Erciyes Üniversitesi Fen Bilimleri Enstitüsü Dergisi **25** (1-2) 272 - 286 (2009)
http://fbe.erciyes.edu.tr/
ISSN 1012-2354

HEAT TRANSFER IN A CROSS-FLOW HEAT RECOVERY VENTILATOR WITH FIN

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ABSTRACT

In this study, a heat recovery ventilator was designed and constructed. Experimental procedure consisted of measuring of supply and exhaust air outlet temperatures. According to the results obtained, the effectiveness of heat recovery ventilator (HRV) was calculated and discussed. From the data including supply and exhaust air outlet temperatures obtained from this experimental work, the effectiveness of HRV was calculated for the several inlet flow conditions. In addition, variation of effectiveness (ε) with the number of transfer units (NTU) of heat recovery unit was calculated.
Keywords: Heat recovery ,Ventilation, Cross flow, Heat recovery ventilator (HRV).

KANATÇIKLI ÇAPRAZ AKIŞLI ISI GERİ KAZANIM VANTİLATÖRÜNDE ISI TRANSFERİ

ÖZET

Bu çalışmada, bir ısı geri kazanım sistemi tasarlandı ve imal edildi. Deneysel işlem egzoz ve taze hava sıcaklıklarının ölçümlerinden oluşmaktadır. Elde edilen sonuçlara göre ısı geri kazanım ventilatörünün etkenliği hesaplandı ve tartışıldı. Bu deneysel çalışmadan toplanan egzoz ve taze hava akışının çıkış sıcaklık verileri ve HRV nin etkenliği birçok giriş akış şartları için hesaplandı. İlave olarak etkenliğin ısı geri kazanım sisteminin NTU değeri ile değişimi hesaplandı.
Anahtar Kelimeler: Isı geri kazanımı, Vantilatör, Çapraz akış, HRV.

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1. INTRODUCTION

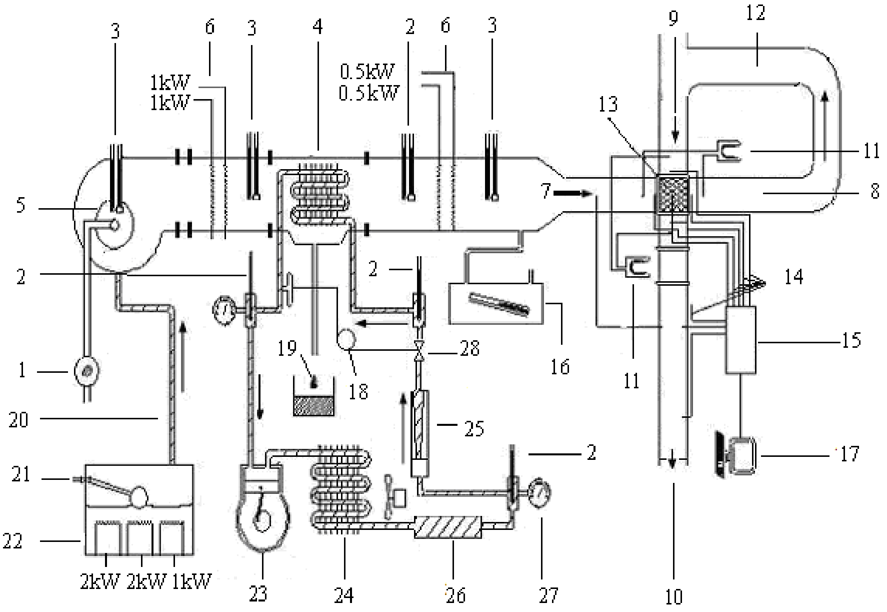
Generally, heat is transferred by means of heat exchangers. As known, there are several types of heat exchangers for different applications according to many kinds of sizes, weights, shapes, flow patterns, etc. respectively. Cross flow heat exchangers are generally used in air/gas heating and cooling systems and can also be used for heat recovery and ventilation systems. The improvements in design of these heat exchangers have attracted many researchers for a long time to develop more compact and less expensive heat exchangers with high-energy performance. The need for high performance thermal systems has led researchers to use different augmentation methods of heat transfer [1, 2, 3, 4]. The relationship between the effectiveness and number of transfer units (ε-NTU) of the cross flow heat exchanger were determined [5]. Bergles reported that [6] heat transfer enhancement at heat exchangers may be achieved by numerous techniques, and these techniques can be classified into three groups: passive, active and compound techniques. In the active techniques, heat transfer is improved by giving additional flow energy into the fluid. In the passive techniques, however, this improvement is acquired without giving any extra flow energy. In the compound techniques, two or more of the active or passive techniques may be utilized simultaneously to produce an enhancement that is much higher than the techniques operating separately. The effects of supply and exhaust flow rate on the static pressure losses across the heat exchanger core for both supply flow and exhaust flow were experimentally investigated. Finally, for all there configurations of HRV, overall heat transfer coefficient was calculated and discussed [7]. The optimization of heat exchanger area for heat recovery by mixing and, so eliminating, all these thermal and economical parameters except the surface area, depending on the certainty of the operating characteristics of the applications and the most efficient operating condition of the heat exchanger studied [8].

In this study, a heat recovery ventilation operating with unmixed fluids was studied in order to examine the effects of the cross-flow plate type and a recovery system was manufactured in the laboratory. The main purpose of heat recovery ventilation (HRV) systems is to reclaim waste energy from the exhaust air stream and use that energy to heat the incoming fresh air.

2. EXPERIMENTAL APPARATUS AND PROCEDURE

A schematic diagram of the experimental apparatus is shown in Figure1. Experiments were carried out at dimensions $L_a/L_c = 1$ (see Figure 2). The experimental set-up mainly consists of one air filter, one supply fan unit, cross flow compact heat exchanger core, hydrodynamic entrance section, flexible pipe, orifice plate, pressure measurement units, inclined tube alcohol micro manometer, thermocouples, AC574 air conditioning laboratory unit for air supply and data acquisition system. Supply and exhaust air were used as working fluid. The test section and the connections of the piping system were designed in such a way that parts can be changed or repaired easily. The experimental apparatus was operated in the blow out-mode. The test section was constructed with stainless steel plates. The outer surface of the test section was insulated with a layer of glass wool to inhibit heat loss and to obtain a uniform heat flux. In order to obtain a linear velocity distribution in the channels, wire sieves have been placed between the test section and the outlet of the heaters. For the flow and heat transfer tests, surface temperatures, inlet and outlet temperatures were measured. The heat exchanger matrix was manufactured in the form of convergent–divergent vortex generators as shown in Figure 2. The angle of winglet in the flow direction of exhaust fluid is β_h=30°, and the angle of winglet in the flow direction of supply fluid is β_c=60°. The trailing edge of the winglet was located at a distance of x = 5 mm from the inlet. The experimental set up was manufactured of stainless steel plates (t_w =1.5 mm thick) and the dimensions of the heat exchanger is 200x200 mm. Height (b) and length (l) of each channel is 10 mm.

In Figure 2, exhaust flow, supply flow and form of the winglets permutation are shown on the plate of cross flow heat exchanger. There is a region situated between diverging and converging channels (regions between A and B, in Figure 3) where the velocity is too low to enable the resolution of the flow direction. Due to pressure rise at the end of the diverging channel (region A), and pressure drop at the neighboring converging channel (region B), where the velocities are high enough to cause turbulence were investigated [9, 10]. This region is believed to be the mixing zone caused by the slit between divergent and convergent channel pairs.



1- Fan- speed controller	2- Thermometer	3-Thermometer(wet-dry bulb)	4-Evaporator
5- Fan	6- Air Heaters	7- Supply Flow in	8- Supply Flow out
9- Exhaust Flow out	10-Exhaust Flow in	11- U-Manometer	12- Pipe Line
13- Test Section	14- Thermo Couples	15-Data Acquisition Card	16- Inclined Manometer
17- Computer	18- Tank	19- Condense	20- Vapor Connection
21- Main Water Inlet	22- Water Heater	23- Compressor	24- Condenser
25- Flow Meter	26-Drier	27-Pressure Indicator	28-Thermostatic Exp.Valve

Figure 1. Schematic display of the experimental set-up.

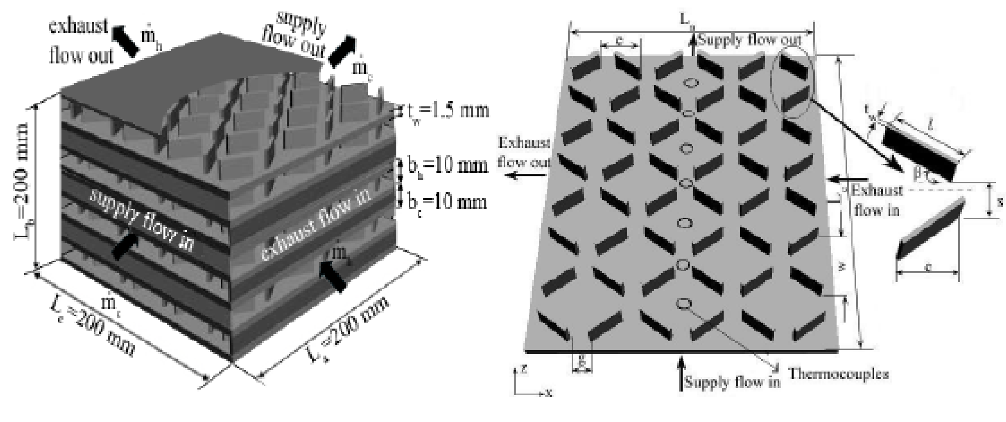


Figure 2. (a) The geometric features of the cross flow heat recovery ventilator matrix

(b) Top view of geometry and placement of winglets in rectangular channel.

As the convergent–divergent longitudinal fins act as plates in the flowing fluid, each new edge starts new boundary layer which is very thin, thus high heat transfer coefficients can be obtained. While the exhaust flow is passing through one direction of the channels, the supply flow is passing through the other direction. Heat is transferred from the exhaust air to the supply air during this process. The flow rate of the exhaust air flow was controlled by adjusting the clack valve, and measured by the manometers with an accuracy of ±0.3%. Pressure taps for measuring the pressure losses were mounted across the orifice plates at the inlet and outlet ends of the test section. Supply air flow rates were measured via flow meters with an accuracy of ±0.22% of full scale. The supply air flow rate was adjusted by a digital fan speed controller, and conditions were measured by wet and dry bulb thermometers. Thermocouples were mounted at different locations on the surface test section. In order to determine the effectiveness of the heat exchanger temperature and air velocity of fluids in the test section were measured with thermocouples and pressure taps continuously. Temperatures were measured with 0.25mm diameter Teflon coated type T copper-constantan thermocouples. Thus, the measurements were performed at the four locations in the same cross-section and they were also calculated. Accuracy of the thermocouples was ±0.15% of full scale. All the thermocouples and pressure sensors were fully calibrated with a dry box temperature calibrator, with 0.01°C precision. All flow properties were determined at the average bulk temperature. National Instrument Multifunction DAQ Board, AT-MIO 16 of 100 kS/s sampling rate and Lab View software version 5 were used to record the temperatures. The effect of thermal radiation for internal flow was ignored during the experiments due to low temperature differences between wall and winglets.

The experimental work mainly consists of temperature measurements of supply and exhaust air flows at the inlet and outlet. Effectiveness of the heat recovery unit was determined for various inlet flow conditions. Experiments were performed for various temperature and flow rates of the exhaust and supply flow. The system was allowed to reach the steady state condition before any data was recorded. A data acquisition system having a capacity of 40 channels was used for collecting the data. All measurements were collected, processed, stored, analyzed and displayed by a PC through the data acquisition card and software.

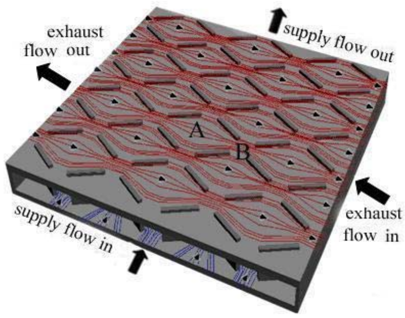


Figure 3. Sketch showing the upper part of the rectangular fins with stream lines.

2.1. Air Conditioning Laboratory Unit AC574

The conditioning of the air takes place in a glass fibre duct 200mm^2 in section, mounted on a steel frame with castors. Air is drawn into the duct by a variable speed centrifugal fan and wet and dry thermometers are placed at the inlet. Air is preheated by two 1kW finned electrical heaters. Next the air flows through the evaporator of the refrigeration plant where it is cooled de-humidified. Two 0.5kW finned electrical heating coils are used for reheating. Seven heaters (two preheaters, two reheaters, three boilers) are individually switched. The steam of humidification is provided by an atmospheric boiler with three heating elements that can be switched to create various rates of steam production. Boiler in the test rig is a humidifier as shown in the Figure 1. The water level is controlled by a float valve and observed through a sight glass. The refrigerant, namely R134a, flows, from the evaporator into the hermetic compressor where it is discharged into an air-cooled condenser. Table 1 was obtained for different heating capacities of preheater, reheater, boilers.

Table 1. Cases according to different volume flow rates, preheater and reheater conditions

Cases	Boiler Position	Fan position	Preheater (kW)	Reheater (kW)
Case-1	Closed	Closed	1 kW	0.5 kW
Case-2	Closed	Fan open	1 kW	Closed
Case-3	Closed	Fan open	2 kW	Closed
Case-4	Open	Fan open	Closed	0.5 kW
Case-5	Open	Fan open	Closed	1 kW

3. DATA REDUCTION

The goal of this experimental study is to determine the effectiveness of the winglets inserted into plates in terms of parameters such as heat transfer. These parameters were analyzed for fully developed flow.

The outlet temperatures and the total heat transfer from the exhaust fluid to the supply fluid can be calculated by means of the $\varepsilon - NTU$ method based on the second law of thermodynamics when the inlet temperatures of the exhaust and the supply fluid, volume flow rates of the fluids, physical properties of fluids and the type of the heat exchanger are specified. The heat exchanger effectiveness ε can be written as,

$$\varepsilon = \frac{Q_a}{Q_{\max}} \tag{1}$$

Maximum heat transfer occurs at the surface temperature of the air inlet in case of discharging in the inner channel of heat exchanger. According to this statement, for the maximum heat transfer the following equation can be expressed as follows

$$Q_{\max} = (\dot{m} c_p)_{\min} (T_{h,i} - T_{c,i}) = C_{\min} (T_{h,i} - T_{c,i}) \tag{2}$$

The symbols *h* and *c* refer to the exhaust and the supply fluids. The actual heat transfer rate Q_a is defined as

$$Q_a = (\dot{m} c_p)_h (T_{h,i} - T_{h,o}) = (\dot{m} c_p)_c (T_{c,o} - T_{c,i}) \tag{3}$$

where defined $(\dot{m} c_p)_h = C_{hot}$, $(\dot{m} c_p)_c = C_{cold}$ are the heat capacities of exhaust and supply flows, and $C_{\min}$ is equal to the smallest value of C_{hot} and C_{cold}. Here $C_{\min}$ means the smaller heat capacity for hot and cold fluids. Effectiveness for $C^* = 1$ can be rewritten again from Eqs. (2) and (3) as

$$\varepsilon = \frac{(T_{h,i} - T_{h,o})}{(T_{h,i} - T_{c,i})} = \frac{(T_{c,o} - T_{c,i})}{(T_{h,i} - T_{c,i})} \tag{4}$$

From equation (4) the outlet temperatures of both fluids can be calculated by using the effectiveness and the inlet temperatures of both fluids. The number of heat transfer units, NTU, is expressed in terms of thermal capacity,

$$NTU = \frac{UA_{heat}}{C_{\min}} \tag{5}$$

The actual heat transfer area and the overall heat transfer coefficient are denoted as A_{heat} and U**,** respectively. In the case in which both flows are unmixed, effectiveness of the cross-flow heat exchangers depends on the C*= (Cmin / Cmax) and NTU parameters are given as

$$\varepsilon = 1 - \exp\left[\left(\frac{1}{C^*} \right) NTU^{0,22} \left\{ \exp\left[-C^* (NTU)^{0,78} \right] - 1 \right\} \right] \tag{6}$$

4. RESULTS and DISCUSSION

In order to calculate the heat transfer effectiveness of the heat recovery ventilator, the temperatures of the supply flow inlet, supply flow exit and exhaust flow exit must be measured. The supply flow exit and the exhaust flow exit temperatures are measured for changing the exhaust flow rates at the each supply flow inlet condition. For this part, supply flow air was heated by 2kW. Figures 4, 5, 6, and 7 show the effect of the exhaust volume flow rate on the supply and exhaust flow exit temperatures for constant supply volume flow rate and constant supply flow inlet temperature. As can be seen from Figures 4, 5, 6 and 7, the exhaust and supply flow exit temperatures decreased with the increase of exhaust flow volume flow rate. Figures 4, 5, 6 and 7 also show that the exhaust and supply flow exit temperatures decreased as the supply volume flow rate increased for a constant exhaust volume flow rate. But, for the case of supply flow of 0,017m^3/s, supply flow exit temperature depicts the irregular behaviour when it compared for the case of supply flow of 0,014 m^3/s.

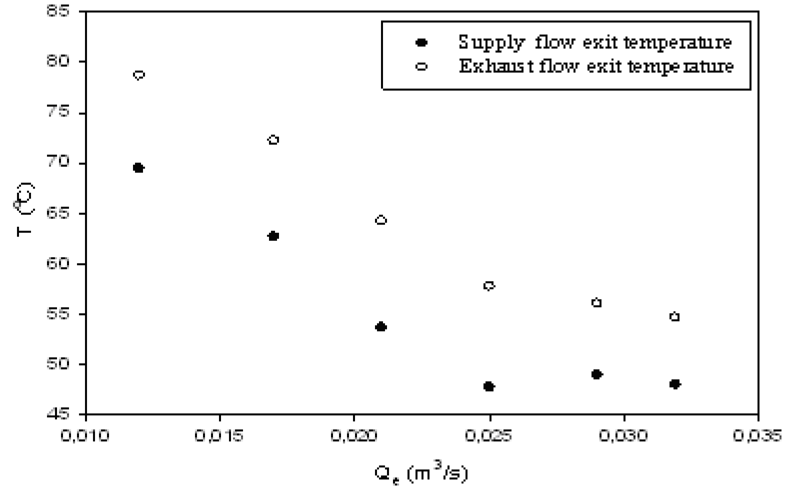


Figure 4. Variation of the exhaust and supply flow exit temperatures (Q_s=0,012 m^3/s, heating 2 kW open).

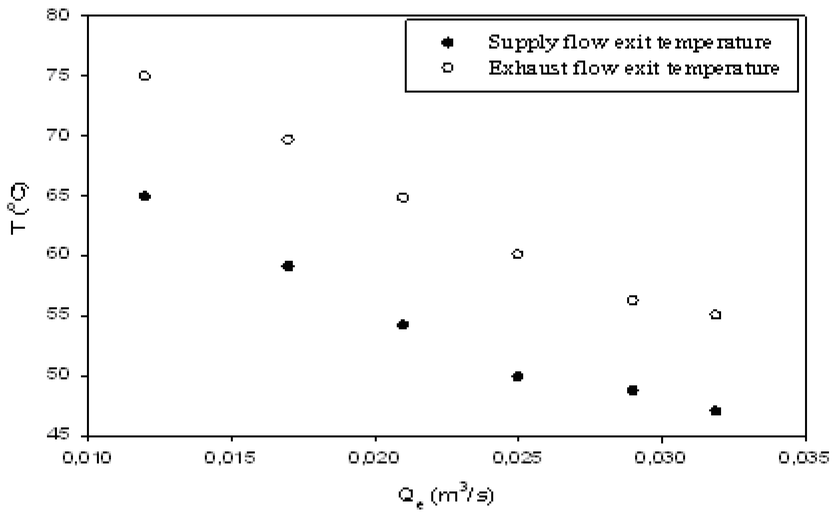


Figure 5. Variation of the exhaust and supply flow exit temperatures (Q_s=0,014 m^3/s, heating 2 kW open)

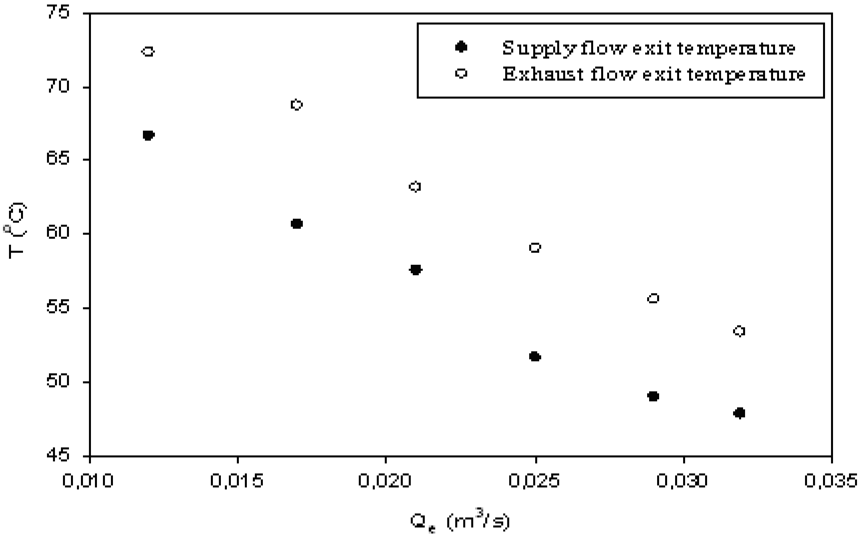


Figure 6. Variation of the exhaust and supply flow exit temperatures (Q_s=0,017 m^3/s, heating 2 kW open)

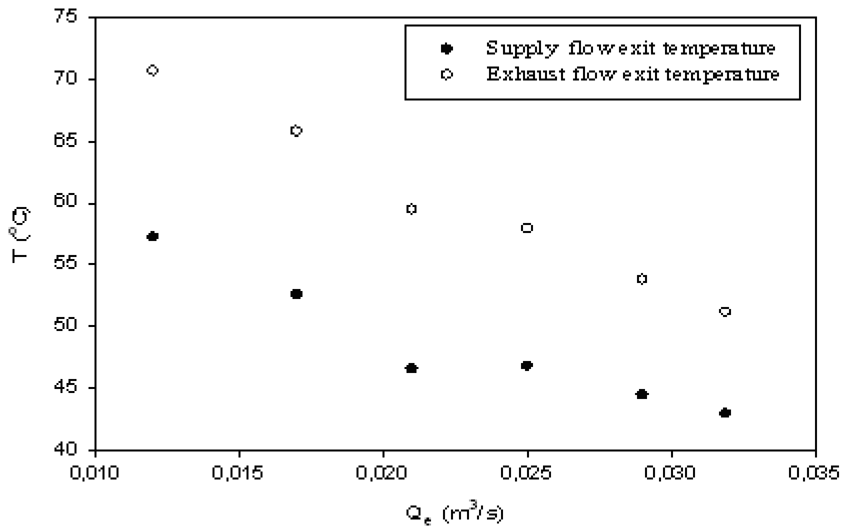


Figure 7. Variation of the exhaust and supply flow exit temperatures (Q_s=0,022 m^3/s, heating 2 kW open)

Figures 8, 9 and 10 show the variation of the heat exchanger effectiveness as a function of the exhaust flow volume flow rates for 1kW, 2kW and 3 kW heater capacity at the diffrerent supply flow volume flow rates (Q_s=0,014 m^3/s, Q_s=0,017 m^3/s, Q_s=0,020 m^3/s). As can be seen Figures 8, 9 and 10 the effectiveness increased with the increase of exhaust flow volume flow rate. It is clearly seen from these there figures that the effectiveness for 3 kW heater power is higher than that of 1kW and 2kW for the same exhaust flow volume flow rates. In addition, the best performance was found for the operating condition of Q_s=0,014m^3/s and 3 kW heating power.

The concept of Number of Transfer Units (NTU) is a well-known method for measuring heat exchanger effectiveness. Notice that the NTU value is a function of the type of heat exchanger as well as its physical size. Figure 11 shows the changes of the effectiveness with NTU for cross flow heat recovery unit. It is observed from Figure 11 that the effectiveness increased with the increase of NTU.

In the present study for NTU=1.55, the effectiveness change between $\varepsilon = 0.63$ with 0.78 for five different cases as it is seen from Figure 11. The experimental effectiveness values for balanced cross-flow arrangement (C* = 0.75) are more suitable to compare the results with theoretical values and the literature.

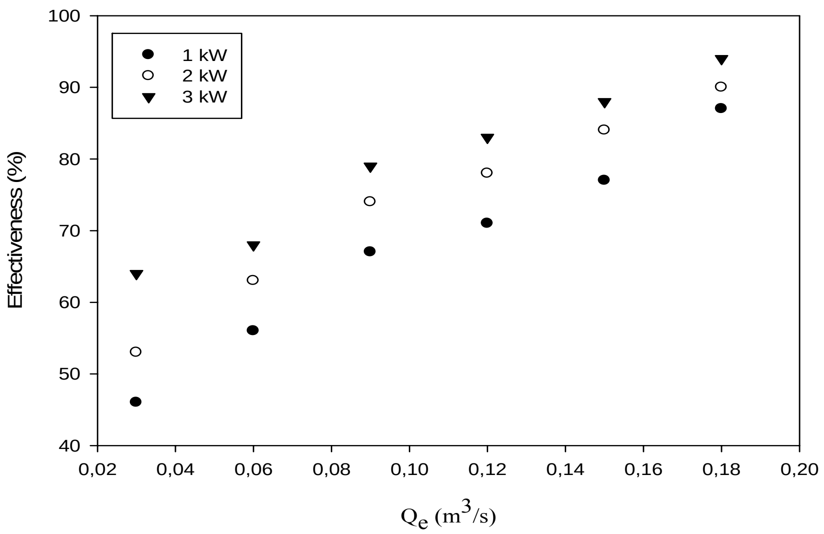


Figure 8. Variation of heat exchanger effectiveness for different heater power (Q_s=0,014m^3/s)

While the boiler (case-4) is open at the same Reynolds numbers, effectiveness decreased about 1.19 times with NTU. While the boiler (case-3) is closed at the same Reynolds number, effectiveness increased with NTU.

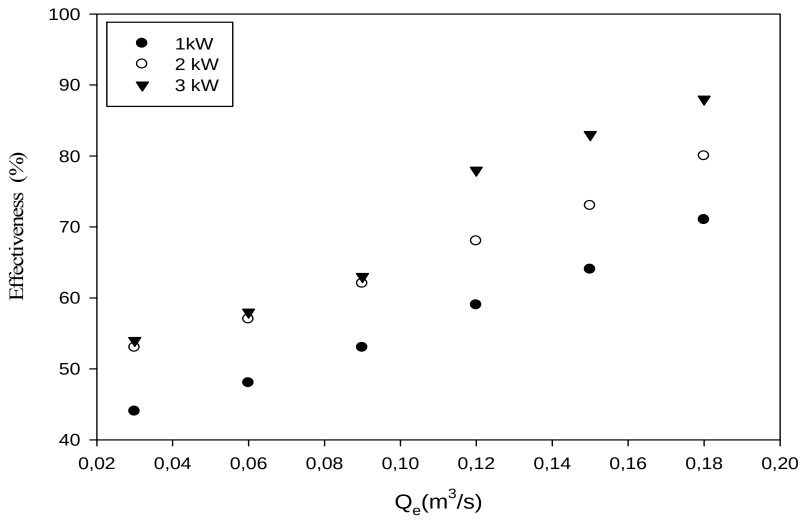


Figure 9. Variation of the heat exchanger effectiveness for different heater power (Q_s=0,017 m^3/s)

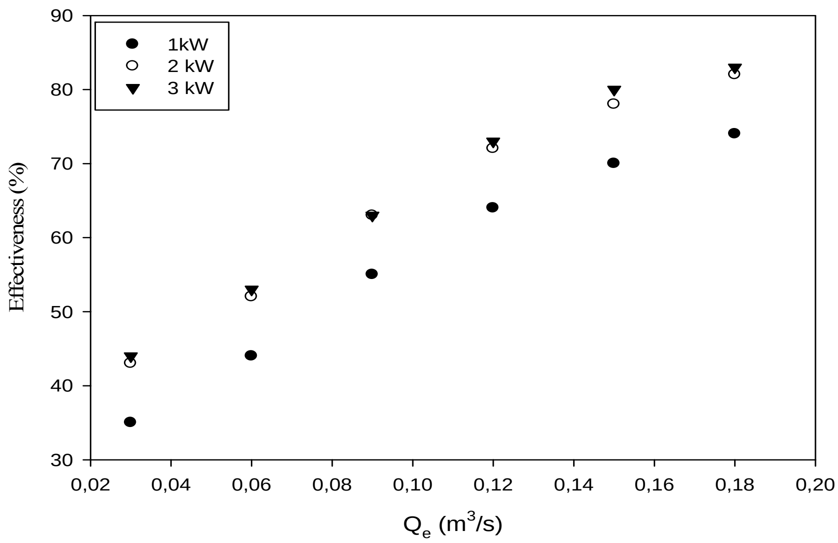


Figure 10. Variation of the heat exchanger effectiveness for different heater power (Q_s=0,020 m^3/s)

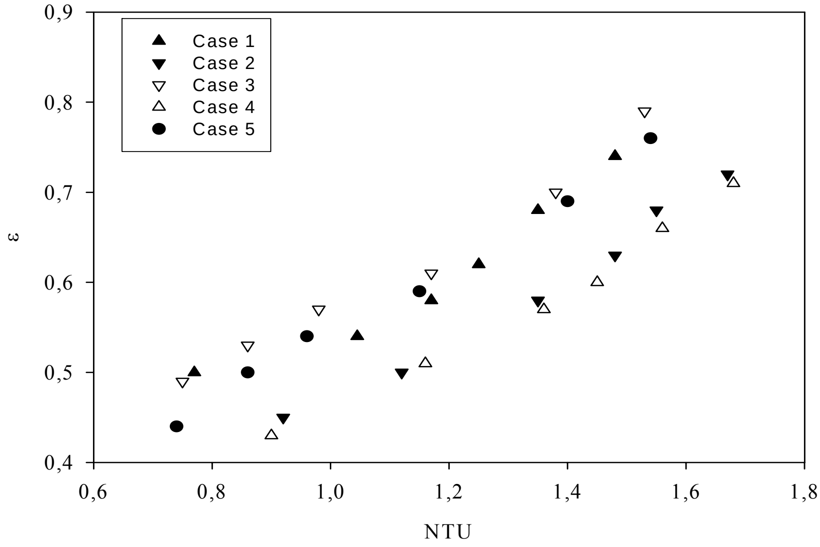


Figure 11. Variation of effectiveness with *NTU* of the cross flow heat exchanger, both fluids unmixed.

5. CONCLUSIONS

A comprehensive experimental study has been carried out on the convergent and divergent plate type fins located in a cross-flow heat exchanger with various volume

flow rates. Variation of heat transfer, and effectiveness with NTU were also investigated.

Converging and diverging rectangular plate fins were used. The following conclusions can be derived:

- The supply flow exit and the exhaust flow exit temperatures were measured for changing the exhaust flow rates at the each supply flow inlet condition. For this part, supply flow air was heated by 2kW. Changes in the exhaust and supply flow exit temperatures with the exhaust flow volume flow rate are depicted in Figs. 4, 5, 6 and 7. As can be seen, the exhaust and supply flow exit temperatures decrease with the increase of the exhaust flow volume flow rate. The highest heat transfer rate was observed for the volume flow rate of

Q_s=0,012 m^3/s and heating capacity of 2 kW. The effect of the secondary flow was expected to be in the intermediate region between wing cascades, which is due to pressure and velocity differences across the passage between the converging and diverging pair of winglets.

- ε-NTU technique was used to evaluate the performance of the plate-fin array system. As a heat exchanger is used with small values of NTU a large size can be justified economically. According to the results obtained by using Eq.5, number of transfer units (NTU) is varies between 0.75 and 1.69. For this reason, the experimental effectiveness values for balanced cross-flow arrangement (C* = 0.75) are very compatible with for the theoretical values. Thus, a heat exchanger with a very high effectiveness may be desirable from a heat transfer, exergy and economical point of views.
- Because the CDLVGs plate fin profile is formed as nozzle and diffuser, renewing of boundary layer in this winglet type plays an important role on effectiveness of heat exchanger. The velocity was affected due to pressure variations in secondary flow regions between converging and diverging slices. The influence of velocity causes turbulence for the whole flow. The main reason for turbulence can be explained by two behaviors of pressure condition, the rise at the end of diverging channel and the drop at the converging slices. The heat recovery efficiency decreased with increasing air velocity. It was also found that the pressure loss reduces as velocity increases, but the reduction is small.

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