2; ROAL=2700; CPAL=0.896;
  a=0.00344; AHALF=0.158; MS=90; F=134.16;

Function Us(x,y:Real):DOUBLE;
 begin
  Us:=exp(y*ln(x));
 end;
```

```
{Main Program}
BEGIN
Clrscr;
gotoxy(10,11);WRITE('OUTPUT File: ');READLN(NAME1);
gotoxy(10,12);WRITE('INPUT File: ');READLN(NAME2);
ASSIGN (CAL1,NAME1);
REWRITE (CAL1);
ASSIGN (CAL2,NAME2);
RESET (CAL2);

clrscr;
gotoxy(10,8);Write ('Rotational Speed:=  ');Readln (CAL2,N1);
gotoxy(10,10);Write ('Inlet Temperature  1:=  '); Readln (CAL2,TG1);
gotoxy(10,11);Write ('Outlet Ten.perature 1:=  '); Readln (CAL2,TC1);
gotoxy(10,12);Write ('Inlet Temperature  2:=  '); Readln (CAL2,TG2);
gotoxy(10,13);Write ('Outlet Temperature 2:=  '); Readln (CAL2,TC2);

{Volume Flow Rate}
VG1:=1.042;VG2:=0.933; {Flow Rate}
{VG1:=0.874;VG2:=0.753; {Flow Rate}
{VG1:=0.576;VG2:=0.493; {Flow Rate}

{**************************************************************}
   AF:=((Us(3,0.5))*A*A)/4;
   C:=3*A;
   DEQ:=4*AF/C;
   DMAX:=DEQ;
   FL:=F/2;
{**************************************************************}
```

```
T1:=(TG1+TC1)/2;
T2:=(TG2+TC2)/2;
RO1:=1.2642-(T1*3.805E-3);
RO2:=1.2642-(T2*3.805E-3);
CP1:=1.006+(T1*5E-5);
CP2:=1.006+(T2*5E-5);
LAM1:=0.02457+(T1*7.3E-5);
LAM2:=0.02457+(T2*7.3E-5);
VIS1:=1.334E-5+(T1*8.95E-8);
VIS2:=1.334E-5+(T2*8.95E-8);
PR1:=0.697-(T1*1.5E-4);
PR2:=0.697-(T2*1.5E-4);
A1:=(19.1+(T1*0.135))*1E-6;
A2:=(19.1+(T2*0.135))*1E-6;

{General Calculations}
D1:=DEQ/DMAX;
N:=(AF/((3.1415/4)*Us(DEQ,2)));
FIS:=0.5155*US(D1,2)/(3-D1);
DFIM:=(0.007*US(D1,8))/((1+10*US(D1,-28))*(US((1+64*US(10,-
8)*US(D1,28)),0.5)));
DFI:=DFIM*((0.95*US((N-1),0.5))/(1+0.038*US((N-1),3)));
FI:=1+((FIS-1)/(1+1/(N-1)))+DFI;
NUS:=FI*3.657;
KIS:=(3/8)*D1*D1*(3-D1);
KI:=1+((KIS-1)/(1+0.33*US(D1,2.25)/(N-1)));
ITA:=1+(((3*US((D1/2),(7/8))/(1+D1))-1)/(1+0.25/(N-1)));
KAS:=(12/5)*US((3-D1),2)*(((9/7)*((3-D1)/(7-3*D1)))-1/(5-2*D1));
KA:=1.33/(1+((1.33/KAS)-1)/((1+0.74*US(D1,2))/(N-1)));
```

```
{Calculation of Pressure Losses}
V1:=(VG1+VG1)/2;
U1:=V1/AHALF;
RE1:=U1*DEQ/VIS1;
X1:=L/(DEQ*RE1);
DELTP1:=64*KI*X1+((13.766*US(X1,0.5))/US((1+13.95*KI*US(X1,0.5))+(13.76
6/KA*US(X1,1.5)),1/3));
PE1:=DELTP1*RO1*U1*U1/2;
HEADLOSS1:=(PE1/(9.81*1000));
V2:=(VG2+VG2)/2;
U2:=V2/AHALF;
RE2:=U2*DEQ/VIS2;
X2:=L/(DEQ*RE2);
DELTP2:=64*KI*X2+((13.766*US(X2,0.5))/US((1+13.95*KI*US(X2,0.5))+(13.76
6/KA*US(X2,1.5)),1/3));
PE2:=DELTP2*RO2*U2*U2/2;
HEADLOSS2:=(PE2/(9.81*1000));
{Calculation of Heat Transfer on the Rotary Heat Exchanger}
C1:=RO1*CP1*V1;
C2:=RO2*CP2*V2;
CMIN:=C2; CMAX:=C1;
CY:=CMIN/CMAX;
Z1:=L/(DEQ*RE1*PR1);
NU1:=NUS*US((1+((4.212*KI*US(ITA,3))/(Z1*US(NUS,3)))-
0.8*US(((KI*US(ITA,3))/(Z1*US(NUS,3))),2/3)),1/3);
ALFA1:=NU1*LAM1/DEQ;
Z2:=L/(DEQ*RE2*PR2);
NU2:=NUS*US((1+((4.212*KI*US(ITA,3))/(Z2*US(NUS,3)))-
0.8*US(((KI*US(ITA,3))/(Z2*US(NUS,3))),2/3)),1/3);
ALFA2:=NU2*LAM2/DEQ;
```

```
KF:=1/(1/(ALFA1*FL)+1/(ALFA2*FL));
NTU:=(KF/CMIN)/1000;
{Calculation of the Analytical Effectiveness}
EPSO:=(1-EXP(-NTU*(1-CY)))/(1-CY*EXP(-NTU*(1-CY)));
DS:=3.1415*N1/30;
CR:=MS*CPAL*DS;
CRY:=CR/CMIN;
FIR:=(0.24*CRY)/US((1+(0.02*US(CRY,1.82))+US(0.24*CRY,3)),1/3);
ANALYEFF:=EPSO*FIR;
{Calculation of the Experimental Effectiveness}
Q1:=C1*(TC1-TG1);
Q2:=C2*(TG2-TC2);
Q:=(Q1+Q2)/2;
EXPEFF1:=Q1/(CMIN*(TG2-TC1));
EXPEFF2:=Q2/(CMIN*(TG2-TG1));
EXPEFF:=Q/(CMIN*(TG2-TG1));
{Calculated values written into OUTPUT File}
write(CAL1,'N1= ',n1:6:3);
write(CAL1,'  Ntu= ',ntu:6:3);
write(CAL1,'  EPSO= ',EPSO:6:3);
write(CAL1,'  CR= ',CR:6:3);
write(CAL1,'  CRY= ',CRY:6:3);
write(CAL1,'  FIR= ',FIR:6:3);
write(CAL1,'  EFF_ANALYTIC= ',ANALYEFF:6:3);
write(CAL1,'  EFF_EXPERIMENTAL=',EXPEFF1:6:3);
READLN;
CLOSE(CAL1);
CLOSE(CAL2);
END.
```

Appendix-E

A Fortran Computer Program to calculate the numerical effectiveness of the rotary-type heat exchanger:

This program is written by Ünal (1996) for the calculation of numerical effectiveness of the counter-flow heat exchanger.

```
C     eta1 and eat2 were the numerical effectiveness and must be equal to each other.
      IMPLICIT REAL*8(A-H,O-Z)
      DIMENSION Tf(1001,101),Tw(1001,101)
      DIMENSION Tf1(1001,101),Tw1(1001,101)
      DIMENSION b(101),ad(101),bd(101)
      DIMENSION Tfm(101),Tfm1(101),Twm(101),Twm1(101)
      REAL*8 Ntu

      OPEN(4,FILE='D1',FORM='FORMATTED',STATUS='modify')
C
      M=1000
      N=100
      akat1=4./3.
      akat2=1./3.
      akat3=4./3.
      akat4=1./3.
C
      dty=1./M
      dxy=1./N
c
      NS=1             ! Calculated for NS different rotational speed
      DO 50 NN=1,NS
```

```
      WRITE(*,*) 'Input the rotational speed (rad/s)'
      read(*,*) devir
C     ================
      Ntu=3.114                  !Eqn. (108) used by Ünal (1996).
      R1=25.77
      R=R1/devir                 ! Eqn. (19) used by Ünal (1996).
      Fof1=0.000598
      Fof=Fof1/devir             ! Eqn. (21) used by Ünal (1996).
      Fow1=0.002365
      Fow=Fow1/devir             ! Eqn. (22) used by Ünal (1996).
      Cry1=6.11
      Cry=Cry1*devir             ! Eqn. (23) used by Ünal (1996).
C     Main Program
        M1=M/2
        DO 100 J=1, M+1
          DO 100 I=1, N+1
            Tf(J,I)=0.5
            Tw(J,I)=0.5
            Twm(I)=0.5
 100      CONTINUE
        DO 110 J=2, M/2
          Tf(J,1)=1.
          Tf(M/2+J,1)=0.
 110      CONTINUE
        Tf(1,1     )=1.0
        Tf(1,N+1   )=0.0
        Tf(M/2+1,1  )=1.0
        Tf(M/2+1,N+1)=0.0
        Tf(M+1,1    )=0.0
        Tf(M+1,N+1  )=1.0
```

```
      DO 120 J=1, M+1
        DO 120 I=1, N+1
          Tf1(J,I)=Tf(J,I)
          Tw1(J,I)=Tw(J,I)
120     CONTINUE
      ISAY=0
200     Tf1maxh=0.
      Tw1maxh=0.
      ISAY=ISAY+1
      ISAY1=0
      ISAY2=0
C     ----------- Calculation of Fluid Temperature -------
210     Tfmaxh=0.
      ISAY1=ISAY1+1
      DO 220 J=1, M
        ak11=4.*dxy*dxy/R
        ak12=dxy*dty
        ak13=2.*Fof*dty/R
        ak14=2.*Ntu*dty*dxy*dxy
        ak1=-(ak12+ak13)
        ak2= ak11+2*ak13+ak14
        ak3= ak12-ak13
        DO 230 I=2, N
          b(I)=(ak11-ak14)*Tf(J,I)-ak12*(Tf(J,I+1)-Tf(J,I-1))
     *         +ak13*(Tf(J,I+1)-2*Tf(J,I)+Tf(J,I-1))
     *         +ak14*(Tw(J+1,I)+Tw(J,I))
230       CONTINUE
        ad(2)=ak2
        bd(2)=b(2)-ak1*Tf(J+1,1)
        ak3d=ak3
```

```
      DO 240 I=3, N
        IF ((J.EQ.M/2).OR.(J.EQ.M)) THEN
          ad(2)=ak2+ak1*akat1
          bd(2)=b(2)
          IF (I.EQ.3) THEN
            ak3=ak3d-ak1*akat2
          ELSEIF (I.GT.3) THEN
            ak3=ak3d
          ENDIF
        ENDIF
        IF (I.LT.N) THEN
          ad(I)=ak2-ak3*ak1/ad(I-1)
          bd(I)=b(I)-bd(I-1)*ak1/ad(I-1)
        ELSEIF(I.EQ.N) THEN
          ad(I)=(ak2+akat1*ak3)-ak3*(ak1-akat2*ak3)/ad(I-1)
          bd(I)=b(I)-bd(I-1)*(ak1-akat2*ak3)/ad(I-1)
        ENDIF
240     CONTINUE
      I=N+1
250     I=I-1
        IF ((J.EQ.M/2).OR.(J.EQ.M)) THEN
          ad(2)=ak2+ak1*akat1
          bd(2)=b(2)
          IF (I.EQ.2) THEN
            ak3=ak3d-ak1*akat2
          ELSEIF (I.GT.2) THEN
            ak3=ak3d
          ENDIF
        ENDIF
      Tfji=Tf(J+1,I)
```

```
      IF (I.EQ.N) THEN
        Tf(J+1,I)=bd(I)/ad(I)
      ELSEIF (I.LT.N) THEN
        Tf(J+1,I)=(bd(I)-ak3*Tf(J+1,I+1))/ad(I)
      ENDIF
      Tfhata=abs((Tf(J+1,I)-Tfji)/Tf(J+1,I))*100
      IF(Tfhata.GT.Tfmaxh) Tfmaxh=Tfhata
      IF(I.GT.2) GOTO 250
      Tf(J+1,N+1)=akat1*Tf(J+1,N)-akat2*Tf(J+1,N-1)
      IF ((J.EQ.M/2).OR.(J.EQ.M)) THEN
        Tf(J+1,1)=akat1*Tf(J+1,2)-akat2*Tf(J+1,3)
      ENDIF
      IF (J.EQ.M/2) THEN
c         Tf(M/2+1,N+1)=0.0
        DO 260 I=1, N+1
          Tfm(I)=Tf(M/2+1,I)
260       CONTINUE
        DO 270 I=1, N+1
          Tf(M/2+1,I)=Tfm(N+2-I)
          Tw(M/2+1,I)=Twm(N+2-I)
270       CONTINUE
      ENDIF
220   CONTINUE
c     Tf(M+1,N+1)=1.0
      DO 280 I=1, N+1
        Tfm1(I)=Tf(M+1,I)
280   CONTINUE
      DO 290 I=1, N+1
        Tf(1,I)=Tfm1(N+2-I)
        Tf(M/2+1,I)=Tfm(I)
```

```
      Tw(M/2+1,I)=Twm(I)
290   CONTINUE
     IF (Tfmaxh.GT.1E-1) GOTO 210
    DO 300 J=1, M+1
      DO 300 I=1, N+1
        IF (Tf(J,I).GT.0) THEN
        Tf1hata=abs((Tf(J,I)-Tf1(J,I))/Tf(J,I))*100
        IF(Tf1hata.GT.Tf1maxh) Tf1maxh=Tf1hata
      ENDIF
      Tf1(J,I)=Tf(J,I)
300   CONTINUE
C   ----------- Calculation of Wall Temperature ---------
500   Twmaxh=0.
    ISAY2=ISAY2+1
    DO 510 J=1, M
      ak21=dty*dxy*dxy*Ntu/Cry
      ak22=2*dxy*dxy
      ak23=Fow*dty
      ak1=-ak23
      ak2= ak21+ak22+2*ak23
      ak3=-ak23
      DO 520 I=2, N
        b(I)=(ak22-ak21)*Tw(J,I)
  *         +ak23*(Tw(J,I+1)-2*Tw(J,I)+Tw(J,I-1))
  *         +ak21*(Tf(J+1,I)+Tf(J,I))
520     CONTINUE
      ad(2)=ak2+akat3*ak1
      bd(2)=b(2)
      DO 530 I=3,N
        IF (I.EQ.3) THEN
```

```
      ad(I)=ak2-(ak3-akat4*ak1)*ak1/ad(I-1)
      bd(I)=b(I)-bd(I-1)*ak1/ad(I-1)
    ELSEIF(I.GT.3.AND.I.LT.N) THEN
      ad(I)=ak2-ak3*ak1/ad(I-1)
      bd(I)=b(I)-bd(I-1)*ak1/ad(I-1)
    ELSEIF(I.EQ.N) THEN
      ad(I)=(ak2+akat3*ak3)-ak3*(ak1-akat4*ak3)/ad(I-1)
      bd(I)=b(I)-bd(I-1)*(ak1-akat4*ak3)/ad(I-1)
    ENDIF
530 CONTINUE
  I=N+1
540 I=I-1
  Twji=Tw(J+1,I)
  IF (I.EQ.N) THEN
    Tw(J+1,I)=bd(I)/ad(I)
  ELSEIF ((I.GT.2).AND.(I.LT.N)) THEN
    Tw(J+1,I)=(bd(I)-ak3*Tw(J+1,I+1))/ad(I)
  ELSEIF (I.EQ.2) THEN
    Tw(J+1,I)=(bd(I)-(ak3-akat4*ak1)*Tw(J+1,I+1))/ad(I)
  ENDIF
  Twhata=abs((Tw(J+1,I)-Twji)/Tw(J+1,I))*100
  IF(Twhata.GT.Tw1maxh) Twmaxh=Twhata
  IF(I.GT.2) GOTO 540
  Tw(J+1,1)=(akat3*Tw(J+1,2)-akat4*Tw(J+1,3))
  Tw(J+1,N+1)=akat3*Tw(J+1,N)-akat4*Tw(J+1,N-1)
  IF (J.EQ.M/2) THEN
    DO 550 I=1, N+1
      Twm(I)=Tw(M/2+1,I)
550   CONTINUE
    DO 560 I=1, N+1
```

```
          Tw(M/2+1,I)=Twm(N+2-I)
          Tf(M/2+1,I)=Tfm(N+2-I)
560       CONTINUE
        ENDIF
510     CONTINUE
      DO 570 I=1, N+1
        Twm1(I)=Tw(M+1,I)
570     CONTINUE
      DO 580 I=1, N+1
        Tw(1,I)=Twm1(N+2-I)
        Tw(M/2+1,I)=Twm(I)
        Tf(M/2+1,I)=Tfm(I)
580     CONTINUE
     IF(Twmaxh.GT.1E-1) GOTO 500
      DO 600 J=1, M+1
        DO 600 I=1, N+1
          Tw1hata=abs((Tw(J,I)-Tw1(J,I))/Tw(J,I))*100
          IF(Tw1hata.GT.Tw1maxh) Tw1maxh=Tw1hata
          Tw1(J,I)=Tw(J,I)
600     CONTINUE
      WRITE(*,610) Tf1maxh,Tw1maxh,ISAY
610    FORMAT(2D20.3,1I14)
     IF (ISAY.GE.100) GOTO 700
     IF((Tf1maxh.GT.1E-3).OR.(Tw1maxh.GT.1E-3)) GOTO 200
700   write (*,*)
     DO 710 J=1, M+1
       DO 710 I=1, N+1
         IF (Tf(J,I).LT.0) Tf(J,I)=0.
         IF (Tf(J,I).GT.1) Tf(J,I)=1.
         IF (Tw(J,I).LT.0) Tw(J,I)=0.
```

```
      IF (Tw(J,I).GT.1) Tw(J,I)=1.
710   CONTINUE
C     -------- Average of the Outlet Temperature -------------
      Tc1ort=0.
      DO 800 K=2,M1,2
      Tc1ort=Tc1ort+(dty/3)*(Tf(K-1,N+1)+4*Tf(K,N+1)+Tf(K+1,N+1))/0.5
800   CONTINUE
      Tc2ort=0.
      Tff=Tf(M/2+1,N+1)
      Tf(M/2+1,N+1)=Tf(M/2+1,1)
      DO 810 K=M1+2,M,2
      Tc2ort=Tc2ort+(dty/3)*(Tf(K-1,N+1)+4*Tf(K,N+1)+Tf(K+1,N+1))/0.5
810   CONTINUE
      Tf(M/2+1,N+1)=Tff
C     -----------------------------------------------------
      Tg1ort=1.
      Tg2ort=0.
      eta1=(Tg1ort-Tc1ort)*100
      eta2=(Tc2ort-Tg2ort)*100
      etaest=Ntu/(2+Ntu)*100
      GOTO 845
815   WRITE(*,20) Ntu
20    FORMAT(1F8.2)
      DO 820 J=1, M+1,10
        IF (J.LE.M/2+1) THEN
          WRITE(*,26) (J-1)*1./M, Tf(J,1), Tw(J,1),
     *            Tf(J,N+1), Tw(J,N+1)
        ELSEIF (J.GT.M/2+1) THEN
          WRITE(*,26) (J-1)*1./M, Tf(J,N+1),Tw(J,N+1),
     *            Tf(J,1),Tw(J,1)
```

```
      ENDIF
820  CONTINUE
     WRITE(*,21)
     WRITE(*,20) Ntu
     DO 830 I=1, N+1
       WRITE (*,23) (I-1)*1./N,
    *            Tf(0*M/8+1,I),Tw(0*M/8+1,I),
    *            Tf(1*M/8+1,I),Tw(1*M/8+1,I),
    *            Tf(2*M/8+1,I),Tw(2*M/8+1,I),
    *            Tf(3*M/8+1,I),Tw(3*M/8+1,I)
830  CONTINUE
     WRITE(*,21)
     WRITE(*,20) Ntu
     DO 840 I=1, N+1
       WRITE (*,23) (I-1)*1./N,
    *            Tf(4*M/8+1,I),Tw(4*M/8+1,I),
    *            Tf(5*M/8+1,N+2-I),Tw(5*M/8+1,N+2-I),
    *            Tf(6*M/8+1,N+2-I),Tw(6*M/8+1,N+2-I),
    *            Tf(7*M/8+1,N+2-I),Tw(7*M/8+1,N+2-I)
840  CONTINUE
     WRITE(*,21)
23   FORMAT(9F14.6)
21   FORMAT(6F14.6)
26   FORMAT(5F14.6)
845  WRITE(*,24) R,Cry,Fof,Fow,Ntu,DEVIR,eta1,eta2,etaest,isay
     WRITE(4,24) R,Cry,Fof,Fow,Ntu,DEVIR,eta1,eta2,etaest,isay
24   FORMAT(9F8.2,I15)
50   CONTINUE
     CLOSE(4)
     END
```