DOKUZ EYLUL UNIVERSITY
GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES

NUMERICAL AND EXPERIMENTAL INVESTIGATIONS OF HEAT TRANSFER IN A CROSS-FLOW HEAT RECOVERY SYSTEM

by
Bengisu KORKMAZ

November, 2013
İZMİR

NUMERICAL AND EXPERIMENTAL INVESTIGATIONS OF HEAT TRANSFER IN A CROSS-FLOW HEAT RECOVERY SYSTEM

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by
Bengisu KORKMAZ

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M.Sc THESIS EXAMINATION RESULT FORM

We have read the thesis entitled **"NUMERICAL AND EXPERIMENTAL INVESTIGATIONS OF HEAT TRANSFER IN A CROSS-FLOW HEAT RECOVERY SYSTEM"** completed by **BENGİSU KORKMAZ** under supervision of **PROF. DR. DİLEK KUMLUTAŞ** and we certify that in our opinion it is fully adequate, in scope and in quality, as a thesis for the degree of Master of Science.

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ABSTRACT

Nowadays, ventilation is obligatory in buildings where human beings spend a large part of their life. To maintain indoor air quality, dirty indoor air is exhausted and fresh air is taken from outdoors.

In modern systems, dirty indoor air with high energy is not exhausted directly. By the help of air to air heat exchangers, fresh air from outdoors is pre-heated or pre cooled by the help of exhausted air thus decreases the energy used to condition indoor air. In HVAC Systems, between 35 percent and 90 percent ratio of waste heat energy recovery is possible. Plate type, rotary, heat pipe and coil heat exchangers are widely used as heat recovery exchangers.

In this research, the efficiency of heat recovery exchanger was researched numerically. The dimensions of the exchanger which was designed by a manufacturer company are 600x600x600mm. This heat exchanger was investigated for the real boundary conditions and turbulent flow with the help of the program Ansys Workbench. According to fin characteristics and flow patterns, created models were analyzed.

Heat exchanger which developed by manufacturer company was experimented on testing laboratory which was founded by the company. The results of the tests and theoretical research are shown compatibility of numerical analysis. As the results are compatible with each other, the finite element method for heat recovery efficiency of the exchanger was found to be a reliable method.

Keywords: Heat recovery ventilator, CFD, cross-flow, HRV.

ÇAPRAZ AKIŞLI ISI GERİ KAZANIM SİSTEMİNDE ISI TRANSFERİNİN SAYISAL VE DENEYSEL OLARAK İNCELENMESİ

ÖZ

Günümüzde insanların hayatlarının büyük kısmını geçirdiği kapalı mekanlarda havalandırma yapılması zorunlu hale gelmiştir. İç hava kalitesini koruyabilmek için iç ortamdan emilen atık hava dışarı atılarak, şartlandırılmış taze hava ortama verilmektedir.

Modern sistemlerde ise iç ortamdan alınan yüksek enerjili hava direkt dışarı atılmamakta, atık havanın enerjisinden yararlanılarak taze havanın ön ısıtması ya da ön soğutması gerçekleştirilmektedir. HVAC sistemlerinde atık ısının yüzde 35 ile yüzde 90 oranları arasındaki kısmının geri kazanılması mümkündür. Plakalı, rotorlu, ısı borulu ve serpantinli eşanjörler, yaygın olarak kullanılan ısı geri kazanım eşanjörleridir.

Bu çalışmada, üretici bir firmadan bilgileri alınan 600x600x600 mm boyutlarındaki ısı geri kazanım eşanjörünün verimi teorik, sayısal ve deneysel olarak araştırılmıştır. Bu eşanjör gerçek sınır şartları ve türbülanslı akış için Ansys Workbench programı yardımıyla incelenmiştir. Farklı akış şekilleri ve kanat dizaynlarına göre oluşturulan ısı geri kazanım eşanjörleri numerik olarak analiz edilmiştir.

Üretici bir firmanın geliştirdiği eşanjör, firma tarafından kurulmuş olan test düzeneğinde aynı sınır şartları için deneye tabii tutulmuştur. Deney sonuçlarının ve teorik sonuçların sayısal çalışmayla uyumluluğu tespit edilmiştir. Sonuçlar birbirine uyumlu olduğundan, ısı geri kazanım cihazı verimliliğinin bulunmasında sonlu elemanlar yönteminin güvenilir bir yol olduğu görülmüştür.

Anahtar Kelimeler: Isı geri kazanımlı havalandırma cihazı, CFD, çapraz-akış, IGK.

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CHAPTER ONE
INTRODUCTION

Ventilation is a necessity for the health and comfort of occupants of all buildings. Ventilation supplies air for occupants to breathe and removes moisture, odors, and indoor pollutants like carbon dioxide. In modern buildings the design, installation and control systems of these functions are integrated into one or more HVAC systems.

A heat-recovery ventilator (HRV) is similar to a balanced ventilation system, except it uses the heat in the outgoing stale air to warm up the fresh air. A typical unit features two fans—one to take out household air and the other to bring in fresh air.

Kocaoglu, Caliskan, & Zırzakıran (2009), made experiments to measure the effectiveness of the cross-flow heat recovery ventilator. The cross flow heat exchanger with fin was designed by them. Supply and exhaust air temperatures are measured. For several inlet flow conditions, the effectiveness of HRV (heat recovery ventilator) was calculated. In addition, variation of effectiveness (ε) with the number of transfer units (NTU) of heat recovery unit was compared. According to these experiments, 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.

Zhang, & Jiang (2007), researched heat and mass transfer model for a energy recovery ventilator (ERV) with porous hydrophilic membrane one. The coupled heat and mass transfer mechanisms for the system are discussed. Through finite difference simulations, the temperature and humidity in the unit are calculated. It is found that for the present cross-flow arrangement, the membrane area is not effectively used either for sensible heat or for moisture transfer. With counter flow arrangement, the sensible effectiveness, the moisture effectiveness, and the enthalpy effectiveness can be even higher. The temperature difference between two surfaces of the membrane can be neglected. The distribution of the contours of humidity difference affects the vapor osmosis rate through the membrane. The permeability is very high near the inlets of two fluids but relatively lower at other sites. This suggests that there is a ridge in the direction of left-lower to upper-right where permeability is relatively higher.

Another investigation was made by Jesper, Jorgen, & Svend (2005). They research mechanical ventilation with heat recovery in cold climates. Based on measurements and calculations of existing building ventilation systems with heat recovery, it could be concluded that there is a need for developing new heat recovery systems with both high temperature efficiency and protection against ice formation, if the systems were to be used effectively in cold climates. Measurements from typical single-family houses and on a heat exchanger in the laboratory had shown that ice formation occurs when the outside temperature reaches a level below –5°C for a short period of time. Measurements also showed that the inlet temperature at times reaches levels below 15°C, which means that draught can occur.

Demir, Atayılmaz, Ağra, & Teke (2010), researched for the best type of the heat exchangers for heat recovery. In this study a new model has been developed for determining the area and type of the most appropriate waste heat recovery heat exchanger for maximum net gain. A non-dimensional E number has been defined based on known technical and economic parameters such as the life-time, unit area cost of the heat exchanger, lower heating value of the fuel, overall heat transfer coefficient of heat exchanger, boiler efficiency, operation time per year, heat exchanger effectiveness, ratio of heat capacities, annual variation of the temperature of fluids supplied to the heat exchanger and present worth factor. Parallel flows, counter flow, cross flow (C_{max} mixed, C_{min} unmixed), shell and tube, cross flow (C_{max} unmixed, C_{min} mixed) and cross flow (both fluid unmixed) are compared according to their efficiency. According to the calculations, counter flow heat recovery exchanger was given maximum net gain.

In literature survey, theoretical and experimental researches for heat exchangers were examined. None of these articles have numerical investigations about heat recovery exchangers that are used for HVAC applications. In this thesis, the heat recovery exchanger was investigated experientially, theoretically and also numerically and these results were compared with each other. The heat exchanger design data was given by manufacturer company. In addition, heat exchangers were developed according to different flow patterns. By comparing efficiency values, the most efficient heat recovery exchanger was determined.

CHAPTER TWO
CLASSIFICATION OF HEAT EXCHANGERS

Heat exchangers are devices that provide the flow of thermal energy between two or more fluid at different temperatures. Heat exchangers are used in various applications. In power production, industrial processes, air-conditioning and refrigeration, heat exchangers are used extensively. In Figure 2.1, classification of heat exchangers according to 5 main criteria is shown (Kakac, 2002).

1. Recuperators and regenerators
2. Transfer processes: direct contact and indirect contact
3. Geometry of construction: tubes, plates, and extended surfaces
4. Heat transfer mechanisms
5. Flow arrangements: parallel, counter, and cross flows

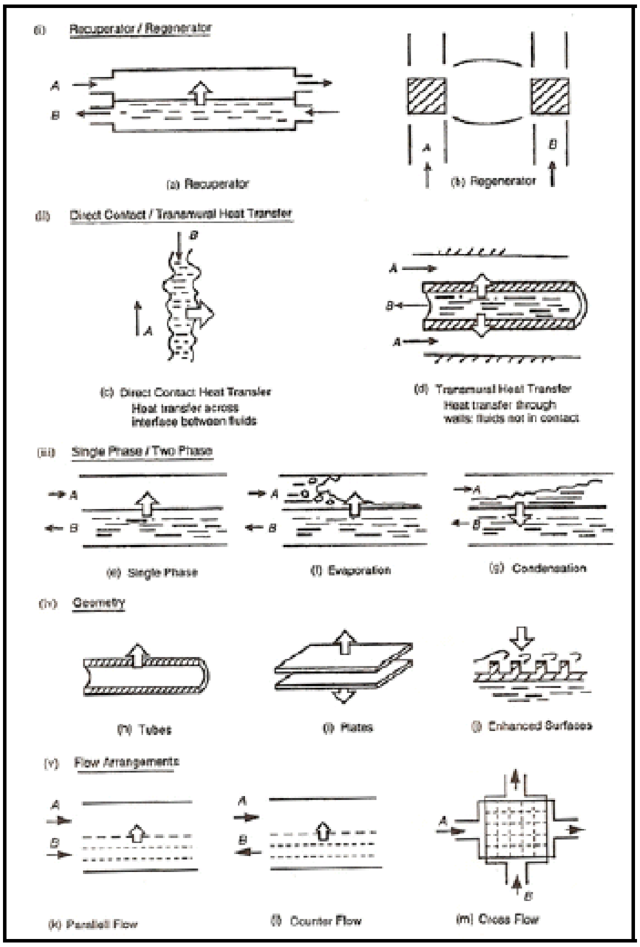


Figure 2.1 Classification of heat exchangers (Kakac, 2002)

2.1 Recuperation and Regeneration

A conventional heat exchanger, shown diagrammatically in Figure 2.1a, with heat transfer between two fluids, is called as recuperator, because hot stream A recovers (recuperates) some of the heat from stream B. The heat transfer occurs through a separating wall or through the interface between the streams as in the case of the direct contact type heat exchangers. (Figure 2.1c) Some of the recuperative-type exchangers are shown in Figure 2.2

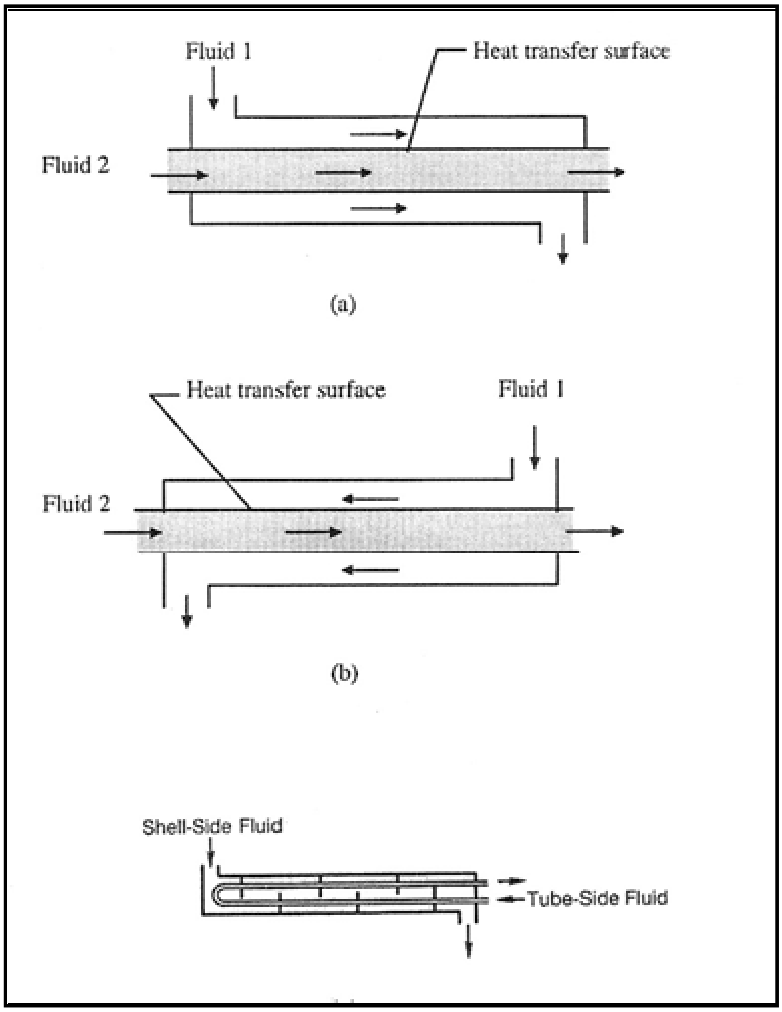


Figure 2.2 Indirect contact types of heat exchangers. (a), (b) Double-pipe type, (c) shell and tube type (Kakac, 2002)

In regenerators or in storage type heat exchangers, the same flow passage (matrix) is alternately occupied by one of the two fluids. The hot fluid stores the thermal energy in the matrix, during the cold fluid flow through the same passage at a later time, stored energy will be extracted from the matrix. Therefore, thermal energy is not transferred through the wall as in direct transfer type heat exchanger.(Figure 2.1b)

Regenerators can be classified as follows:

1. Rotary regenerator
   (a) Disk-type
   (b) Drum-type
2. Fixed-matrix regenerator

The disk-type and drum-type regenerators are shown in Figure 2.3, schematically. The heat transfer surface is in a disk form and fluids flow axially in disk-type regenerators. In drum-type regenerators, the matrix is in a hollow drum form and fluids flow radially.

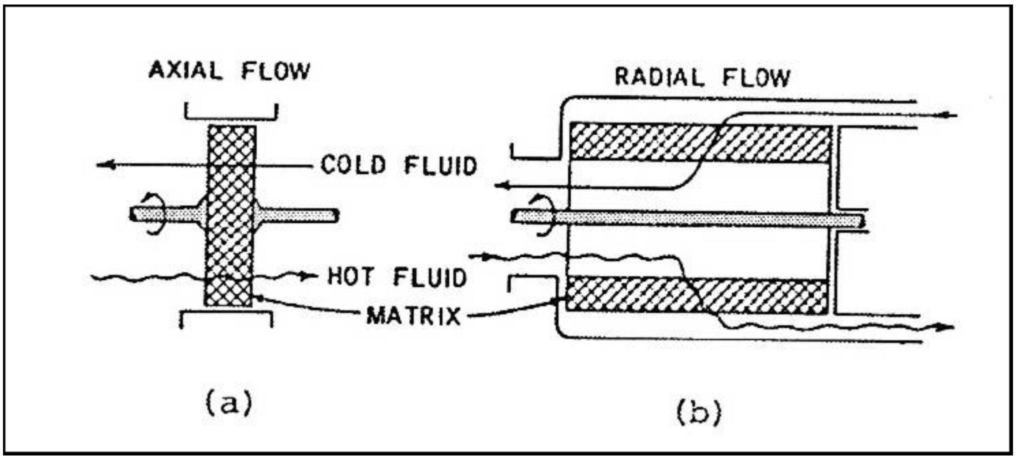


Figure 2.3 Rotary regenerators. (a) Disk type. (b) Drum type

These regenerators are periodic flow heat exchangers. In rotary regenerators, the operation is continuous. In order to achieve this, the matrix moves periodically in and out of the fixed stream of gases. A rotary regenerator for air heating is illustrated in Figure 2.4

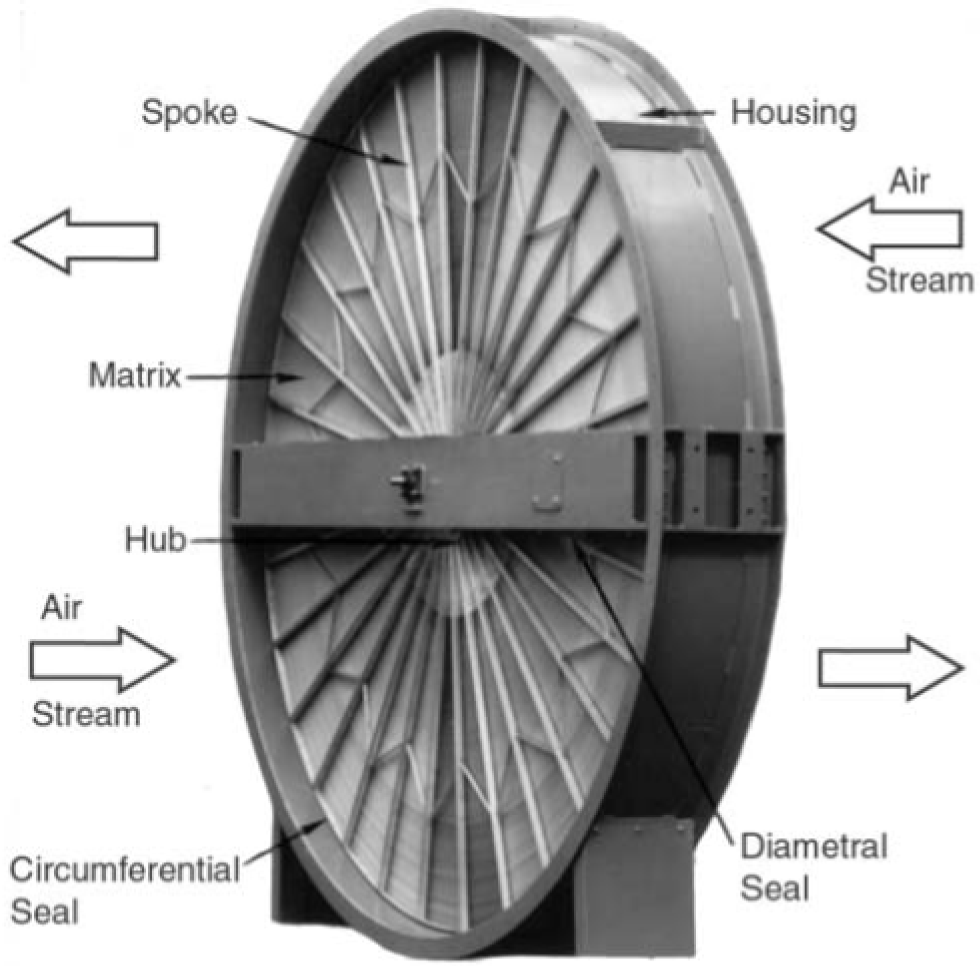


Figure 2.4 Rotary heat wheel or a rotary regenerator made from a polyester film (Shah, 2003)

2.2 Transfer Processes

Heat exchangers are classified as direct contact type and indirect contact type (transmural heat transfer) according to transfer processes

In direct contact type heat exchangers, heat is transferred through a direct contact between the cold and hot fluids. As shown in Figure 2.1c, since there is no wall between hot and cold streams, the heat transfer occurs through the interface of two streams. The streams are two immiscible liquids, a gas-liquid pair, or a solid particle fluid combination in direct contact type heat exchangers. Cooling towers spray and tray condensers are good examples of such heat exchangers.

Indirect contact and direct contact type heat exchangers are also called recuperators. Tubular (double-pipe, shell and tube), plate, and extended surface heat exchangers; cooling towers; and tray condensers are examples of recuperators.

2.3 Geometry of Construction

Direct transfer type heat exchangers (transmural heat exchanger) are often described in terms of their construction features. The major construction types are tubular, plate and extended surface heat exchangers.

2.3.1 Tubular Heat Exchangers

Tubular heat exchangers are built of circular tubes. One fluid flows inside the tubes and the other flows on the outside of the tubes. Tube diameter, the number of the tubes, the tube length, the pitch of the tubes, and the tube arrangements can be changed. Therefore, there is considerable flexibility in their design.

Tubular heat exchangers can be further classified as follows:

1 Double-pipe heat exchangers

2 Shell-and-tube heat exchangers

3 Spiral tube type heat exchangers

2.3.1.1 Double-pipe Heat Exchangers

A typical double-pipe heat exchanger consists of one pipe placed concentrically inside another of larger diameter with appropriate fittings to direct the flow from one section to the next, as shown in Figure 2.5. Double-pipe heat exchangers can be arranged in various series and parallel arrangements to meet pressure drop and mean temperature difference requirements. In sensible heating or cooling of process fluids where the small heat transfer areas (to 50 m^2) are required, double-pipe heat exchangers are used extensively. Double-pipe heat exchangers can be built in modular concept (i.e., in the form of hairpins).

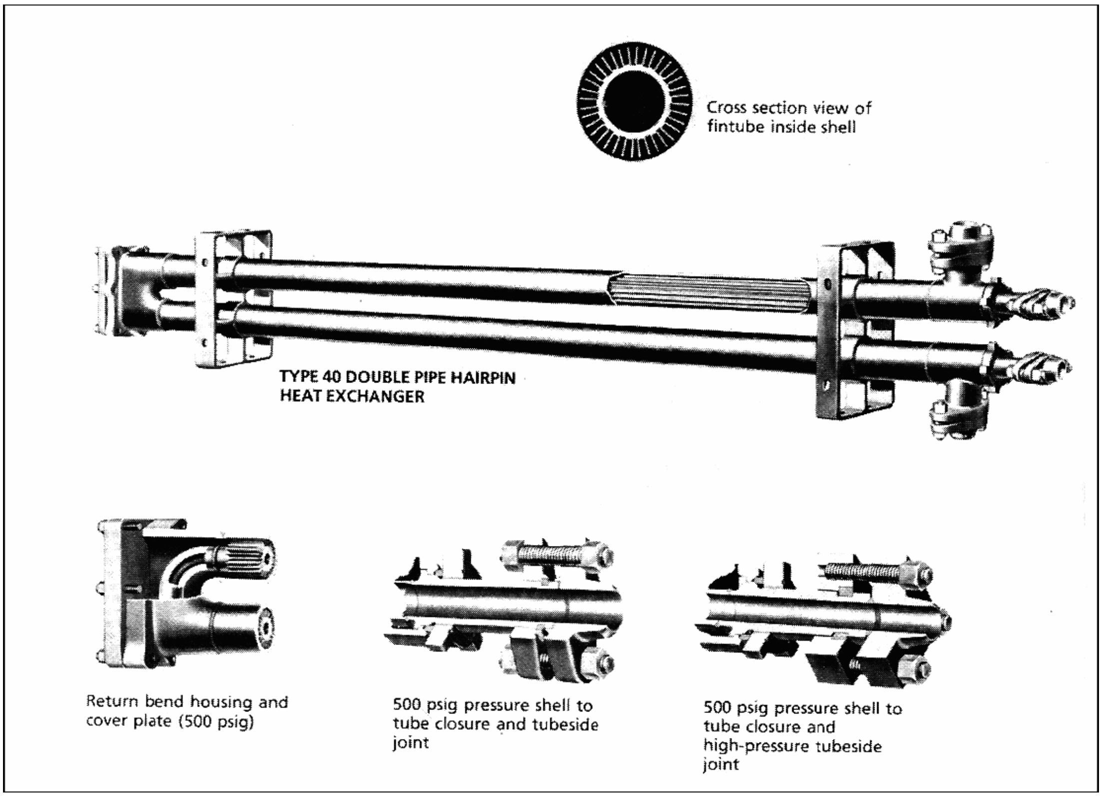


Figure 2.5 Double-pipe hairpin heat exchanger

2.3.1.2 Shell and Tube Heat Exchangers

Shell and tube heat exchangers are built of round tubes mounted in large cylindrical shells with the tube axis parallel to that of the shell. They are widely used as oil coolers, power condensers, preheaters in power plants, steam generators in nuclear power plants, and in process and chemical industry applications. The simplest form of a horizontal shell and tube type condenser with various components shown in Figure 2.6. One fluid flows through the tubes while the other flows on the shell side, across or along the tubes. The baffles are used to promote a better heat transfer coefficient on the shell side and to support the tubes. In a baffled shell and tube heat exchanger, the shell side fluid flows across between pairs or baffles and then flows parallel to the tubes as it flows from one baffle compartment to the next. There are many different shell and tube heat exchangers depending on the application. The most representative tube bundle types are used in shell and tube heat exchangers are shown in Figure 2.7 and 2.8. Since only one tube sheet is used, the U tube is the least expensive construction. But the tube side cannot be mechanically cleaned because of the sharp U-bend.

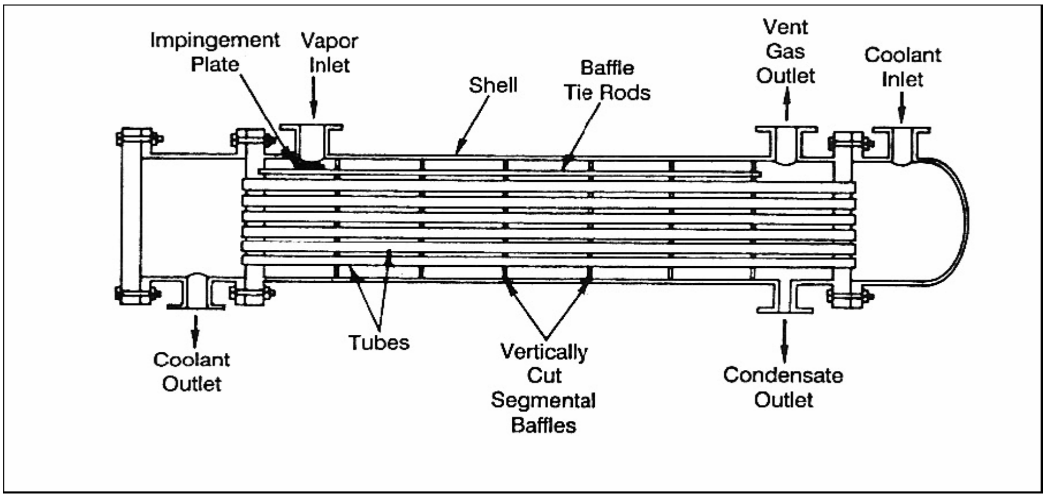


Figure 2.6 Shell and tube heat exchanger as a shell side condenser (Kakac, 2002)

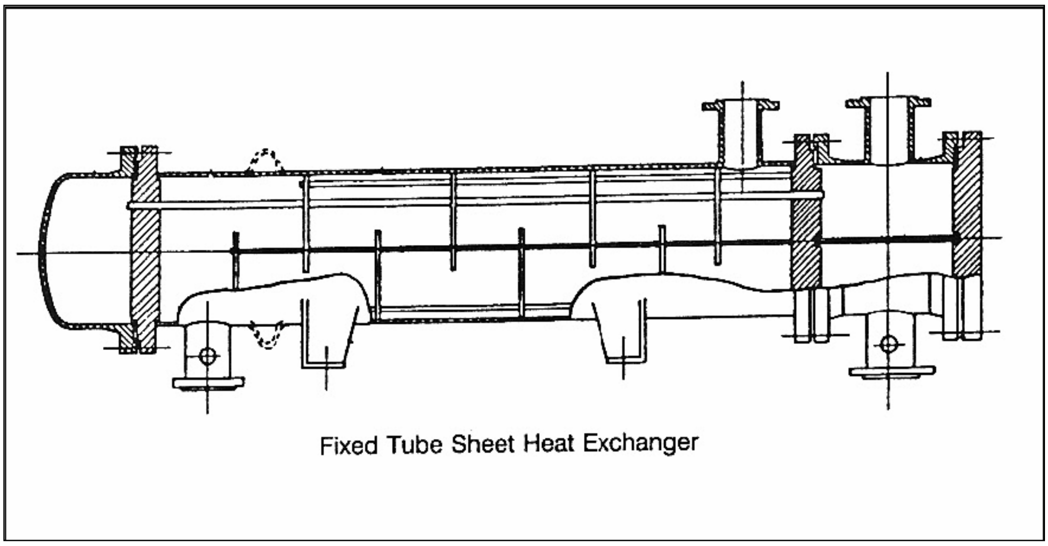


Figure 2.7 Two-pass tube, baffled single-pass shell, shell and tube heat exchanger (Kakac, 2002)

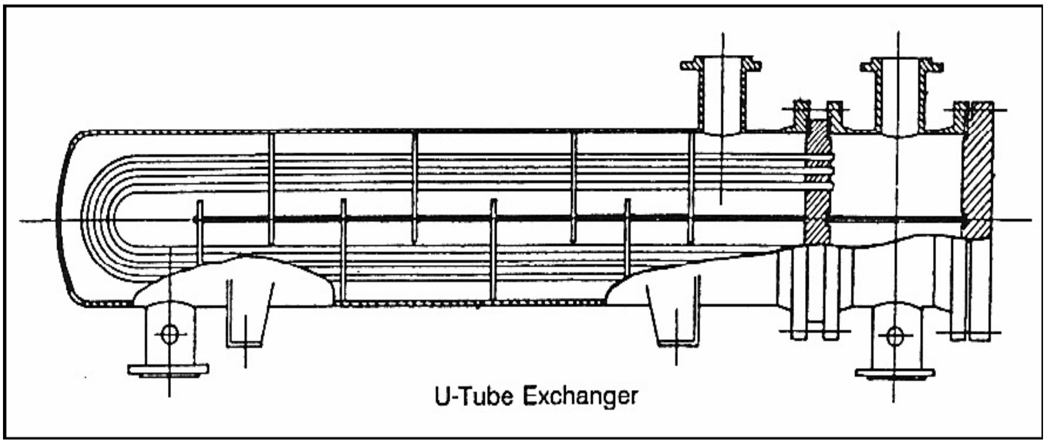


Figure 2.8 U-tube, baffled single-pass shell, shell and tube heat exchanger (Kakac, 2003)

2.3.1.3 Spiral-Tube Heat Exchangers

This consists of spirally wound coils placed in a shell or designed as co-axial condensers that are used in refrigeration systems. The heat transfer coefficient is higher in a spiral tube than in a straight tube. Spiral-tube heat exchangers are suitable for thermal expansion and clean fluids, since cleaning is almost impossible.

2.3.2 Plate Heat Exchangers

Plate heat exchangers are made of thin plates forming flow channels. The fluid streams are separated by flat plates which are smooth or between which lie corrugated fins. Plate heat exchangers are used for transferring heat for any combination of gas, liquid, and two-phase streams. These heat exchangers can further be classified as gasketed plate, spiral plate, or lamella.

2.3.2.1 Gasketed-Plate Heat Exchangers

A typical gasketed plate heat exchanger is shown in Figures 2.9 and 2.10. A gasketed plate heat exchanger consists of a series of thin plates with corrugations or a wavy surface that separates the fluids. The plates come with corner parts arranged so that the two media between which heat is to be exchanged flow through alternate interplate apaches. Appropriate to design and gasketing permit a stack of plates to be held together by compression bolts joining the end plates. Gaskets prevent leakage to the outside and direct the fluids in the plates as desired. The flow pattern is generally chosen so that the media flow counter current to each other.

The plate heat exchangers are usually limited to a fluid steam with a pressure below 25 bar and a temperature below about 250 C. Since the flow passages are quite small, strong eddying gives high heat transfer coefficients, high pressure drops, and high local shear which minimizes fouling. These exchangers provide a relatively compact and lightweight heat transfer surface. They are temperature and pressure-limited due to the construction details and gasketing.

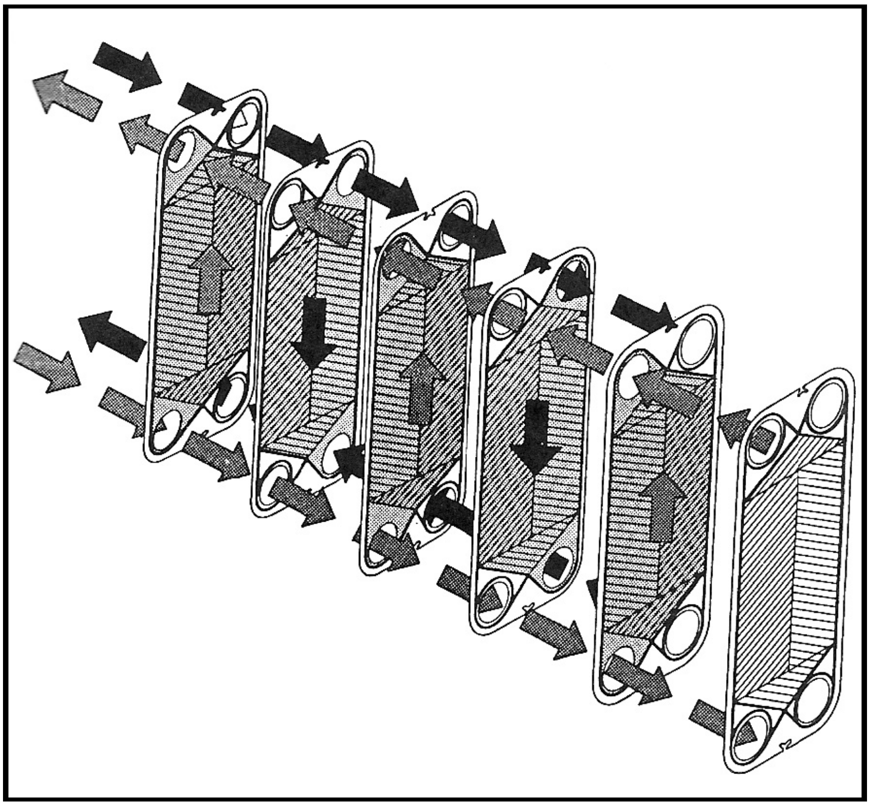


Figure 2.9 Gasketed-plate heat exchanger and flow paths (Kakac, 2002)

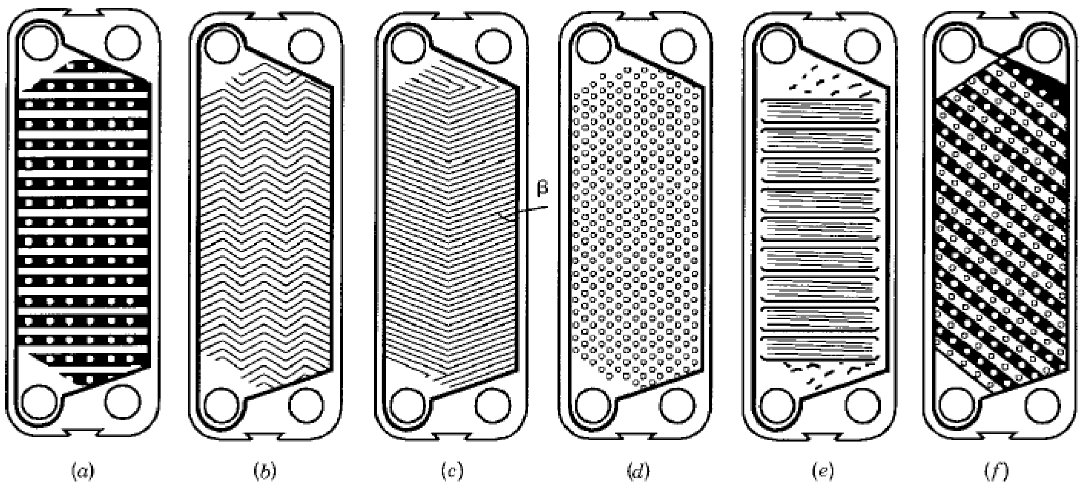


Figure 2.10 Plate patterns: (a) washboard; (b) zigzag; (c) chevron or herringbone; (d) protrusions and depressions; (e) washboard with secondary corrugations; (f) oblique washboard (Shah, 2003)

Some advantages of plate heat exchangers are as follows. They can easily be taken apart into their individual components for cleaning, inspection, and maintenance. The heat transfer surface area can readily be changed or rearranged for a different task or for anticipated changing loads, through the flexibility of plate size, corrugation patterns, and pass arrangements. High shear rates and shear stresses, secondary flow, high turbulence, and mixing due to plate corrugation patterns reduce fouling to about 10 to 25% of that of a shell-and-tube exchanger, and enhance heat transfer (Shah, 2003).

Very high heat transfer coefficients are achieved due to the breakup and reattachment of boundary layers, swirl or vortex flow generation, and small hydraulic diameter flow passages. Because of high heat transfer coefficients, reduced fouling, the absence of bypass and leakage streams, and pure counter flow arrangements, the surface area required for a plate exchanger is one-half to one-third that of a shell-and tube exchanger for a given heat duty, thus reducing the cost, overall volume, and space requirement for the exchanger (Shah, 2003).

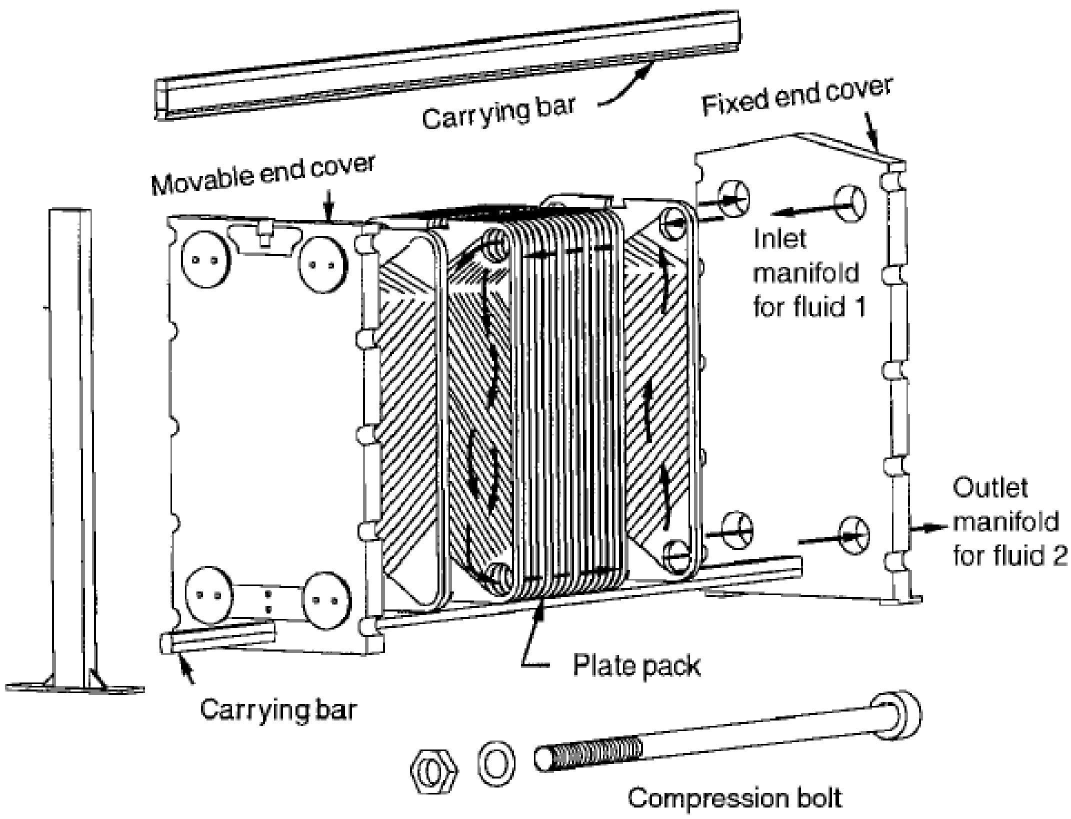


Figure 2.11 Gasketed plate- and-frame heat exchanger (Shah, 2003)

2.3.2.2 Spiral Plate Heat Exchangers

Spiral plate heat exchangers are formed by Rolling two long, parallel plates into a spiral using a mandrel and welding the edges of adjacent plates to form channels. (Figure 2.12) The distance between the metal surfaces in both spiral channels is maintained by means of distance pins welded to the meral sheet. The length of the distance pins may vary between 5 and 20 mm. It is therefore the smallest possible to choose between different channel spacings according to the flow rate. This means that the ideal flow conditions, and therefore the smallest possible heating surfaces, are obtained (Kakac, 2002).

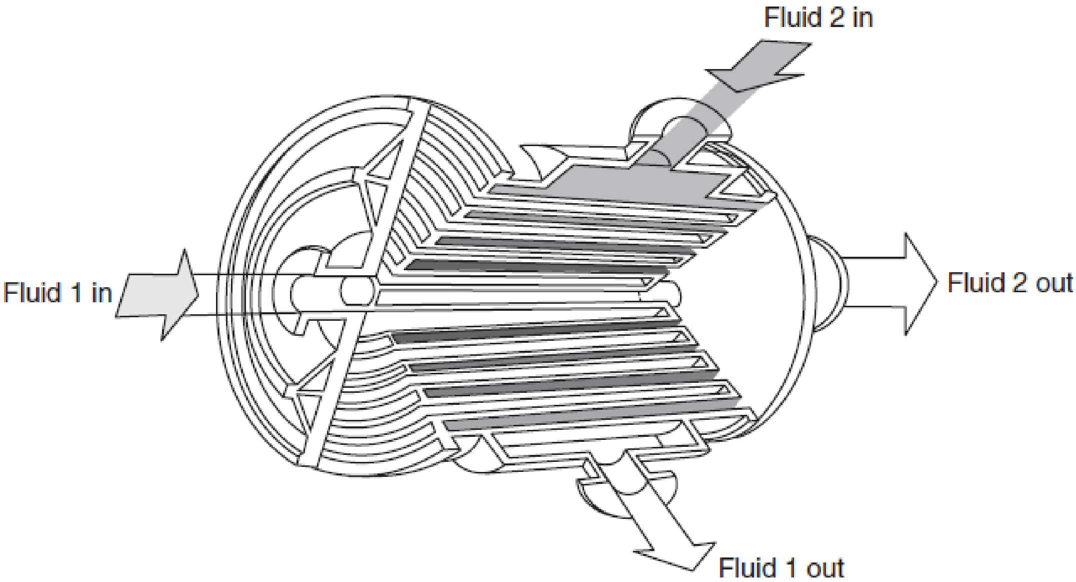


Figure 2.12 Counter-flow spiral heat exchanger

Two spiral paths introduce a secondary flow, increasing the heat transfer and reducting fouling deposits. These heat exchangers are quite compact, but are relatively expensive due to their specialized fabrication. Sizes range from 0.5 to 500m² heat transfer surface in one single spiral body.

The spiral heat exchanger is particularly effective in handling sludges, viscous liquids, and liquids with solids in suspension including slurries. A cross flow type spiral heat exchanger is shown in Figure 2.13.

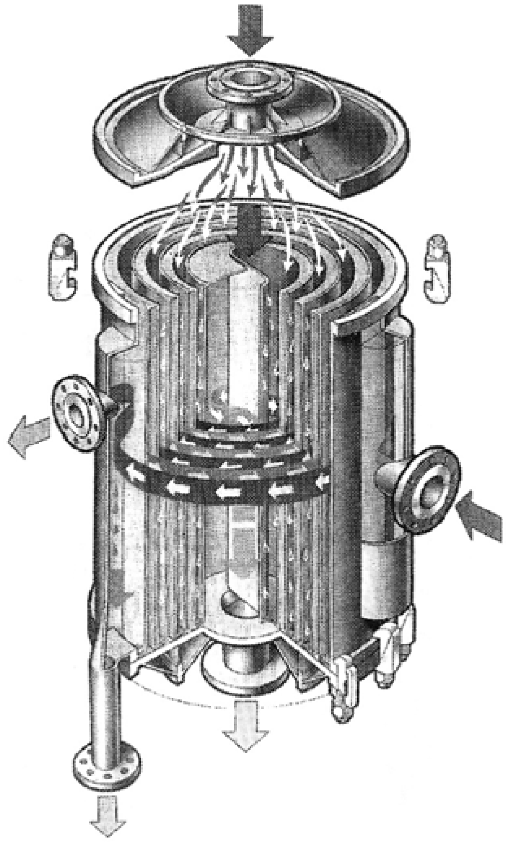


Figure 2.13 Cross-flow spiral heat exchanger (Kakac, 2002)

2.3.2.3 Lamella Heat Exchanger

The lamella (Ramen) type heat exchangers consists of a set of parallel, welded, thin plate channels or lamella (flat tubes or rectangular channels) placed longitudinally in a shell. (Figure 2.14) It is a modification of the floating-head type of shell and tube heat exchanger. These flattened tubes (lamellas) are made up of two strips of plates, profiled and spot or seam welded together in a continuous operation. The lamellas are welded together at both ends by joining the ends with steel bars in between, depending on the space required between lamellas. Both ends of the lamella bundle are joined by peripheral welds to the channel cover, which at the outer ends is welded to the inlet and outlet nozzle. The lamella side is thus completely sealed in by welds.

Lamella heat exchangers can be arranged for true countercurrent flow, since there are no shell side baffles. Because of high turbulence, uniform flow distribution, and smooth surfaces, the lamellas do not foul easily. They can be used up to 35 bar, 200°C for Teflon gaskets, and 500°C for asbestos gaskets.

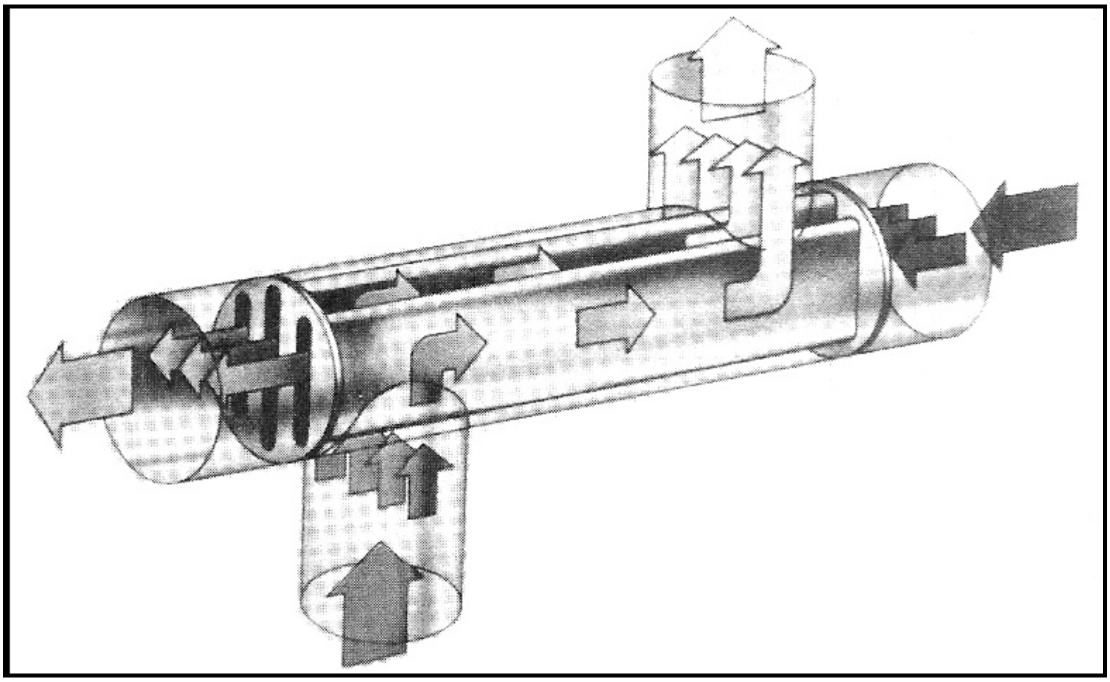


Figure 2.14 Lamella heat exchanger (Kakac, 2002)

2.3.3 Extended Surface Heat Exchangers

Extended surface heat exchangers are devices with fins or appendages on the primary heat transfer surface (tubular or plate) with the object of increasing heat transfer area. As it is well known that the heat transfer coefficient on the gas side is much lower than those on the liquid side, finned heat exchanger surfaces are used on the gas side to increase the heat transfer area. Fins are widely used in gas-to-gas and gas-liquid heat exchangers whenever the heat transfer coefficient on one or both sides is low and there is a need for a compact heat exchanger. The two most common types of extended surface heat exchangers are plate-fin heat exchangers and tube-fin heat exchangers.

2.3.3.1 Plate-Fin Heat Exchangers

Plate-fin type heat exchangers are primarily used in gas-to-gas applications and tube-fin type heat exchangers are used in liquid-air applications. Since mass and volume reduction is important in most of the applications, compact heat exchangers are widely used in air-conditioning, refrigeration and process industries. Basic construction of a plate-fin heat exchanger is shown in Figure 2.14. The fluids are separated by flat plates between which are sandwiched corrugated fins. Figure 2.13 shows the arrangement for parallel flow or counter flow and cross flow between the streams.

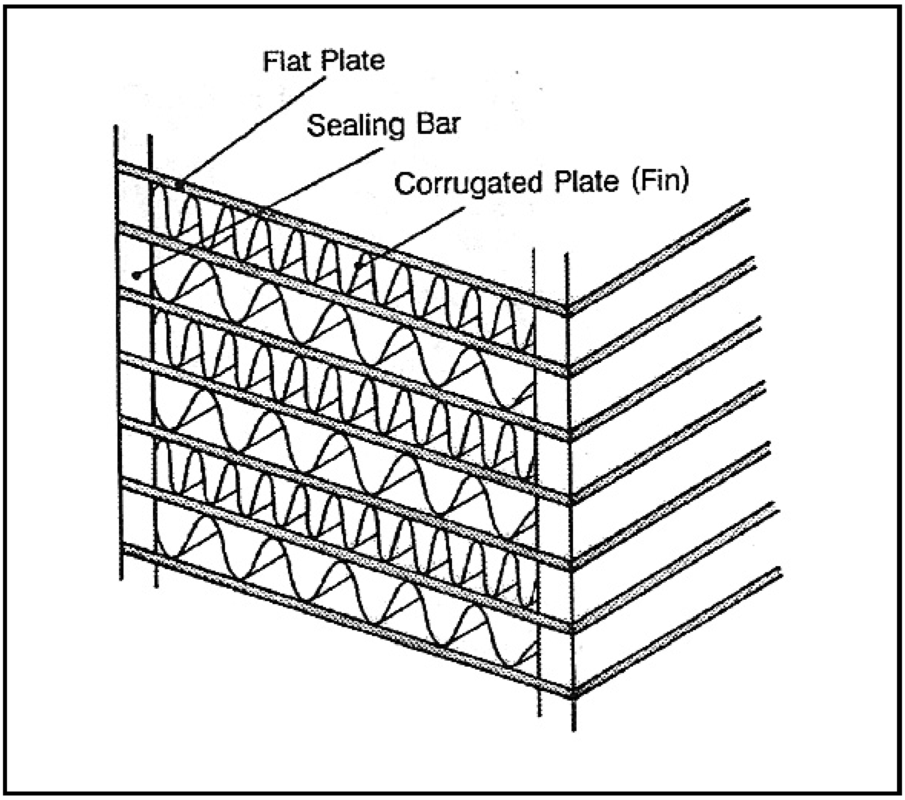


Figure 2.15 Basic construction of a plate-fin heat exchanger

These heat exchangers are very compact units with a heat transfer area per unit volume of around 2000m2/m3. Special manifold devices are provided at the inlet to these exchangers to provide good flow distributions across the plates and from plate to plate. The plates are typically 0.5 to 1.0 mm thick and the fins are 0.15 to 0.75 mm thick. The whole exchanger is made of aluminum alloy, and the various components are brazed together in a salt bat or in a vacuum furnace.

The corrugated sheets that are sandwiched between the plates serve both to give extra heat transfer area and to give structural support to the flat plates. The most common types of corrugated sheets are shown in Figure 2.16.

1. Plain fin
2. Plain-perforated fin
3. Serrated (interrupted, louver) fin
4. Herringbone or wavy fin

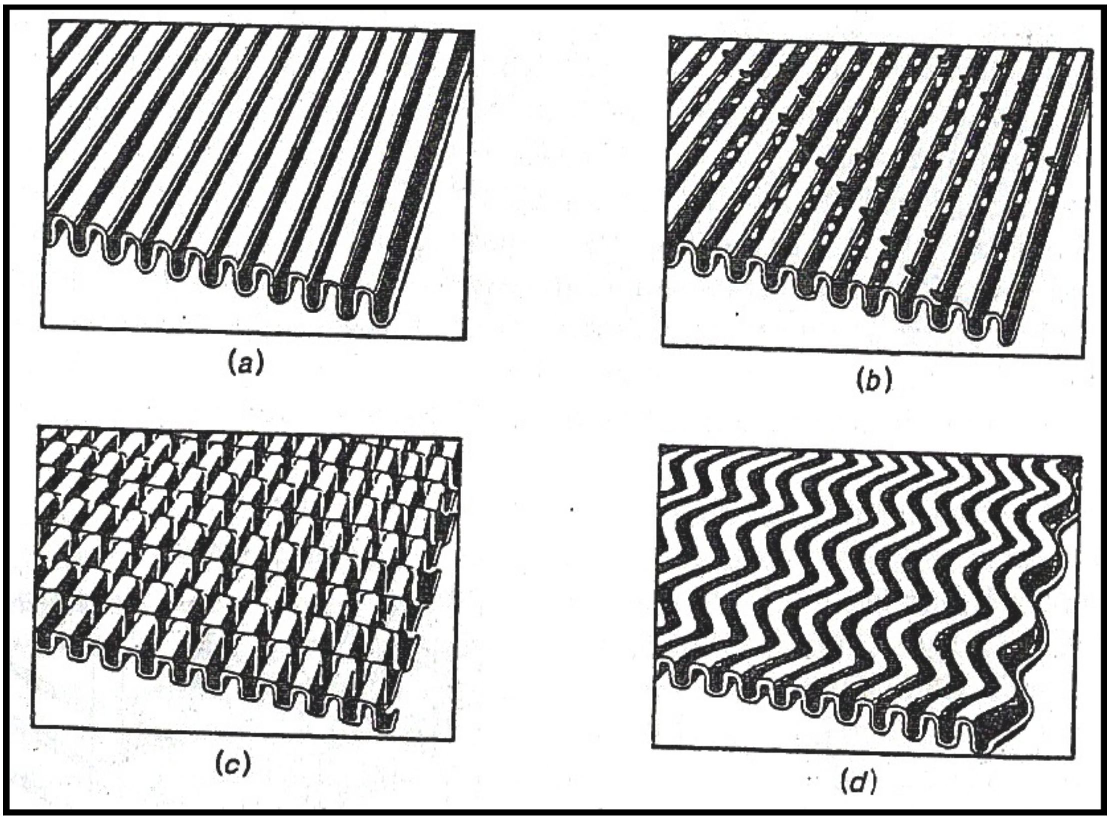


Figure 2.16 Fin types in plate-fin heat exchangers. (a) Plain, (b) perforated, (c) serrated, (d) herringbone (Kakac 2002)

The flow channels in plate-fin exchangers are small which means that the mass velocity also has to be small (10 to 300 kg/m2s) to avoid excessive pressure drops. This may make the channel prone to fouling which, when combined with the fact that they cannot be mechanically cleaned, means that plate-fin exchangers are restricted to clean fluids. Plate-fin exchangers are frequently used for condensation duties in air liquefaction plants.

Plate-fin heat exchangers have been established for use in gas turbines, conventional and nuclear power plants, propulsion engineering (airplanes, trucks and automobiles), refrigeration, heating, ventilating and air conditioning, waste heat recovery systems, in chemical industry, and fort he cooling of electronic devices.

2.3.3.2 Tubular-Fin Heat Exchangers

Tubular-fin heat exchangers are used as gas-to-liquid heat exchangers. Since the gas side heat transfer coefficients are generally much lower than the liquid side, fins are required. As shown in Figures 2.17 and 2.18, a tubular-fin heat exchanger consists of an array of tubes with fins fixed on the outside. The fins may be normal on individual tubes, transverse or helical, or longitudinal (Figure 2.17). Longitudinal fins are commonly used in double-pipe or shell and tube heat exchangers with no baffles. As shown in Figure 2.16, continuous plate-fin sheets may be fixed on the array of round, rectangular, or elliptical tubes. Plate fin and tube heat exchangers are commonly used in air-conditioning and refrigerating systems.

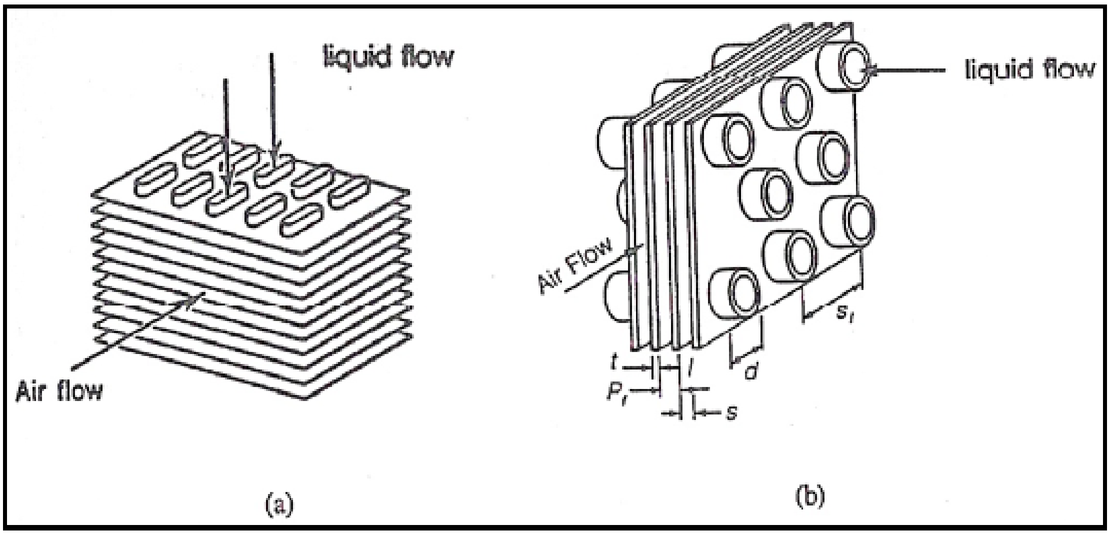


Figure 2.17 Tube-fin heat exchangers. (a) Flattened tube-fin, (b) round tube-fin. (Kakac, 2002)

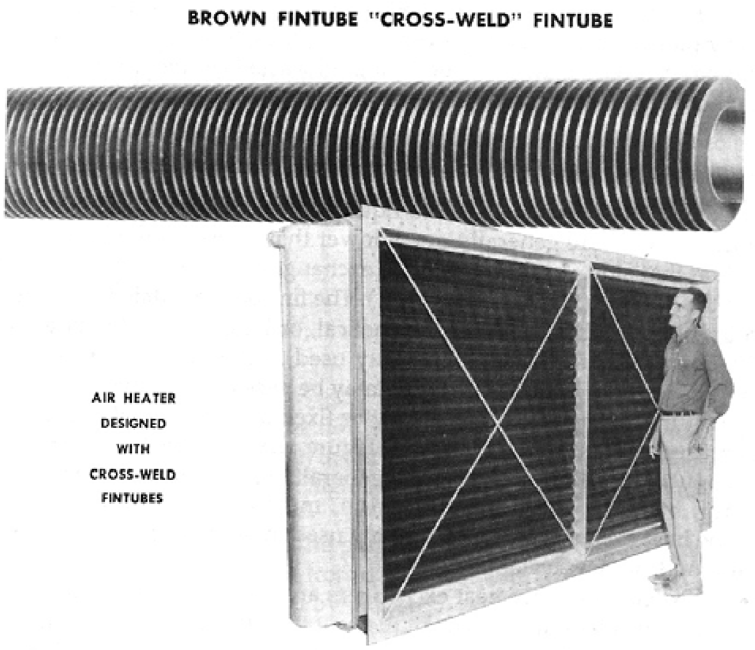


Figure 2.18 Fin-tube air heater (Kakac, 2002)

2.4 Heat Transfer Mechanisms

Heat exchanger equipment can also be classified according to the heat transfer mechanisms as:

1. Single-phase convection on both sides
2. Single-phase convection on one side two-phase convection on the other side
3. Two-phase convection on both sides

The principles of these types are illustrated in Figures 2.1e,f, and g. Figures 2.1f and g illustrate two possible modes of two-phase flow heat exchangers. In Figure 2.1f, fluid A is being evaporated receiving heat from fluid B; and in Figure 2.1g, fluid A is being condensed, giving up heat to fluid B (Kakac, 2002).

In heat exchanger like economizers and air heaters in boilers, compressors intercoolers, automotive radiators, regenerators, oil coolers, space heaters, etc., single-phase convection occurs on both sides.

Condensers, boilers, and steam generators used in pressurized water reactors, power plants, evaporators, and radiators used in air-conditioning an space heating include the mechanisms of condensation, boiling and radiation on one of the surfaces of the heat exchanger. Two-phase heat exchanger could also occur on each side of the heat exchanger such as condensing on one side and boiling on the other side of the heat exchanger surface. However, without phase change, we may also have a two-phase flow heat transfer mode as in the case of fluidized beds where a mixture of gas and solid particles transports heat to or from a heat transfer surface.

2.5 Flow Arrangement

Heat exchangers may be classified according to the fluid-flow path through the heat exchanger (Kakac, 2002). Three basic flow arrangements are:

1. Parallel flow
2. Counter flow
3. Cross flow

In parallel flow (co current) heat exchangers, the two fluid streams enter together at one end, flow through the same direction, and leave together at the other end. (Figure 2.19a) In counter flow (countercurrent) heat exchangers, two fluid streams flow in opposite direction (Figure 2.19b). In single-cross flow heat exchangers, one fluid flows through the heat exchanger surface at right angles to the flow path of the other fluid. Cross flow arrangements with both fluids unmixed, and one fluid mixed and the other fluid unmixed are shown in Figures 2.19c and 2.19d, respectively.

Multi pass cross flow configurations can also be organized by having the basic arrangement in a series. For example, in a U-baffled tube single-pass shell-and-tube heat exchanger, one fluid flows through the U-tube while the other fluid flows first downward and then upward, across the flow path of the other fluid steam, which is also referred to as cross-counter, cross-parallel flow arrangements.

Multi pass flow arrangements are frequently used in heat exchanger designs, especially in shell-and-tune heat exchangers with baffles (Figure 2.7-2.8). The main difference between the flow arrangements lies in the temperature distribution along the length of the heat exchanger and the relative amounts of heat transfer under given temperature specification for specified heat exchanger surfaces (as will be shown later for given flow and specified temperatures, a counter flow heat exchanger requires minimum area, a parallel-flow heat exchanger requires maximum area, a parallel plate-flow heat exchanger requires maximum area, and a cross-flow heat exchanger requires in area between).

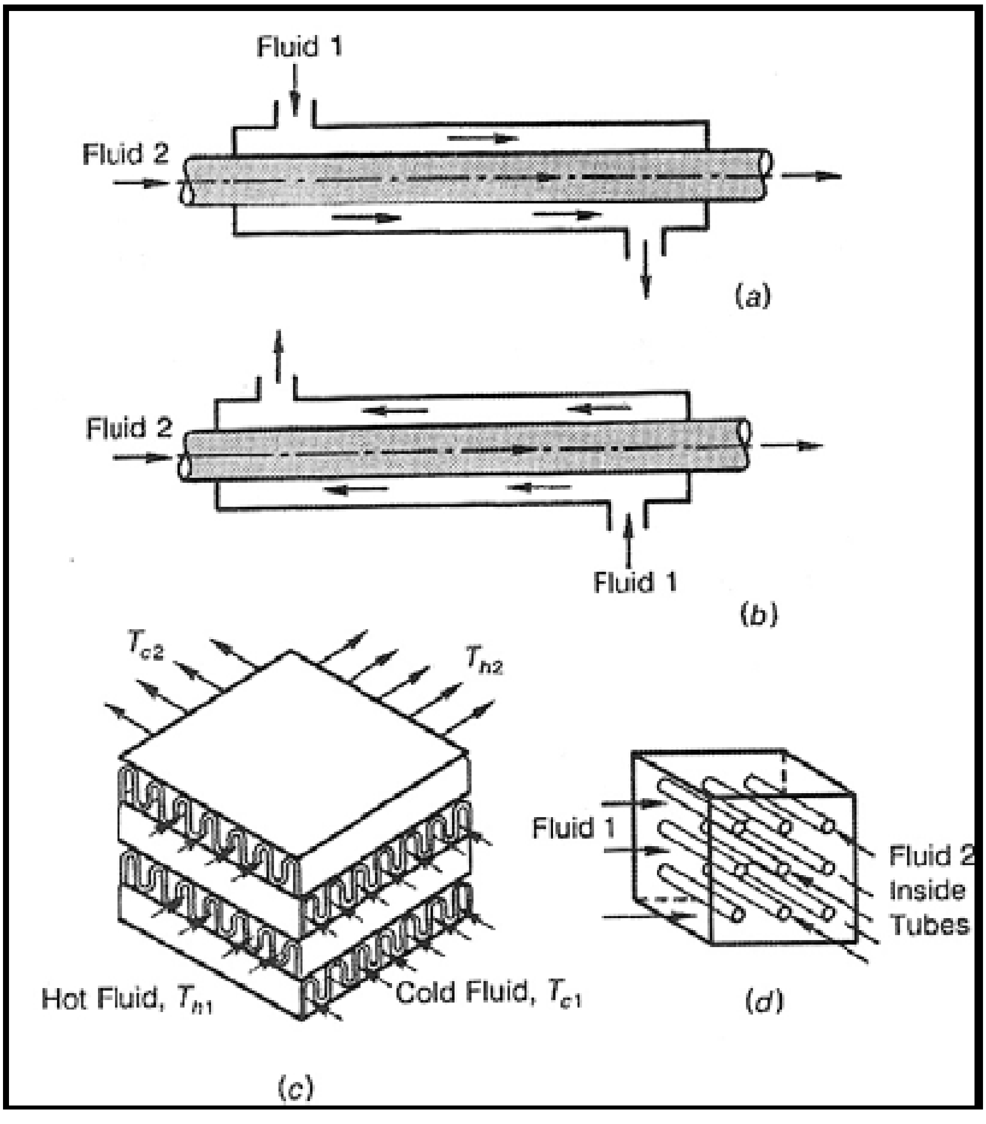


Figure 2.19 Heat exchanger classification according to flow arrangement.(a) Parallel flow, (b) Counter flow, (c) cross flow-both fluids unmixed, (d) cross flow-fluid 1 mixed, fluid2 unmixed. (Kakac, 2002)

In the cross-flow arrangement, the flow may be called mixed or unmixed, depending on the design. Figure 2.19c shows an arrangement in which both hot and cold fluids flow through individual flow channels with no fluid mixing between adjacent flow channels. In this case, each fluid steam is said to be unmixed.

In flow arrangement shown in Figure 2.19d, fluid 2 flows inside the tubes and is therefore not free to move in transverse direction ; fluid 2 is therefore considered unmixed, while fluid 1 is free to move in the transverse direction and mix itself. Consequently, this heat exchanger is called an unmixed-mixed cross-flow heat exchanger. For extended surface heat exchangers, it is also possible to have the basic cross-flow arrangements in a series to form multipass arrangements as cross-counter flow and cross-parallel flow. This usually helps to increase overall effectiveness of the heat exchanger.

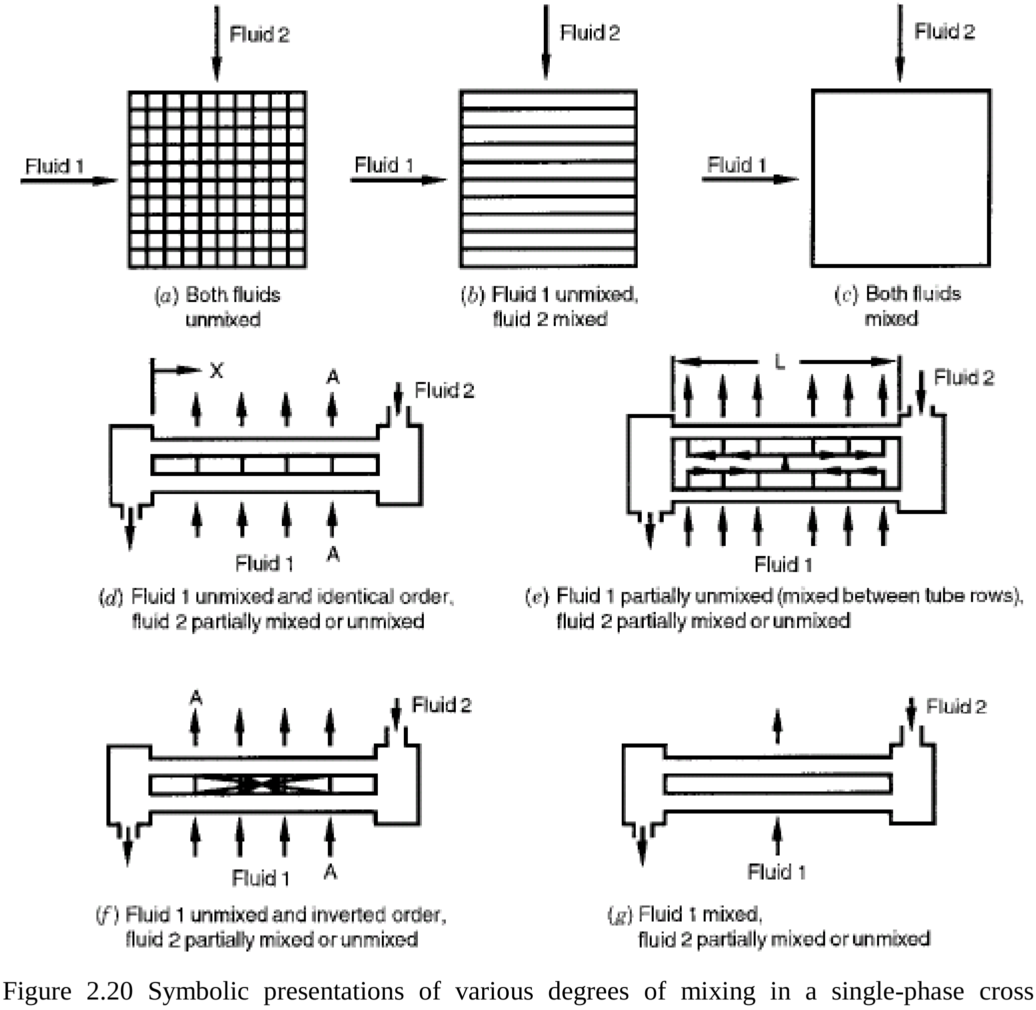


Figure 2.20 Symbolic presentations of various degrees of mixing in a single-phase cross flow exchanger (Shah,2003).

CHAPTER THREE
HEAT RECOVERY VENTILATION IN HVAC SYSTEMS

3.1 History of HVAC

HVAC (Heating, ventilating, and air conditioning) is based on inventions and discoveries made by Nikolay Lyov, Michael Faraday, Willis Carrier, Reuben Trane, James Joule, William Rankine, Sadi Carnot, and many others.

The invention of the components of HVAC systems went hand-in-hand with the industrial revolution, and new methods of modernization, higher efficiency, and system control are constantly introduced by companies and inventors all over the world. The three central functions of heating, ventilating, and air-conditioning are interrelated, providing thermal comfort, acceptable indoor air quality, within reasonable installation, operation, and maintenance costs. HVAC systems can provide ventilation, reduce air infiltration, and maintain pressure relationships between spaces. How air is delivered to, and removed from spaces is known as room air distribution.

In modern buildings the design, installation, and control systems of these functions are integrated into one or more HVAC systems. For very small buildings, contractors normally "size" and select HVAC systems and equipment. For larger buildings, building services designers and engineers, such as mechanical, architectural or building services engineers analyze, design, and specify the HVAC systems, and specialty mechanical contractors build and commission them. Building permits and code-compliance inspections of the installations are normally required for all sizes of buildings. (http://en.wikipedia.org/wiki/HVAC)

3.2 Ventilation

Ventilation is a necessity for the health and comfort of occupants of all buildings. Ventilation supplies air for occupants to breathe and removes moisture, odors, and indoor pollutants like carbon dioxide. Ventilation design for apartment buildings is inherently more complex than what is required for single-family homes. Most apartments have limited exposure of walls and windows to the outside environment.

3.2.1 Natural Ventilation

Natural ventilation systems are normally driven by the buoyancy force, where the temperature difference between inside and outside, or in special designs the wind pressure, drives the ventilation. The advantage of using natural ventilation is that there is no electricity consumption in the system (no fans) and lower installation costs compared with traditional mechanical ventilation systems. The disadvantages are that it is often difficult to control natural ventilation, i.e. the air change rate, and attention must be given to check if the required ventilation rate is fulfilled at all times.

Natural ventilation with heat recovery is rarely seen and very difficult to construct because of the conflict between the use of the temperature difference as a driving force and the equalization of temperature in the heat exchanger. Because of the large temperature difference between the inside and outside air in cold climates, natural ventilation will result in very large ventilation heat losses, and preheating the inlet air will be necessary if draught is to be avoided. Traditional preheating systems, i.e. using heating coils, are assumed to be unacceptable in this project due to the extra energy consumption that this implies. Other methods of preheating the air could be achieved by solar heating, but solar radiation is not always available.

3.2.2 Mechanical Ventilation

"Mechanical" or "forced" ventilation is used to control indoor air quality. Excess humidity, odors, and contaminants can often be controlled via dilution or replacement with outside air. When infiltration and natural ventilation systems are inadequate (as determined either by code or experience), mechanical ventilation should be installed. Mechanical ventilation systems are capable of providing a controlled rate of air exchange and should respond to the varying needs of occupants and pollutant loads, irrespective of climate vagaries (http://en.wikipedia.org/wiki/HVAC).

Mechanical ventilation with efficient heat recovery consists of two fans, a heat exchanger, filters, ducts, inlet and outlet diffusers and a controlling system. Using a heat exchanger with high efficiency will typically reduce the ventilation heat loss by 80-90% and the total heat loss by 30-60%, depending on the insulation level of the house etc. Outlets are normally placed in the rooms where moisture, odour and other pollutants of the indoor environment are produced, i.e. the kitchen, bathrooms and scullery, and the inlet diffusers are placed in rooms where the people are present over periods of time, i.e. the living room and bedrooms. In this way the moisture and odour from cooking and bathing etc. is removed effectively without polluting the surrounding rooms, and fresh air is blown into the building, providing a good indoor climate.

The disadvantages of using mechanical ventilation systems are higher installation costs, necessary space for the components and ducts and very importantly, the electricity consumption by the fans. In the design phase for a ventilation system the attention must always be on minimizing the pressure loss in the system, as this is directly proportional to the electricity used by the fans (Svendsen & Rose & Kragh, 2005).

3.3 Heat Recovery Ventilation

In principle, heat recovery ventilator, HRV systems consist of two electrically driven fans (one for exhaust air and one for fresh air), heat recovery exchanger, fresh and exhaust air filters, air ducts, diffusers and control system. As a result of HRV systems usage, it is possible to reduce indoor heat loss due to ventilation between 50-75% according to room insulation and reduce total heat loss between 20-60%.

Air-to-air energy recovery systems may be categorized according to their application as (1) process-to-process, (2) process-to-comfort, or (3) comfort-to-comfort. Typical air-to-air energy recovery applications are listed in Table 3.1.

Table 3.1 Applications for air-to air energy recovery (Ashrae, 2008)

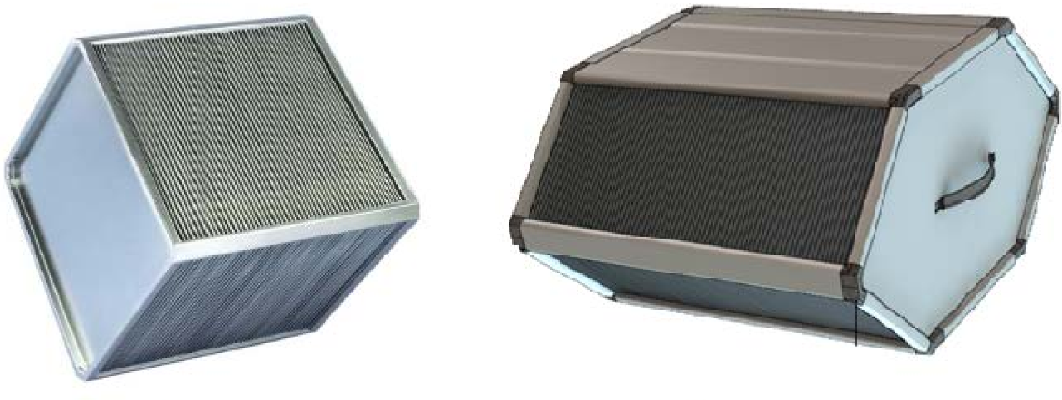
Methods	**Typical Application**
Process-to-process And Process-to-comfort	Dryers Ovens Flue stacks Burners Furnaces Incinerators Paint exhaust Welding exhaust
Comfort-to-comfort	Swimming pools Locker rooms Residential Operating rooms Nursing homes Animal ventilation Plant ventilation General exhaust Smoking exhaust

3.4 Types of Heat Recovery Exchangers

There are 4 well known heat exchanger type that are used in heat recovery ventilation system for comfort applications.

3.4.1 Plate Type Heat Recovery Exchanger

Plate exchangers are available in many configurations, materials, sizes, and flow patterns. Many have modules that can be arranged to handle almost any airflow, effectiveness, and pressure drop requirement. Plates are formed with spacers or separators (like ribs, dimples, ovals) constructed into the plates or with external separators (like supports, braces, corrugations). Air stream separations are sealed by folding, multiple folding, gluing, cementing, welding, or any combination of these, depending on the application and manufacturer. Ease of access for examining and cleaning heat transfer surfaces depends on the configuration and installation.



(a) (b)

Figure 3.1 Types of plate heat recovery exchanger; (a) cross-flow plate heat recovery, (b) counter flow plate heat recovery

Plate spacing ranges from 2.5 to 12.5 mm, depending on the design and application. Heat is transferred directly from the warm airstreams through the separating plates into the cool airstreams. Usually design, construction, and cost restrictions result in the selection of cross-flow exchangers, but additional counter flow patterns can increase heat transfer effectiveness (Ashrae, 2008).

Figure 3.2 Cross flow heat recovery exchanger installation sample (Imeksan Heat Recovery Ventilator Catalogue)

3.4.2 Rotary Air-to-Air Heat Exchangers

A rotary air-to-air energy exchanger, or rotary enthalpy wheel, has a revolving cylinder filled with an air-permeable medium having a large internal surface area. Adjacent supply and exhaust airstreams each flow through half the exchanger in a counter flow pattern. Heat transfer media may be selected to recover sensible heat only or total (sensible plus latent) heat.

Air contaminants, dew point, exhaust air temperature, and supply air properties influence the choice of materials for the casing, rotor structure, and medium of a rotary energy exchanger. Aluminum, steel, and polymers are the usual structural, casing, and rotor materials for normal comfort ventilating systems. Exchanger media are fabricated from metal, mineral, or synthetic materials and provide either random or directionally oriented flow through their structures

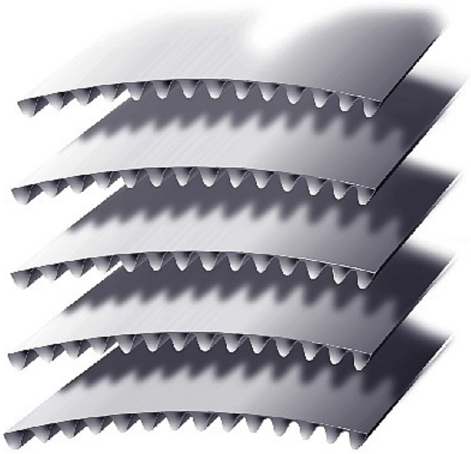


Figure 3.3 Specially joined Aluminum plates (Im Makine Rotary Heat Exchanger Catalogue)

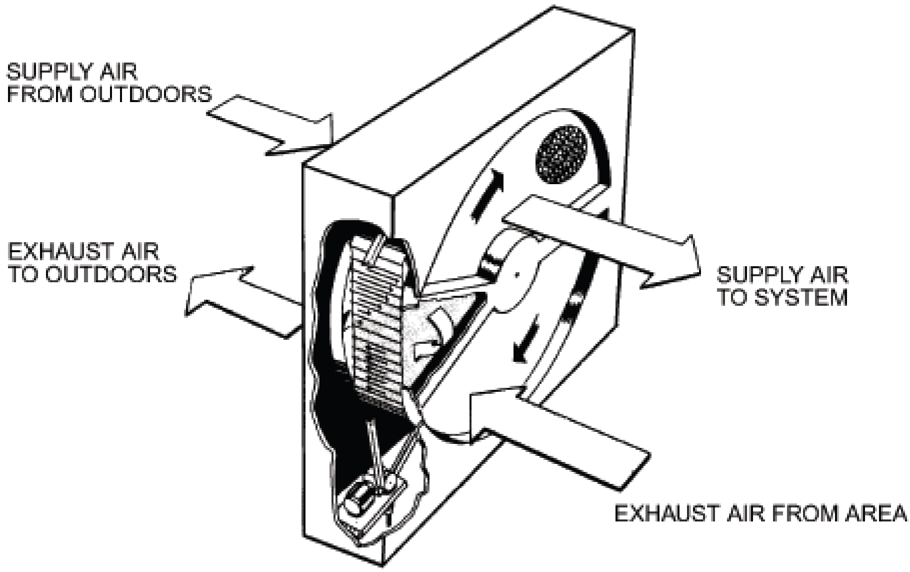


Figure 3.4 Rotary air-to-air energy recovery (Ashrae, 2008)

3.4.3 Heat Pipe Heat Recovery Exchangers

Figure 3.5 shows a typical heat pipe assembly. Hot air flowing over the evaporator end of the heat pipe vaporizes the working fluid. A vapor pressure gradient drives the vapor to the condenser end of the heat pipe tube, where the vapor condenses, releasing the latent energy of vaporization. The condensed fluid is wicked or flows back to the evaporator, where it is revaporized, thus completing the cycle. Thus the heat pipe's working fluid operates in a closed-loop evaporation/condensation cycle that continues as long as there is a temperature difference to drive the process. Using this mechanism, the heat transfer rate along a heat pipe is up to 1000 times greater than through copper.

HVAC systems use copper or aluminum heat pipe tubes with aluminum fins. Fin designs include continuous corrugated plate, continuous plain, and spiral. Modifying fin design and tube spacing changes pressure drop at a given face velocity.

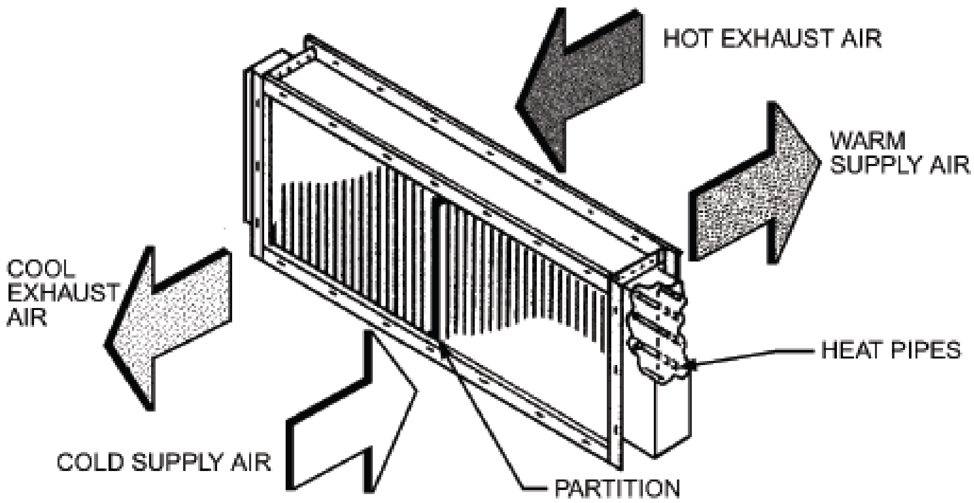


Figure 3.5 Heat pipe assembly (Ashrae, 2008)

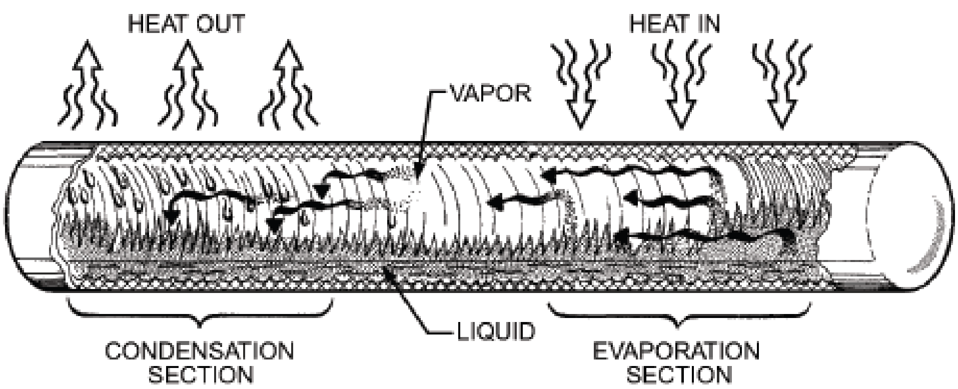


Figure 3.6 Heat pipe operation (Ashrae, 2008)

3.4.4 Coil Energy Recovery (Runaround) Loops

A typical coil energy recovery loop places extended surface, finned-tube water coils in the supply and exhaust airstreams of a building or process. (Figure 3.7) The coils are connected in a closed loop by counter flow piping through which an intermediate heat transfer fluid (typically water or antifreeze solution) is pumped.

The coil energy recovery loop cannot transfer moisture between airstreams; however, indirect evaporative cooling can reduce the exhaust air temperature, which significantly reduces cooling loads. For the most cost-effective operation, with equal airflow rates and no condensation, typical effectiveness values range from 45 to 65%. The highest effectiveness does not necessarily give the greatest net life-cycle cost saving (Ashrae, 2008).

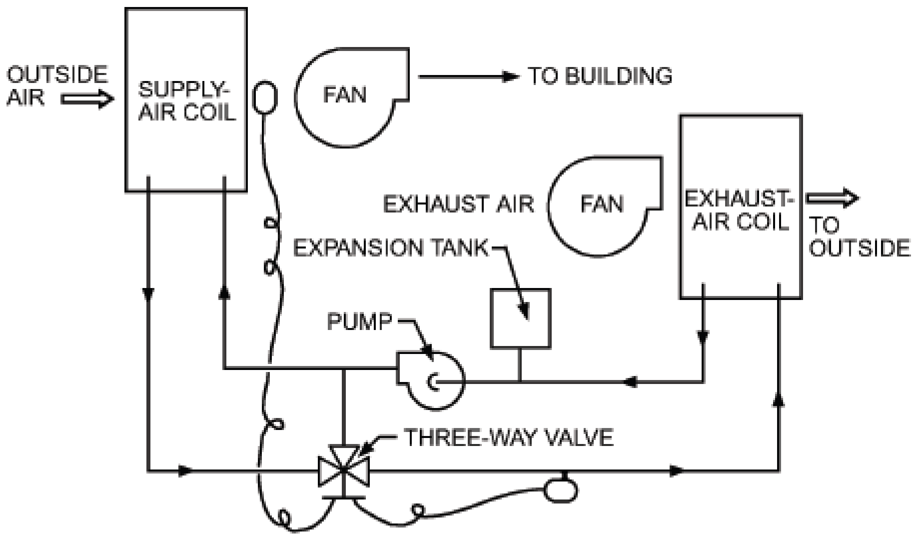


Figure 3.7 Coil energy recovery loop (Ashrae, 2008)

3.5 Plate Type Heat Recovery in HVAC Applications

Plate heat exchangers are obtained from collecting plates repeatedly. The plates are made of different materials and plates’ ends are clamped together within the framework. Depending on the type of plate material, heat exchangers can do latent heat transfer and dehumidification.

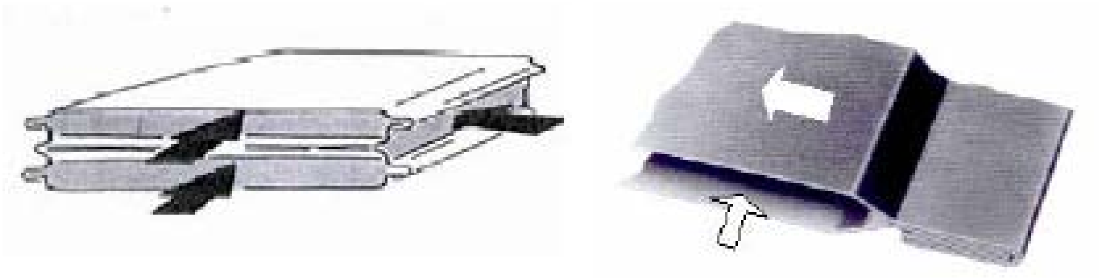


Figure 3.8 Junction of plates in plate heat recovery exchanger

As shown in the Figure 3.8, top and bottom plates are clamped the middle two plates from right and left and the middle two plates are clamped each other from front and back. Thus, the heat energy of air flowing from front to back, may be possible to transfer by the middle two plates to the air moving through the left to right.

Two plates can be connected with each other; single, double, triple, triple clamps with silicone-backed according to the desired level of impermeability. In the heat exchanger block, up to 4500 Pascal pressure difference can be achieved certain tightness and resistance to deformation by the using materials and the clamp application technique.

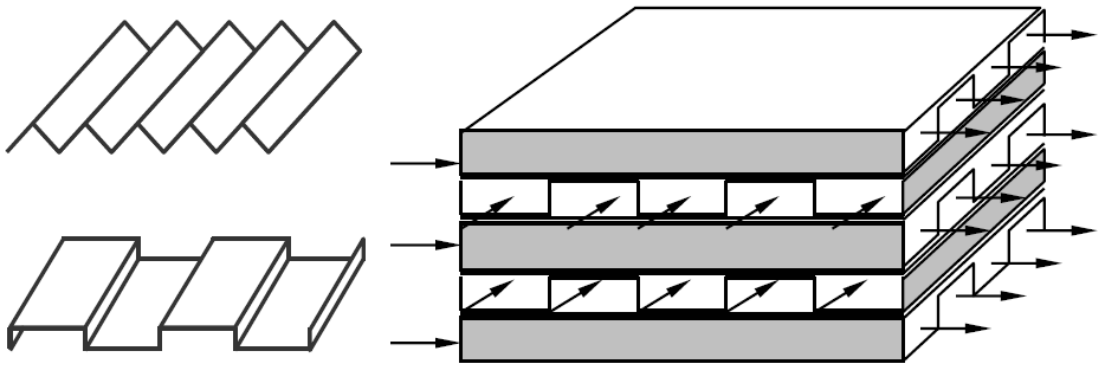


Figure 3.9 Heat exchanger with corrugated sheets (Bulgurcu, 2001)

Efficiency of a plate heat exchanger is related to; air velocity, surface form of the plates, plate range and the pressure loss. If the efficiency increase, air velocity and pressure will drop, dimension of heat exchanger will increase and become more expensive. If the pressure loss increase, the air velocity will increase, heat exchanger will be reduced, will be cheaper, but efficiency will decrease. In applicable industrial HVAC designs, efficiency keeping around %55, the loss of pressure in the 100-250 remained in the range of Pascal and not exceed 300 Pascal is proposed.

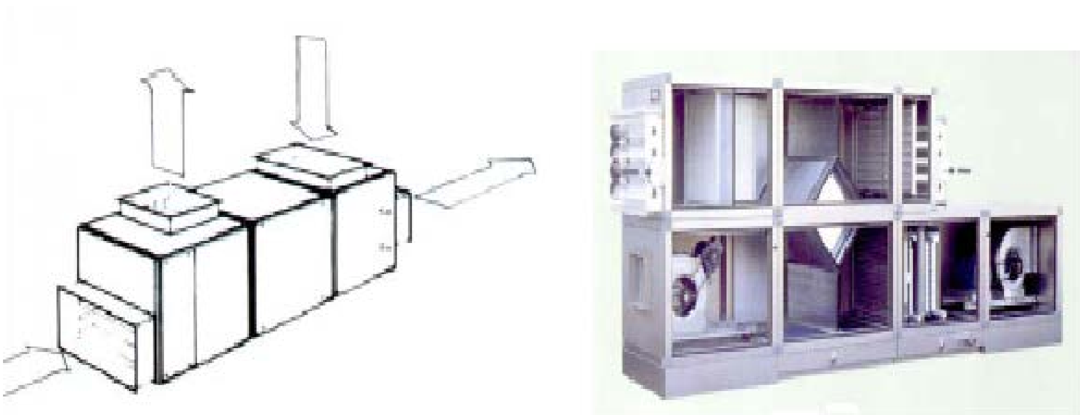
Plate heat exchangers for HVAC applications organized recycling proposal is shown in Table 3.2.

Table 3.2 Plate heat exchanger applications for HVAC

Subject	**Plate Material**	**Application Form**
Industrial air conditioning and ventilating	Aluminum, epoxy-coated aluminum, stainless steel	Pre-heater and cooler in air handling unit
Local application Pub, disco, restaurant	Kraft, PVC aluminum	Split type air conditioning systems
Hospital	Epoxy-coated aluminum, stainless steel	Unwanted areas of the exhaust mixture, in laminar box
Indoor swimming pools	Epoxy-coated aluminum, stainless steel	Pre-heater, cooler and humidity controller in part where the flow of fresh air in air handling unit
Underground Garage	Aluminum, epoxy-coated aluminum	Heater in air handling unit or in duct
Textile and chemistry industry	Aluminum, epoxy-coated aluminum, stainless steel	Pre-heater, cooler and humidity controller in part where the flow of fresh air in air handling unit
Drying Processes	Aluminum, stainless steel, PVC	Pre-heater in air drying applications
Printing plants	Epoxy-coated aluminum, stainless steel	Pre-heater and cooler

3.5.1 In Air Handling Units (AHU)

Air flow range is between 5000 and 80 000 m³ / h in AHU applications. Central air conditioning heat exchangers are placed into a flat or can be used as a diagonal. Diagonal layout is an application which could be described as a two-storey air-conditioning plant as shown in the Figure 3.10b. Double storey AHU practices, fresh and exhausts air damper circuit by-pass to put the unit in the transition period permitted by atmospheric conditions or circumstances where it passes in front more energy than necessary. Heat recovery exchangers' the main function in AHU is to behave as pre-heating or pre-cooler. Considering an air handling unit in the heat recovery applications has all the functions (filtering, mixing, heating, cooling, humidification, etc..), air handling unit costs by 40% ~ 60% of the unit from the system checks the total energy of 5% ~ 15% at the level increases.



a b

Figure 3.10 Placement of heat recovery exchanger; (a)In a single-storied air-conditioning plants, (b) In a double-ply air-conditioning plants (Sahan, 1999)

Heat recovery exchangers can also be used in single-storied air handling units, as shown in the Figure3.10a. These applications are designed as compact devices with last-cooler which works with 100% fresh air. Height of the AHU is more than the normal air-conditioning plant, less than two-ply. In single storied AHU or in double-ply AHU, heat flux on exhaust and fresh air ducts is shown in Figure 3.11.

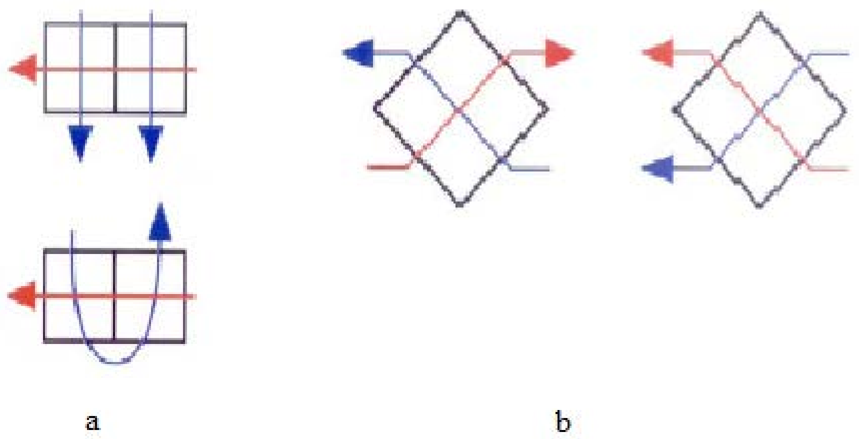


Figure 3.11 Flow patterns of heat recovery exchanger (a) Single-storied air handling unit, (b) Double-ply air handling unit (Sahan, 1999)

3.5.2 On Ventilation Ducts

Plate heat recovery exchangers can be mounted on duct through mechanism as shown in Figure 3.12. In this kind of application, the most important issue is to prevent suffusing of condensate along the duct and take off the condensate out of the duct.

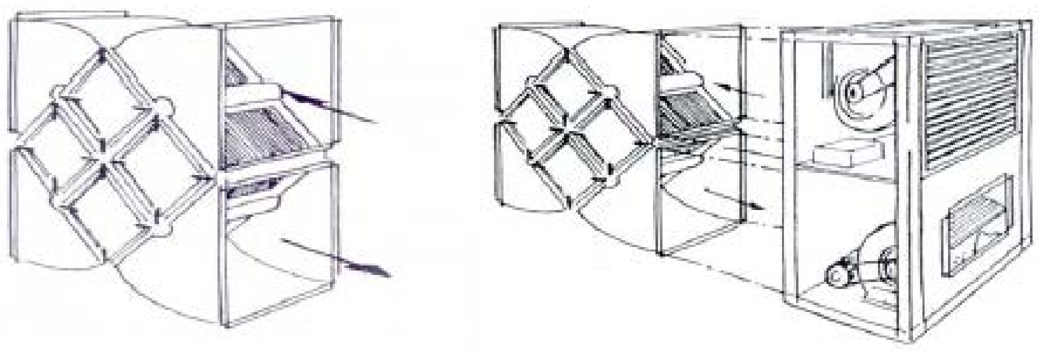


Figure 3.12 Heat recovery exchangers mounted on ventilation ducts. (Sahan, 1999)

The double-ply air handling units' height can create problems. In these applications, placement on duct is overcome this problem. The heat recovery exchanger must be mounted on intake ducts before air handling unit.

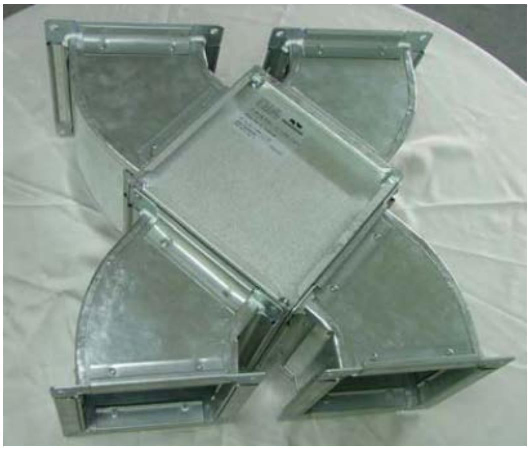


Figure 3.13 Heat recovery exchangers mounted on ventilation ducts. (Immak Heat Recovery Exchanger Catalogue)

3.5.3 Heat Recovery Device

Applications which have 500 ~ 6000 m^3 / h air flow rate in the range are widely used. Less air flow rate can be used as fan-coil units with ducts, great air flow rate can be used instead of the roof air-conditioning equipment. The lacking heating capacity is supplied by electric heaters or heating coils (80/60 or 60/50 ° C). The lacking cooling capacity is supplied by DX evaporator or cooling coils (12/7 ° C). Indoor air practical air conditioner running at 100% in places, resulting in meeting the need for fresh air, outside air to the burden caused by the lack of capacity are the ideal solution to eliminate. Materials with special license plates to be elected, is to prevent moisture loss of volume.

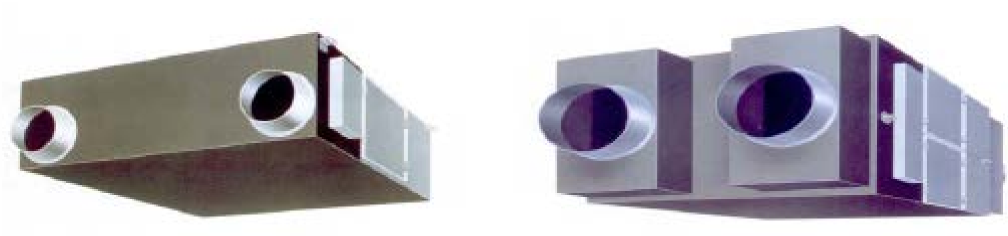


Figure 3.14 Heat recovery device. (Immak Heat Recovery Exchanger Catalogue)

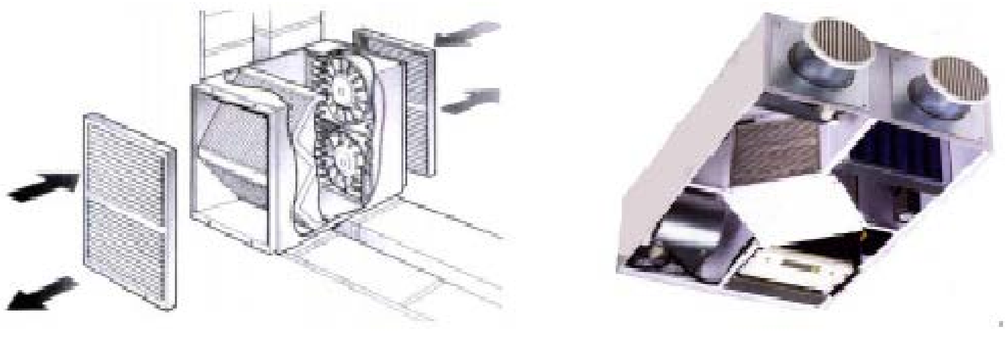
3.5.4 Roof Type Applications

Some places, such as warehouses, hangars, underground garages and large factory, ventilate because of the fresh air requirements. These heat recovery units are designed to use exhaust air energy. These units are used for heating and controlling humidity rather than cooling. Therefore, heat recovery plates are made of kraft and PVC.

Figure 3.15 Roof type applications (Eneko, Heat Recovery Ventilation Catalogue)

3.5.5 Residential Applications

Housing type developed devices for HVAC applications, as well as in air conditioning equipment, window air conditioners are devices or duct-type air conditioner device features. Window type devices (Frivent) 50 to 600 m³ / h air flow rate range, while others (HRV, heat recovery ventilator) 500 and 6000 m³ / h are used in a range.



a b

Figure 3.16 Residential applications (a) Frivent, (b) Heat Recovery Ventilator (HRV) (Sahan, 1999)

CHAPTER FOUR
BASIC DESIGN METHODS OF HEAT EXCHANGERS

4.1 Heat Exchanger Design Methodology

Design is an activity aimed at providing complete descriptions of an engineering system, part of a system, or just of a single system component. These descriptions represent an unambiguous specification of the system/component structure, size, and performance, as well as other characteristics important for subsequent manufacturing and utilization. This can be accomplished using a well-defined design methodology.

From the formulation of the scope of this activity, it must be clear that the design methodology has a very complex structure. Moreover, a design methodology for a heat exchanger as a component must be consistent with the life-cycle design of a system. Lifecycle design assumes considerations organized in the following stages.

- Problem formulation
- Concept development (selection of workable designs, preliminary design)
- Detailed exchanger design (design calculations and other pertinent considerations)
- Manufacturing
- Utilization considerations (operation, phase-out, disposal)

A methodology for designing a new (single) heat exchanger is illustrated in Figure 4.1; it is based on experience and presented by Kays, & London (1998), Taborek (1988), and Shah (1982) for compact and shell-and-tube exchangers. This design procedure may be characterized as a case study (one case at a time) method. Major design considerations include:

- Process and design specifications
- Thermal and hydraulic design
- Mechanical design

- Manufacturing considerations and cost
- Trade-off factors and system-based optimization (Shah, 2003)

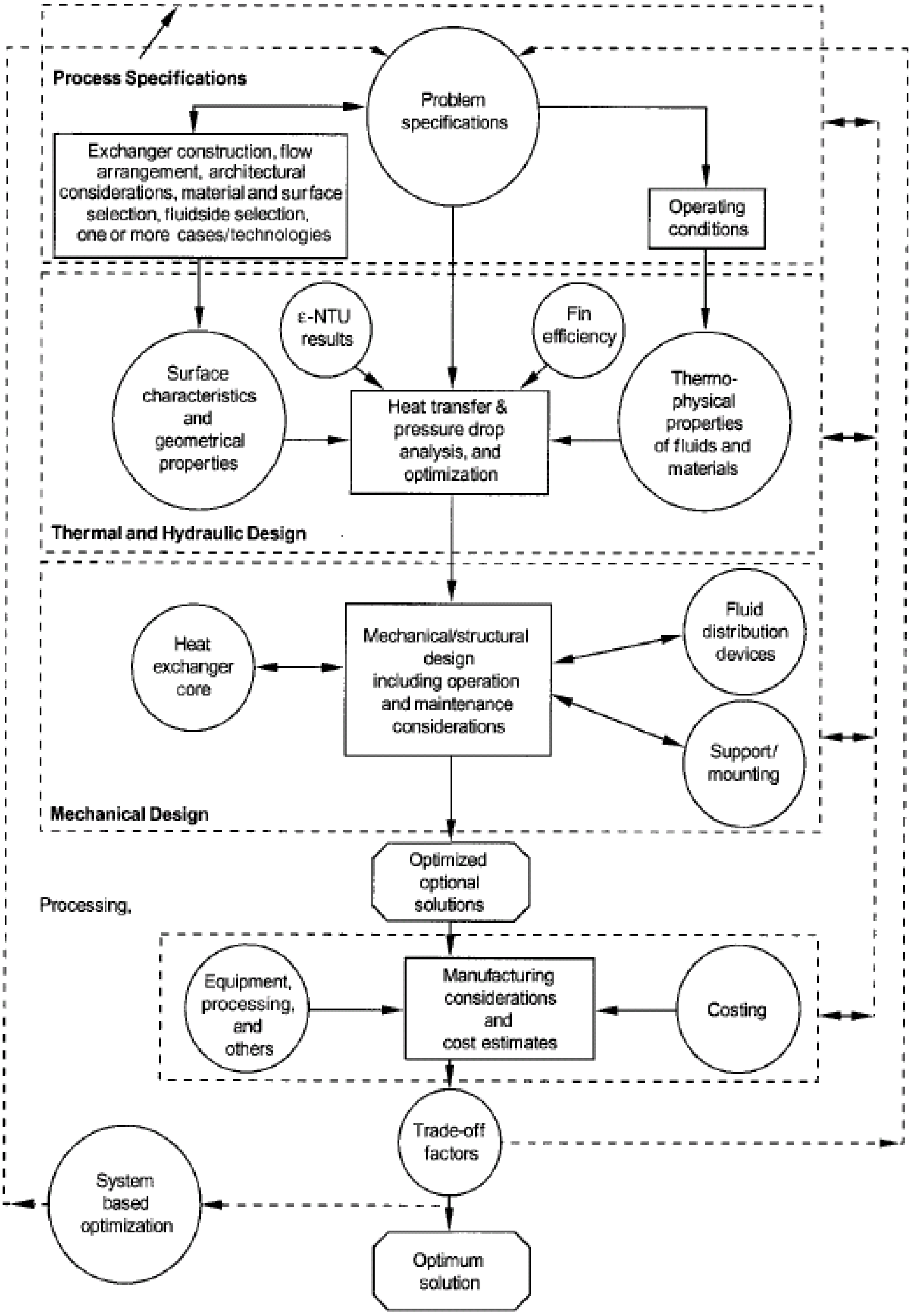


Figure 4.1 Heat exchanger design methodology (Shah, 2003)

4.2 Basic Equations in Design

The goal of heat exchanger design is to relate the inlet and outlet temperatures, the overall heat transfer coefficient, and the geometry of the heat exchanger, to the rate of heat transfer between the two fluids. The two most common heat exchanger design problems are those of rating and sizing.

From the first law of the thermodynamics for an open system, under steady-state, steady flow conditions, with negligible potential and kinetic energy changes, the change of enthalpy of one of the fluid stream is Figure 4.2.

$Q = m\ (h_2 - h_1)$

(4.1)

Q: Heat transfer capacity (W)

m: Mass flow (kg/s)

h_1: Specific enthalpy (j/kg)

h_2: Specific enthalpy (j/kg)

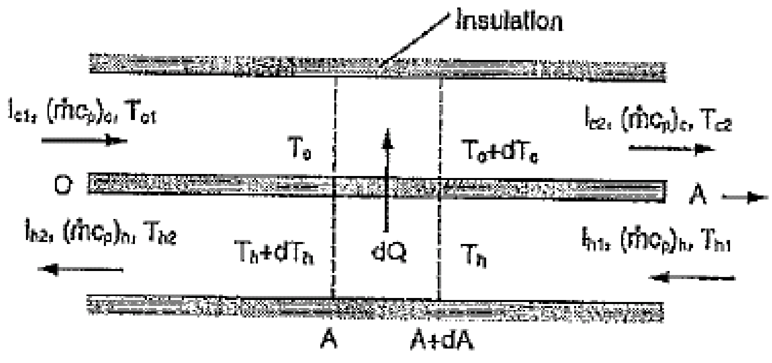


Figure 4.2 Overall energy balance for the hot and cold fluids of a two-fluid heat exchanger (Kakac, 2002)

Equation 4.1 holds for all processes of Figure 4.3. Note that δQ is negative for the hot fluid. If there is negligible heat transfer between the exchanger and its surroundings (adiabatic process), integration of Equation 4.1 for hot and cold fluids gives.

$Q = m\,(h_{c2} - h_{c1})$

(4.2)

$Q = m\,(h_{h1} - h_{h1})$

(4.3)

Q: Heat transfer rate (W)

m: Mass flow (kg/s)

h_{c1}: Cold fluid stream inlet specific enthalpy (j/kg)

h_{c2}: Cold fluid stream outlet specific enthalpy (j/kg)

h_{h1}: Hot fluid stream inlet specific enthalpy (j/kg)

h_{h2}: Hot fluid stream outlet specific enthalpy (j/kg)

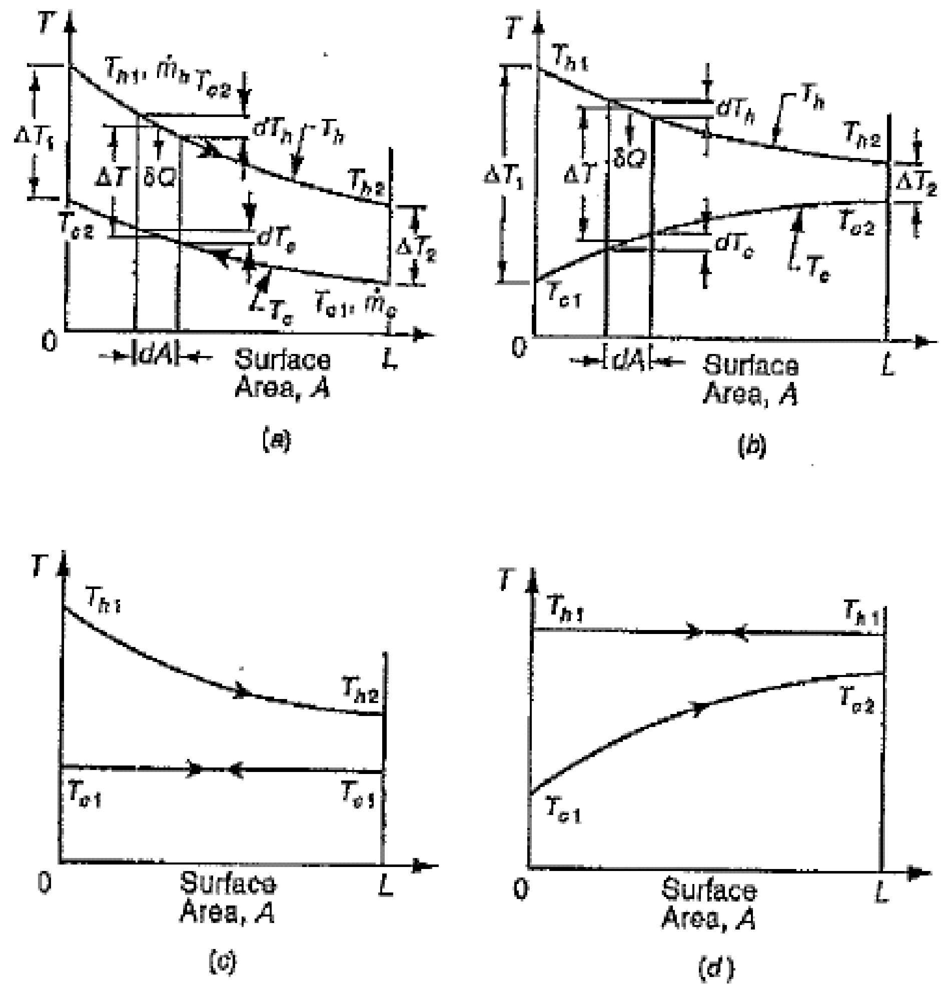


Figure 4.3 Fluid temperature variation in parallel flow, counter flow, evaporator and condenser heat exchangers, (a) Counter flow; (b) parallel flow; (c) cold fluid evaporating at constant temperature; (d) hot fluid condensing at constant temperature (Kakac, 2002)

If the fluids do not undergo a phase change and have constant specific heat, then Equations (4.4) and (4.5) can be written as:

$Q = (m\ c_p)_c\ (T_{c2} - T_{c1})$

(4.4)

$Q = (m\ c_p)_h\ (T_{h1} - T_{h2})$

(4.5)

$c_{p,h}$: Hot fluid specific heat at constant pressure

$c_{p,c}$: Cold fluid specific heat at constant pressure

T_{c1}: Cold fluid inlet temperature

T_{c2}: Cold fluid outlet temperature

T_{h1}: Hot fluid inlet temperature

T_{h2}: Hot fluid outlet temperature

As can be seen from Figure 4.3, the temperature difference between the hot and cold fluids ($\Delta T = T_h - T_c$) varies with position in the heat exchanger. Therefore, in the heat transfer analysis of heat exchangers, it is convenient to establish an appropriate mean value of the temperature between the hot and cold fluids such that the total heat transfer rate Q between the fluids can be determined from the following equation:

$$Q = U\ A\ \Delta T_m \qquad (4.6)$$

Q: Heat transfer rate (W)

A: The total hot-side or cold-side heat transfer area (m^2)

U: Average overall heat transfer coefficient based on area (W/m^2K)

ΔT: A function of T_{h1}, T_{h2}, T_{c1}, T_{c2} (K)

4.3 Overall Heat Transfer Coefficient

To understand the exchanger overall heat transfer rate Equation (4.6), consider the thermal circuit model of Figure 4.4. Scale or fouling deposit layers are also shown on each side of the wall.

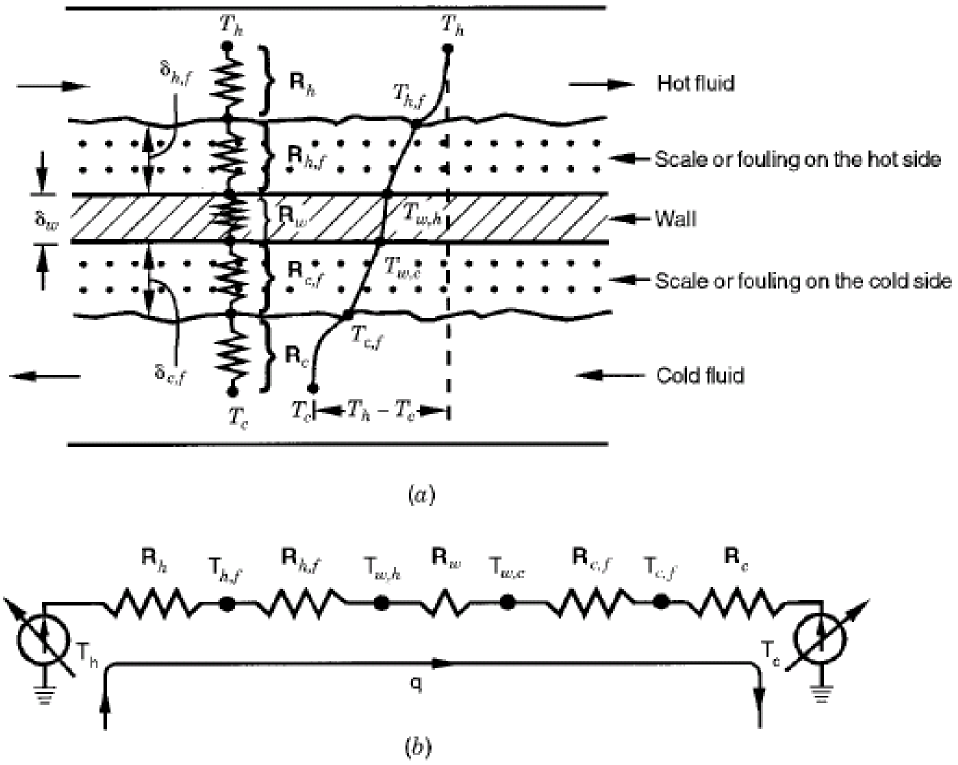


Figure 4.4 (a) Thermal resistances; (b) thermal circuit for a heat exchanger (Shah, 2003).

In the steady state, heat is transferred from the hot fluid to the cold fluid by the following processes: convection to the hot fluid wall, conduction through the wall, and subsequent convection from the wall to the cold fluid. In many heat exchangers, a fouling film is formed as a result of accumulation of scale or rust formation, deposits from the fluid, chemical reaction products between the fluid and the wall material, and/or biological growth. This undesired fouling film usually has a low thermal conductivity and can increase the thermal resistance to heat flow from the hot fluid to the cold fluid. Where the overall differential thermal resistance dRo consists of component resistances in series (similar to those shown in Fig. 4.4b for a heat exchanger):

$$\frac{1}{UdA} = dR_0 = dR_h + dR_{h,f} + dR_w + dR_c + dR_{c,f} \tag{4.7}$$

U: Overall heat transfer coefficient (W/m^2K)

R_0: Overall differential thermal resistance

R_h: Hot-fluid-side convection resistance

$R_{h,f}$: Hot-fluid-side fouling resistance

R_c: Cold-fluid-side convection resistance

$R_{c,f}$: Cold-fluid-side fouling resistance

$$\frac{1}{UdA} = \frac{1}{(\eta_0 hdA)_h} + \frac{1}{(\eta_0 h_f dA)_h} + dR_W + \frac{1}{(\eta_0 hdA)_c} + \frac{1}{(\eta_0 h_f dA)_c} \tag{4.8}$$

$$1/h_f = \Delta\delta_f / k_f \tag{4.9}$$

$\Delta\delta_f$: The thickness of the fouling layer

k_f: The thermal conductivity of the fouling layer

For constant and uniform U and h's throughout the exchanger, Equations (4.8) and (4.11) are identical (Shah, 2003).

$$\frac{1}{UA} = R_0 = R_h + R_{h,f} + R_w + R_c + R_{c,f} \tag{4.10}$$

$$\frac{1}{UA} = \frac{1}{(\eta_0 hA)_h} + \frac{1}{(\eta_0 h_f A)_h} + dR_W + \frac{1}{(\eta_0 hA)_c} + \frac{1}{(\eta_0 h_f A)_c} \tag{4.11}$$

The wall thermal resistance R_w for flat walls is given by, (Shah, 2003)

$$R_w = \begin{cases} \dfrac{\delta_w}{k_w A_w} & \text{for flat walls with single layer wall} \\[2ex] \sum_j \left(\dfrac{\delta_w}{k_w A_w}\right)_j & \text{for flat walls with a multiple-layer wall} \end{cases} \tag{4.12}$$

For a cylindrical (tubular) wall, it is given as (Shah, 2003),

$$\begin{cases} \dfrac{\ln(d_o / d_i)}{2\pi L N_t} & \text{for } N_t \text{ circular tubes with single layer wall} \end{cases}$$

$R_w =$ (4.13)

$$\frac{1}{2\pi L N_t}\sum_j \frac{\ln(d_{j+1}/d_j)}{k_{WJ}}$$ for N_t circular tubes with a multiple-layer wall

$\delta_{w:}$ The wall plate thickness (mm)

A_w: The total wall area of all flat walls for heat conduction (m^2)

k_w: The thermal conductivity of the wall material (W/mK)

d_o: The tube outside diameter (mm)

d_i: The tube inside diameter (mm)

L: The tube or exchanger length (mm)

N_t : The number of tubes (mm)

A flat (or plain) wall is generally associated with a plate-fin or an all-prime-surface plate exchanger. In this case,

$A_w = L_1 L_2 N_p$

(4.14)

$L_{1:}$ The length of separating planes

$L_{2:}$ The width of separating planes

N_p: The total number of separating planes

4.4 Thermal Effectiveness (ε)

Effectiveness, ε is a measure of thermal performance of a heat exchanger. It is defined for a given heat exchanger of any flow arrangement as a ratio of the actual heat transfer rate from the hot fluid to the cold fluid to the maximum possible heat transfer rate q_{max} thermodynamically permitted:

$$\varepsilon = \frac{\text{Actual Heat Transfer Rate}}{\text{Theoretical Maximum Heat Transfer Rate}} \quad (4.15)$$

$$\varepsilon = \frac{q}{q_{max}} \quad (4.16)$$

Here it is idealized that there are no flow leakages from one fluid to the other fluid, and vice versa. If there are flow leakages in the exchanger, q represents the total enthalpy gain (or loss) of the C_{min} fluid corresponding to its actual flow rate in the outlet (and not inlet) stream. q_{max} would be obtained in a ‘‘perfect’’ counterflow heat exchanger (recuperator) of infinite surface area, zero longitudinal wall heat conduction, and zero flow leakages from one fluid to the other fluid, operating with fluid flow rates and fluid inlet temperatures the same as those of the actual heat exchanger. This perfect exchanger is the ‘‘meterbar’’ used in measuring the degree of perfection of actual exchanger performance. The value of ε ranges from 0 to 1. Thus ε is like an efficiency factor and has thermodynamic significance. As shown below, in such a perfect heat exchanger, the exit temperature of the fluid with the smaller heat capacity will reach the entering temperature of the larger heat capacity fluid.

Based on a counterflow heat exchanger having infinite surface area, an overall energy balance for the two fluid streams is ;

$$q = C_h (T_{h,i} - T_{h,o}) = C_c (T_{c,o} - T_{c,i})$$
(4.17)

C_h : Heat capacity rate of hot fluid
C_c : Heat capacity rate of cold fluid

Based on this equation, for $C_h < C_c$, $(T_{h,i} - T_{h,o}) > (T_{c,o} - T_{c,i})$. The temperature drop on the hot fluid side will thus be higher, and over the infinite flow length the hot fluid temperature will approach the inlet temperature of the cold fluid as shown by the two bottom curves in Figure 4.5, resulting in $T_{h,o} = T_{c,i}$. Thus for an infinite area counterflowexchanger with $C_h < C_c$, we get q_{max} as ;

$$q_{max} = C_h (T_{h,i} - T_{c,i}) = C_h \Delta T_{max}$$
(4.18)

Similarly, for $C_h = C_c = C$;

$q_{max} = C_h (T_{h,i} - T_{c,i}) = C_c (T_{h,i} - T_{c,i}) = C \Delta T_{max}$
(4.19)

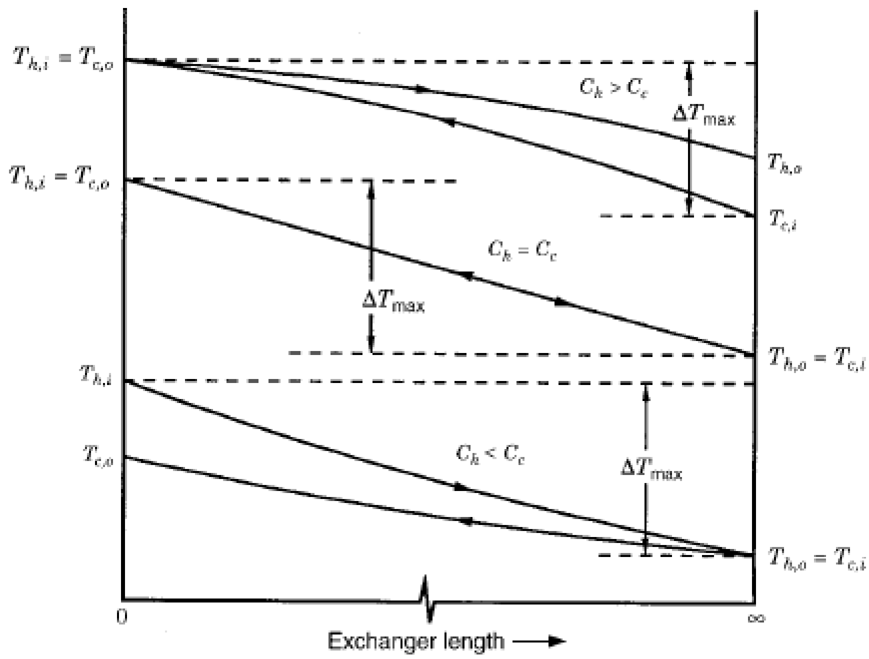


Figure 4.5 Temperature distributions in a counterflow exchanger of infinite surface area (Shah, 2003)

Based on Equation (4.17), for $C_h > C_c$, $(T_{c,o} - T_{c,i}) > (T_{h,i} - T_{h,o})$. Hence, Tc,o will approach $T_{h,i}$ over the infinite length, and therefore,

$q_{max} = C_c (T_{h,i} - T_{c,i}) = C_c \Delta T_{max}$
(4.20)

Or, more generally, based on Equations. (4.18) throught (4.20),

$q_{max} = C_{min} (T_{h,i} - T_{c,i}) = C_{min} \Delta T_{max}$
(4.21)

where,

$$C_{min} = \begin{cases} C_c & \text{for } C_c < C_h \\ C_h & \text{for } C_h < C_c \end{cases} \qquad (4.22)$$

Using the value of actual heat transfer rate q from the energy conservation equation and qmax from Equation (4.21), the exchanger effectiveness of Equation (4.16) valid for all flow arrangements of the two fluids is given by,

$$\varepsilon = \frac{C_h(T_{h,i} - T_{h,o})}{C_{\min}(T_{h,i} - T_{c,i})} = \frac{C_c(T_{c,o} - T_{c,i})}{C_{\min}(T_{h,i} - T_{c,i})} \qquad (4.23)$$

4.5 Heat Capacity Rate Ratio (C^*)

C^* is simply a ratio of the smaller to larger heat capacity rate for the two fluid streams so that $C^* \leq 1$. A heat exchanger is considered balanced when $C^* = 1$;

$$C^* = \frac{C_{\min}}{C_{mac}} = \frac{(\dot{m}c_p)_{\min}}{(mc_p)_{mac}} = \begin{cases} (T_{c,o} - T_{c,i})/(T_{h,i} - T_{h,o}) \Rightarrow \; for C_h = C_{\min} \\ (T_{h,i} - T_{h,o})/(T_{c,o} - T_{c,i}) \Rightarrow \; for C_c = C_{\min} \end{cases} \qquad (4.24)$$

C* is a heat exchanger operating parameter since it is dependent on mass flow rates and/or temperatures of the fluids in the exchanger. The C_{max} fluid experiences a smaller temperature change than the temperature change for the Cmin fluid in the absence of extraneous heat losses, as can be found from the energy balance:

$$q = C_h \, \Delta T_h = C_c \, \Delta T_c \qquad (4.25)$$

where the temperature ranges ΔTh and ΔTc are:

$$\Delta T_h = T_{h,i} - T_{h,o}$$
(4.26)
$$\Delta T_c = T_{c,o} - T_{c,i}$$
(4.27)

4.6 LMTD Method for Heat Exchanger Analysis

4.6.1 Parallel and Counterflow Heat Exchangers

In the heat transfer analysis of heat exchangers, the total heat transfer rate Q through the heat exchanger is the quantity of primary interest. In a simple counterflow or parallel flow heat exchanger (Figure 4.3a and b), the form of ΔT_m in Equation 4.6 may be determined by applying an energy balance to a differential area element dA in the hot and cold fluids. The temperature of the hot fluid drop will be dT_h. The temperature of the cold fluid drop will be dT_c, over the element dA for counterflow, but it will increase by dT_c for parallel flow if the hot fluid direction is taken as positive. For adiabatic, steady-state, steady flow, energy balance yields,

$$\delta Q = - C_h \, dT_h = \pm C_c \, dT_c$$
(4.28)

The log mean temperature difference (LMTD) is derived in all basic heat transfer texts. It may be written for a parallel flow or counterflow arrangement. The LMTD has the form:

$$\Delta T_{LMTD} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \tag{4.29}$$

$$Q = UA\frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} \tag{4.30}$$

ΔT_1: The temperature difference between two fluids at one end of the heat exchanger
ΔT_2: The temperature difference between two fluids at the other end of the heat exchanger

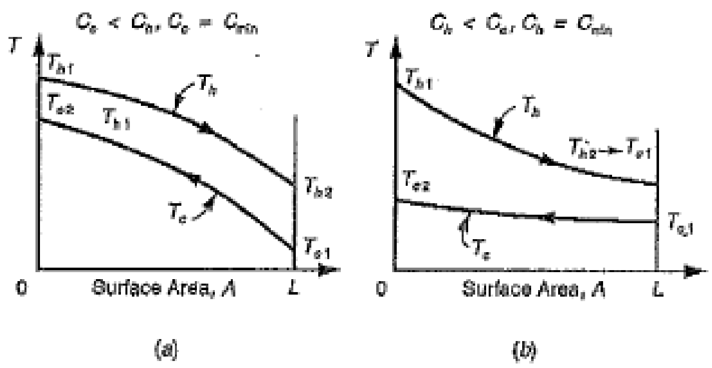


Figure 4.6 Temperature variation for a counterflow heat exchanger (Kakac,2002)

4.6.2 Multipass and Cross-flow Heat Exchangers

The LMTD developed previously is not applicable for the heat transfer analysis of cross-flow and multipass flow heat exchangers (Kakac, 2002).

$$Q = U\, A\, \Delta T_m \tag{4.31}$$

ΔT_m : The effective mean temperature difference (K)

Generally, this function ΔT_m can be determined analytically in terms of the following quantities.

$$\Delta T_{lm,ef} = \frac{(T_{h2} - T_{c1}) - (T_{h1} - T_{c2})}{\ln\left[(T_{h2} - T_{c1})/(T_{h1} - T_{c2})\right]} \tag{4.32}$$

$\Delta T_{lm,ef}$: The log-mean temperature difference for counterflow

$$P = \frac{T_{c2} - T_{c1}}{T_{h1} - T_{c1}} = \frac{\Delta T_c}{\Delta T_{max}} \tag{4.33}$$

P: The temperature effectiveness of the heat exchanger on the cold-fluid side.

$$R = \frac{C_c}{C_h} = \frac{T_{h1} - T_{h2}}{T_{c2} - T_{c1}} \tag{4.34}$$

R: The heat capacity rate ratio

For design purposes, Equation 4.32 can also be used for multipass and cross-flow heat exchangers by multiplying ΔT_{lm}, which would be computed under the assumption of a counterflow arrangement with correction factor F.

$$Q = U\ A\ \Delta T_{lm} = U\ A\ F\ \Delta T_{lm,ef}$$
(4.35)

F is non-dimensional; it depends on the temperature effectiveness P, the heat capacity rate ratio R and the flow arrangement:

$$F = \Phi\ (P, R, \text{flow arrangement}) \tag{4.36}$$

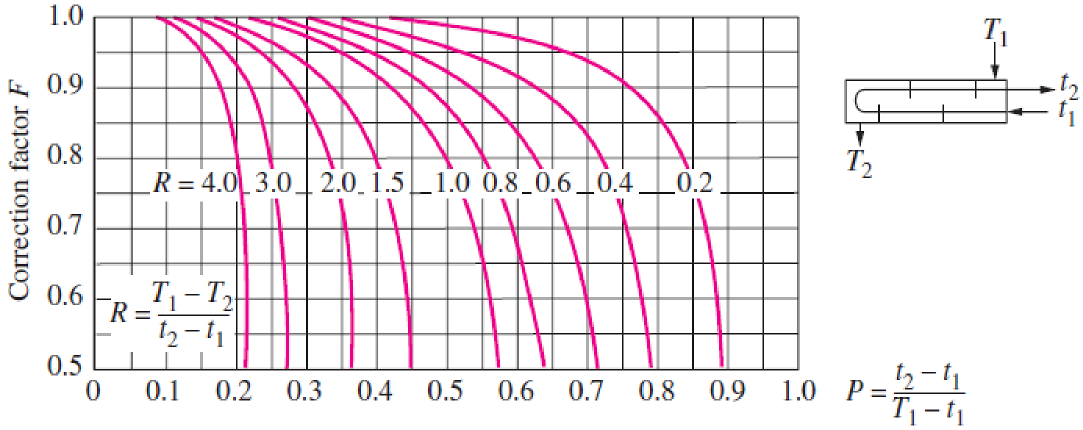


Figure 4.7 Correction factor F chart for one-shell pass and 2, 4, 6 etc. (any multiple of 2), tube passes (Cengel, 2006)

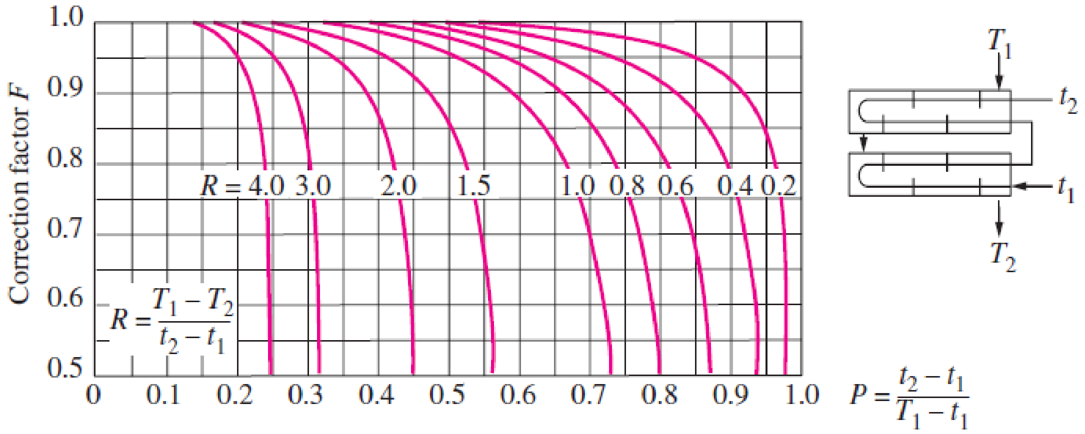


Figure 4.8 Correction factor F chart for two-shell pass and 4, 8, 12 etc. (any multiple of 4), tube passes (Cengel, 2006)

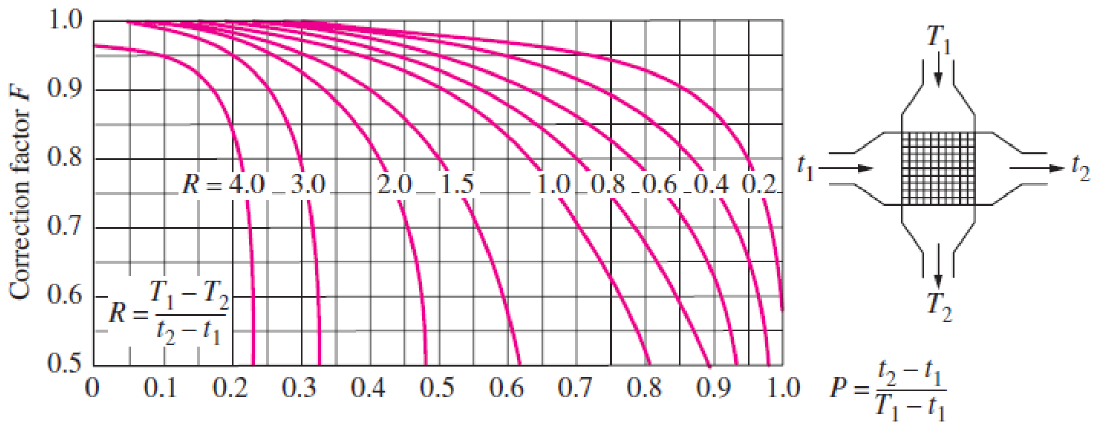


Figure 4.9 Correction factor F chart for single-pass cross-flow with both fluids unmixed (Cengel, 2006)

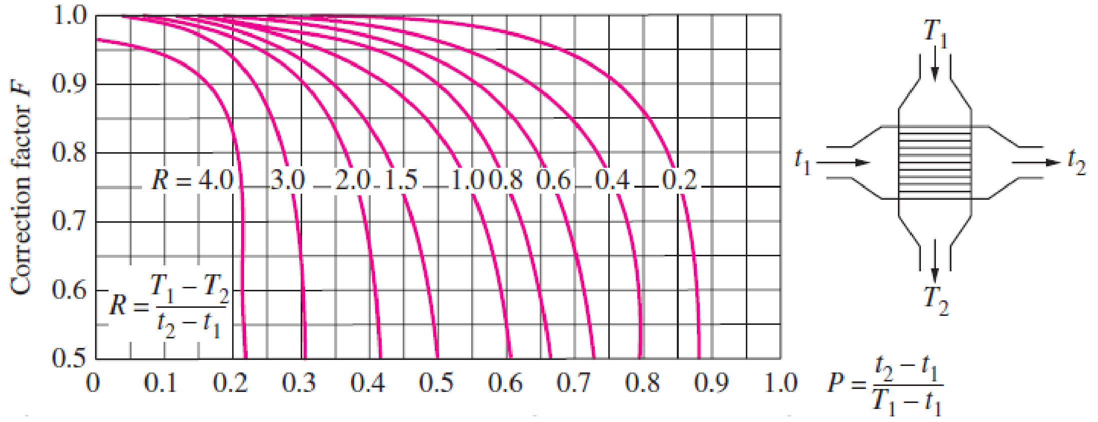


Figure 4.10 Correction factor F chart for single-pass cross-flow with one fluid mixed and the other unmixed (Cengel, 2006)

4.7 The ε-NTU Method for Heat Exchanger Analysis

The number of transfer units NTU is defined as a ratio of the overall thermal conductance to the smaller heat capacity rate:

$$NTU = \frac{UA}{C_{\min}} = \frac{1}{C_{\min}} \int_A U dA \tag{4.37}$$

If U is not a constant, the definition of the second equality applies. NTU may also be interpreted as the relative magnitude of the heat transfer rate compared to the rate of enthalpy change of the smaller heat capacity rate fluid.

$$NTU = \frac{1}{C_{\min}} \frac{1}{R_h + R_{h,f} + R_w + R_{c,f} + R_c} \tag{4.38}$$

NTU designates the nondimensional heat transfer size or thermal size of the exchanger, and therefore it is a design parameter. NTU provides a compound measure of the heat exchanger size through the product of heat transfer surface area A and the overall heat transfer coefficient U.

The effectiveness of a heat exchanger depends on the geometry of the heat exchangers as well as the flow arrangement. Therefore different types of heat exchangers have different effectiveness relations.

4.7.1 Parallel Flow

It can be shown that the effectiveness of a heat exchager is a function of the number of transfer units NTU and the heat capacity ratio C^* (Incropera, 2003).

$$\varepsilon = \frac{1-\exp[-NTU(1+C^*)]}{1+C^*} \tag{4.39}$$

or

$$NTU = -\frac{\ln[1-\varepsilon(1+C^*)]}{1+C^*} \tag{4.40}$$

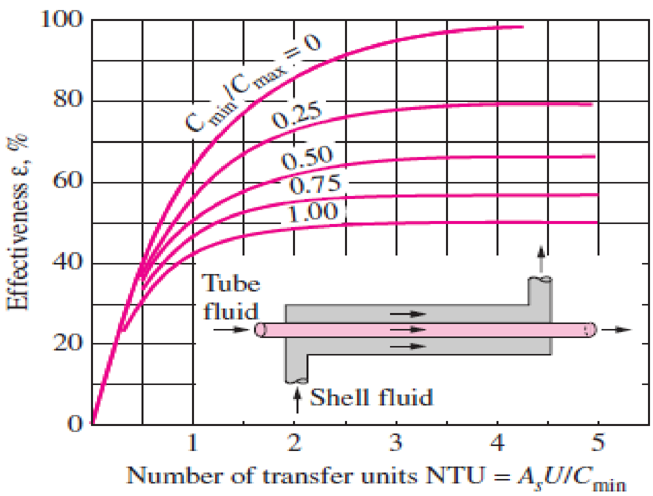


Figure 4.11 Effectiveness of paralel flow heat exchanger (Cengel, 2006)

4.7.2 Counter Flow

The heat transfer effectiveness and NTU relations are given.

$$\varepsilon = \frac{1-\exp[-NTU(1-C^*)]}{1-C^*\exp[-NTU(1-C^*)]} \qquad (C^* < 1) \tag{4.41}$$

$$\varepsilon = \frac{NTU}{1 + NTU} \qquad (C^*=1) \tag{4.42}$$

or

$$NTU = \frac{1}{C^* - 1} \ln\left(\frac{\varepsilon - 1}{\varepsilon C^* - 1}\right) \qquad (C^* < 1) \tag{4.43}$$

$$NTU = \frac{\varepsilon}{1 - \varepsilon} \qquad (C^*=1) \tag{4.44}$$

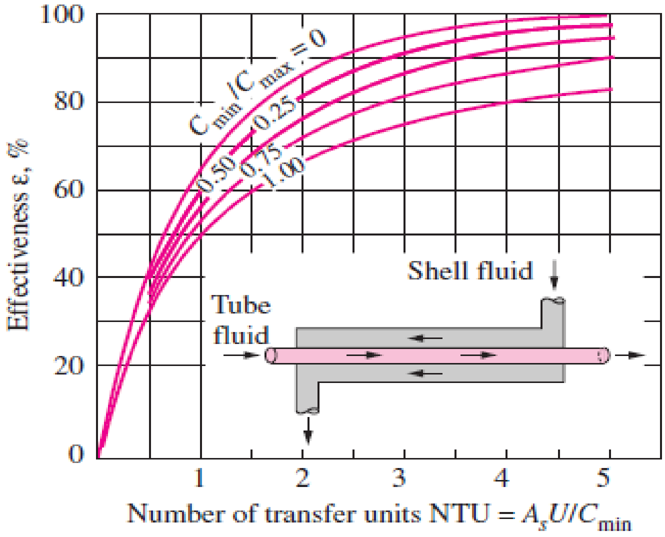


Figure 4.12 Effectiveness of counter flow heat exchanger (Cengel, 2006)

4.7.3 Shell-and-tube Heat Exchanger (One shell pass)

The heat transfer effectiveness and NTU relations are given.

$$\varepsilon = 2\left\{1 + C^* + (1 + (C^*)^2)^{1/2} \times \frac{1 + \exp[-(NTU)_1 (1 + (C^*)^2)^{1/2}]}{1 - \exp[-(NTU)_1 (1 + (C^*)^2)^{1/2}]}\right\}^{-1} \tag{4.45}$$

or,

$$(NTU)_1 = -\left[1 + (C^*)^2\right]^{-1/2} \ln\left(\frac{E - 1}{E + 1}\right) \tag{4.46}$$

where,

$$E = \frac{2/\varepsilon_1 - (1+C^*)}{\left[1+(C^*)^2\right]^{1/2}} \tag{4.47}$$

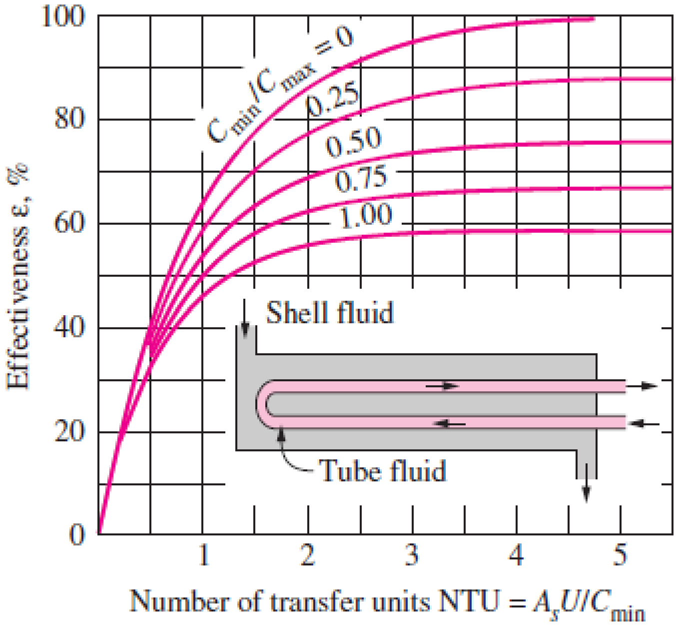


Figure 4.13 Effectiveness of shell and tube heat exchanger (one shell passes) (Cengel, 2006)

4.7.4 Shell-and-tube Heat Exchanger (n shell passes)

The heat transfer effectiveness and NTU relations are given for 2n,4n,.. tube passes shell and tube heat exchanger (Incropera, 2003).

$$\varepsilon = \left[\left(\frac{1-\varepsilon_1 C^*}{1-\varepsilon_1}\right)^n - 1\right]\left[\left(\frac{1-\varepsilon_1 C^*}{1-\varepsilon_1}\right)^n - C^*\right]^{-1} \tag{4.48}$$

or,

NTU = n (NTU)1 (4.49)

where,

$$F = \left(\frac{\varepsilon C^* - 1}{\varepsilon - 1} \right)^{1/n} \tag{4.50}$$

and,

$$\varepsilon_1 = \frac{F - 1}{F - C^*} \tag{4.51}$$

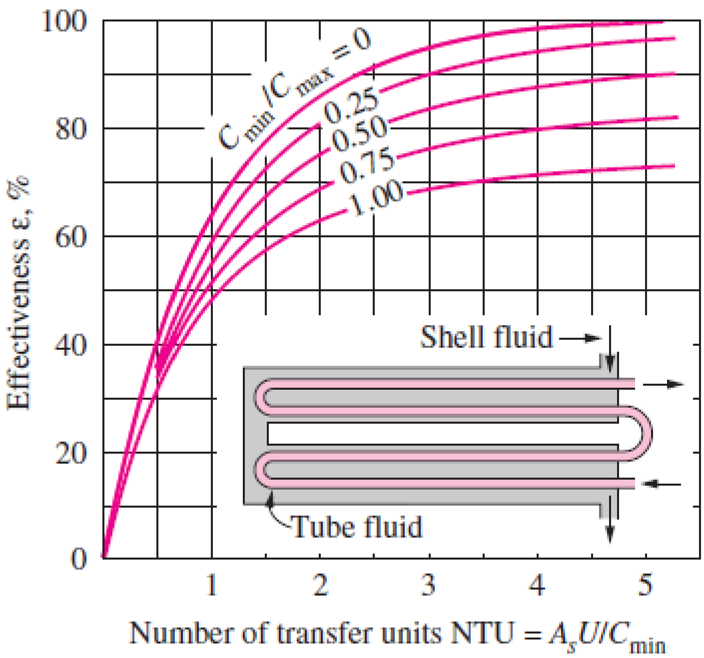


Figure 4.14 Effectiveness of shell and tube heat exchanger (two shell passes and 4,8,12,…tube passes (Cengel,2006)

4.7.5 Cross-flow Heat Exchanger with Both Fluids Unmixed (Single pass)

The heat transfer effectiveness and NTU relations are given for cross-flow heat exchanger (Incropera, 2003).

$$\varepsilon = 1 - \exp\left[\left(\frac{1}{C^*} \right) (NTU)^{0,22} \left\{ \exp[-C^* (NTU)^{0,78}] - 1 \right\} \right] \tag{4.52}$$

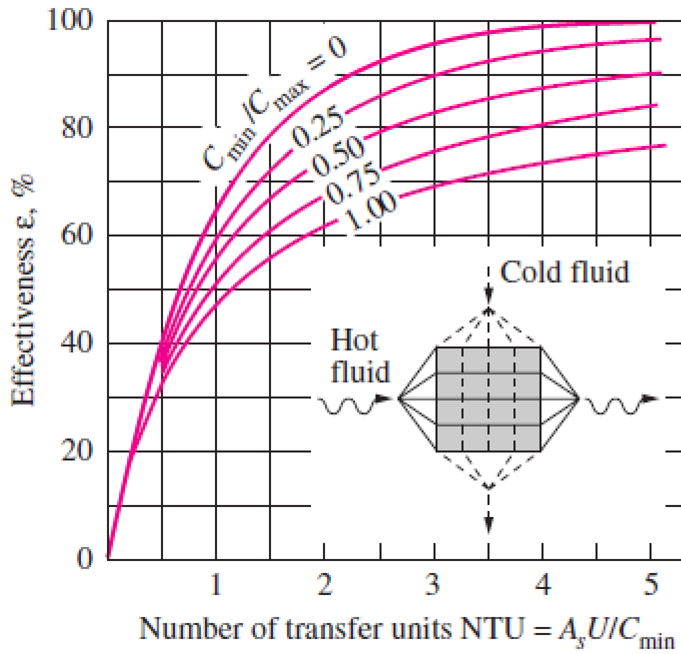


Figure 4.15 Effectiveness of cross-flow heat exchanger with both fluids mixed (Cengel, 2006)

4.7.6 Cross-Flow Heat Exchanger with One Fluid Mixed and The Other Unmixed (Single Pass)

The heat transfer effectiveness and NTU relations are given for cross-flow heat exchanger (Incropera, 2003).

Considering C_{max} (mixed), C_{min} (Unmixed);

$$\varepsilon = \left(\frac{1}{C^*}\right)(1 - \exp\left\{- C^*[1 - \exp(- \exp(-NTU)]\right\} \tag{4.53}$$

or,

$$NTU = -\ln\left[1 + \left(\frac{1}{C^*}\right)\ln(1 - \varepsilon C^*)\right] \tag{4.54}$$

Considering C_{min} (mixed), C_{max} (Unmixed);

$$\varepsilon = 1 - \exp(-(C^*)^{-1}\left\{1 - \exp[-C^*(NTU)]\right\}) \tag{4.55}$$

or,

$$NTU = -\left(\frac{1}{C^*}\right)\ln\left[C^*\ln(1-\varepsilon)+1\right] \tag{4.56}$$

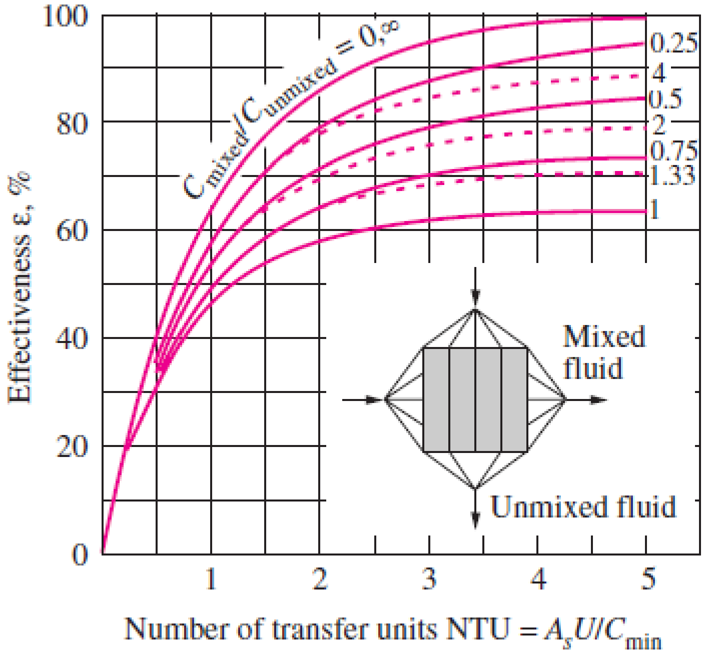


Figure 4.16 Effectiveness of cross-flow heat exchanger with one fluid mixed and the other unmixed (Cengel, 2006)

4.7.7 Cross-flow Heat Exchanger with Both Fluids Mixed (Single pass)

The heat transfer effectiveness and NTU relations are given for cross-flow heat exchanger (Shah, 2003).

$$\varepsilon = \frac{1}{\dfrac{1}{1-\exp(-NTU)} + \dfrac{C^*}{1-\exp(-NTU \times C^*)} - \dfrac{1}{NTU}} \tag{4.57}$$

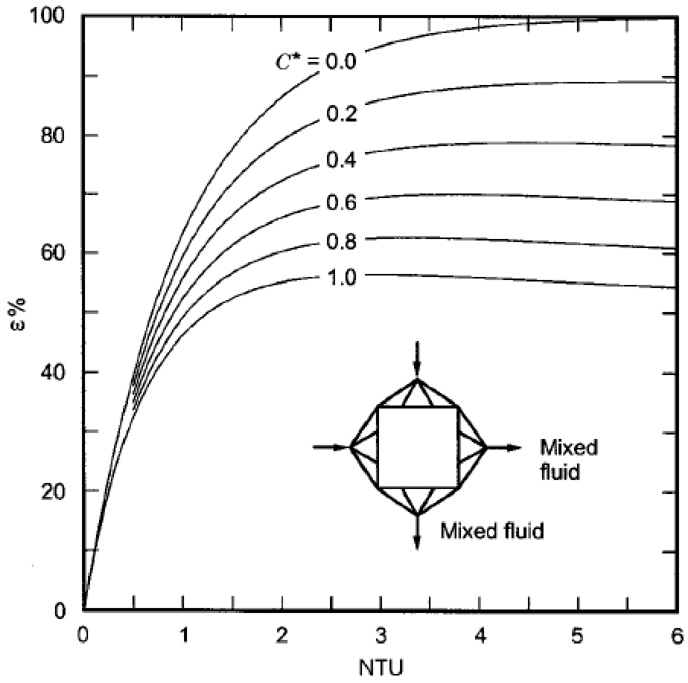


Figure 4.17 Effectiveness of cross-flow heat exchanger with both fluids mixed (Shah, 2003)

The air volume flow rate is determined as 1,08 m^3/h for cross-flow heat recovery exchanger at 6m/s velocity. Overall heat transfer coefficient is calculated as:

U = 14,56 W/m^2K

The heat transfer effectiveness and NTU results are given for researched cross-flow heat exchanger.

NTU = 1,16
ε = 48%

4.8 Pressure Drop

Air side pressure drop through the heat exchanger fins calculated from (Shah, 2003).

$$\Delta P = \frac{G^2}{2\rho_i g_c}\left[1 - \sigma^2 + K_c + 2\left(\frac{\rho_i}{\rho_o} - 1\right) + f\frac{L}{r_h}\rho_i\left(\frac{1}{\rho}\right)_m - (1 - \sigma - K_e)\frac{\rho_i}{\rho_o}\right] \quad (4.58)$$

The mean density is the harmonic mean value given (Shah, 2003).

$$\frac{1}{\rho_m} = \frac{1}{2}\left(\frac{1}{\rho_i} + \frac{1}{\rho_o}\right) \tag{4.59}$$

ρ: average air density (kg/m3)

ρ_i: air density at inlet temperature (kg/m3)

ρ_o: air density at film temperature (kg/m3)

The ratio between minimum flow area to frontal is called as ε and given (Shah, 2003).

$$\sigma = \frac{A_0}{A_{fr}} \tag{4.60}$$

$$f = f_{cp}\left(\frac{T_w}{T_m}\right)^m \tag{4.61}$$

f : fanning friction factor for gases

CHAPTER FIVE
EXPERIMENTAL STUDY

5.1 Test Procedure of Heat Recovery Exchanger

There exist two methods for air to air heat exchangers that are used in heat recovery ventilators for commercial use. The first one is developed by American Society of Heating, Refrigeration and Air Conditioning Engineers, ASHRAE with the standard number 84. According to this standard an evaluation program by Air Conditioning and Refrigeration Industry, ARI has been developed and the manufacturers are treated with this certification program 1060-2005. For European market, a certification program is also been evolved by European Committee of Air Handling and Air Conditioning Equipment Manufacturers, Eurovent according to the European Norm, EN308 and named “Air to Air heat exchanger performance”. Eurovent program requires design software for the whole range of manufactured heat recovery exchangers.

5.2 Heat Recovery Exchanger Test Unit

According to TS EN 308/October 1997 Standard, the test stand for air to air heat recovery exchanger was installed by a manufacturer company. The test stand is shown in Figure 5.1. The exhaust and fresh air are conditioned by two air handling units. In left of the cabin, cold air handling unit is installed and in right of the cabin, warm air handling unit is installed. The heat recovery exchanger is put into heat exchanger case. In warm air handling unit, hot water coil (90/70°C) is used. Hot water coil is connected to boiler. Cold water coil (7/12°C) is used in cold air handling unit. Cold water is derived from the chiller.

Figure 5.1 Air to air heat exchanger test system

Figure 5.2 Cold air handling unit

Plug fan is used in air handling unit. Fan speed is controlled as shown in Figure 5.3. This sensitive sensor detected if the system performs at desired velocity or not. The sensors are placed in cold and warm air handling unit.

Figure 5.3 Plug fan speed control in cold air handling unit

Temperature is controlled by two ways. One of them is manual control that is shown in Figure 5.4. There are two thermometers with four channels. In fresh and exhaust air inlet and outlet ducts, four sensors is placed. Manual results are read as figure.

Figure 5.4 Manual temperature controls

The other method to control temperature is the sensors which are connected to microprocessor. The sensors transfer these values to microprocessor. Also humidity and pressure control sensors are placed in inlet and outlet ducts.

Figure 5.5 Microprocessor and air handling unit control panel

Microprocessor and air handling unit control panel are shown in Figure 5.5. There is a control panel for all air handling unit. Control panel has a frequency converter for controlling temperature.

5.3 Experimental Phases

Heat recovery exchanger is attached to its own mould and placed in experiment cabin. The size of the heat recovery exchanger is 560 x 540 mm and fin spacing is 6.25 mm.

Figure 5.6 Experiment cabin

Main control processor is shown in Figure 5.7. Before starting the test, test settings are changed. Test parameters are heat exchanger front area, intended maximum temperature oscillation, outer air and inner mass flow rate precision, minimum resolution period, maximum test period. First step is entering these data.

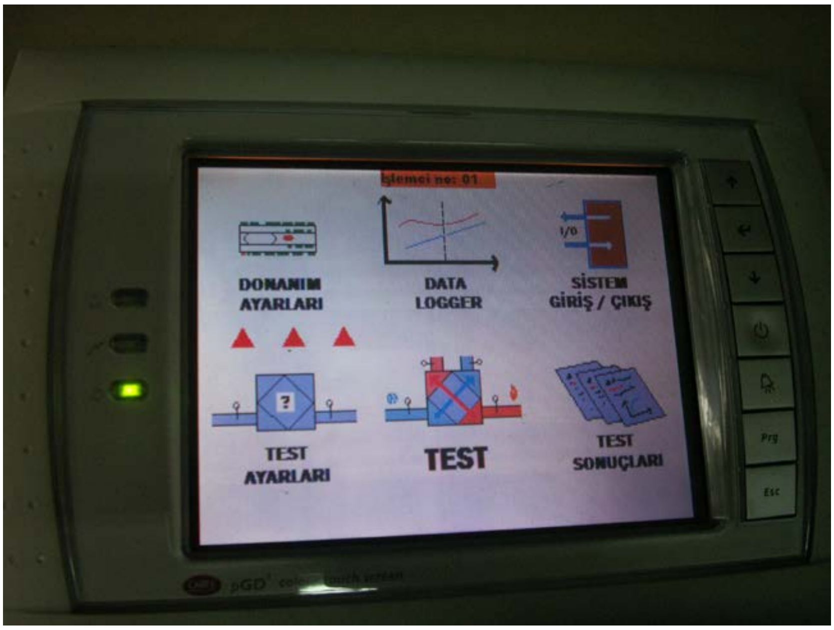


Figure 5.7 Main control processor

Figure 5.8 First step of the test parameters

As second step, intended test velocities are entered as shown in Figure 5.9. The inner and outer velocities are thought as the same. 3 m/s and 6 m/s velocity values are chosen as variable speed values.

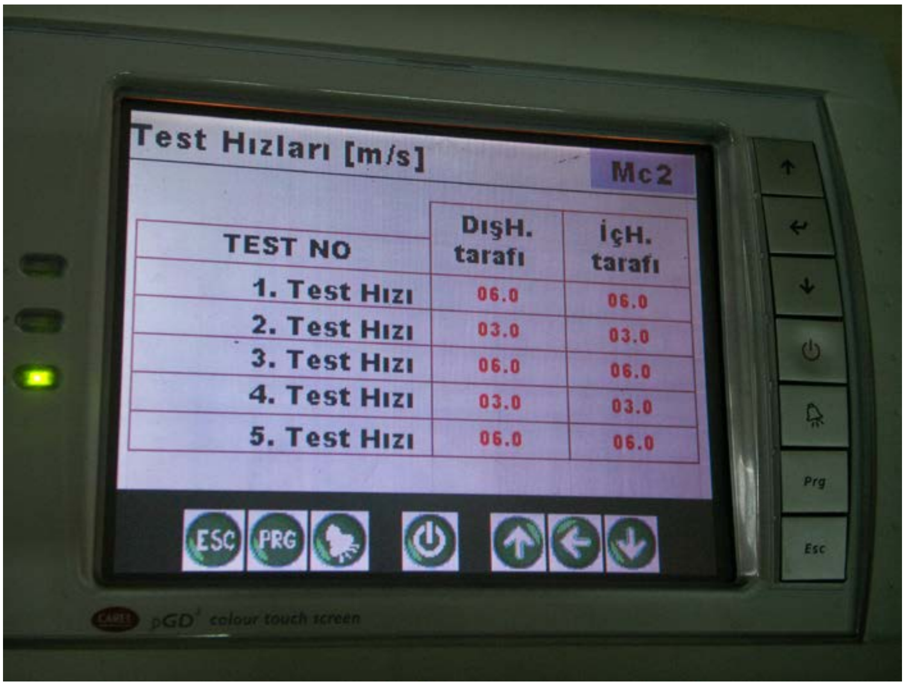


Figure 5.9 Second step, entering test velocities (m/s)

Intended temperature values are entered to the test processor for third step as shown in Figure 5.10. Ambient air temperature is permitted as 15.8°C and exhaust air temperature is permitted as 38.5°C. Thereafter test is started.

Figure 5.10 Third step, entering test temperatures (°C)

The values from sensitive sensors are seen on monitor. When the values which are received from fan speed sensors, catches the intended values, red light changes to yellow as shown in Figure 5.11. Thereafter, the intended temperatures are tried to catch by means of air handling units.

Figure 5.11 Test monitor

When all intended values are supplied, all red light changes to yellow as shown in Figure 5.12. If these values are protected for minimum resolution period (100s for this experiment), the test is over as shown in Figure 5.12.

Figure 5.12 Test results that all intended values are supplied

5.4 Test Results

The heat recovery outlet temperature and efficiency values, total experiment period are seen on the result display as shown in Figure 5.12. The efficiency and pressure drop values are seen for all experiment on fresh and exhaust air side as shown in Figure 5.13 and 5.14, respectively.

Figure 5.13 The efficiency and pressure drop results on external air side

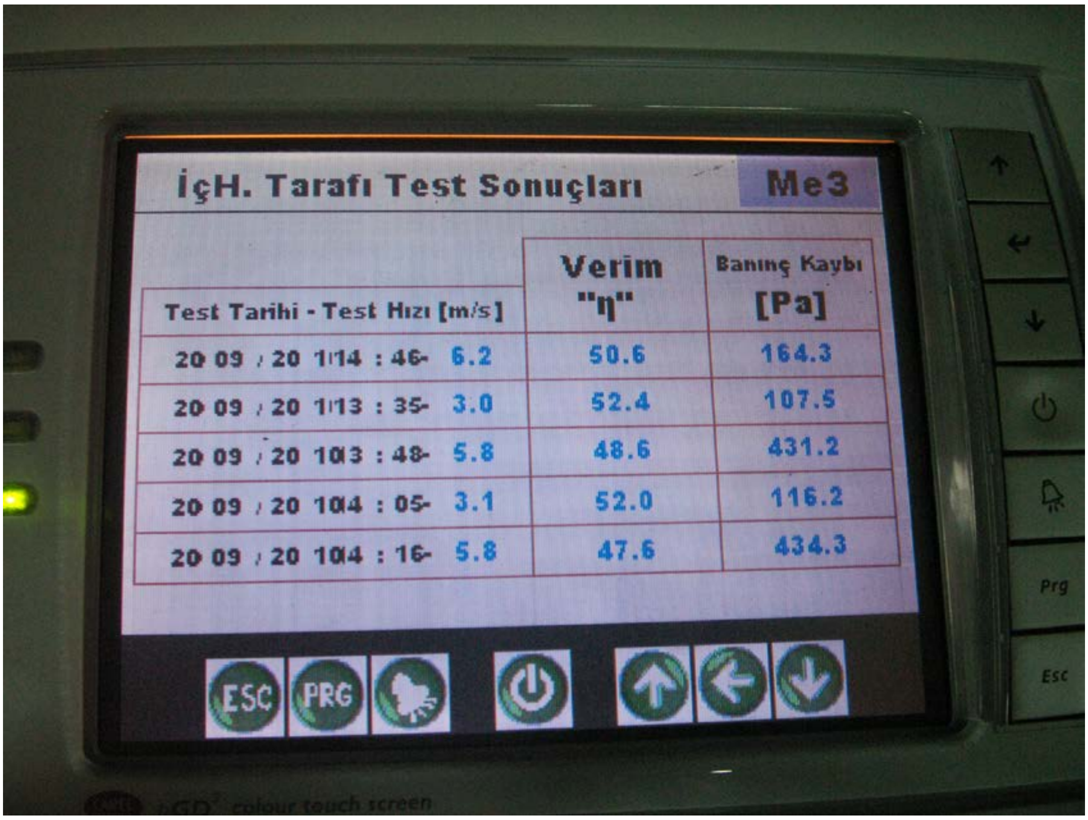


Figure 5.14 The efficiency and pressure drop results on internal air side

CHAPTER SIX
NUMERICAL ANALYSIS

6.1 Model Development

The numerical study is based on a cross-flow heat recovery exchanger as shown in Figure 6.1. This heat recovery exchanger is a product of a manufacturer company. The size of the heat exchanger used for numerical study is H×L×W = 600×600×600 mm. The heat recovery exchanger is mounted into air handling unit.

Figure 6.1 Cross-flow heat exchanger model

Plates are made of aluminum with thickness of 0.2 mm. The type of the plates is shown in Figure 6.2. The distance between two plates is 6.25 mm. Plates touch each other through the circular top surface of the cone. The plates are clamped to each other. To giving proper boundary conditions to these edges, the docking is simplified.

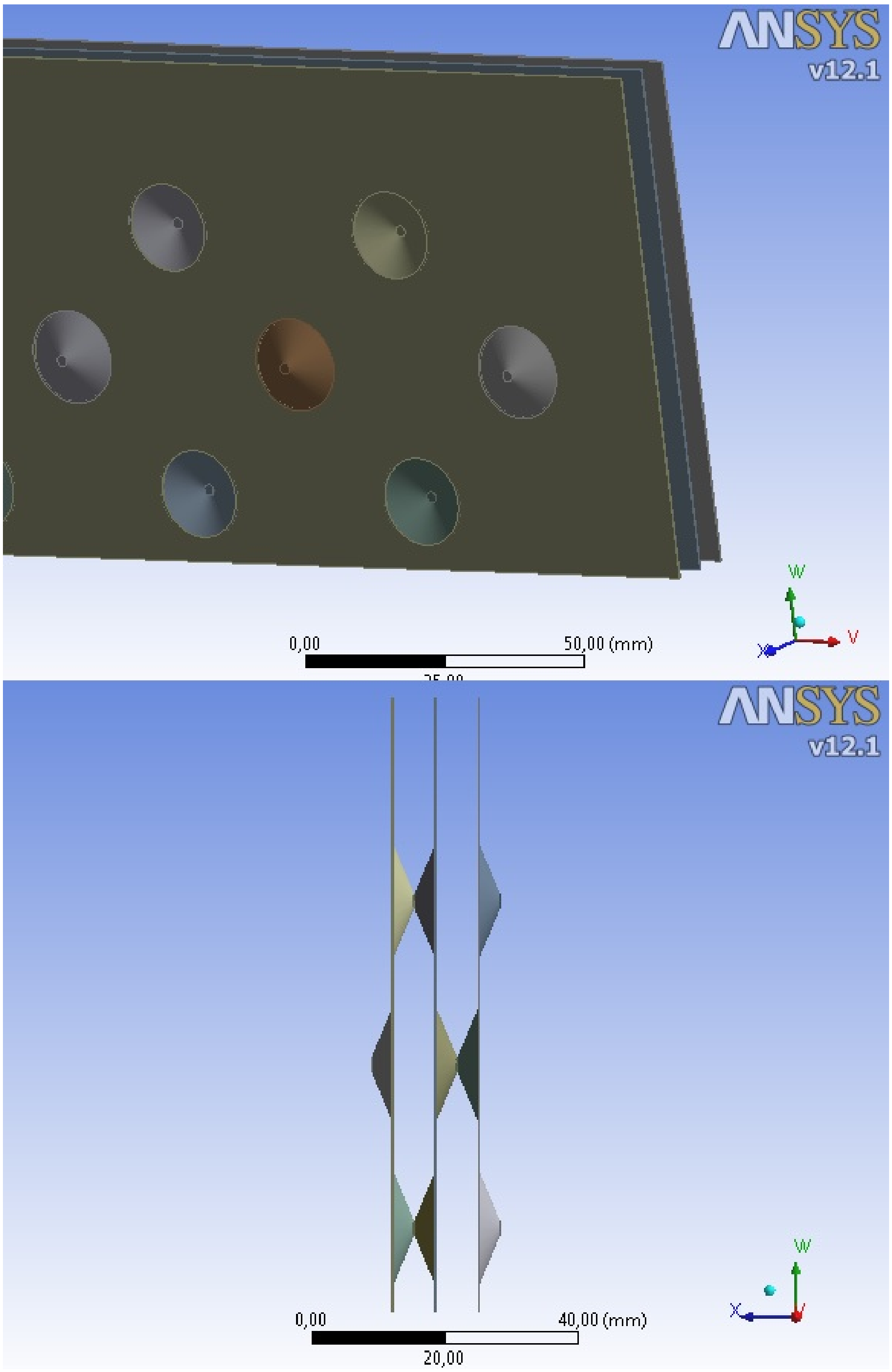


Figure 6.2 Fin type of the exchanger

The model was prepared by the means of a computer aided design program and imported as parasolid into Design Modeler. The model consists of seven volumes. These are three plates, two air volumes between the plates and two half-air volumes.

The three heat recovery exchanger models are analyzed for different geometrical parameters shown in Table 6.1, numerically. The geometrical parameters are taken from commercially available products.

Table 6.1 Geometrical parameters

Specifications	**Design 1**	**Design 2**	**Design 3**
Dimensions	600*600*600 mm	600*600*600 mm	600*600*600 mm
Exchanger Type	Mixed - Mixed	Unmixed-Unmixed	Unmixed-Unmixed
Fin pitch	6,25 mm	6,25 mm	6,25 mm
Fin surface	Louvered Fin	Smooth Fin	Louvered Fin
Flow type	Turbulent Flow	Turbulent Flow	Turbulent Flow

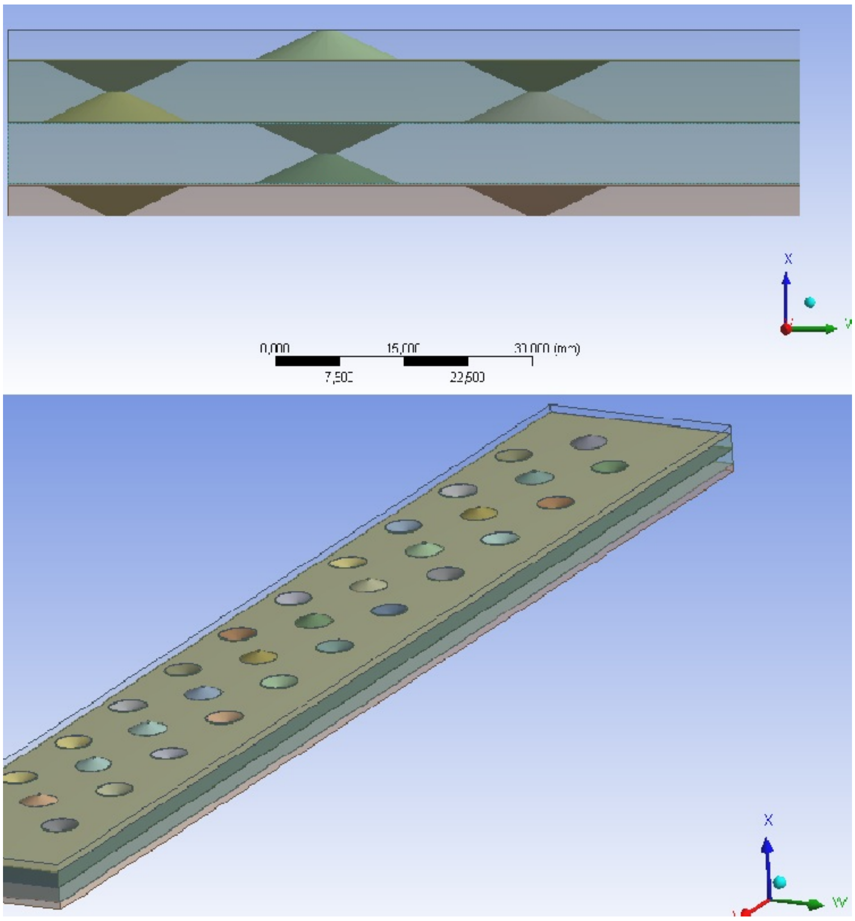


Figure 6.3 Partial view of Design-1

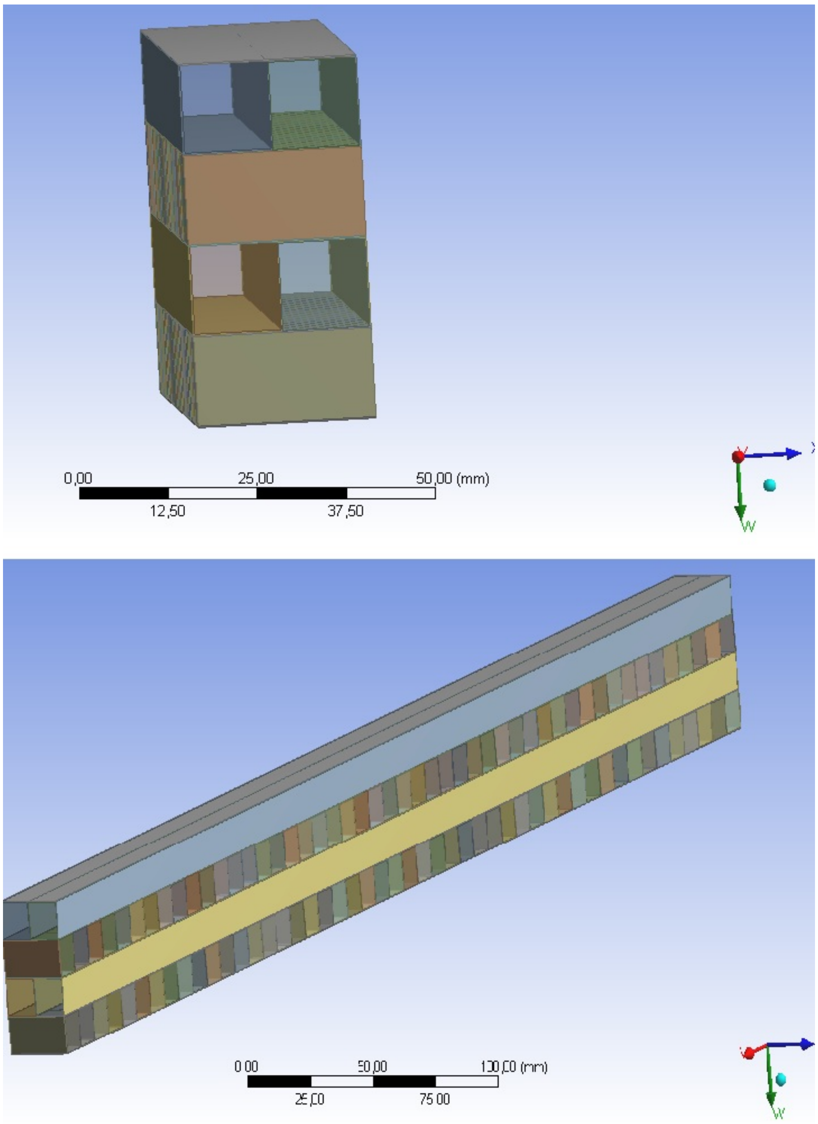


Figure 6.4 Partial view of Design-2

The heat flow arrangements are the different from each other. In Design-2, air flows through the channel in contrast to Design-1. Otherwise because of the louvered surface, the area of Design-1 is larger than Design-2. In Design-3, the air flows through the channel also, and the model has the louvered surface as shown in Figure 6.5. So the Design-3's area is larger than the others.

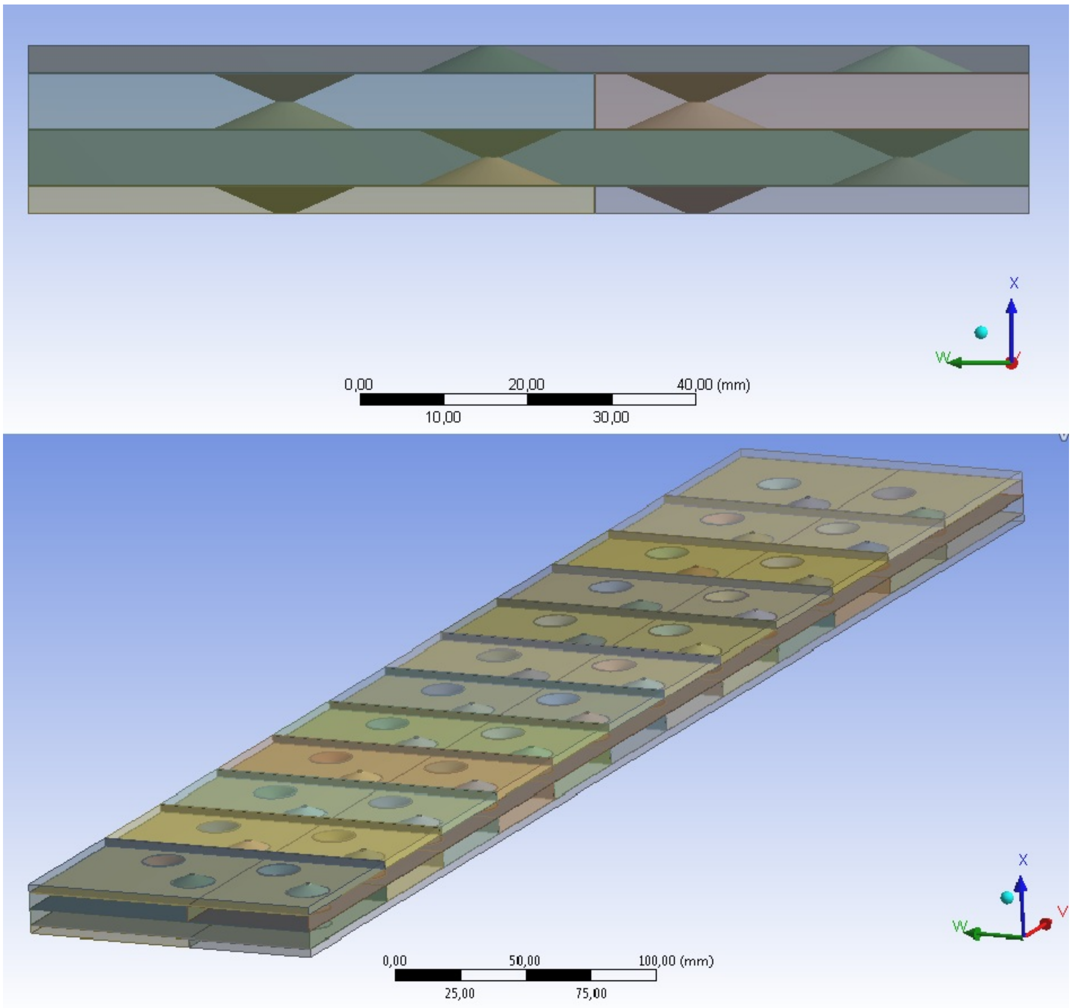


Figure 6.5 Partial view of Design-3

6.2 Mesh Creation

Meshing the geometry is the second step of the numerical. Faces and volumes are meshed, respectively. There are two methods as extruded 2D mesh and advanced front and inflation. To improve mesh quality in thin surfaces, plates and cones are meshed separately. In smooth plates, extruded 2D mesh method is used. In air volumes and fins conics advanced front method is used.

In Design-1, smooth plate's volume mesh display is shown in Figure 6.6. The number of elements is 37.786 which consist of 30.636 triangles and 7150 quads. Maximum edge length ratio is 170.533. Total area is 0.0974355 m^2.

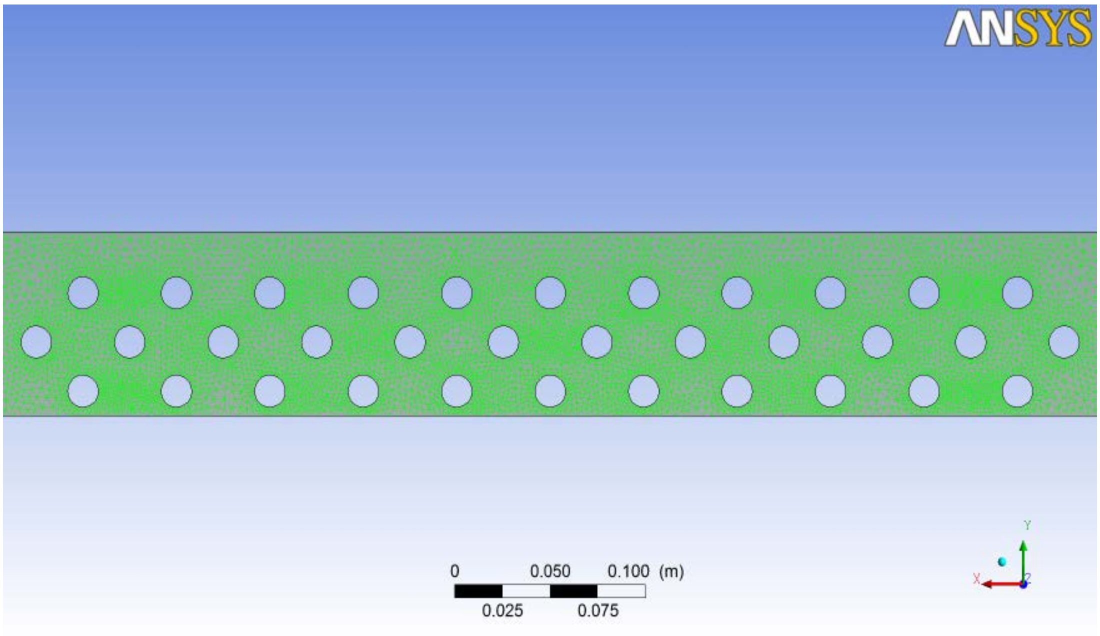


Figure 6.6 Smooth plate's volume mesh display

One fin has 34 conics. Conics volume mesh display is shown in Figure 6.7. The number of elements is 87.386 which consist of 87.386 triangles. Maximum edge length ratio is 7.00934. Total area is 0.0166971 m^2.

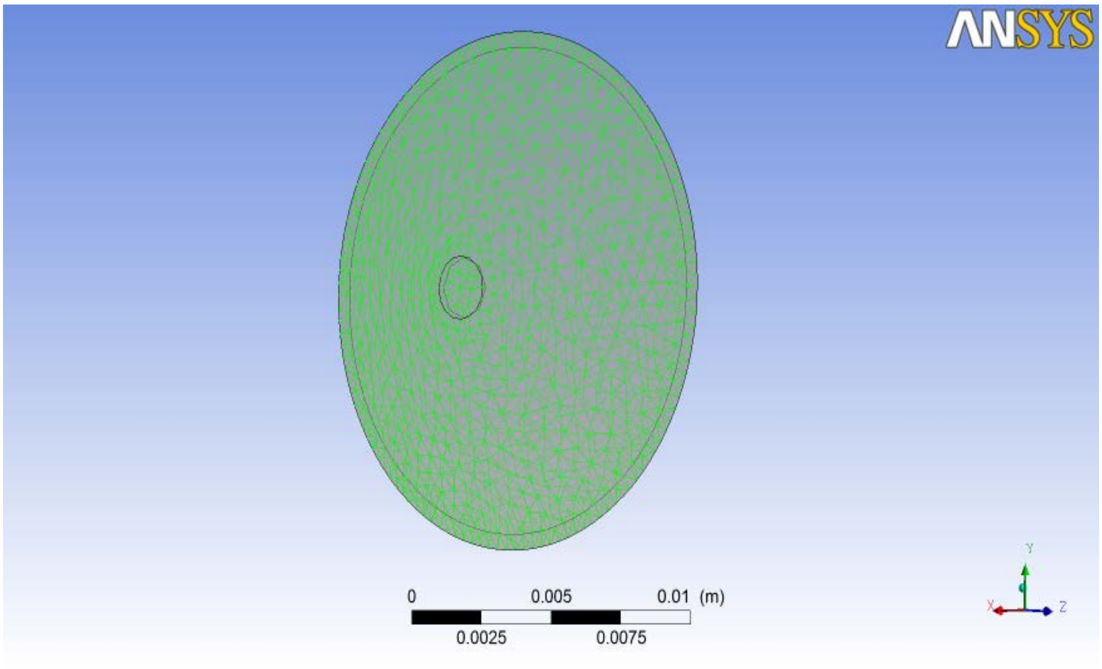


Figure 6.7 Conics volume mesh display

Half-air volume's volume mesh display is shown in Figure 6.8. The number of elements is 63.019 which consist of 63.019 triangles. Maximum edge length ratio is 5.44374. Total area is 0.115328 m^2.

Air volume between fins's volume mesh display is shown in Figure 6.9. The number of elements is 57.696 which consist of 57.696 triangles. Maximum edge length ratio is 5.8626. Total area is 0.120413 m^2.

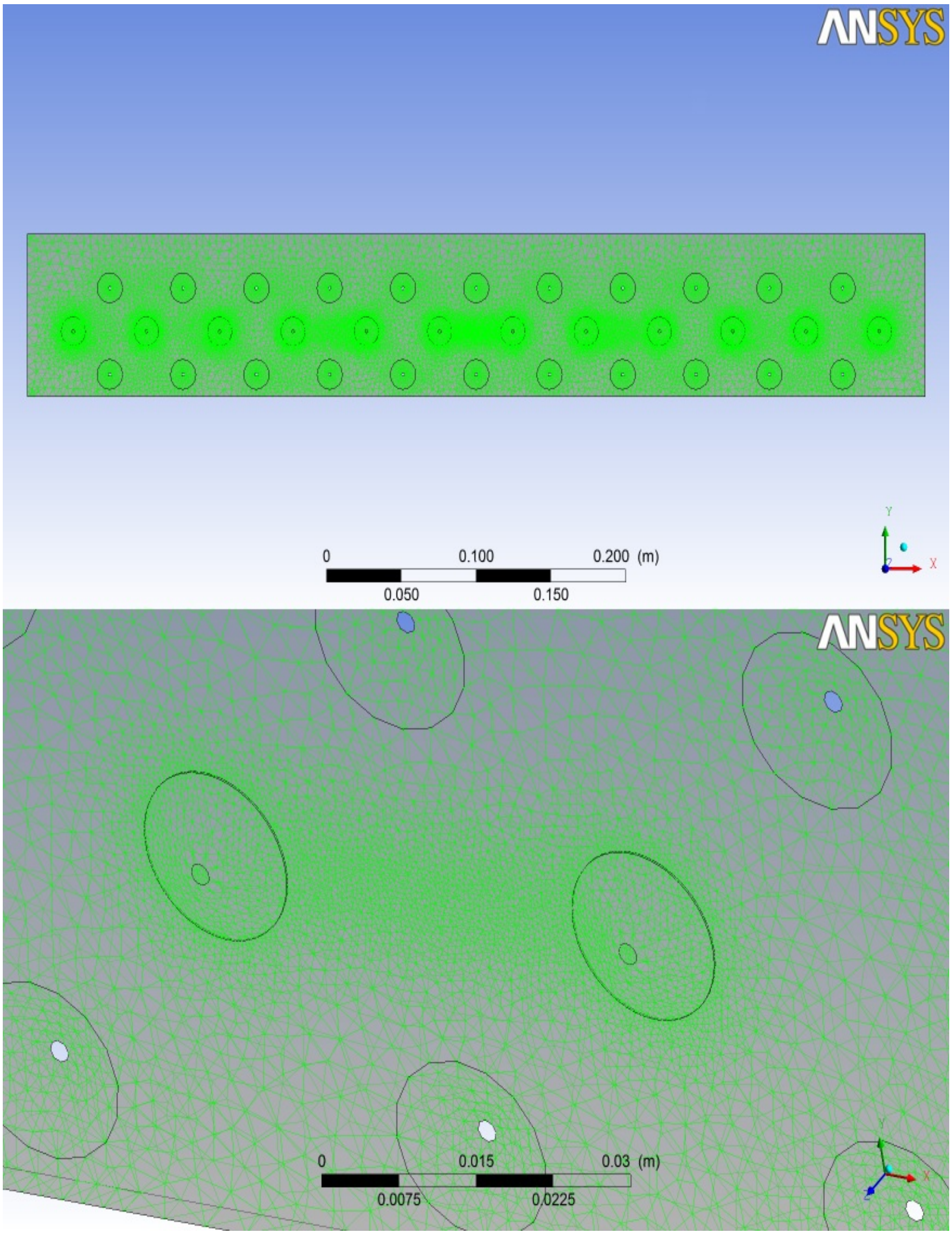


Figure 6.8 Half-air volume's volume mesh display

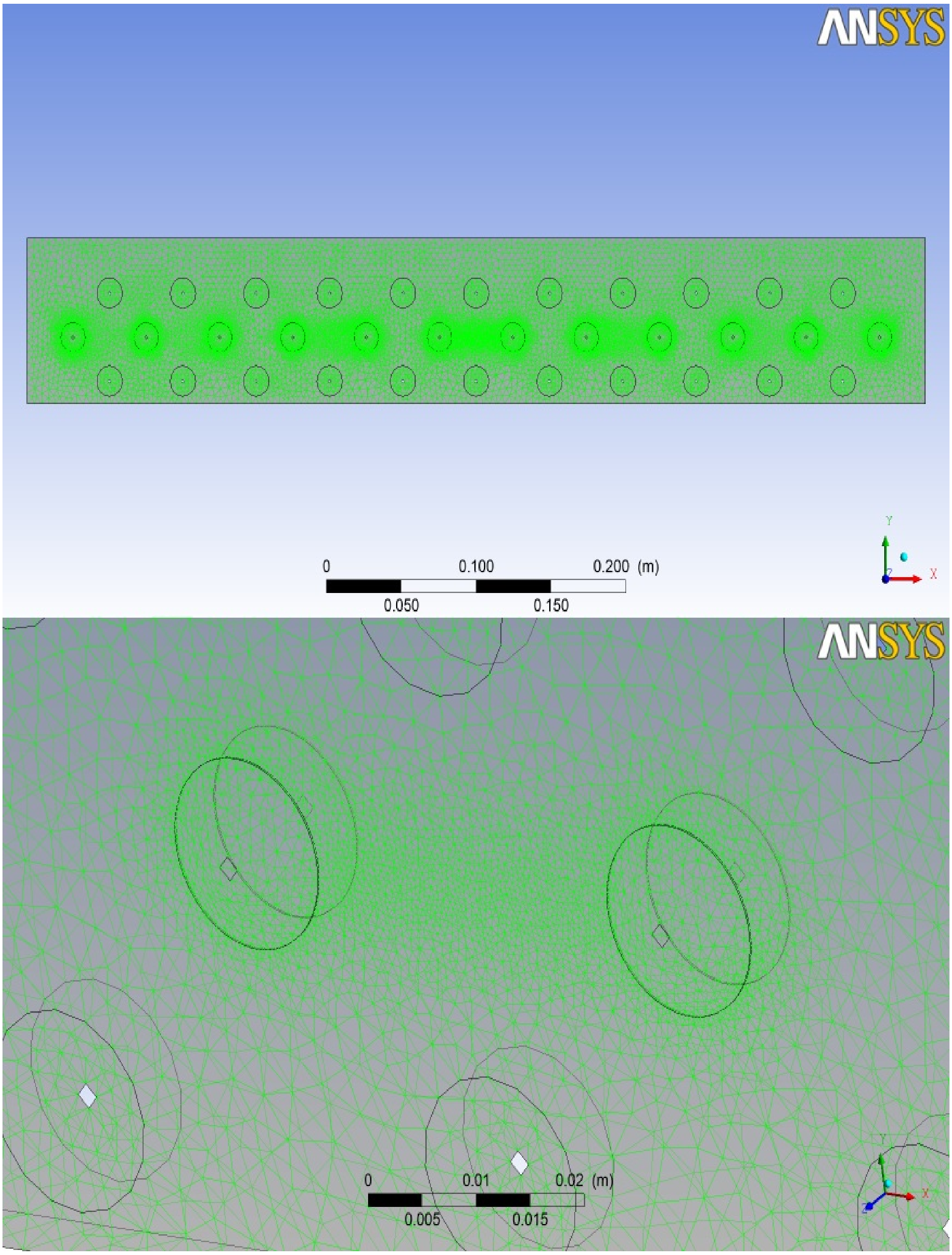


Figure 6.9 Air volume’s volume mesh display

In Design-2, due to the parallelism, extruded 2D mesh method is used. Smooth plates, air volumes are meshed separately. The number of elements is 2.371.535. The model's mesh display is shown in Figure 6.10.

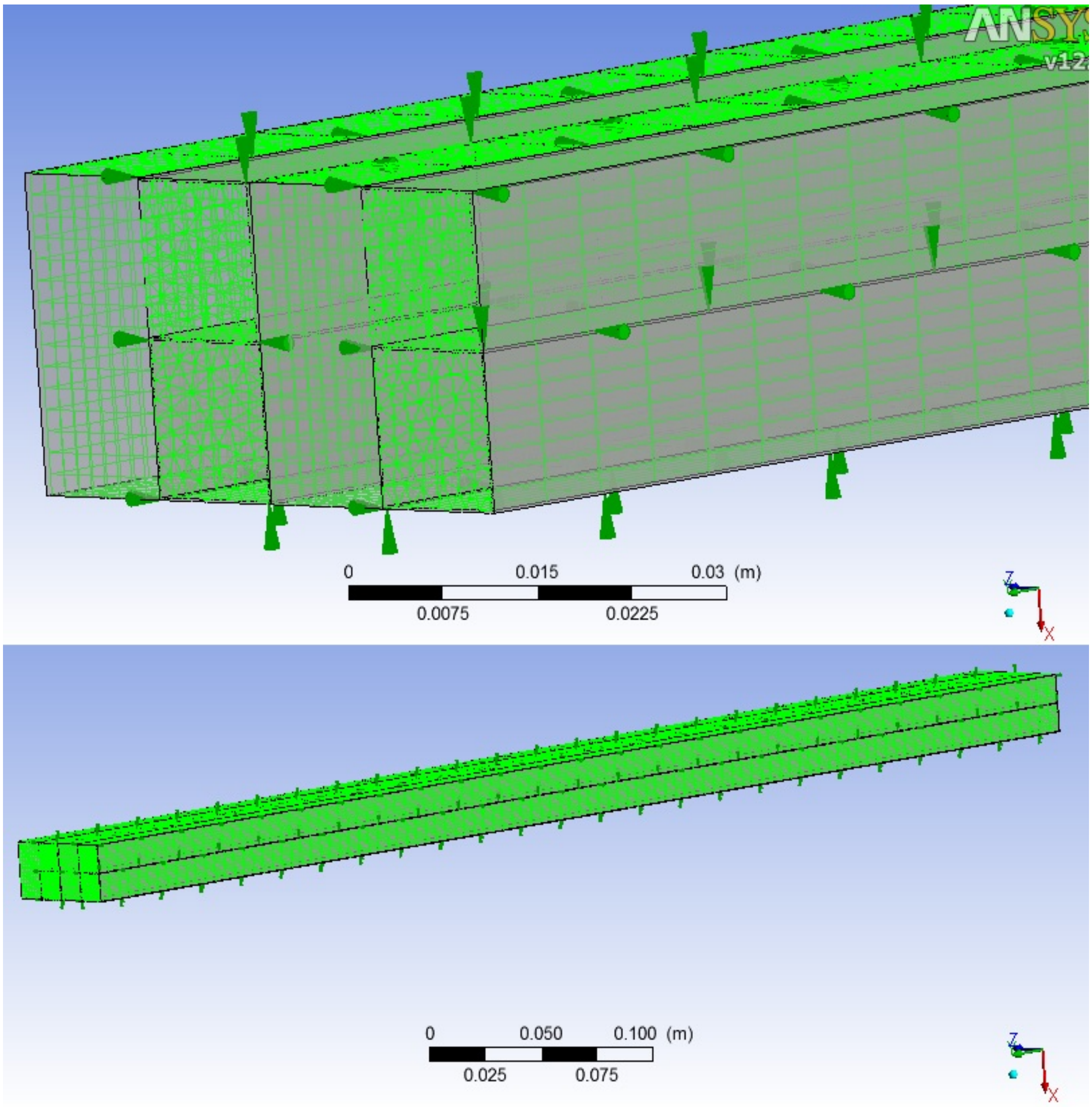


Figure 6.10 Design-2 mesh display

In Design-3, two methods as extruded 2D mesh and advanced front and inflation are used. To improve mesh quality, plates and cones are meshed separately such as Design-1. In smooth plates, extruded 2D mesh method is used. In addition, fin parts separating channels are meshed apart from the smooth fins. In air volumes and fins conics advanced front method is used. The number of elements is 4.888.523. The model's mesh display is shown in Figure 6.11.

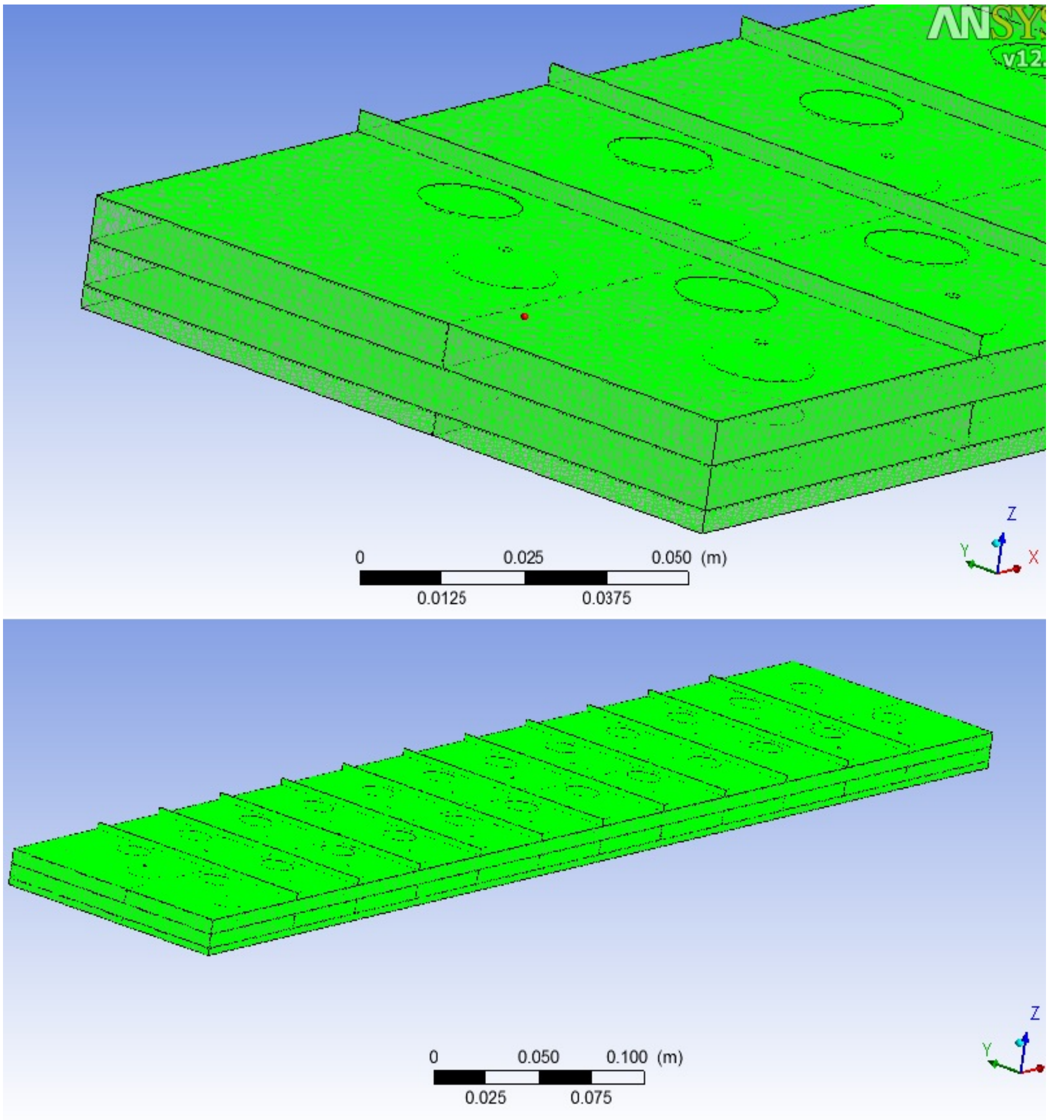


Figure 6.11 Design-3 mesh display

6.3 Boundary Conditions

The boundary conditions used in finite element analysis are listed in Table 6.2. Symmetrical boundary conditions are given to the top surfaces of to two half-air volumes due to the symmetry.

Table 6.2 Boundary conditions

Zone	Type	Properties	Inputs
Fin	Wall	Material	Aluminum
Inner surface of hot air	Inlet	Normal Speed	6 m/s
		Turbulence	Medium
		Static Temperature	288.8 K
Outer surface of hot air	Outlet	Average static Pressure	0 Pa
Inner surface of cold air	Inlet	Normal Speed	6 m/s
		Turbulence	Medium
		Static Temperature	311.5 K
Outer surface of cold air	Outlet	Average static Pressure	0 Pa

6.4 Solver

3D version of the Ansys Workbench is selected in order to analyze heat transfer and efficiency of the heat recovery exchanger. The flow is assumed to be steady and incompressible. Standard k-epsilon turbulence model selected because of high flow velocity and Reynolds number.

Three dimensional, steady-state analyses' iterations have continued until residuals reach $10*E^{-6}$ and domain imbalances come below % 0.000001. This took 4 hours and 32 minutes with 200 iterations.

CHAPTER SEVEN
RESULTS AND DISCUSSIONS

7.1 Results and Discussions

The heat transfer in air-to-air heat recovery exchanger was investigated theoretically, experimentally and numerically in this study. The heat recovery exchanger with 600*600*600 mm dimensions were analyzed in a computational package program. Design parameters were taken from manufacturer.

To create boundary and initial conditions, the heat recovery exchangers' actual working conditions in HVAC applications were considered. In numerical study, according to flow type and fin geometries, three different models were examined. The geometrical characteristics were described in previous chapter. At same boundary conditions and geometrical dimensions, the heat recovery exchanger models were analyzes by means of the computational package program.

To verify the numerical results, experimental study was made in manufacturer test laboratory that installed according to TS EN 308/October 1997 Standard.

7.1.1 Static Temperature Distribution Results

Static temperature distribution of the cold air in Design-1 at 6 m/s is shown in Figure 7.1. That is shown that the temperature of the cold air increases from inlet to outlet properly. Static temperature distribution of the hot air in Design-1 at 6 m/s is shown in Figure 7.2. As can be seen from the figure, the temperature decreases from inlet to outlet properly, in contrast to cold air behavior. Due to the small part of the heat exchanger investigated, temperature variation in cold air side is larger than temperature variation in hot air side. The 288.8 K ambient air comes through the heat exchanger and starts to warm up while it is passing between heat exchangers' fins. Heated air passes out from the cold air outlet surface at 298.8 K.

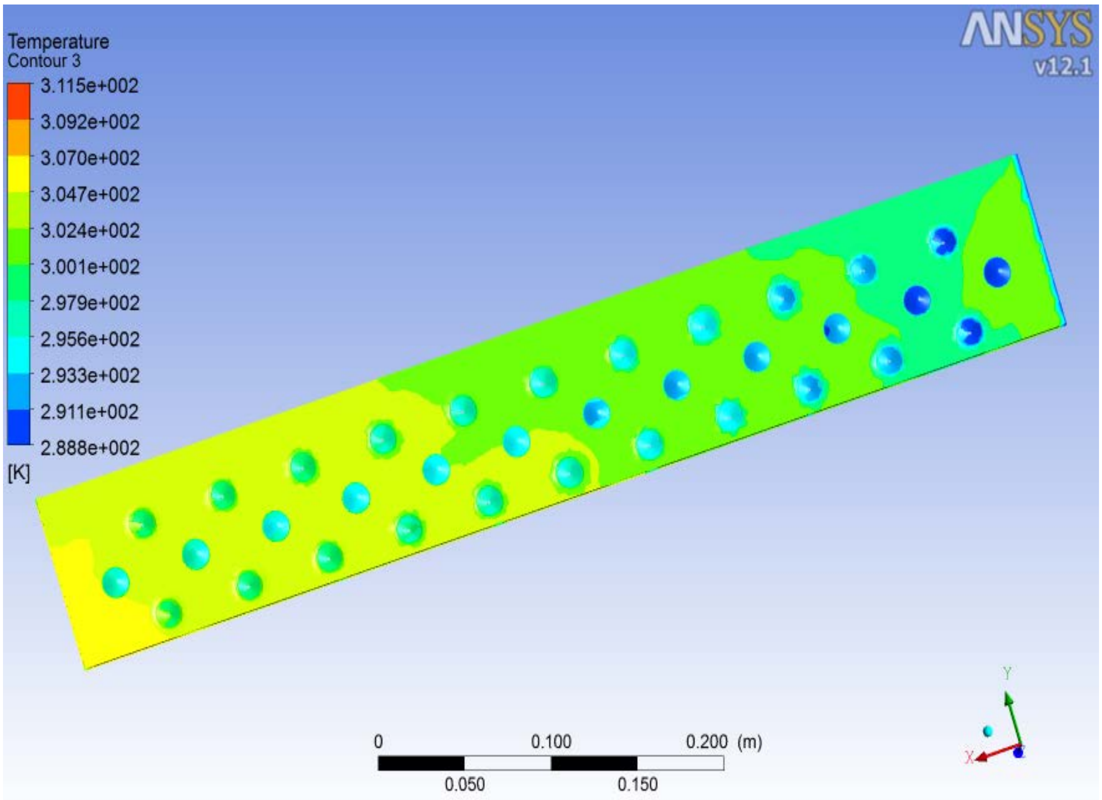


Figure 7.1 Static temperature distribution of the cold air (Design-1)

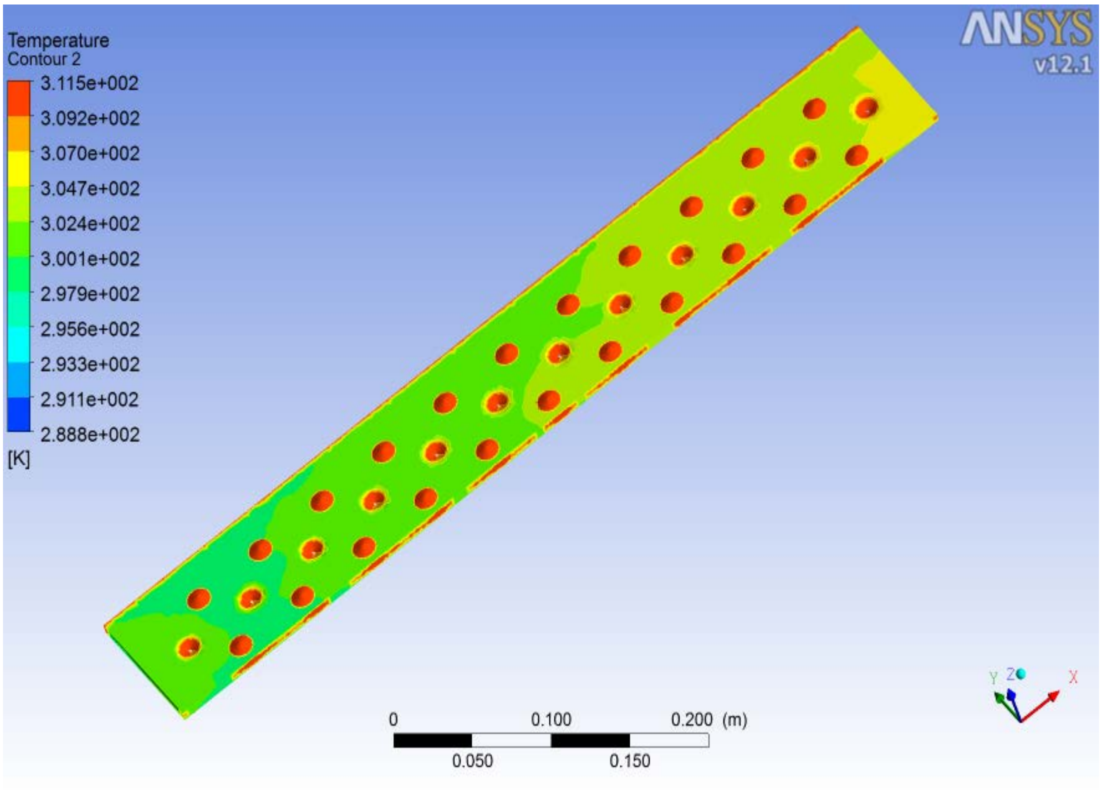


Figure 7.2 Static temperature distribution of the warm air (Design-1)

Static temperature distribution of the cold air in Design-2, which has smooth fins, at 6 m/s is shown in Figure 7.3. Temperature distribution at inner surface of hot air is more homogenous than temperature distribution at outlet surface of the hot air. The homogenous distribution at inlet surface of hot air arisen from the fix inlet temperature of 311.5 K.

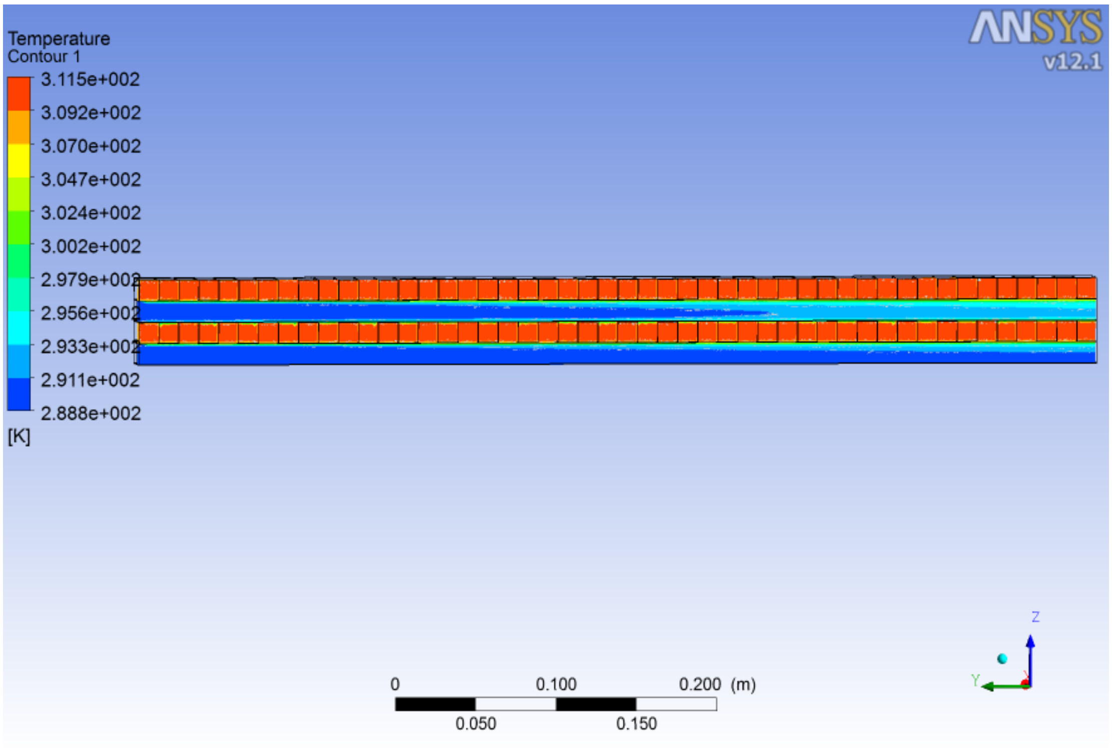


Figure 7.3 Static temperature distribution of Design-2

Static temperature distribution of the cold air in Design-3, which has louvered fins with unmixed-unmixed flow, at 6 m/s is shown in Figure 7.4. The homogenous variance at cold and hot air inlet surface is conspicuous. Otherwise as can be seen from the figure, the cold air temperature increases uniformly.

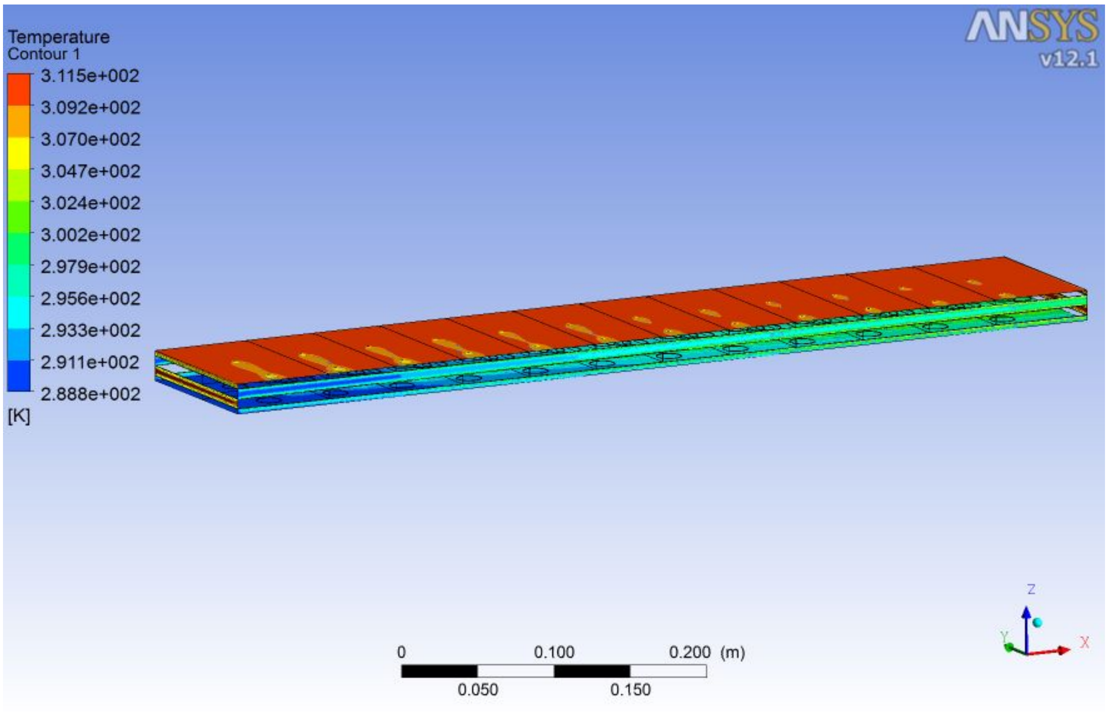


Figure 7.4 Static temperature distribution of Design-3

7.1.2 Velocity Distribution Results

Vector form of velocity magnitudes at the cold air in Design-1 at 6 m/s is shown in Figure 7.5. The difference of the inlet and outlet velocities is negligible. In addition, velocity vector flow paths inlet to outlet zones are providing us to understand louvered fin effects easily.

Vector form of velocity magnitudes at the hot air in Design-1 at 6 m/s is shown in Figure 7.6. Uniform vector distribution is seen through the inlet surface to the outlet surface. By means of the baffled fin surface, the increasement of heat transfer area is seen in figure.

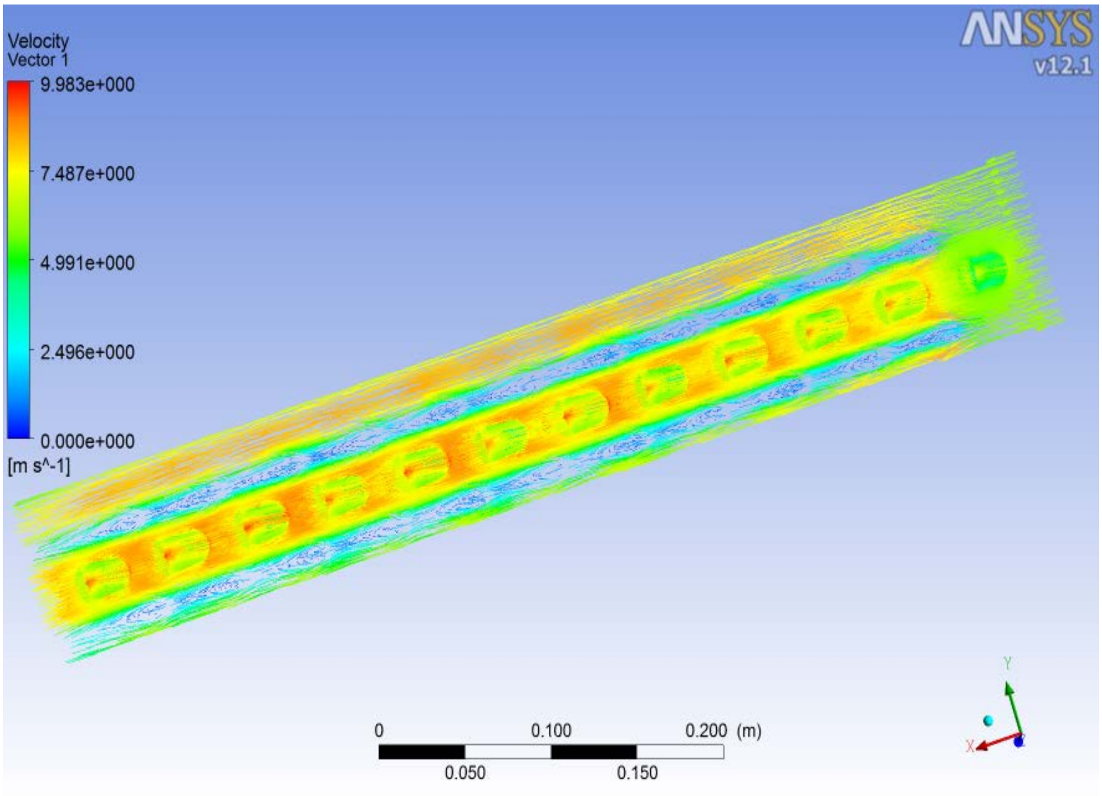


Figure 7.5 Vector form of velocity magnitudes at the cold air (Design-1)

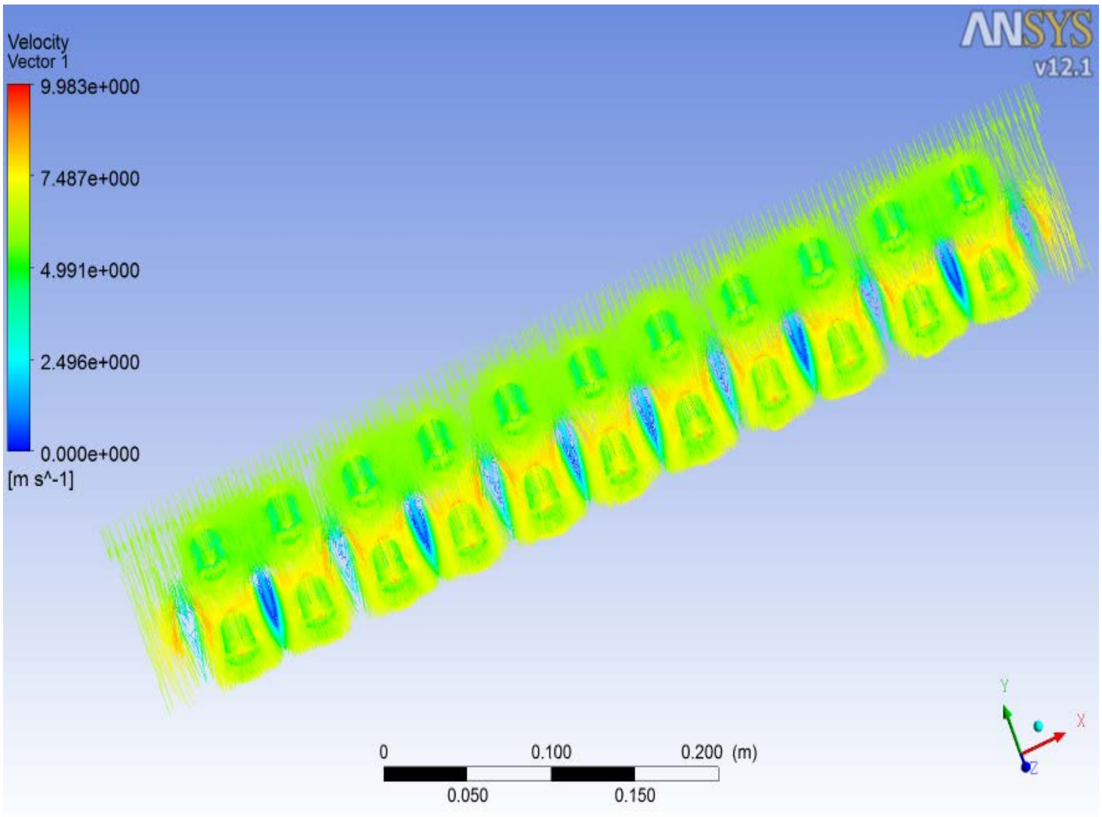


Figure 7.6 Vector form of velocity magnitudes at the warm air (Design-1)

Flow characteristics of cold and hot air inside the heat recovery exchanger are shown in Figure 7.7 with velocity vectors in Design-2 at 6 m/s. Vector forms are shown us flow direction and vortexes at air volumes. Due to the form of the fin surface, it is seen that the velocity vector paths straightness. The outlet velocity at hot air side is similar to the hot air inlet velocity, since a small percentage of the heat exchanger is researched at hot air side. For examining all of the cold air side flux, the inlet velocity increases from 6 m/s to 7 m/s.

Vector form of velocity magnitudes at the cold air in Design-3 at 6 m/s is shown in Figure 7.8. The flow characteristics of cold and hot air inside the heat recovery exchanger are similar to Design-1 because of the louvered fin. As distinct from Design-1, air volumes are separated from each other by means of inner plates. Thus, unmixed-unmixed flux occurs.

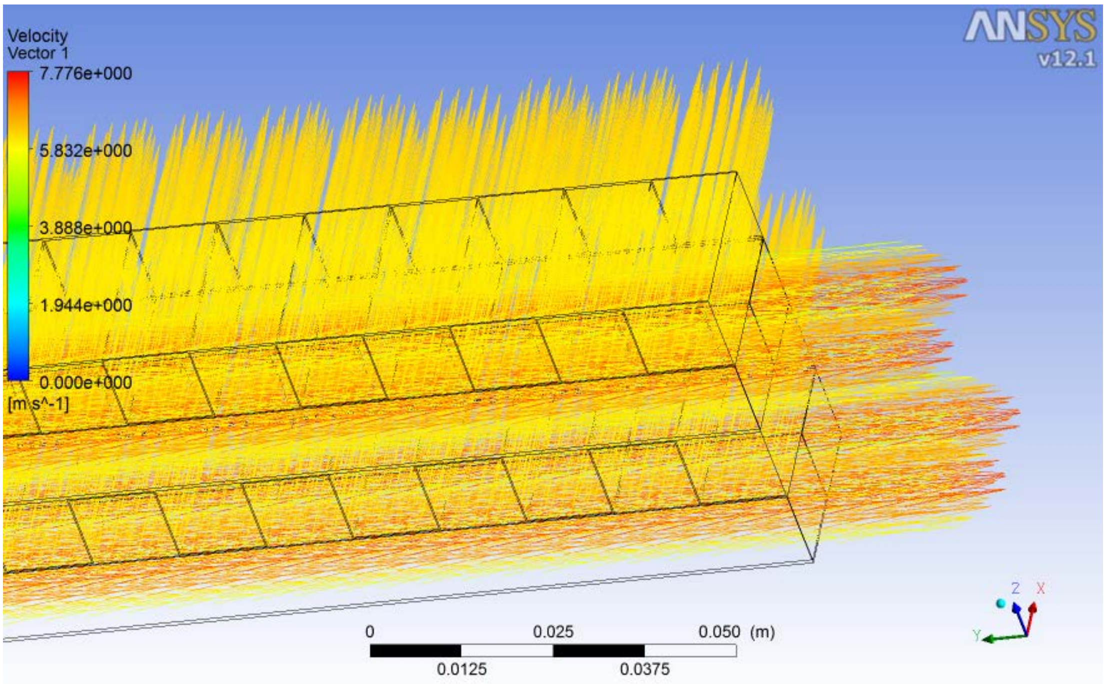


Figure 7.7 Vector form of velocity magnitudes (Design-2)

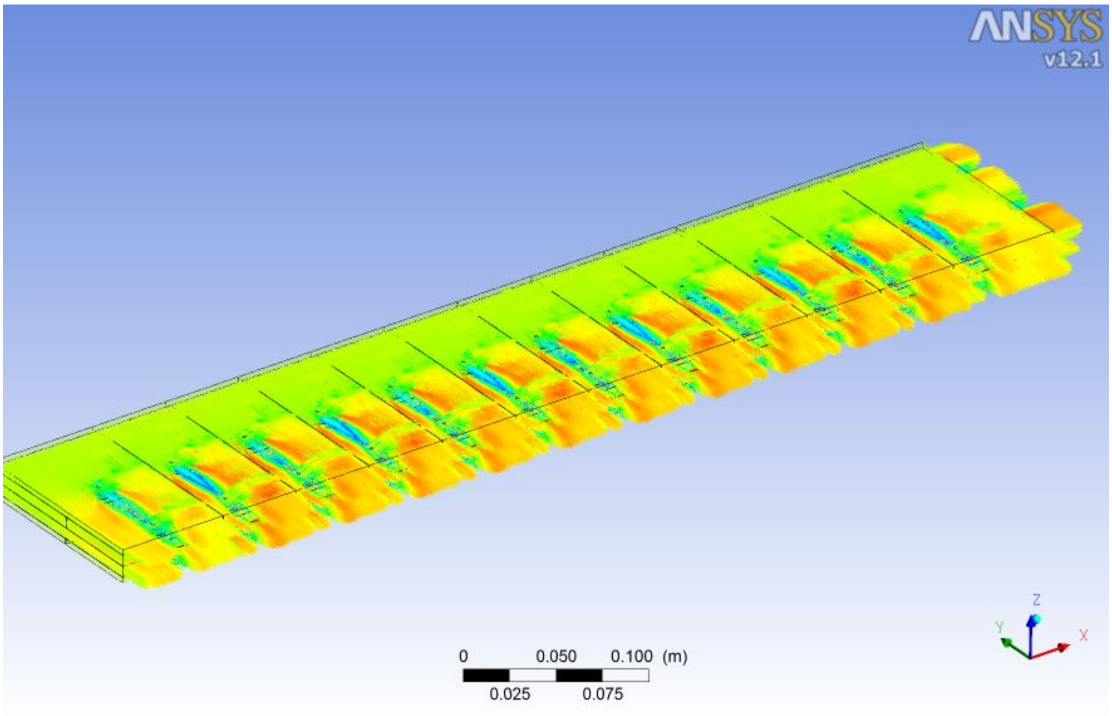


Figure 7.8 Vector form of velocity magnitudes (Design-3)

7.1.3 Comparison of Numerical and Theoretical Results

In theoretical study, the mixed-mixed type heat recovery exchanger is examined for different velocities. The conditions and design characteristics are given by the manufacturer company. The theoretical and numerical results are compared to find out the accuracy of the results.

For different inlet velocity values, the variation of cold air outlet temperature on fresh air side and hot air outlet temperature on exhaust air side is shown in Figure 7.9 and 7.10. As can be seen in figures, average relative difference between theoretical and numerical results is approximately 1°C. It shows the accuracy of the numerical analysis.

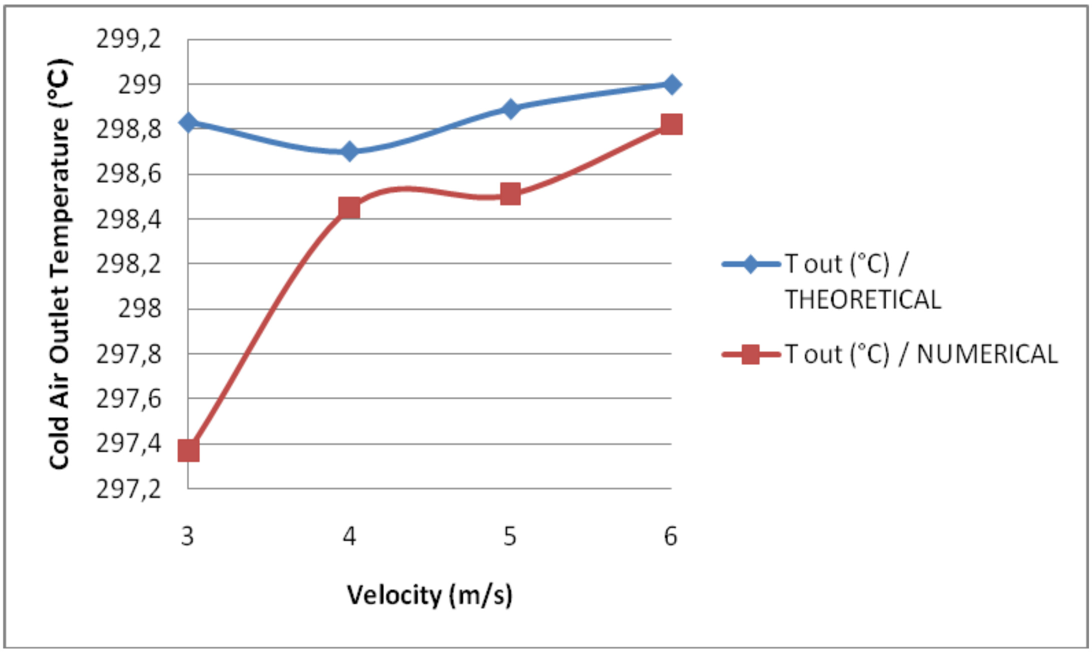


Figure 7.9 Cold air outlet temperature comparisons between CFD and theoretical results (Design-1)

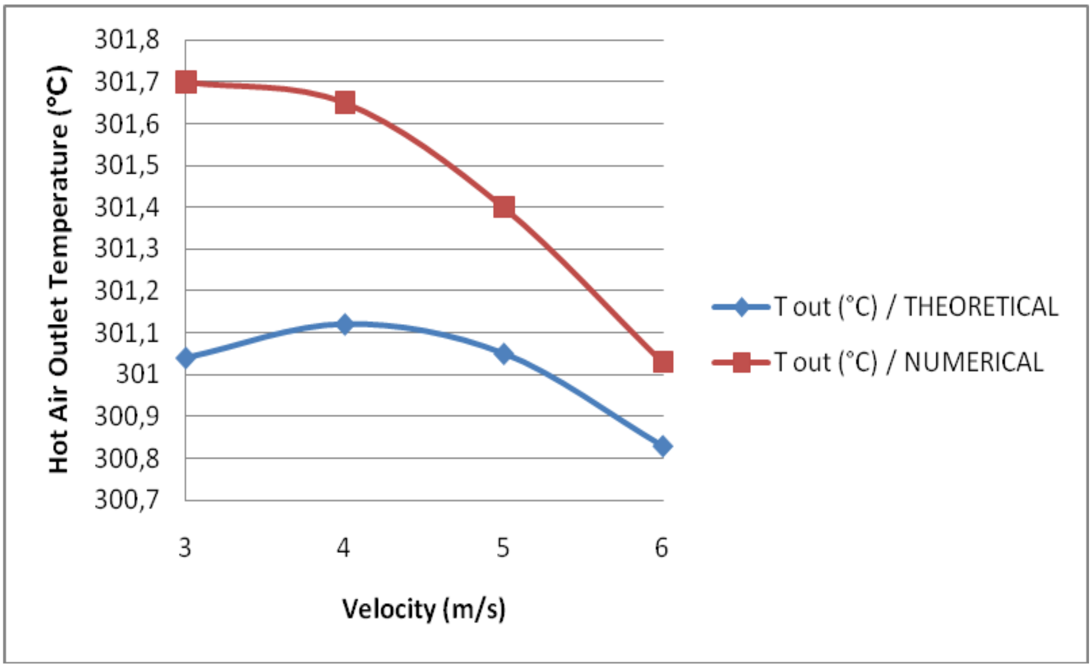


Figure 7.10 Hot air outlet temperature comparisons between CFD and theoretical results (Design-1)

Depending on air outlet temperatures, the efficiency of the heat recovery exchanger is calculated. The comparison of the heat recovery exchanger efficiency on fresh air side and on exhaust air side is shown in Figure 7.11 and 7.12. The average differentness of the theoretical and numerical analysis is approximately %2. This difference is negligible.

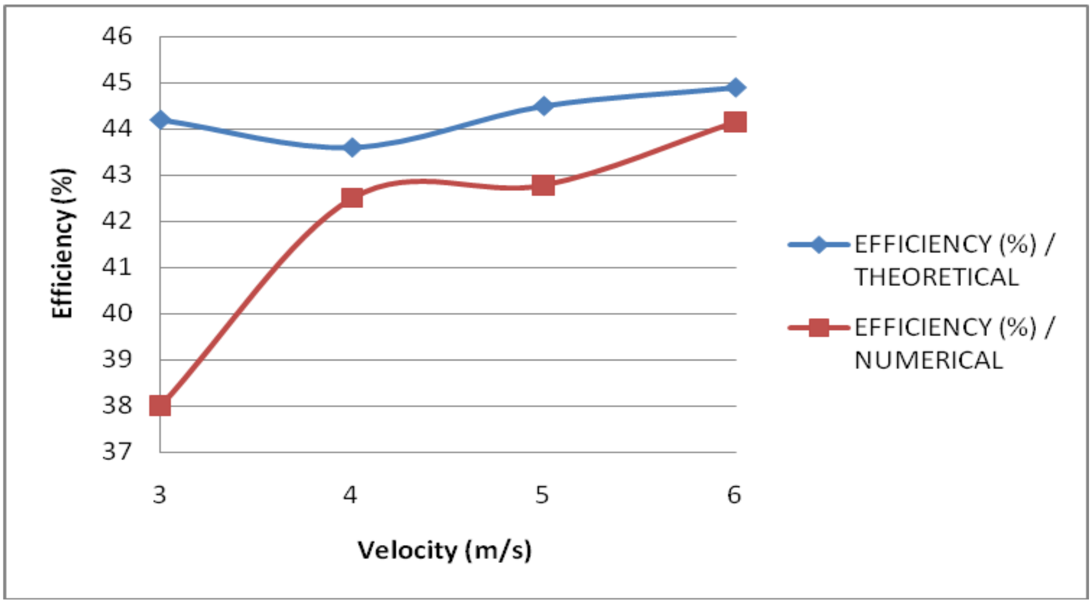


Figure 7.11 Efficiency comparisons between CFD and theoretical results on cold air side (Design-1)

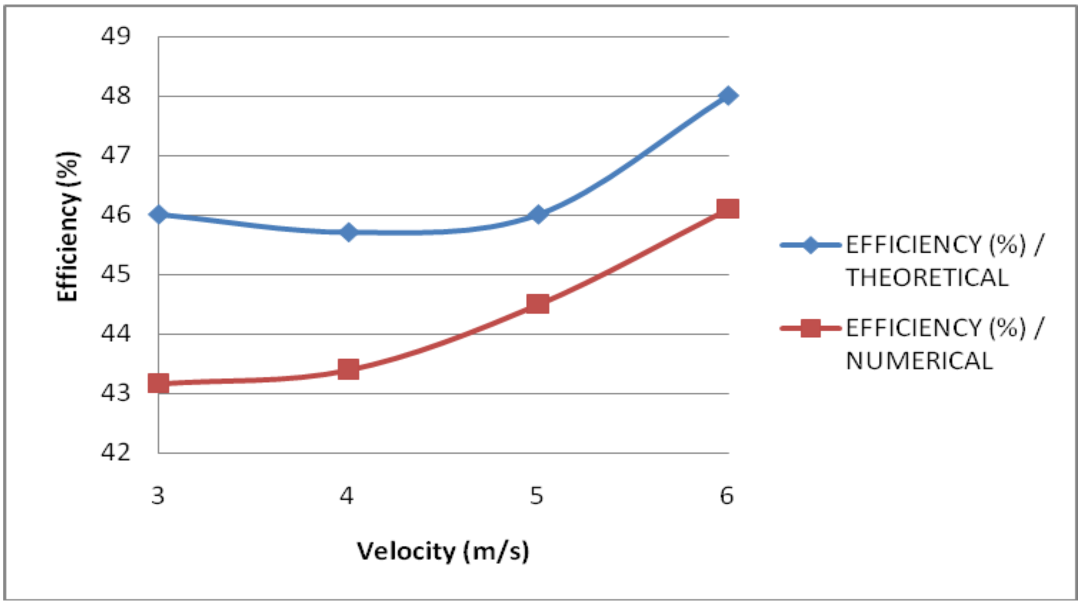


Figure 7.12 Efficiency comparisons between CFD and theoretical results on hot air side (Design-1)

7.1.3 Comparison of Numerical Results

For all of the designs researched in previous chapter, temperature variation in x-direction is analyzed at 6 m/s numerically. Variation of cold and hot air temperature is shown in Figure 7.13 and 7.14, respectively. The results in Design-1 and Design-3 are similar to each other. The temperatures in Design-2 are less about 4°C on average because of the lack of louvered fin. Louvered fin provide acceleration of the heat transfer area.

On cold air side, the temperature increases uniform along x-direction. Because of the heat transfer between cold air and hot air volumes, on hot air side, the temperature decreases along x-direction.

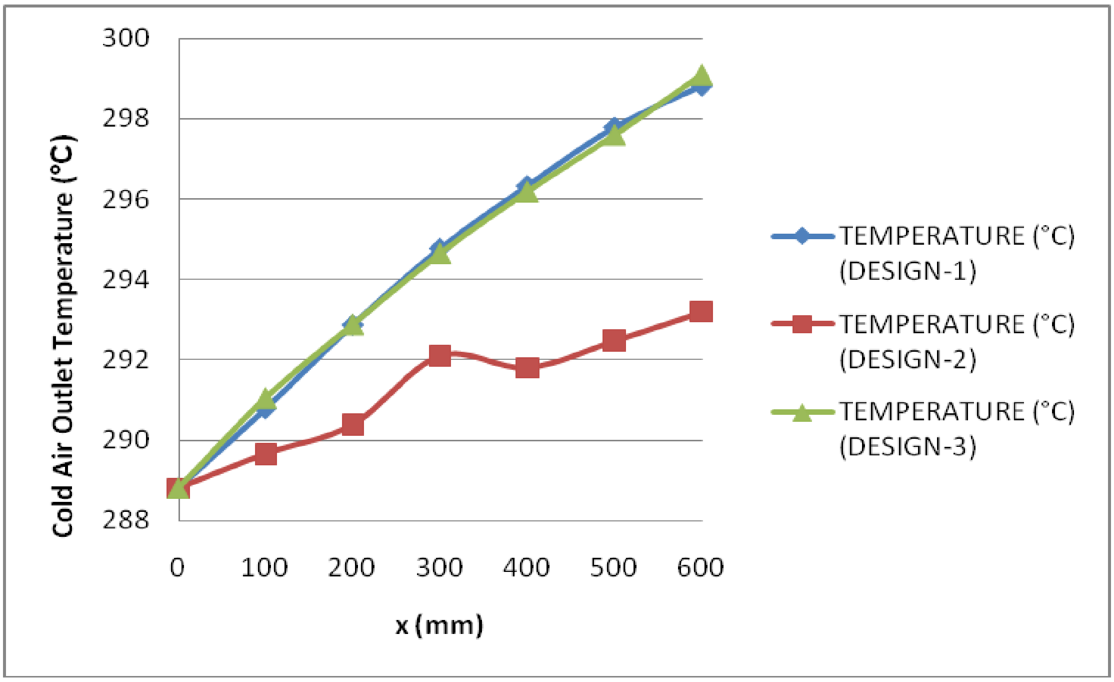


Figure 7.13 Variation of cold air temperature in x-direction

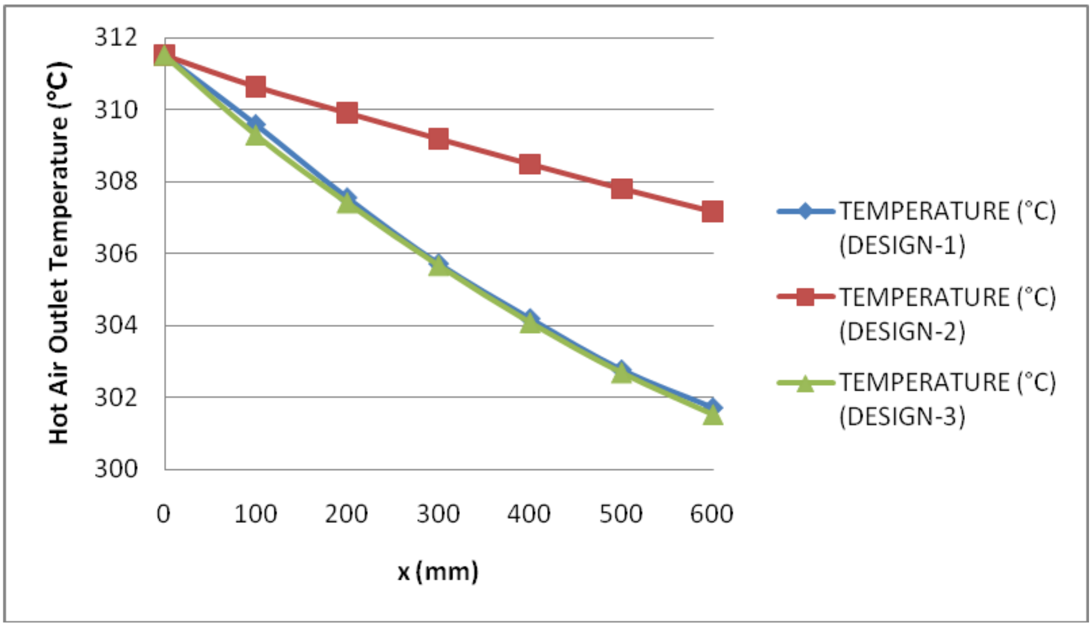


Figure 7.14 Variation of hot air temperature in x-direction

The variation of cold and hot air pressure in x- direction at 6 m/s is shown in Figure 7.15 and 7.16, respectively. The relative pressure of the outlet surfaces is considered as 0 Pa; the inner pressure values are seen.

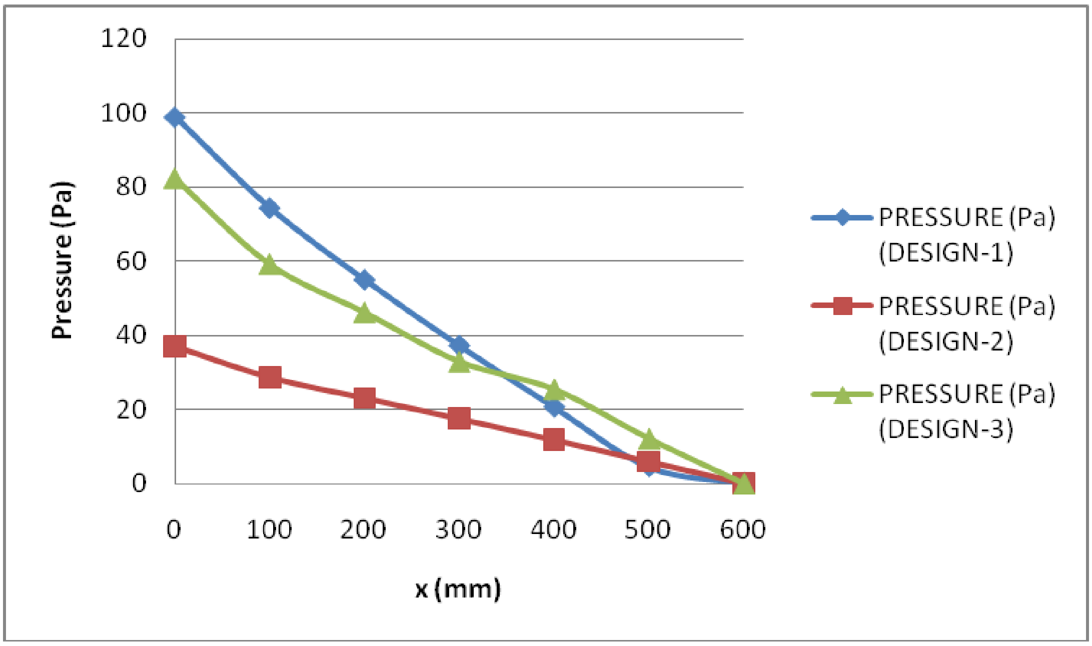


Figure 7.15 Variation of cold air pressure in x-direction

The maximum pressure drop occurs in Design-1. The cold and hot air flow thought smooth duct between two plates. Therefore, the minimum pressure drop occurs in Design-3. The velocity vector paths, fins surface and fins dimension are similar, for this reason, the pressure drop on cold air side and on hot air side is approximately same, for all designs.

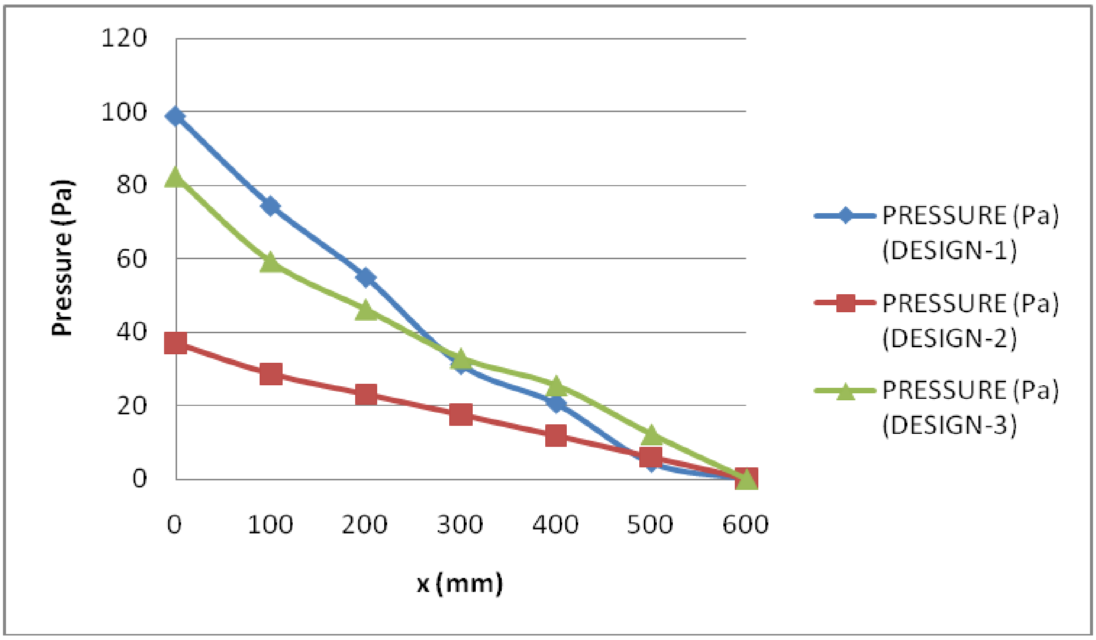


Figure 7.16 Variation of hot air pressure in x-direction

The variation of cold and hot air outlet temperatures are compared according to different velocities. The results in Design-1 and Design-3 are similar to each other as shown in Figure 7.17 and 7.18. The outlet temperatures in Design-2 are less about 5°C because of the lack of louvered fin.

As shown in figure, the cold air outlet temperature increases with increasing inlet velocity. Also, the hot air outlet temperature decreases with increasing inlet velocity. In other words, heat transfer rate increases with increasing velocity. On the other hand, heat transfer rate does not change discernibly, since the range of velocity magnitude is not wide in HVAC applications

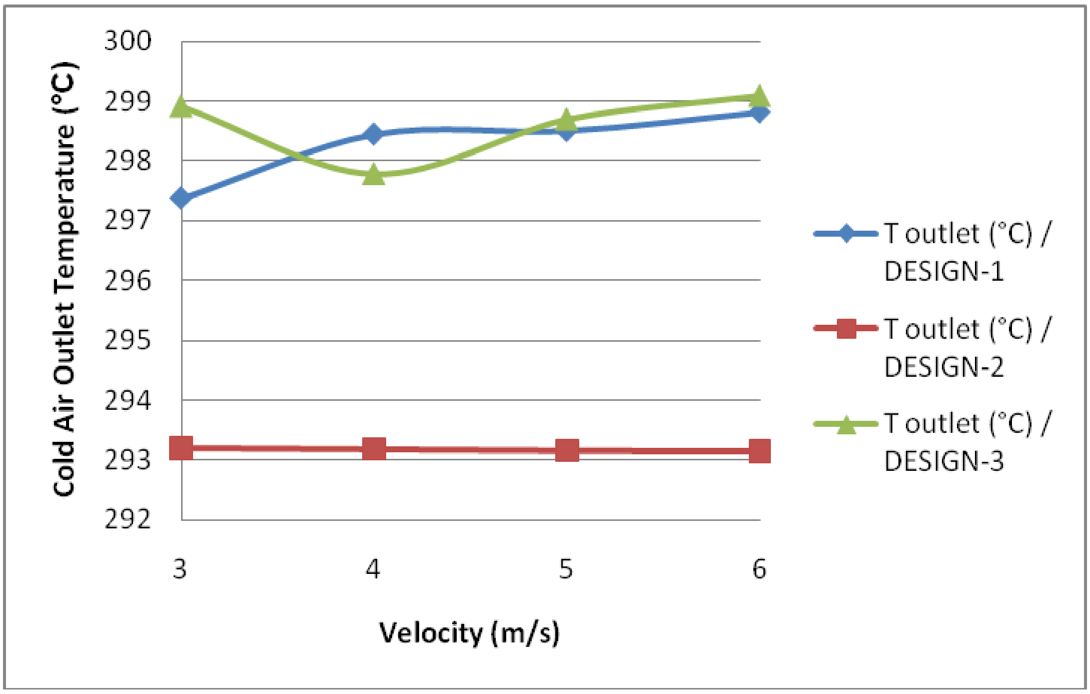


Figure 7.17 Variation of cold air outlet temperatures at different velocities

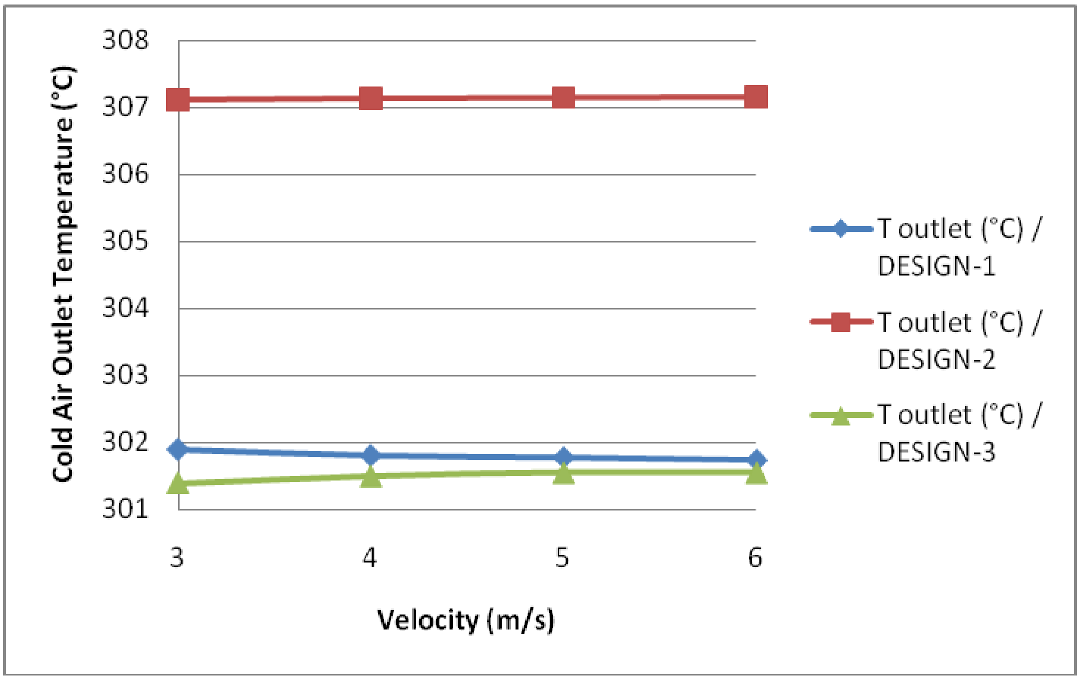


Figure 7.18 Variation of hot air outlet temperatures at different velocities

The variation of cold and hot air pressure drop are compared according to different velocities. Maximum pressure drop at all velocities occurs in Design-1 as shown in Figure 7.19 and 7.20. Pressure drop increases with increasing velocity. According to velocity variation, the increment in pressure drop is significant.

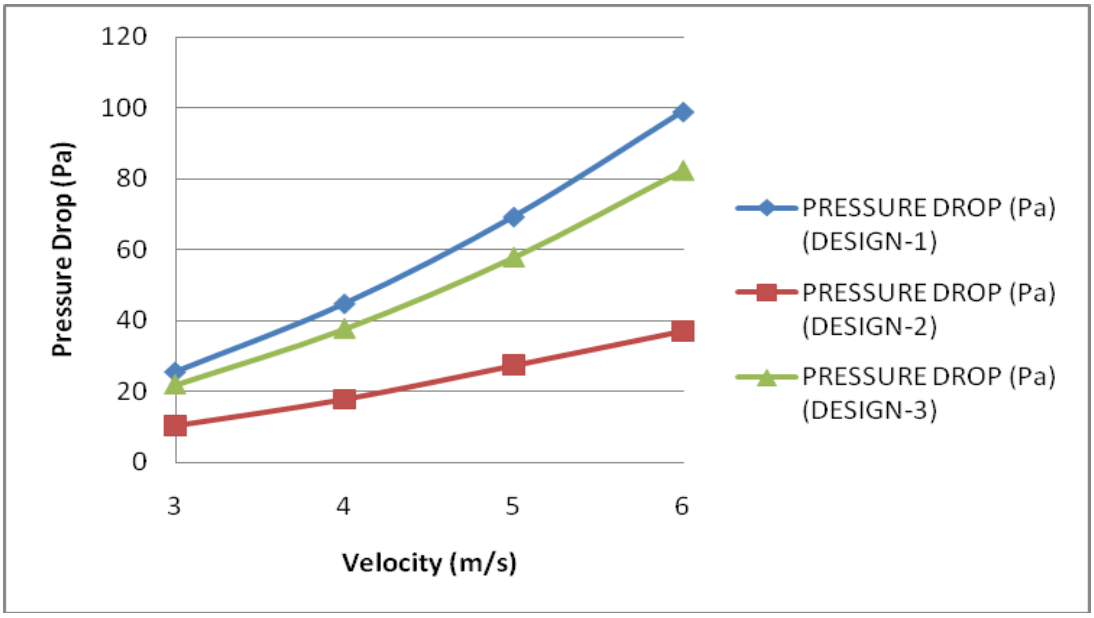


Figure 7.19 Variation of pressure drop on cold air side at different velocities

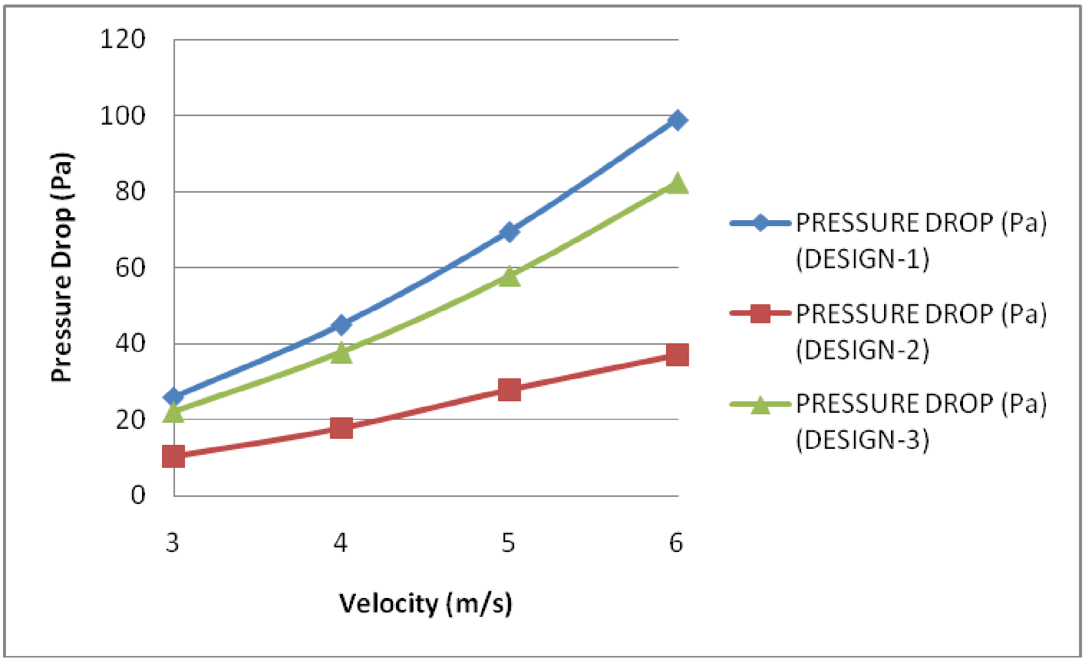


Figure 7.20 Variation of pressure drop on hot air side at different velocities

CHAPTER EIGHT
CONCLUSION

In this study, a literature survey about heat recovery exchangers was performed. Types of heat exchangers were researched. Applications, types and utilities of heat recovery exchanger are explained. In literature researches, numerical analysis with computational package program haven't made about heat recovery exchanger that are used for HVAC applications

In theoretical study, the efficiency of mixed-mixed type heat recovery exchanger is researched. For different velocities, the efficiency of heat exchanger was calculated. The dimensions of heat recovery exchanger theoretically 600*600*600 mm. The design characteristics and boundary conditions were given by manufacturer. The heat exchangers that have same exchanger area, same boundary conditions are compared each other according to flux type.

As a result, the efficiency of heat recovery exchanger which has 79.12 m^2 of area is 51.9%, if the flux is unmixed-unmixed. Whereas the efficiency of heat recovery exchanger is 44.9%, if the flux is mixed-mixed. The efficiency of unmixed-mixed type heat recovery exchanger is 49.7%. As seen from the results, the most efficient heat recovery exchanger is unmixed-unmixed type.

In experimental study, the mixed-mixed type heat recovery exchanger which has dimension of 540*560 mm is examined. The fin spacing is 6.5mm and fin thickness is 0.2 mm. As a boundary condition, 3m/s and 6m/s were considered as velocity values. At 6m/s speed, the hot air outlet temperature is 27.8 °C and relative humidity was 37%. The cold air outlet temperature was 26 °C and relative humidity is 37.9%. Whereas at 3m/s speed, the hot air outlet temperature is 26.5 °C and relative humidity was 37.7%. The cold air outlet temperature is 26.5 °C and relative humidity was 37.8%.

At 6m/s speed, the efficiency of heat recovery exchanger was calculated as approximately 49.7%. Whereas at 3 m/s speed, the efficiency of heat recovery exchanger was calculated as approximately 52.4%.

At 3m/s speed, pressure drop on external air side was calculated as 136.2 Pa. Whereas at 6m/s speed, pressure drop was calculated as 520.6 Pa. The pressure drop increases where the velocity increases. Because the appreciation of pressure drop is undesirable situation, the higher velocities should not be preferred.

In numerical study, the heat recovery exchanger was examined at computational package program by finite elements methods. The mixed-mixed type heat exchanger, which was examined theoretically, was drawn by the means of a computer aided design program. For different velocities, the theoretical and numerical results are compared. The difference between outlet temperature on cold and hot air side is approximately 1°C. Also the average differentness of the theoretical and numerical analysis is approximately %2. This difference is negligible. It is observed that theoretical and numerical results were accord with each other.

In addition, three different models were created according to flow patterns and geometrical designs. These models were analyses at same boundary conditions at a commercial computational fluid dynamic package program. Unmixed-unmixed type heat exchanger, which has smooth surface, has minimum efficiency because of the absence of the louvered surface. Two heat recovery exchangers, which have louvered and staggered arrangement, have more similar results. Results show us that unmixed-unmixed flow type is more efficient than mixed-mixed type heat exchanger. It shows the accuracy of the numerical analysis.

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