REPUBLIC OF TURKEY
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APPLIED SCIENCE

EFFECTS OF MECHANICAL VIBRATION ON THE HEAT TRANSFER CHARACTERISTICS OF TUBULAR TURBULENT FLOW

Master Thesis

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Submission date: 20 /02/2018
Presentation date: 16/03/2018

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March-2018

ACKNOWLEDGMENT

I thank all who in one way or another contributed to the completion of this thesis. First, I give thanks to God for protection and ability to do work.

I would like to express my sincere gratitude to my supervisor Associate Professor Nevin ÇELİK for her patience, kind support, immense knowledge, motivation, directions and thorough guidance during my research work. Her guidance helped me in all the time of research. At many stages of this project, I benefited from her advice, particularly so when exploring new ideas. Her positive outlook and confidence in my research inspired me and gave me confidence. Her careful editing contributed enormously to the production of this thesis.

I would like to thank all of my friends, who have supported me throughout the entire process, both by keeping me harmonious and helping me putting pieces together. Your friendship makes my life a wonderful experience. I cannot list all of the names here, but you are always in my mind. I will be grateful forever for your kindness. I also would like to thank the research assistants from Firat University; Sinan KAPAN and Murat ŞEN who helped me to do my experiments.

Last but not the least, I have to thank my parents for their love, encouraged me, prayed for me, and supported me throughout my life. Thank you, both for giving me the strength to reach for the stars and chase my dreams. My special thanks to my little daughter who bore my absence for two years I was absent from you, but my soul was beside you. You are my support in my life, my only love Vina. My brother and sisters deserve my wholehearted thanks as well.

Sincerely

Azez Majeed Mohammed

Elazığ, 2018

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SUMMARY

In this thesis, the effects of vibration on tabulated turbulent flow are experimentally analyzed. The experiments are performed to investigate the effects of sinusoidal, chirp and random noise vibrations on heat transfer characteristics of internal flow in a concentric type direct flow heat exchanger. Air is used as the working fluid with inlet Reynolds numbers varying between 10,000 and 50,000. The various values of frequencies of the mechanical vibration generator and vibration accelerations are the other important parameters in the experiments.

An experimental setup which was already set in Heat Transfer Laboratory is used in this work. The turbulators used in the concentric heat exchanger are somehow corrugated tapes. In the experimental assembly, hot water vapor is send to the annulus between the inner pipe and the outer pipe. This saturated water vapor is in contact with the outer surface of the inner pipe at all times. Thus, the constant temperature boundary condition is met on the outer surface of the inner pipe. The temperature and pressure measurements are performed to obtain heat transfer and pressure drop variations. Air is passed through the inner pipe throughout the experiment using a radial fan. The fan setting is changed with the help of an inverter to obtain different flow velocities.

The influence of the vibration is tested by using a mechanical vibration test unit. The frequency and accelerations (amplitude) are changed in a wide range area. The sine, chirp and random-noise signal types are imposed on the heat exchanger. Hence effects of the vibration on heat transfer are shown by means of Nusselt number. The simultaneously increment of pressure loss is also measured. Furthermore, the effects of vibration on the smooth tube internal turbulent flow and heat transfer are analyzed for comparison.

As a results it was observed that, regardless of the variation of the frequency, amplitude or signal type, applying vibration to the turbulator increases the heat transfer and pressure loss simultaneously, with high percentages. For example, the heat transfer increases with the percentages of 116%, 105% and 64%, respectively for *Type 1*, *Type 2* and *Type 3*. The enhancement percentages of the friction factor is about 35%, 61% and 95% respectively for *Type 1*, *Type 2* and *Type 3*.

Keywords: Mechanical vibration, heat transfer enhancement, turbulator, heat exchanger

ÖZET

BORULU TÜRBÜLANSLI AKIŞDA MEKANİK TİTREŞİMİN ISI TRANSFERİ KARAKTERİSTİĞİNE ETKİSİ

Bu tezde turbulatorlü bir türbülanslı akıma titreşimin etkisi deneysel olarak incelenmiştir. İç içe geçmiş ısı değiştirgecinde iç akışa sinüs, çörp ve rastgele gürültü sinyalleri gibi farklı titreşim türlerinin etkisini bulmak üzere deneyler yapılmıştır. Deneylerde akışkan olarak Reynolds sayısının 10,000 ila 50,000 değerleri arasındaki hava kullanılmıştır. Mekanik titreşim jeneratörünün frekansı ve titreşim genliği deneylerdeki diğer önemli değişken parametrelerdir.

Bu çalışmada Isı transferi laboratuvarında daha önceden kurulmuş olan bir deney düzeneği kullanılmıştır. İç içe borulu ısı değiştiricisi içerisinde kullanılan türbülatörler üzerinde kıvrımlı şeritler bulunan uzun çubuklar şeklindedir. Deney düzeneğinde içteki borunun dışından sıcak su buharı geçirilmiştir. Bu buhar iç borunun dış yüzeyiyle sürekli temas halinde olduğundan böylece sabit sıcaklık sınır şartı sağlanmıştır. Isı transferini ve basınç farkını elde etmek üzere sıcaklık ve basınç ölçümleri yapılmıştır. İç borunun içerisinden hava radyal fan ile gönderilmiştir. Farklı hava hızları elde etmek için fan bir invertör yardımıyla ayarlanmıştır.

Titreşimin etkisini görmek üzere bir mekanik titreşim test düzeneği kullanılmıştır. frekans ve ivme (genlik) geniş bir aralıkta değiştirilmiştir. Sinüs, çörp ve rastgele gürültü sinyal türleri uygulanarak titreşimin ısı transferi yani Nusselt sayısı üzerindeki etkisi araştırılmıştır. Titreşimsiz durum için sonuçlar elde edilerek her iki sonuç kıyaslanmıştır. Ve neticede ısı transferinin ve basınç kaybının titreşimle beraber arttığı görülmüştür.

Sonuç olarak görülmüştür ki, frekans, genlik ve sinyal tipi değişimleri ne olursa olsun titreşim uygulamak ısı transferini ve basınç kaybını aynı anda yüksek yüzde oranlarında arttırmıştır. Örneğin, ısı transferi test edilen üç türbülatör tipi için *Tip 1*'de 116%, *Tip 2*'de 105% ve *Tip 3*'de 64% şeklinde artmıştır. Sürtünme faktörü için bu artışlar *Tip 1*'de 35%, *Tip 2*'de 61% ve *Tip 3*'de 95% şeklinde olmuştur.

Anahtar Kelimeler: Mekanik titreşim, ısı transfer artımı, turbülatör, ısı değiştirgeci

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NOMENCLATURE

Re	: Reynolds number
Nu	: Nusselt number
Pr	: Prandtl number
K	: Thermal conductivity of air (W/m K)
LMTD	: Logarithmic Average Temperature Difference
Hz	: Frequency unit
MB	: Dynamics Modal Shaker
ρ	: Density (kg/m^3)
μ	: Dynamic viscosity (Ns/m^2)
f	: Friction factor
V	: Volume (m^3)
A_c	: Cross-section area of the inner pipe (m^2)
A_s	: Surface area of the inner pipe (m^2)
C_p	: Specific heat (J/kg K)
T_i	: Inlet temperatures of the fluid (K)
T_o	: Outlet temperatures of the fluid (K)
h_m	: Heat transfer coefficient (W/m^2 K)

1. INTRODUCTION

It is known that the use of turbulators in heat exchangers is an important method of increasing heat and has been the subject of dozens or perhaps hundreds of work to date. Many researchers have been working on turbulator geometries in particular to achieve the highest heat transfer with the lowest pressure loss. Some studies that aim to enhance the heat transfer in the tabulated heat exchangers include vibration effects. Hence this study aims to show the effects of mechanical vibration on heat transfer and pressure drop in a concentric type heat exchanger with turbulators inside it.

In this study, three types of manufactured turbulators are used. Experiments have been carried out at various flow rates in a concentric type heat exchanger in which a constant temperature boundary condition is provided. The flow velocities are determined to be a turbulent flow regime and air flow is passed through the inner tube, where the turbulator is located. The water passes through the annulus of the co-axial pipe helps to keep the outer wall of the inner pipe passing at a constant temperature. Wall temperatures, inlet and outlet fluid temperatures and ambient temperature are measured to check the stability and to use in calculations in the experiments. At the same time, the pressure loss values are calculated by measuring the pressures at the inlet and outlet of the exchanger. The number of experiments to be done is determined with the help of the experimental design method *Taguchi* analysis. Then the effects of heat transfer and pressure loss on the design parameters of the experiments are investigated.

The tests are done with and without mechanical vibration, to see and compare the effects of vibration on heat transfer and pressure drop. The frequency, amplitude (acceleration) and the type of the vibration varied to see their effects on the heat transfer and pressure drop. The Nusselt number and friction factor are extracted as the dimensionless heat transfer and pressure drop results after the experiments with and without mechanical vibration.

2. GENERAL INFORMATION ABOUT HEAT EXCHANGERS AND TURBULATORS

As is known, a heat exchanger is a gadget used to exchange heat energy between at least two fluids, between a strong (solid) surface and fluids, or between solid particles and liquids, at various temperatures and in heat contact. In heat exchangers, there is usually no thermal interaction and external work. Typical applications involve heating or cooling a fluid stream of concern and evaporation or condensation of fluid streams to one or more components. In another applications, the objective probably to regain or reject heat, or sanitize, pasteurize, fractionate, distil, concentrate, crystallize, or controlling a process fluid. In the heat exchanger, heat transfer occurs between liquids in two ways, namely, direct or indirect. The direct method is the thermal transfer between the liquid and this through a separation wall or a transient way outside the wall, such exchangers are referred to as direct transfer type, or simply recuperate. An indirect method is the separation of liquids from each other. It is an ideal method, because liquids do not mix and do not leak.

Heat exchangers are the devices in which heat energy is stored and released through the surface of the exchanger or matrix. This is done by means of a heat exchange between the hot and cold fluids, which is indirect or simply renewals. In these changes due to different pressure of rotation of the valve/switch matrix fluid leakage occurs between the fluid streams from one to the other. The most common examples used frequently for this type of exchanger are pipe and pipe exchangers, motor radiators, condensers, evaporators, air heaters cooling towers. Sometimes the exchanger in which no phase change occurs in any liquid in the exchanger is called a reasonable heat exchanger. It can be sources of heat inside exchangers such as electric heaters and nuclear fuel elements. In some cases, such as hot heaters, boilers or liquid exchangers can occur inside the exchanger combustion and chemical reaction. In some of the engines or heat exchangers such as reactors, engine tanks, surface scrapers, and irritating vessels, used mechanical appliances. In general, heat transfer is carried out in a separation wall from recovery. This occurs by conduction, but this does not mean that, the heat pipe acts as a separation wall in the heat exchanger with a thermal tube, but that distinguishes the liquid conduction inside the heat tube. In general, they facilitate heat transfer, because they work on condensation and evaporation well. However, if the liquids are unbreakable, such as oil, water or any other two types, they do not mix with each other.

In this case, the interface between fluids replaces the surface that conveys the heat, we get rid of the separation wall and this happens if we use a heat exchanger with a direct connection.

Since the discovery of heat exchangers and their use in the field of heat exchangers has played a major role in many industrial applications, such as; power generation, oil industry, chemical processing, cooling, but its uses are not exclusively applications. The aim of the researchers to improve these devices and increase their efficiency urge them to improve and conduct intensive research on them because of the intensification of daily industrial needs and energy developments, over the past few decades have developed their implementation based on their application, type of liquids, heating and cooling flow rates, operating pressures and temperatures, and because of those research and the needs of factories and equipment for those heat exchangers a large number of different types of heat exchangers exist and are available for use, because of the high efficiency and safety and easy to control the temperature in which it exists and change it to the best and do not forget it. Easy maintenance as the cost of those heat exchangers has become reasonably low.

2.1 Classification of the Heat Exchangers

The classification of heat exchangers according to transfer process is showed in Table 2.1.

Table 2.1. Classification of the heat exchangers

Classification according to transfer process

- *Indirect contact type*
  - *Direct transfer type*
    - *Single-phase*
    - *Multi-phase*
  - *Storage type*
  - *Fluidized bed*
- *Direct contact type*
  - *Immiscible fluids*
  - *Gas-liquid*
  - *Liquid-vapor*

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Classification according to number of fluids

- *Tow-fluid*
- *Three-fluid*
- *N-fluid (N > 3)*

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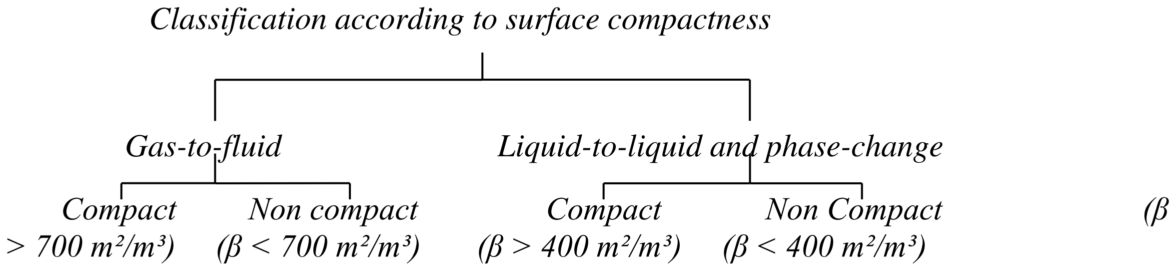
Classification according to surface compactness
Gas-to-fluid
Liquid-to-liquid and phase-change
Compact
Non compact
Compact
Non Compact
(β
> 700 m²/m³)
(β < 700 m²/m³)
(β > 400 m²/m³)
(β < 400 m²/m³)

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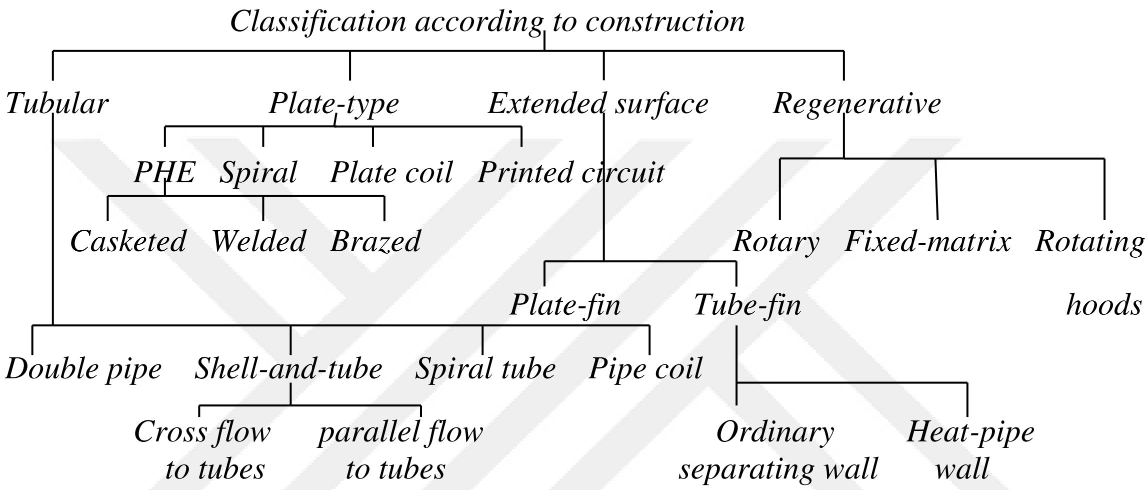
Classification according to construction
Tubular
Plate-type
Extended surface
Regenerative
PHE
Spiral
Plate coil
Printed circuit
Casketed
Welded
Brazed
Rotary
Fixed-matrix
Rotating
hoods
Plate-fin
Tube-fin
Double pipe
Shell-and-tube
Spiral tube
Pipe coil
Cross flow
to tubes
parallel flow
to tubes
Ordinary
separating wall
Heat-pipe
wall

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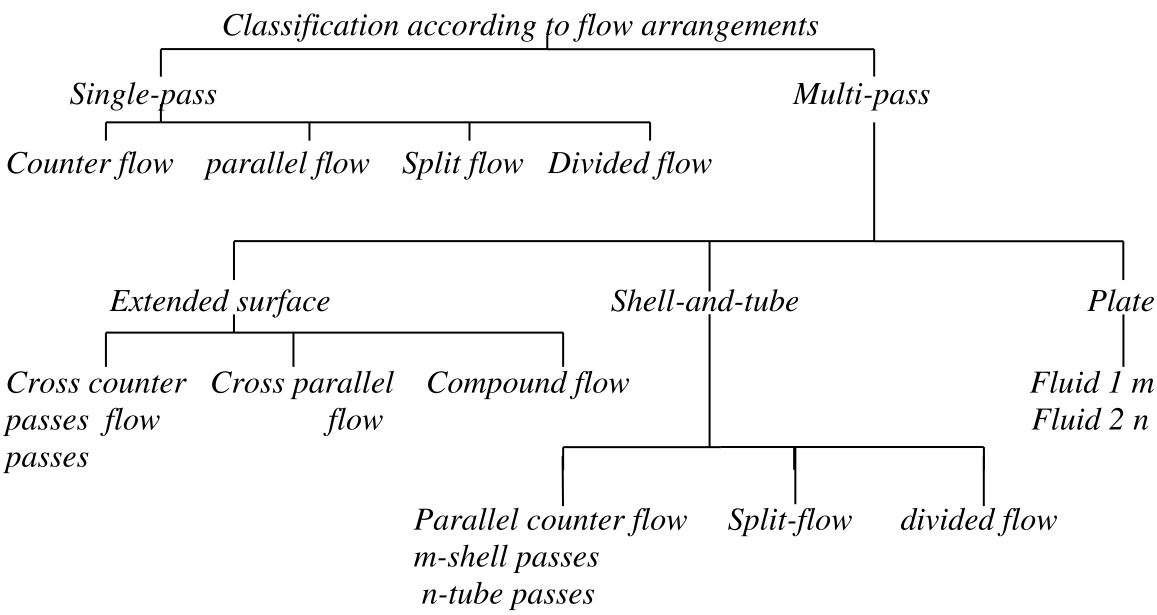

Classification according to flow arrangements
Single-pass
Multi-pass
Counter flow
parallel flow
Split flow
Divided flow
Extended surface
Shell-and-tube
Plate
Cross counter
passes flow
passes
Cross parallel
flow
Compound flow
Fluid 1 m
Fluid 2 n
Parallel counter flow
m-shell passes
n-tube passes
Split-flow
divided flow

……………………………………………………………………………………………

Classification according to heat transfer mechanism

Single-phase convection on both sides | *Single-phase convection on one, side, two-phase convection on other side* | *Two-phase convection on both sides* | *Combined convection and radiative heat transfer*

……………………………………………………………………………………………

2.2 Concentric Type Heat Exchangers

One of the simplest types of heat exchangers is double-pipe heat exchanger in which the liquid fluid flows inside another tube and flows between this tube and another tube surrounds the first tube As was indicated by Williams (2002), the heat exchangers are used in many different fields, for example industries for purposes such as food preparation, air conditioning, material processing and the type used in these areas are double pipe heat exchangers or concentric pipe. Naterer (2002) created a thermal driving force by passing the fluid streams of different temperatures parallel to each other. As we know, the heat exchangers are devices which exchange heat between two liquids of different temperatures separated by a solid wall. In general, heat transfer occurs in three main ways; radiation, conductivity, and convection. The biggest contribution to heat transfer in a heat exchanger is made through convection. With convection when heat is transferred through the wall, the pipes are mixed in a stream and the current stream removes the transferred heat. This maintains the gradient temperature between the two fluids (Deepica, 2016).

The main character that cause to transferring a heat is temperature or a difference between two temperatures. We can define the temperature as the amount of energy contained in the material. To transfer that energy from one substance to another, we use heat exchangers, and these tables can be either gases or liquids. Because it is necessary to control the temperature of incoming and outgoing flows, we use heat exchangers, because heat exchangers work to raise or reduce the temperature of these flows by transferring heat to or from the current (DeNevers, 1991). Additionally, it should not be forgotten that, there are some other conditions such as flow rates, fluid characteristics, fluid composition, etc. It has to be controlled, because a change, even if it is simple in this behavior, changes the temperature of the process and changes the amount of heat transfer. Heat transfer occurs when the heat starts to move. The movement in the heat changes the temperature of the liquid and continues until it reaches a point where the temperature distribution is constant. In order for the temperature to reach this state, the temperature depends on time (Williams, 2002).

Figure 2.2 and Figure 2.3 are the illustrations of the double pipe heat exchangers and direction of the fluid flow inside the pipes in concentric or (double pipe) heat exchanger transfer and pressure drop, there are no general correlations to predict enhancements. However, by changing the number of permits, the coefficient of heat transfer gain can be obtained in small cases of pressure drop.

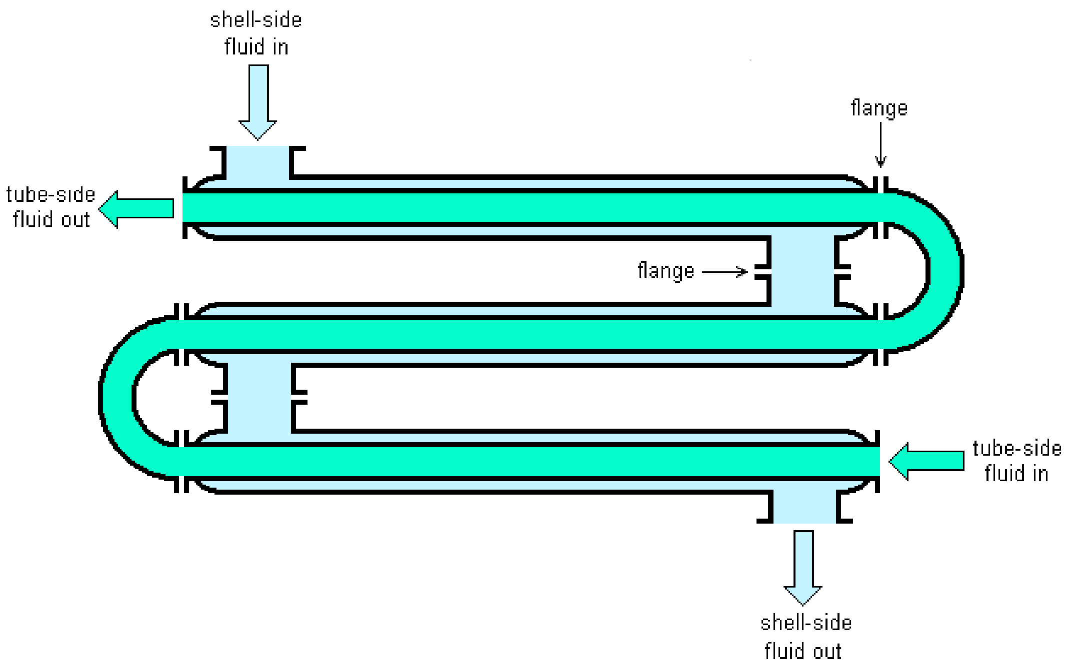


Figure 2.1. Double pipe heat exchanger

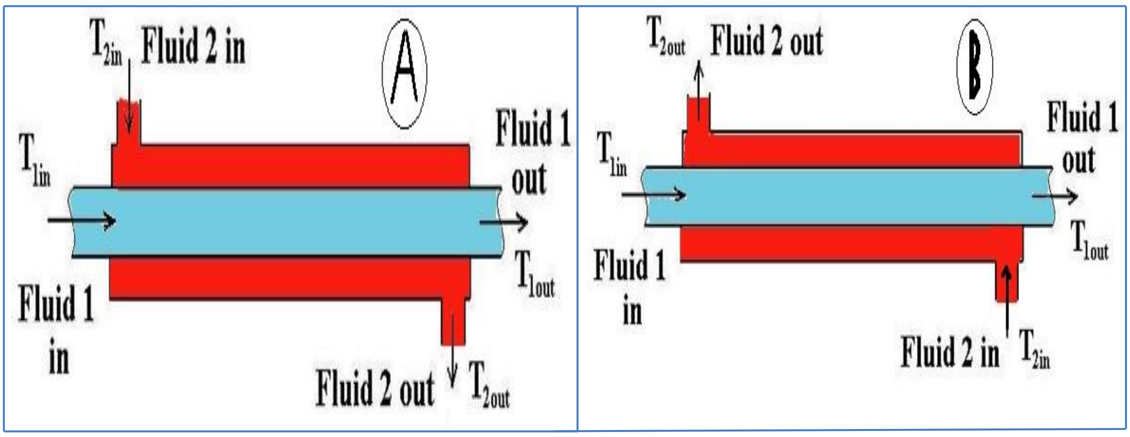


Figure 2.2. Flow direction in double pipe heat exchanger **a)** co-current or parallel flow **b)** counter-current flow

2.3 Turbulators

The device that changes the flow method of liquids from the laminar flow to the turbulent flow are called turbulator. The areas in which turbulent flow is desired, are parts of the air wing surface and industrial applications, such as; heat exchangers and fluid mixing. The term “turbulator” is derived from the word turbulent and is also a term applied to a variety of applications, but as a technical or scientific sense it has no generally acceptable meaning, yet they are approved with machine parts used in rotary drums, sterilization, mixers and granulators, modified fans for agriculture and agriculture, in accordance with the United States and several other countries. Thus, the term became universally known.

In order to enhance the disturbance inside the tube, they inserted the pumps or fixed mixers. In this way, we have obtained the equipment with high efficiency. It is more if it is used with high viscosity liquids inside the laminar flow system. This causes an increase in the heat transfer coefficient to five times. These inputs are most often used to promote boiling alongside heat transfer. However, the inclusion of these inputs is not effective for condensation in general in the tube and is almost at an increase in low pressure, due mainly to the relationship between the geometry of the insert and the resulting increase in heat transfer.

2.3.1 Types of Turbulators

2.3.1.1 Twisted Turbulator

The twisted tape is one of the most important elements of optimization techniques, widely used in heat exchangers. A secondary flow or spiral in the liquid is caused by swirling devices. The diversity of devices can be used to be the reason for this effect which involves the insertion of the tube, the changing of the flow arrangements of the tube and the adjustment of the geometry of the conduit. Ribs, ribs and spiral pipes are examples of modifications to the air ducts. Include the insertion of a twisted tube insertion tape, helical ribbon or engraved type insertion screw and wire coils. Inside spiral flow devices, there is a cyclic liquid of the injector fluid and is considered to be a kind of pipe changing flow arrangement. As well as problems in the design and application of twisted tapes, the tapes have been and are still very popular and because of their outstanding thermal performance in single stage convection, boiling condensation. Figure 2.3 shows a typical view of twisted tapes.

Figure 2.3. A typical view of twisted tapes

Twisted ribbons increase the heat transfer coefficient but increase the low pressure within a small amount. Because of the design and comfort of the application, the swirl tape known to be one of the first vortex flow devices that used in single-phase heat transfer, which has made it used to generate a spiral flow in the liquid over decades. In order to carry a specific thermal load, use the twisted strips in the new heat exchanger to reduce the size of the new heat exchanger, enhance its strength and provide the fixed cost of the equipment. This is a good economic advantage. As we can increase the convection of heat pipe current exchangers and it should be noticed that the twisted bars can be used for renovation purposes as well. One of the most important features of twisted ribbons with multiple tube belts is that they are easily installed, as we can easily remove them, making them easy to clean the tube side in the cans. The detail of twisted type of the topologies clarification is presented in Figure 2.4 as follows.

It was also referred that, the twisted taps are one of the modern techniques to increase the heat transfer within the double-tube heat exchanger. Kumar et al. (1970) used twisted tape turbulators in the water since it does not cause rust when it flows vertically through the steel pipe. The coiled that used was a coiled type of swirl generator which made a significant increase in heat transfer of up to right, but at the same time increased the proportion of the monopoly power to 90%.

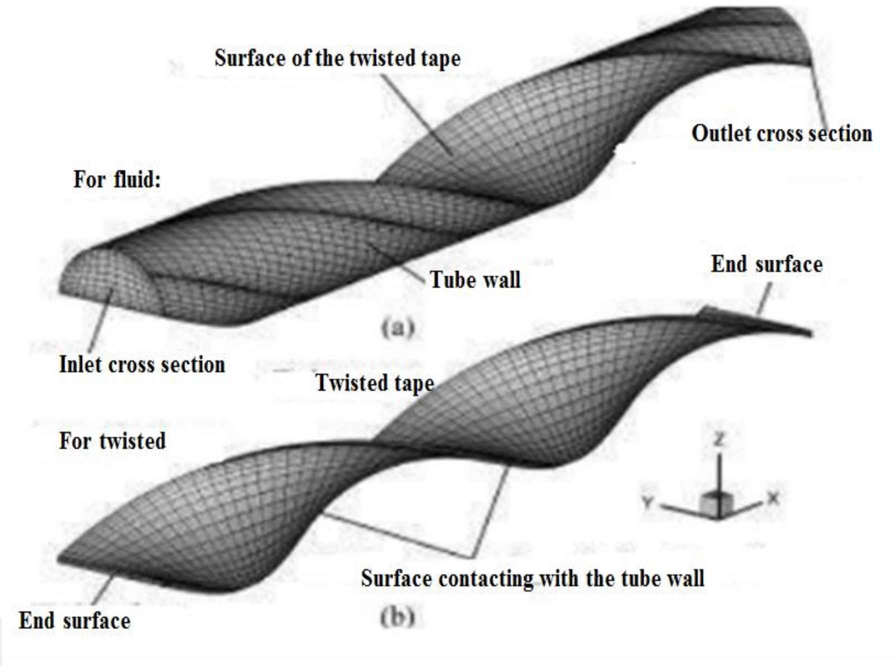


Figure 2.4. Detail of twisted type the topologies clarification

2.3.1.2 V-Type Nozzles Taps

For the purpose of obtaining stable pumping power and improving the efficiency of the heat transfer process, Yakut et al. (2004) presented their experience by using the conical ring turbulator to detect their effect on the turbulent heat transfer, low pressure, and vibration induced by the flow. The view of typical V-type nozzles inserts are shown in Figure 2.5. Yakut and Sahin (2004) found that the highest degree of heat transfer can be obtained for the smallest pitch arranging. This is due to the discovery of the increase ratio between the Nusselt number and the Reynolds number. We can go back to the reason to get these results, because they used circular conical pumps and benefit from the characteristics of the vibration caused by the flow in it and this is why a significant promotion of heat transfer within heat exchangers. For more detail some view of topologies of V-type nozzles inserts are shown in Figure 2.6.

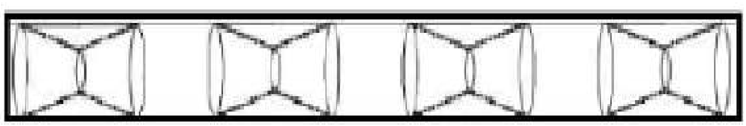


Figure 2.5. View of typical V-type nozzles inserts

Durmus (2004) also found the effect of cutting conical taps on the rate of heat transfer by placing it in the heat exchanger tube. He also studied the effect of heat transfer with four different types of turbulator and used different conical angles.

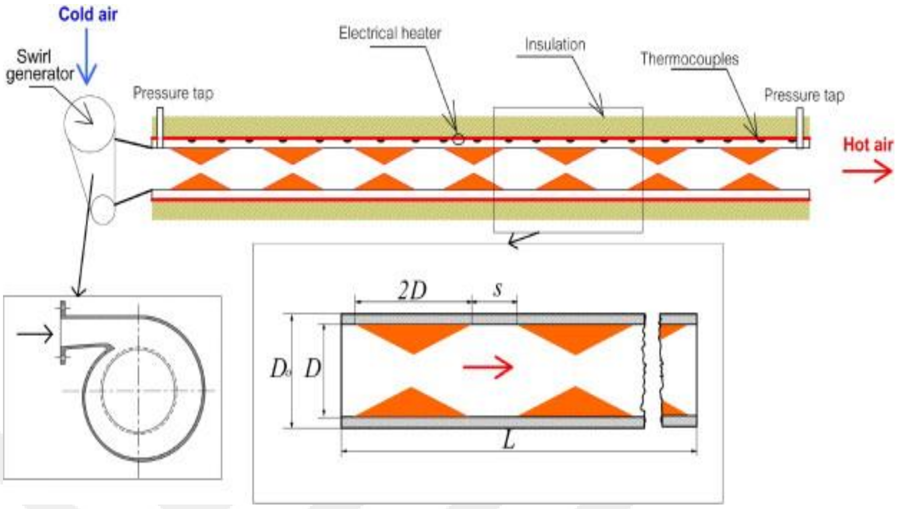


Figure 2.6. Detailed view of topologies of V Nozzles Inserts

2.3.1.3 Helical Type

Helical taps is rings on the base of the turbulence is a type of turbulator and is one of the varieties which is used to promote heat transfer. The efficiency of the helical type in the breakage of the thermal boundary layer, which generally have a good performance in the mixing of liquids and this by generating vortices, which made it a device that is recommended to be included within the tubes of turbulators to promote heat transfer equipment, but must take into account any an increase in heat transfer rate or friction loss because this greatly reflects the energy saving when heat exchange. Eiamsa et al. (2007) inserted helical taps in the core area of the tube as shown in Figure 2.7 to ensure heat transfer and pressure drop characteristics in single phase fluid.

Many researchers have also conducted extensive studies to improve the performance of helical bands, such as the work done by Gul and Evin (2007). In that experimental study short helical spiral generators were used. The helix angles are 30°, 45° and 60° and they used a different range of Reynolds number (5,000 to 30,000). There are also many researchers who have studied numerically; such as Pethkool et al. (2011). They studied the effect of corrugation and the effect of the use of corrugated helical pipes in double pipes in heat exchangers and studied deep heat transfer and reached the result that to increase the rate of heat transfer should be used for modified pipes instead of regular pipes.

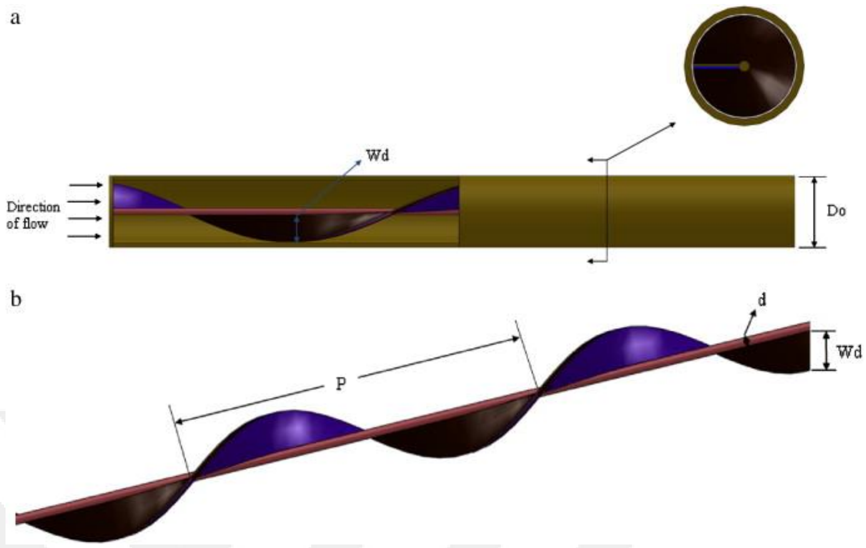


Figure 2.7. Helical tapes and the detail view of topologies

2.3.1.4 *hi TRAN* or Wire Matrix Taps

Occasionally the issue to be understood is straightforward poor heat execution. Despite the fact that, the heat exchanger architect dependably go for high heat exchange coefficient, this can some of the time be hard to accomplish with a regular plain tube outline. Much of the time, this is because of the properties of tube side liquid, for example, high thickness and low heat conductivity. Occasionally low heat exchange rates are an outcome of the game plan of the exchanger, for example, when single pass tube packs are require. Whatever the reason is, poor tube side execution can for the most part be maintained a strategic distance from by considering the utilization of heat exchange upgrade advancements. Designing gadgets, for example, *hi TRAN* framework components in fluidly gives expanded heat exchange with respect to the plain tube. A run of the mill perspective of *hi TRAN* wire lattice is exhibited in Figure 2.8. At the point when liquid move through the plain tube the liquid closest to the divider is subjected to the frictional drag which has the impact of backing off the liquid at the divider this laminar limit layer can altogether diminish the tube side heat exchange coefficient and thusly the execution of heat exchanger. Embedding effectively the profiled *hi TRAN* wire lattice component into the tube will disturb the laminar limit layer, making the extra liquid shear and blending, her by limiting the impact of frictional drag. The *hi TRAN* wire lattice tabulators are especially successful at improving heat move productivity in tubes working at low Reynolds number (laminar to transitional stream). In spite of the fact that heat exchange increment is most noteworthy in the laminar stream locale (up to 16 times) critical advantages can be gotten in the transitional stream administration (up to 12 times) and turbulent administration (up to 3 times). While there is an expansion in frictional protection related with *hi TRAN* framework, the measure of improvement to such an extent that arrangement can be discovered which offer expanded heat exchange at comparable or low weight drop than a plain tube.

Figure 2.8. *hi TRAN* wire matrix and the typical view of it

The component of laminar heat move in flat tube is mind boggling, as they can be constrained, regular, and blended convection. The overwhelming instrument relies upon the condition and physical properties of the liquid being heated or cooled (Holmen, 1992). The liquid is constrained through the tube at sufficiently low speeds the characteristic convection lightness compel still have impact on stream design inside the tube. Meta and Eckert (1964) proposed the constrained blended and free convection administrations in flat tube. Nusselt number remedy by Sieder and Tate (1936) and Oilver (1962) for laminar, constrained, and blended convection are utilized to contrast the outcome from and different heat move parameter in the event that *hi TRAN* tube embeds. Chandrasker et al. (2010-2011) led the investigations which include use of a wire curl embed fitted in round about tube which demonstrated that there was ascend in heat exchange rate with inconsequential ascent in contact factor in plain tube and tube with wire loop embeds.

2.3.1.5 Solded Wire Wound Inserts

Selvam et al. (2012) demonstrated the impact of holding and without holding of wire wound loop lattice turbulator on the heat exchange for a completely created turbulent stream. The commonplace perspective of soled wire wound supplements wire framework is appeared in Figure 2.9. Some tests are led by keeping up steady divider temperature. Tests are performed on three diverse wire snaked curl network turbulator of various pitches of 5, 10 and 15 mm without holding of the turbulator. Three comparable kinds of heat exchangers are manufactured and the wire curled loop grid turbulator with various pitches of 5, 10 and 15 mm are embedded in the heat exchangers and holding is done on the surface of the tube area. Results have shown that, the heat exchange rate improves contrarily with the pitch of the wire looped curl framework tabulators with holding. With a pitch of 5 mm, the turbulator without holding have brought about right around 25.4% improvement when contrasted and plain tube. Then again, for pitches of 10 mm and 15 mm the upgrade were 20.7% and 16.8%, separately.

Figure 2.9. View of typical Soled wire wound when it was inserts wire matrix

2.4 Vibration and Vibration Tests

2.4.1 Vibration

When we discuss utilizing vibration we should first discuss a kind of vibration as general data, we have four sorts of vibration: free vibration, constrained vibration, hose vibration, and constrained damped vibration.

Free vibration is a vibration in which vitality is not added to or expelled from the vibrating framework; it will just shake everlastingly in a similar limit. With the exception of some superconducting electronic oscillators, or maybe an electron in its circle around a nuclear core, there are no free vibrations in nature, all really submerged.

Forced vibration is one that supplements vitality into a vibrating framework, for instance into an outlined component where vitality put away in the spring is put away a little at any given moment in a vibrating component. The vibration will be constrained in sufficiency, and will increment with time until the point that the component is wrecked. The limit of the constrained vibration, the solidness has tumbled to a specific esteem where the vitality misfortune in the much adjusted cycle of the vitality picked up.

Damped vibration is the one that is the vitality loss of the vibrating framework. This misfortune can appear as a mechanical rubbing, for instance in the hub of the pendulum, or an unsettling influence when the vibration framework upsets its border. The disintegration of the broke up vibration will in the long run deteriorate to zero.

Undamped vibration does not experience the ill effects of vitality misfortune. Light and vibrating vibrations have a slight loss of vitality that might be unimportant or not, contingent upon the idea of the vibrator screen. Inertial powers in these frameworks are huge contrasted with cloud or contact powers.

Heavily damped vibrations experience the ill effects of high vitality misfortunes. They are described by high erosion or frictional powers contrasted with the powers of the inertial framework. The vigorously damped framework is one that moves from introductory relocation to relentless state without surpassing, in at least time. For instance, a basic pendulum hanging in a light oil compartment can just tumble from a high beginning stage to hang straight while never swinging to the opposite side. An overdamped framework acts like a fundamentally damped framework; however, it takes more time to achieve harmony. For instance, a straightforward pendulum hanging in a bowl of nectar might be excessively costly. The sufficiency changes are portrayed previously. The vitality of a vibrating framework is identified with the adequacy, more abundancy, and more vitality. The recurrence of a vibrating framework relies upon whether it is constrained or not. A constrained vibration might be made to accept the recurrence of the driving capacity. Unforced vibrations happen at a characteristic recurrence subject to the qualities of the vibrating framework. Essentially, the common recurrence of the vibrating framework increments is with the hardness of the components and reductions with its mass. Entirely, the vibration recurrence is just characterized if the amplitude is steady. In the event that the amplitude changes, the wave form contains a vast scope of frequencies.

2.4.2 Vibration Tests

In this thesis it is tried to find the effect of vibration on the heat transfer in the double pipe heat exchanger, for that we used mechanical vibration and three different types of vibrating test (sinusoidal or sine, random noise and chirp signals).

2.4.2.1 Sinusoidal or Sine Vibration Testing

Sinusoidal or Sine Vibration has the state of a sine wave as seen in Figure 2.10. The parameters used to characterize sinusoidal vibration testing are amplitude (generally speeding up or removal), recurrence, clear rate, and the quantity of ranges. The amplitude is set in the recurrence go.

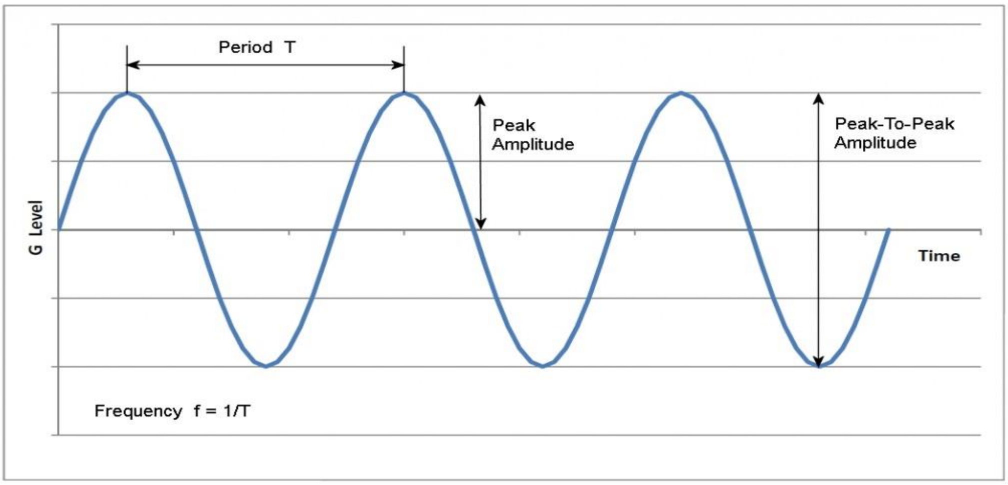


Figure 2.10. The shape of a Sinusoidal vibration wave

An ordinary sinusoidal vibration test profile is appeared in Figure 2.11. The abundancy can be steady or variable within a sine vibration test, the vibration waveforms are cleared through a scope of frequencies; in any case, they are of discrete abundancy, recurrence, and stage at any moment in time. An essential perception is that the relocation increments with the lessening of recurrence for a specific increasing speed. At low frequencies, the uprooting may surpass the breaking points of the test gear. This is the reason a few particulars utilize uprooting to extend in the low-recurrence band.

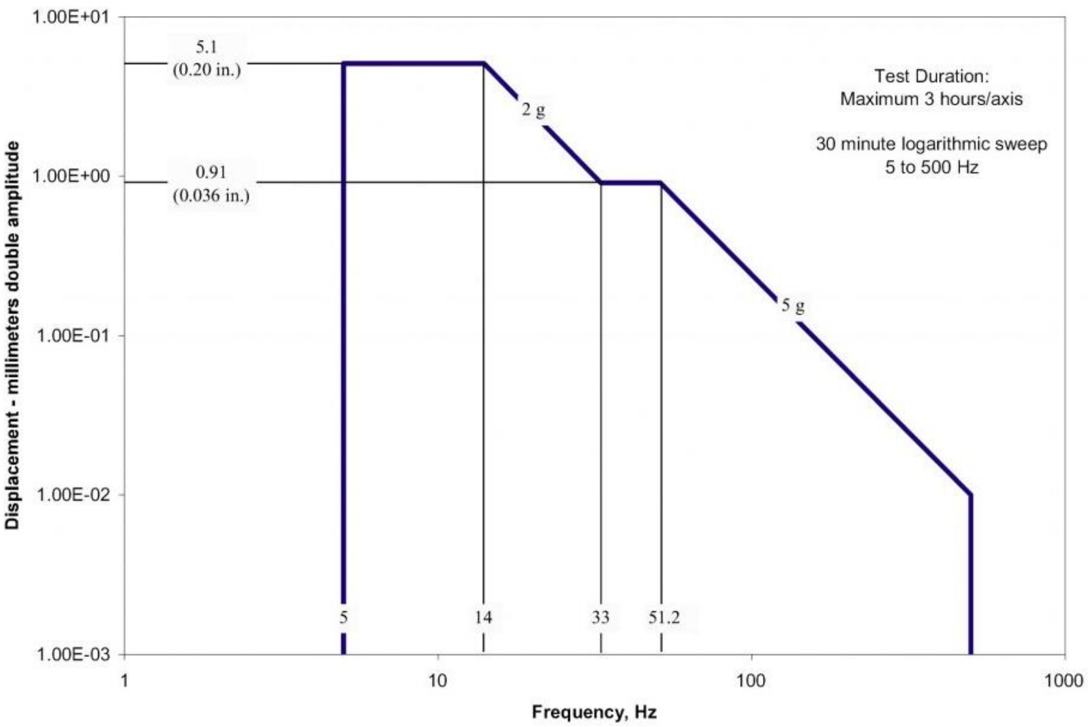


Figure 2.11. Test profiles of Sinusoidal wave

Sine vibration cannot be discovered for the most part in reality aside from in the event that you appended your item to devices or a gear like a responding compressor or engine activity at a settled recurrence, Why is it? It regards discover resonances (enhancement limit in the gadget tried), it is a basic development, it likewise creates a steady speeding up as indicated by the recurrence. Likewise, it was most likely moved from the old test strategies before advanced PC controllers. Nevertheless, sine vibration is not identified with field life unless the item is presented to settled frequencies amid its lifetime.

The definition and some parameters of a sine vibration test are presented as follow:

Amplitude: The abundancy is largely resolved to test the vibration of the sine as removal or speeding up. In the Delserro Engineering Solutions (DES) explore, speed is occasionally utilized as a part of the determination. As appeared in Figure 2.10, the adequacy can be communicated as pinnacle or crest to top. At the point when the relocation is utilized to decide limit, it is characterized in both of the pinnacle units in SA (mmSA) or Peak-to-Peak (mmDA) units in DA. SA speaks to a capacitance (pinnacle) and da speaks to a twofold capacitance (top to-crest). At the point when speeding up is utilized to decide the capacitance, its units are commonly G or millimeters per square second (mm/s^2) or meters every second (m/s^2).

Frequency: Frequency is characterized as cycles every second. Its' units are Hertz (Hz). Frequency is equivalent to the corresponding of the period.

G: One G is equivalent to the speeding up delivered by earth's gravity and is equivalent to 386.1 inch/s^2 or 9.8 m/s^2.

Octave: is the interim between one recurrence and another contrasting by 2:1.

Period: The time it takes to finish 1 cycle. Its' units are seconds. Period is not regularly utilized as a part of the meaning of a sine test. It is recorded here due to its connection to recurrence. Period is the proportional of the recurrence.

Resonance: is the frequency in which the extent of the amplitude of the instrument under test is contrasted and the limit of the vibration table. The ringer is for the most part known as an amplification of 2:1 or more.

Sweep and Sweep Cycles: A scope is characterized as a cross starting with one recurrence then onto the next. A scope cycle fluctuates starting with one recurrence then onto the next and after that back to the gazing recurrence. For example in Figure 2.10., a compass could be a cross from either 5 to 500 Hz or from 500 to 5 Hz. The output cycle will change from 5 Hz to 500 Hz, at that point back to 5 Hz. A few particulars require examines while others require befuddling filter cycles.

Sweep Rate: The normal at which the recurrence is transmitted. The units for the range rate are regularly octave/min or Hz/min. The octave every moment is the logarithmic scope rate while the Hz/min is the straight compass rate (http://www.desolutions.com).

2.4.2.2 Random Vibration Testing

Arbitrary vibration is a variable waveform. The thickness is characterized utilizing the Power Spectral Density (PSD). While the sinusoidal vibration happens at unmistakable frequencies, the arbitrary vibration contains every one of the frequencies all the while. What's more, stage changes happen after some time with irregular vibrations. Sine evening out and vibration tests cannot be irregular.

Vibrations in reality are normally irregular. The vibrations of autos, planes, and rockets are largely irregular. An arbitrary vibration test might be identified with the administration life if the field vibrations are known. Since the irregular vibration contains all frequencies at the same time, as appeared in Figure 2.12 the type of Random vibration's wave every reverberation of the item will be energized all the while, which might be more regrettable than energize exclusively as in a sinusoidal test.

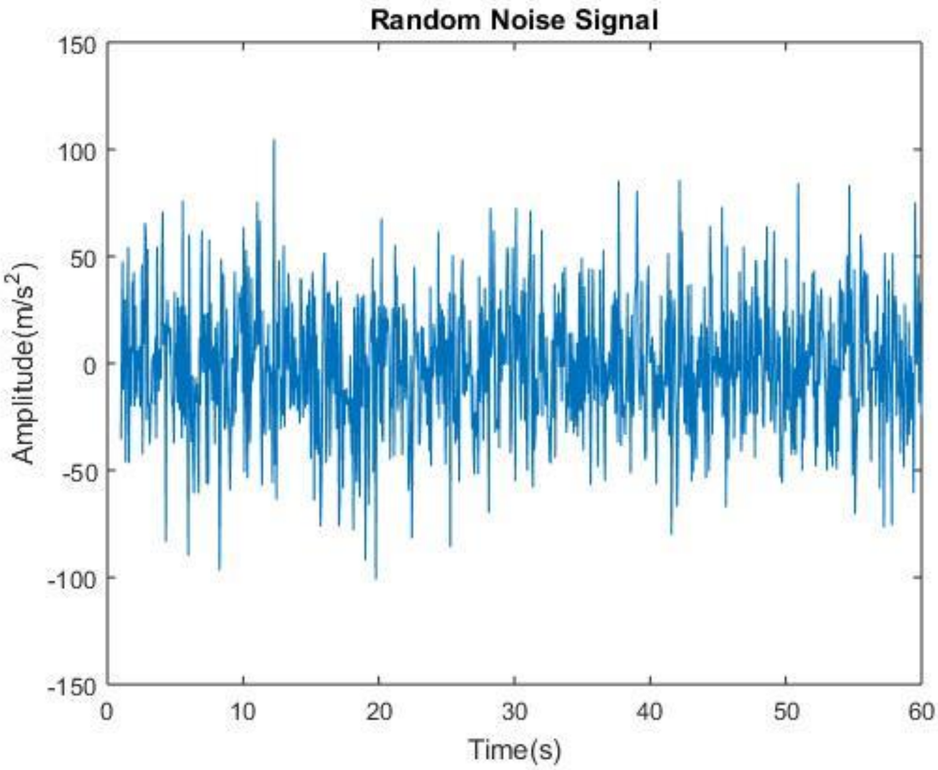


Figure 2.12. Vibration of a random wave form

The run of the mill irregular vibration design PSD is appeared in Figure 2.13. It is known to PSD the frequencies. The square foundation of the territory under the ebb and flow PSD bends delivers a Grms. The assurance of the Grms is not sufficient on the grounds that an extensive variety of spectra can prompt the same Grms.

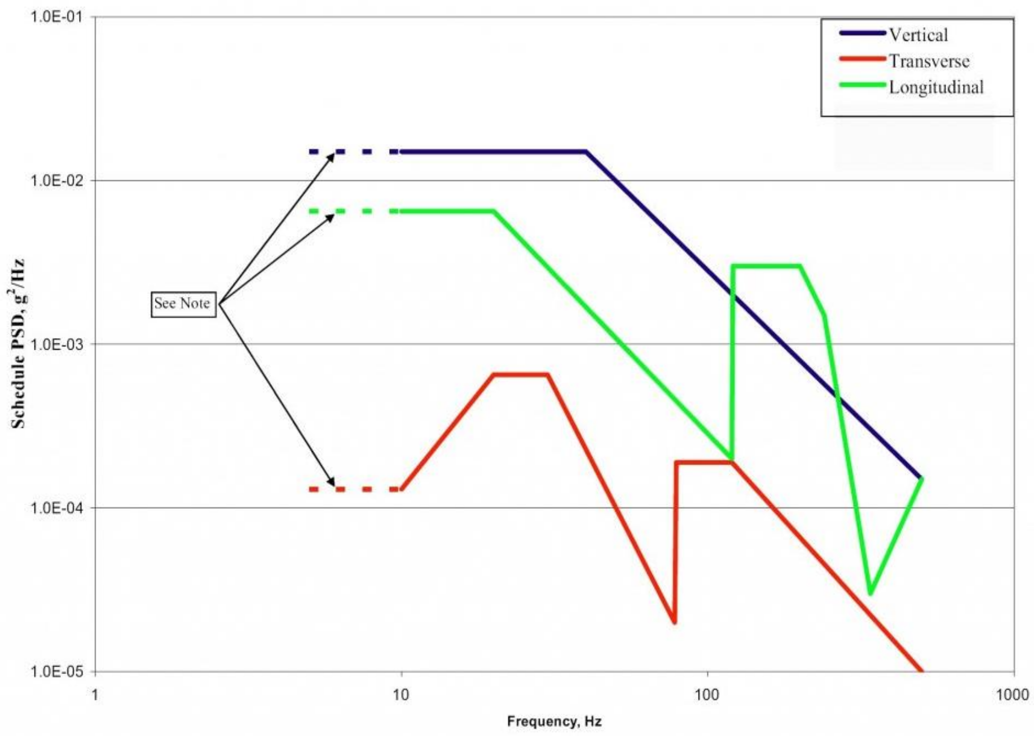


Figure 2.13. Typical random vibration profile from

A portion of the parameters and definitions for irregular vibration tests are:

Implications: Since irregular vibration changes consistently with time (this is the reason it is called arbitrary), the controller takes tests or pictures of vibration information after some time, normal progressive examples. The averaging happens in each band of determination.

Average Weighting Factor: is an exponential weighting factor that characterizes how quick the controller responds to changes. The controller responds quicker for little weighting factors versus slower for bigger weighting factors.

Statistical degrees of freedom (SIDOF): The quantity of autonomous esteems (measures) used to get a characterization as indicated by a specific recurrence. Higher SEDOF implies more advances are being taken.

- For the measurement channel,

$$SIDOF = 2K$$

- For control channels,

$$SIDOF = 2K\,(2N - 1)\,n$$

- *K*: Number of averages per control loop
- *N*: Averaging weighting factor

- *n*: Number of control channels

Grms: Grms is accustomed to deciding the general vitality or the level of speeding up of the irregular vibration. The Grms (square of the normal root) is figured by taking the square base of the zone under the bend of PSD.

Kurtosis: The fourth snapshot of the Potential Density Function (PDF).It measures the high G substance of the flag. The kurtosis of a Gaussian PDF is 3.

Number of lines: The recurrence scope of the test isolated by the determination of the range is the quantity of lines. A vibration controller or range analyzer will play out its estimations for each limited band.

Open Loop/Closed Loop: Closed Loop implies that the controller will constantly alter the drive flag to represent changes in the reaction of the gadget being tried that are returned by the control quickening. Open circle implies the drive flag will be settled or the controller will quit modifying the drive flag paying little mind to changes in the reaction of the gadget under test.

PDF: A likelihood Density Function: A histogram demonstrating the likelihood of event and the dispersion of information.

Power Spectral Density (PSD) or Acceleration Spectral Density (ASD): It defines the force of the irregular vibration flag versus recurrence. Its units are generally G^2/Hz or $(m/s^2)^2$/Hz.

Sigma (σ) and Sigma Clipping: Sigma is the standard deviation of a factual PDF. Gaussian PDF dissemination is expected for irregular vibration, which takes the state of a chime-molded bend. Since the abundancy or power of the irregular vibration will change after some time, the time spent in different capacitive amplitude is estimated utilizing the VDP. Figure 2.14 shows Gaussian PDF. The vertical hub will be 1/G, the even hub will be sigma, and μ is the normal that is zero to control the shaker. For the Gaussian conveyance, 68.2% of the G top triggers happen between ± 1 sigma, 95.4% between ± 2 sigma, and 99.7% between ± 3 sigma.

Vibration controllers enable you to cut the pinnacle abundancy capacity utilizing the Sigma Clipping. Various determinations make it conceivable to characterize a shot at ± 3 sigma. The recurrence, *G*, octave, period, and reverberation are the same as those predefined in the sinus test. The Gaussian Probability Density Function (PDF) is appeared in Figure 2.14 (http://www.desolutions.com).

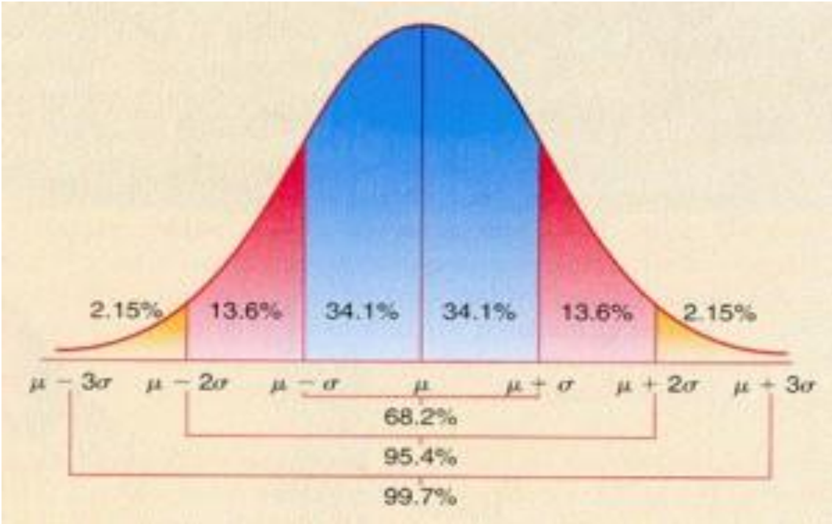


Figure 2.14. Gaussian PDF

2.4.2.3 Chirp Vibration

Chirps are omnipresent in nature and man-made frameworks. Give us initial a chance to count a couple of cases where the idea of peep normally develops.

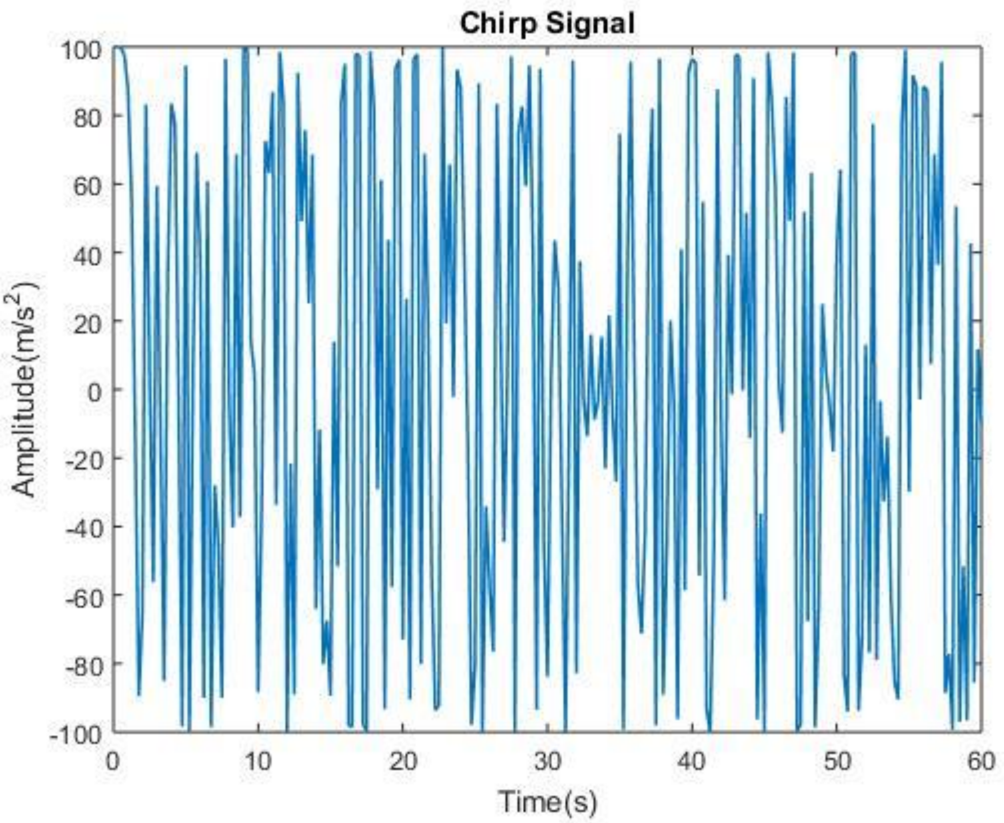


Figure 2.15. Random vibration wave form

Audio signals: Chirps are normally experienced in numerous sound signs, going from flying creature melodies and music (glissando) to creature vocalization (frogs, whales) and discourse. The alleged sinusoidal models are a run of the mill endeavor to speaking to sound flags as a superposition of chirper like segments (McAulay and Quatieri, 1986).

Radar and sonar systems: Cartilage signals are regularly found in common sonar frameworks. Most sorts of bats utilize an ultrasound framework in view of projections that can demonstrate the parameters to straightforwardly screen the echolocation execution (Nachtigal and Moore, 1986). Such a circumstance nearly looks like those of man-made radar and sonar frameworks, where trills are of basic utilize excessively (Rihaczek, 1969).

Wave physics: Low-recurrence chirps, (for example, e.g., PC1 motions. motions (Kodera et al., 1976) can be seen in the ionosphere as shrieking atmospherics. (Story, 1953). Numerous time-changing oscillatory frameworks bring forth chirp like practices: an excellent (and very much archived) case of a peep is given by the gravitational waves anticipated that would be transmitted by enormous astrophysical questions, for example, blending parallels (Schutz, 1989, Chassande-Mottin and Flandrin, 1999). Another illustration is given by breaking waves on a seashore, that have a wavelength balanced by the submerged prole of the ground, henceforth offering ascend to 2D chirps . From an alternate point of view, non-symphonious waves engendering in a dispersive medium are normally chirp by a distorting system (Sessarego et al., 1990).

Mechanics and vibrations: A worldview for a chirp is in certainty the note played by a diapason (or a harmony, or a pipe) with a period fluctuating length. Aside from music, such a marvel can be watched, e.g., in vibration signals recorded on auto motors, because of the time-changing volume of the gas starts chamber (Carstens-Behrens et al., 1999).

Spirals in turbulence: One of the numerous photos of turbulence is that of a gathering of spiraling reasonable structures (Moatt, 1993) when advected by a mean of and estimated at a given point in space, spatio-worldly segments of such questions are viewed as trills.

Biology and medicine: Other types of sound structuration of waves as trills emerge in biomedical signs, e.g., in EEG (epileptic seizure) (Bozek-Kuzmicki et al., 1994) or uterine EMG pregnancy withdrawals (Devedeux and Duchene, 1994).

Critical phenomena: In various basic wonders (Sornette, 2000) it has been proven that, widespread particular practices (regularly, control law divergences) are finished by a trill segment identified with quickening motions (e.g., gathering of antecedents on account of seismic tremors, theoretical rises on account of budgetary accidents (Johansen and Sornette, 1999).

Special functions: Finally, trills have likewise been appeared to exist in simply numerical questions, for example, weier-strass (Sornette, 1998, Berry and Lewis, 1980) or Riemann (Jaard and Meyer, 1996, Flandrin, 1989), capacities and the chirp structure of (minimally bolstered and least stage) Daubechies wavelets (Daubechies, 1992) of expansive request.

2.5 Effects of Vibration on Heat Transfer

Vibration is the movement of particles or bodies connected to the body or system displaced by the equilibrium mode. When we move the system from its balanced and steady position, vibration occurs, but the systems of the systems after changing their fixed position tend to return to their stable state. This is what happens in systems such as elastic forces, such as a spring-related mass or gravitational forces, such as a simple pendulum. Dukibati and Srinivas, 2012). Moreover, it should not be forgotten that, in the number of natural phenomena such as the movement of oceanic tides and rotary machines and fixed machinery in different structures such as buildings, ships and in vehicles often occur vibrations regardless of the nature, shape and size of these elements or systems (Galway, 2010). Vibration and vibration quality on the specific properties of the highly tested systems because the sources of vibration and their types are very complex. In addition, there is a strong coupling between the concepts of mechanical vibration and the spread of vibrations and sound signals across the earth and air to create potential sources of discomfort, and even physical damage to persons and structures adjacent to the source of vibration (Dym and Den Hartog, 2014).

Free, forced and self-excited are the three categories that classify vibration on them. If the vibration occurs without any external influence, or with a subtle impediment in the absence of external energies, we call it free vibration, but if there is an external force that moves the system and this force constantly raises doubt on the energy supply of the system here we have obtained the forced vibration and the addition of vibrations Forced or inevitable forced. Vibrations that depend on periodic and deterministic oscillations and also the system in which it moves back and forth through its position of balance are what we call self-excited vibrations, the back and forth motion of machines or machine components, and component that moves back and forth or oscillates can called mechanical vibration (Dukkipati, 2007).

After we got some general information about vibration and mechanical vibration, how it occurs and knew about types of vibration, also the more important thing that we have to pay attention in this study is vibration in heat exchanger and this effect on it.

In fact Mechanical vibration causes heat transfer and fluid flow destabilization, and ignoring the influence of vibration on the heat transfer process could lead to non-economic problems. However vibration or mechanical vibration in The vibration caused by the flow causes damage within the heat transfer devices and the tubes and causes these vibrations to generate noise and that is what always happens in heat exchangers and in particular heat exchangers and shell.

In recent years, the effect of vibrational phenomena on heat transfer has aroused great interest. In many cases, the heat transfer equipment is the same in case of vibration or vibration can be adjusted without affecting its performance. For example, a refrigerated supply tank oxidized in a liquid fuel rocket on a commercial vehicle, a plane on a trip, etc. But there is a kind of vibration that we do on our own and that aims to increase the efficiency of heat exchangers, like the vibration of the tubes, vibration of the boundary layer or surface vibration (Sastry, 1982).

The disturbance caused by vibration of the boundary layer mechanically is a major reason to make the boundary layer thinner. Surface vibration is one of the most active techniques used and has been considered by researchers. This is for the purpose of promoting heat transfer because the surface vibrations increase the density of the disturbance in the border layers. This results in an increase in heat transfer rate because these vibrations it mixes liquids so that it can transmit heat better and thus produces an increase in heat transfer coefficient. In addition, the vibration of the pipes in the heat exchangers is an important factor in the heat exchanger process. Vibrations are caused by unstable fluid flows that occur in the flow. These are the turbulent pressure pulses, the spiral start and the separation of the tubes in the transverse flow, the hydro elastic interaction of the heat transfer elements with the whole, the acoustic phenomena (https://www.thermopedia.com/content/1242/).

The greatest effect of unstable hydrodynamic forces is observed in the tubes with dissociation flows. Flow separation occurs from the tube surfaces where there is an element of the transverse velocity of the flow and mainly affects the resistance of the pipe vibration in the heat exchangers. The dynamic effect of the flow on the vibrating tube depends on the flow velocity and the vibration characteristics of the tube. With the separate transverse flow on the tube bench, the reference velocity is assumed to be the flow velocity in the narrowest section of the bank at the plane tube (https://www.thermopedia.com/content/1242/).

Equipment designed to contain components that operate at high temperatures and strongly vibrate are known to use heat transfer rates as a sign of vibration intensity.

2.6 Literature Review

In literature there are numerous studies related to the tabulated heat exchangers, however there are limited works that investigate the effects of vibration in the heat exchangers. In this section the selected papers will be introduced.

Pethkool et al. (2011) experimentally investigated the augmentation of heat transfer in a concentric tube heat exchanger using helically corrugated tubes. The effects of pitch to diameter ratio (*P/D* = 0.18,0.22 and 0.27) and rib height to diameter ratio (*e/D* = 0.02,0.04 and 0.06) on heat transfer, isothermal friction factor and the thermal performance factor of the Reynolds group was examined from 5,500 to 60,000 and the average increase in heat transfer was found to be between 123% and 323%. It was also found that the maximum thermal performance factor was 2.3 at *P/D* = 0.27 and *E/D* = 0.06 when Reynolds decreased.

Shinde et al. (2012) showed the comparative thermal analysis of Helix changer with the segmental heat exchanger. Thermal analysis for conventional shell and tube heat exchanger and helix changer for five different baffle inclination angles (α) was observed that continual helical baffles can eliminate dead regions in shell side. The very low pressure was found in α >35°. Continuous spiral barrier heat exchangers constituted the highest heat transfer coefficient with respect to a segmental heat exchanger.

In a pilot study (Kim et al., 2007), the effect of tube vibration on critical heat fluxes was observed to find the difference between critical heat flux and flux induced vibration. The experiment was conducted for different values of mass flow, amplitude, and frequency. They used the frequency range from 0 to 70 Hz. It was also suggested that there was an experimental relationship with tubular vibrations that predicts the critical growth of heat flux with reasonable accuracy at an average error rate of 27.75%.

Lee et al. (2004) carried out experiments with vertical annulus tube. They examined the effects of mechanical vibrations on critical heat fluxes under electrically heated condition and find the total increase of 16.4% in critical heat fluxes under mechanical vibrations. It was observed that reinforced turbulent mixing effect of vibration enhanced the critical heat fluxes values.

Go (2003) studied the effect of micro-fins oscillating due to flow-induced vibration. The fluid taken was air and velocities of air was 4.4 m/s and 5.5 m/s. Also, the micro-fin array was fabricated over the heat sinks. As the fluid moves over them, a vibration was induced which causes heat transfer enhancement. Data so obtained was compared to the plain heat sink. It was concluded that micro fins provided up to an 11.5% enhancement over a plain heat sink.

Yasir (2011) investigated effects on heat transfer coefficient and heat flow characteristics of sub cooling and the saturated boiling process. Acoustic vibration levels ranging from 5 kHz to 15 kHz were used, and the results show an increase in the heat transfer coefficient in the flow of larger amounts of boiling under the radiator. This increases up to 14% of the low to high frequencies used in the experiment. Similarly for the saturated boiling laboratory with two different heat fluxes of 15 kW/m^2 and 29 kW/m^2 and found that increasing the frequency increases the heat transfer coefficient by increasing the mass flow.

Other studies conducted in the last century to investigate the effects of vibration in cylinders on the rate of heat transfer, and away from the traditional methods of Shine and Jarvis (1963) used different methods do not resemble the previous study when they used the cylinders were between 0.032 and 0.072 inches they installed the test cylinders horizontally on both ends and the cylinder's enthusiasm at a point near one end, and used sine waves in the air where the cylinders vibrated at the height of a sine wave in the air.

In order to study the relationship between vibration and Reynolds number, they extended the research done by Shane and Jarvis (1963). This time they studied the effect of vibration on the number of Reynolds at two degrees of temperature of the surface of the cylinder and at the different test sites along the cylinder, Reynolds number to express the vibration intensity of 2 *American Force Bear/p* but the identification of the vibration intensity of the cylinder diameter was also considered as 2 *American power* is actually the average speed of the movement of the cylinder and this means the correlation between the increase between the number of Reynolds and vibration intensity.

McAdams (1954) studied the effects of vibration on heat transfer coefficient and the relation between increased vibration and increase in temperature coefficient, he found that the change in heat transfer coefficient is generally equivalent to the forced load curve, which means the relation of the direct relationship Where the frequency band was used from 15 to

75 cycles per second, and the amplitude range used in this experiment was between 0.002 to 0.99

Mehrabian et al. (2001) investigated heat transfer mechanisms in double-pipe heat exchangers and reported higher heat transfer coefficients in the laminar flow system. The phenomena that contribute to the promotion of heat transfer are mainly natural convection, secondary flows and surface roughness. There are, of course, other reasons for this improvement that must be studied empirically. The question is what part of this improvement is due to natural convection, secondary flows and surface roughness that remain unanswered.

Pak et al. (1972) They used a horizontal cylinder and exposed the entire system to both surface and fluid vibrations at the same time, so they studied the effects of vibration on heat transfer from the horizontal cylinder and noticed a significant improvement in heat transfer up to 200% due to vibrations and also found that there is no A relationship between the number of nuclei and measured measures.

Lemlinch (1955) studied the effects of vibration on heat transfer involving natural convection from electrically heated wires, of three different diameters, subjected to transverse' vibrations in air. Marked improvement in the coefficient of heat transfer even to the extent of quadrupling the film was obtained by vibrations of 39 to 122 cycles per second. In addition, a pair of dimensional bonds was introduced as a function of the number of Reynolds oscillations for vibration in air or other binary atoms.

Fand and Peebles (1962) argued that the physical mechanism of the interaction between the free load of a horizontal hot cylinder and the horizontal transverse vibrations is essentially the same as if the acoustic vibrations or mechanically updated.

Weaver and Fitzpatrick (1998) studied the vibration problems caused by the single-phase phase flow of the cortex in tree and tube heat exchangers and suggested preventive approaches against vibration damage from tubes.

Pettigrew and Taylor (1994) reviewed the main mechanisms of vibration induced by the two-phase flow and the impacts of dynamic parameters on the tube vibration. They revealed the damping properties of the shell-and-tube heat exchanger vibration induced by the two-phase flow by using a semi-empirical formula and established a practical design criterion by integrating the dynamic parameters related to the dumping into the empirical formula Pettigrew and Taylor (2004).

For the effect of mechanical vibrations, experimental and numerical results reported by Gould et al. (1966) showed that the heat transfer rate increased approximately linearly with the vibration amplitude.

In another work Shokouhmand et al. (2008) studied the effect of flexible tube vibration on pressure drop and heat transfer in heat exchangers considering viscous dissipation effects.

Shi et al. (2014) numerically studied the heat transfer of a two dimensional model in the channel using flow-induced vibration in various Reynolds numbers. Results showed that the maximum heat transfer enhancement of 90.1% was obtained at $Re = 204$.

The paragraph that was above talked about in detail about a number of experiments and experiments conducted by the researchers for the purpose of studying and analyzing the effect of vibration in different methods on the coefficient and rate of heat transfer within the heat exchanger and the use of different types of turbulator and its effect on the rate of heat transfer, Through what the researchers reached, we produced some important points

- The greater the severity of critical vibration, the higher the heat transfer rate
- The most character that effect on vibration is the temperature and the temperature of the test independent of vibration.
- The mode of vibration play a masseur effect of the heat transfer rate, and it is dependent directly on the intensity of vibration Eckert and Drake (1959) and Neely (1964).

3. EXPERIMENTAL SETUP

3.1 Experimental Procedure

This thesis was carried out on a pre-established heat exchanger experimental system at Firat University, Department of Mechanical Engineering, Heat Transfer Laboratory. The schematic diagram of the test setup is given in Figure 3.1.

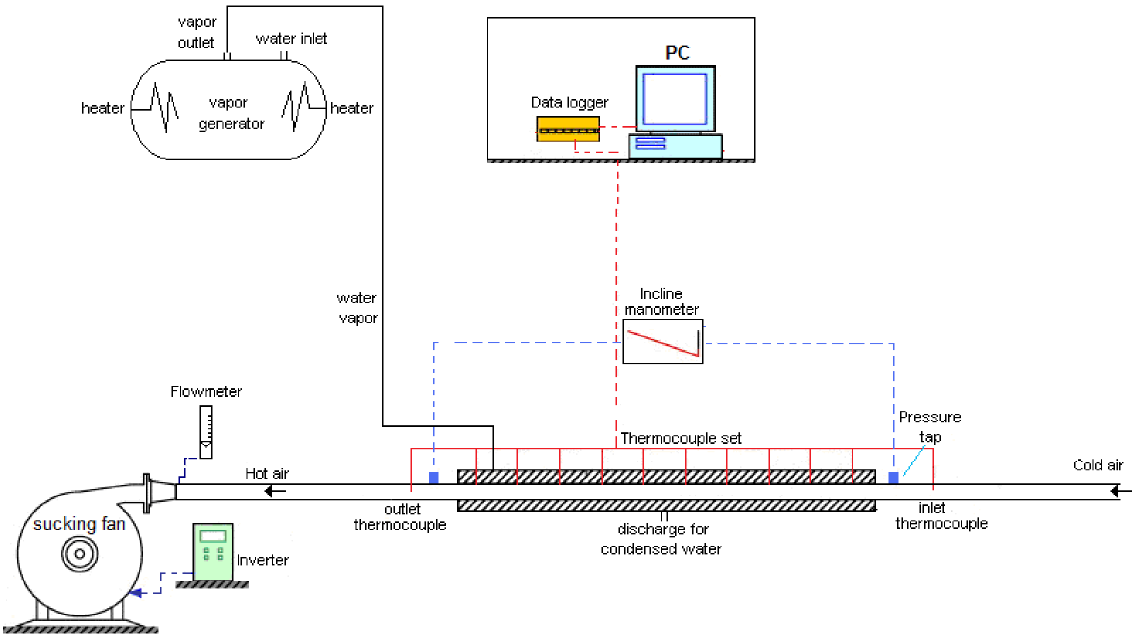
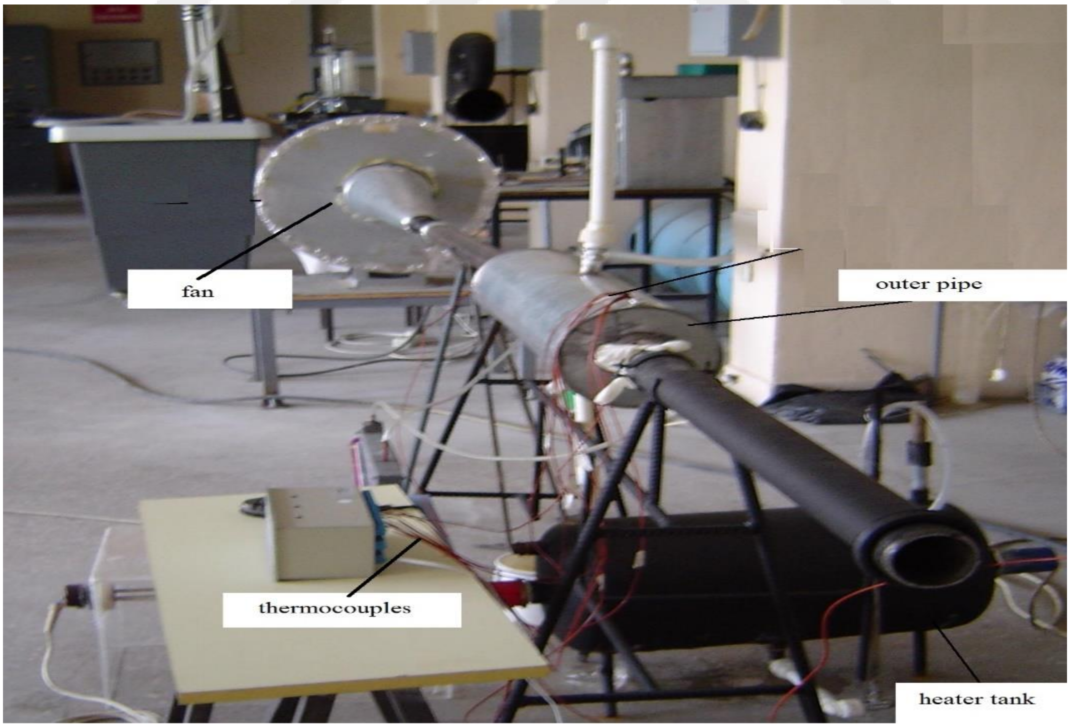


Figure 3.1. Schematic view of whole setup (Celik et al., 2018)

The turbulators used in the coaxial heat exchangers have various geometric designs. Schematic views and photographs of the plates are given in Figure 3.2 and Figure 3.3.

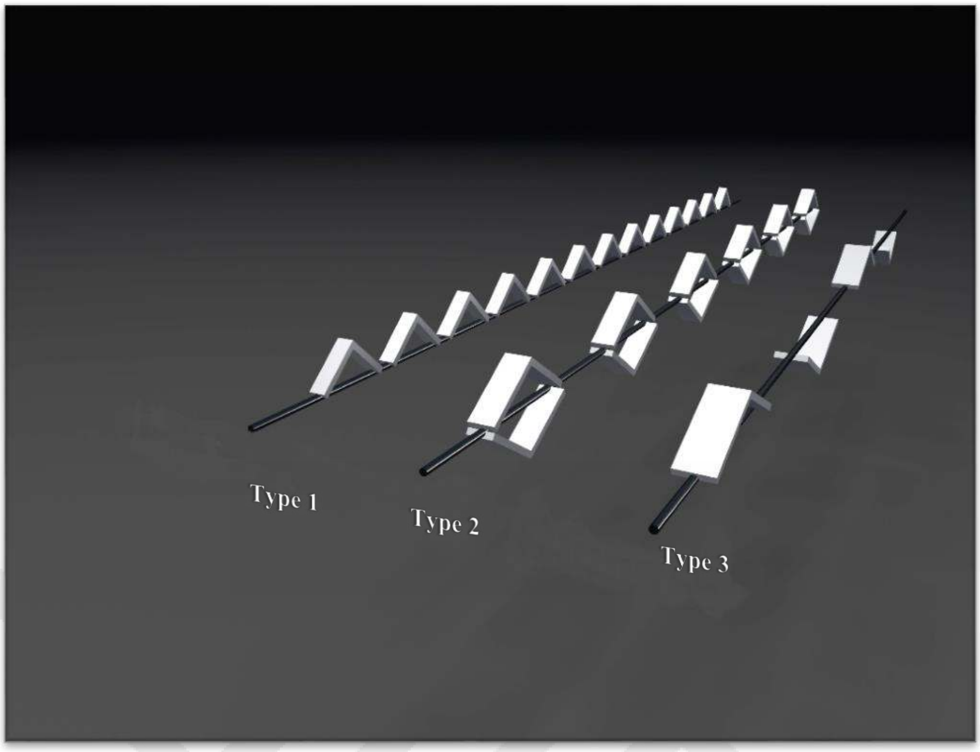
In the test setup, hot water vapor was continuously fed into the gap between the inner pipe and the outer pipe. The outer surface of this saturated water vapor and the inner pipe are in continuous contact. Thus, a constant temperature boundary condition is provided on the outer surface of the inner pipe (T_w = 100 ° C). In the steam boiler used in the experiment, there are 2 heaters in each 1500 W power.

a)

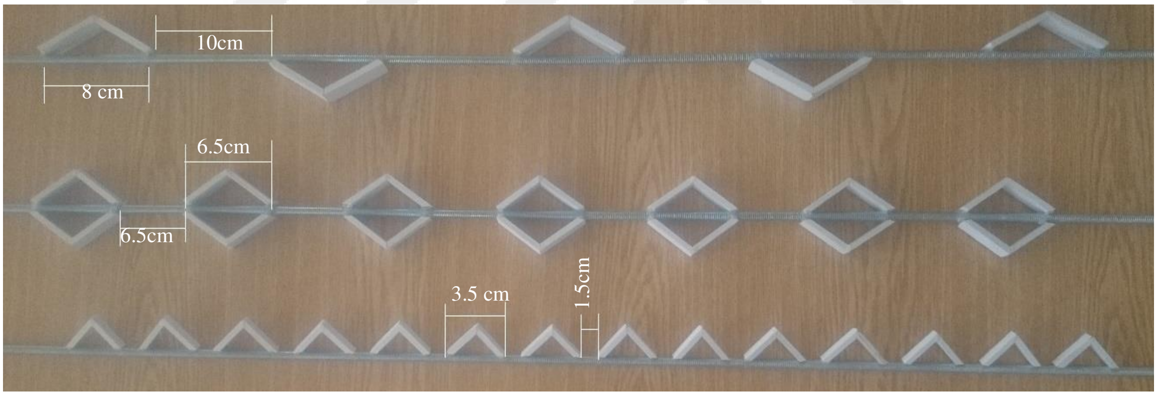


b)

Figure 3.2. Picture of whole setup from two views



a)



b)

Figure 3.3. Tested turbulators, **a)** Schematic view **b)** Photo of the turbulators

In the experiments, temperature measurements were made with thermocouples of 0.5 mm thickness, Teflon insulated, *T* type Cu-Cons (copper-constantan). One of the heat couples is adhered to the outer surface of the inner pipe with epoxy-metal adhesives in coaxial pipe system. Heat dissipation has been neglected, either by plate thickness or, if necessary, by adhesive thickening. The other side of the thermal pair is connected to a computer-connected data collector. Temperature measurements were taken at 10 points, and pipe inlet and outlet temperatures and ambient temperature were recorded continuously. A constant wall temperature is provided and the temperature is expected until the continuous regime is reached.

Temperature values measured with thermal pairs can be saved directly to the computer via data collector connection. The data collector is the CAMPBELL CR10 X model. The data collector has a fully programmable and controllable structure. Especially when it is possible to set the time, to have a fast response time, to have a flexible program and to make general changes in the program, computer or electronic engineering information does not require the use of general coding. Figure 3.4 shows the image of the data collector main unit and Figure 3.5 shows the image of the multiplexer to which the thermal couples are connected.

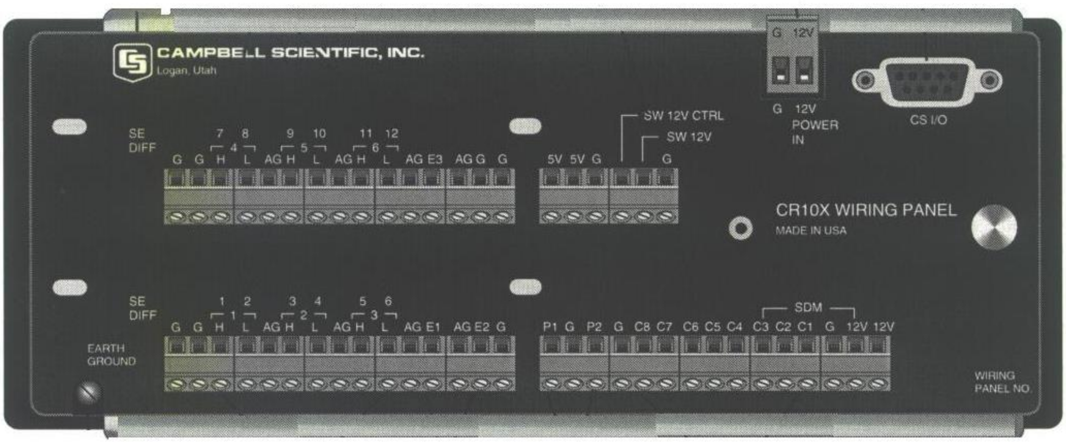


Figure 3.4. Data logger unit

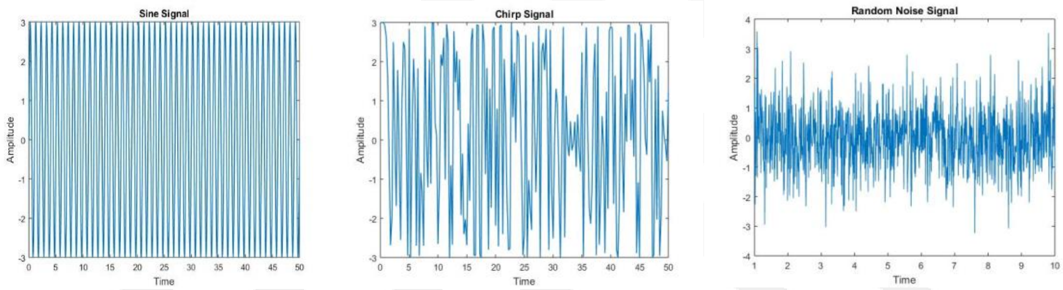
Figure 3.5. Data logger amplificatory and measurement logs

During the test, air was passed through the inner tube by means of a radial fan. With the aid of an inverter, the fan setting is changed so that flow at high speed is achieved. The average velocity of the air was determined by a wind gauge in a position perpendicular to the flow, which can be fully inserted into the pipe.

A digital pressure gauge is mounted on the pressure sockets fitted to the inlet and outlet of the pipe. Thus, in all experiments, the pressure difference was determined as mbar.

Inside the concentric tubes forming the heat exchanger; it is 90 cm long, 6.5 cm inside diameter, 6.7 cm outside diameter, 0.2 cm thick copper wire. Copper sheet is obtained by properly twisting and welding. The inner pipe is made from as smooth as possible. The outer pipe is 75 cm long, 25 cm in diameter, 1 mm thick galvanized sheet. In order to prevent the loss of heat, 2 cm thick insulation was made around the outer pipe.

Three different kinds of vibrations as sine, chirp and random noise signals with three different amplitude values were used for the experimental studies. The samples of vibrations are given in Figures 3.6 (a-c). The frequency values were applied as 100, 300 and 600 Hz for sine signals and with 100, 300 and 600 Hz lower frequency and with 1000 Hz upper frequency for chirp and random noise signals.



a) **b)** **c)**

Figure 3.6. Samples of **a)** sine signal, **b)** chirp signal, **c)** random noise signal

All signals were generated by using signal generator of OR36 Vibration Analyzer. The generated signals were amplified by using MB Dynamics Signal Amplifier and the turbulators were excited by using MB Dynamics Modal Shaker with longitudinal vibrations. The amplitude of vibrations were measured by using a transducer and applied as 100, 300 and 600 m/s^2. The schematic view of vibration experimental setup is given in Figure 3.7 and the photo of the device attached to the setup is presented in Figure 3.8.

As the experimental design parameters, dimensionless Reynolds number (*Re*), turbulator geometry (three types of the tabulators will be called hereafter *Type 1*, *Type 2* and *Type 3*) were selected. The number of *Re* was tested at 3 different values in the experiment (~10,000, ~30,000 and ~50,000).

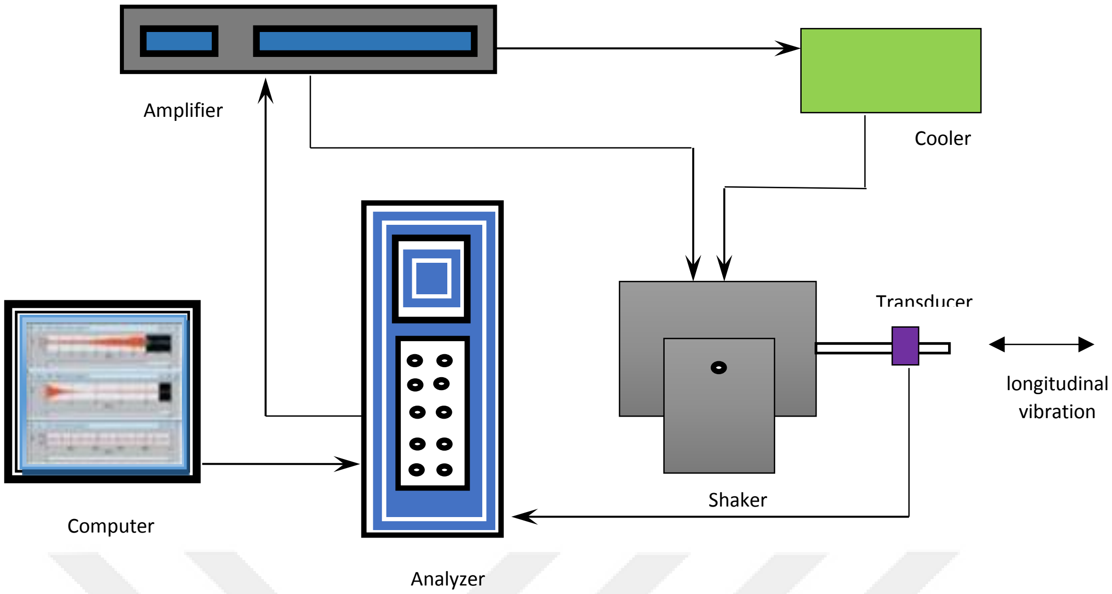


Figure 3.7. Vibration experimental setup

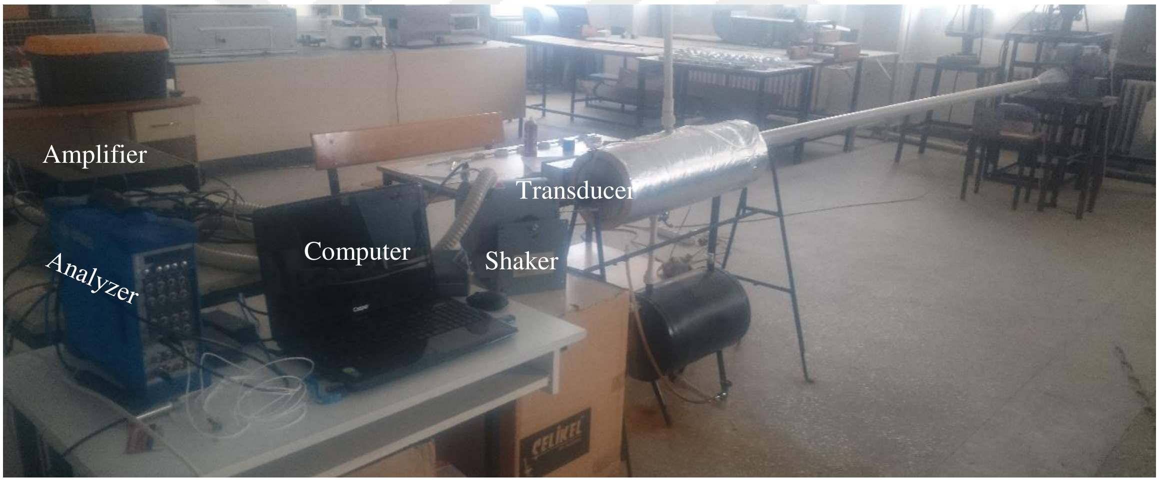


Figure 3.8. Photo of vibration device

As mentioned the design parameters affected the heat transfer are assumed to be (A) Reynolds number, (B) frequency, (C) amplitude, (D) type of the turbulator and (E) type of the vibration. Each of these 5 parameters varied 3 times. However a simple design of experiment methodology was used to minimize the number of the experiments, namely Taguchi method. Detailed information about Taguchi analysis can be found in the literature and books. For this reason, this section deals with the adaptation of the method to the study rather than the introduction of the design method. *Genichi Taguchi* suggested a method to develop orthogonal arrays which produce good results with a low number of trials. Instead of changing factor levels one by one, orthogonal arrays suggest changing factors simultaneously. In this way, the Taguchi approach received broad acceptance in engineering, particularly in the field of production (Celik et al., 2018).

As mentioned earlier, the selected experimental design parameters were Reynolds number, frequency, amplitude, type of turbulators and type of vibration. The Reynolds number was tested with 3 different values within the experiment (10,000, 30,000 and 50,000). The frequency was tested with 3 different values (100, 300, 600 Hz). The amplitude was tested with 3 different values (100, 300 and 600 m/s^2). Finally the type of turbulators was tested with 3 different values (*Type 1*, *Type 2* and *Type 3*) and the type of vibration varied 3 times (sine, chirp and random noise). Since there are totally 5 design parameters (coded as A, B, C, D and E) and the levels of each design parameter were 3, then the total number of experiments should normally be $3^5 = 243$ in a full factorial design. However, an orthogonal array of 27 was created in this study instead of 243 and hence 27 experiments were conducted. Table 3.1. shows the experiments to be performed with the L_{27} array and the factors used in the experiments and their levels. The validity of this assumption will be controlled by confirmation experiments performed under optimum operating conditions and presented later in the article.

Table 3.1. The levels of the factors

A (Re number)	**B (frequency)**	**C (amplitude)**	**D (type of turbulator)**	**E (type of vibration)**
1	1	1	1	1
1	1	1	1	2
1	1	1	1	3
1	2	2	2	1
1	2	2	2	2
1	2	2	2	3
1	3	3	3	1
1	3	3	3	2
1	3	3	3	3
2	1	2	3	1
2	1	2	3	2
2	1	2	3	3
2	2	3	1	1
2	2	3	1	2

2	2	3	1	3
2	3	1	2	1
2	3	1	2	2
2	3	1	2	3
3	1	3	2	1
3	1	3	2	2
3	1	3	2	3
3	2	1	3	1
3	2	1	3	2
3	2	1	3	3
3	3	2	1	1
3	3	2	1	2
3	3	2	1	3

3.2 Data Reduction

Reynolds number, an important parameter in the experiments, was calculated based on air velocity and inner pipe diameter. Accordingly:

$$Re = \frac{\rho V d}{\mu} \tag{3.1}$$

where, ρ and μ represent the density and the dynamic viscosity of air. The values were selected according to average inlet and outlet temperatures $\left(T_b = \frac{T_i + T_o}{2}\right)$.

After converting pressure values obtained using digital manometer into *Pascal*, the values were placed in the Darcy equation to calculate the dimensionless friction factor.

$$f = \frac{2 \Delta P d}{l \rho V^2} \tag{3.2}$$

The experimental results related to heat transfer are expressed in terms of Nusselt number. It is firstly necessary to create energy balance in order to calculate the Nusselt number under fully-developed turbulent flow conditions with constant wall temperature. To this end, the heat gain of the air is calculated using the temperature difference between the inner pipe inlet and outlet temperatures. Then, the heat convection coefficient is found based on the assumption that the entire heat is transferred with convection. Therefore;

$$\rho V A_c C_p (T_o - T_i) = h_m A_s (LMTD) \quad (3.3)$$

where; ρ is the density of the fluid, V is the mean velocity of the fluid, A_c is the cross-section of the inner pipe, C_p is the specific heat of the fluid, T_i and T_o are the inlet and outlet temperatures of the fluid, h_m is the average heat transfer coefficient, A_s is the surface area of the inner pipe. *LMTD* is the logarithmic average temperature difference. *LMTD* is the temperature difference between the wall temperature, which is calculated by averaging measurements from 10 points, and the inlet-outlet temperatures.

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{ln\frac{\Delta T_1}{\Delta T_2}} = \left(\frac{(T_w - T_\mathrm{i}) - (T_w - T_O)}{\ln\frac{T_w - T_\mathrm{i}}{T_w - T_o}}\right) \quad (3.4)$$

If Eq. (3.4) is rearranged then heat transfer coefficient becomes as follows:

$$h_m = \frac{\rho V \frac{\pi d^2}{4} C_p (T_0 - T_i)}{(\pi d L)(LMTD)} \quad (3.5)$$

Thus, the Nusselt number is found as:

$$Nu = \frac{h_m d}{k} \quad (3.6)$$

where k is the thermal conductivity of air.

4. RESULTS AND DISCUSSION

In this chapter, first the verification of the experimental results will be presented. Then the tests with turbulators but without vibration will be shown. The heat transfer and pressure loss results will be presented by means of Nusselt number and friction factor. Both the Nusselt number and the friction factor are shown in graphs with the variation of Reynolds number in each graph. The graphs will show the comparison of the tested turbulators and empty pipe.

4.1 Verification of the Results

The presentation begins with a display of the validation of the results for smooth empty pipe. The Nusselt number *Nu* and friction factor *f* obtained from the present smooth pipe are, respectively, compared to well-known correlation of Dittus-Boelter (1930) and Blasius (1908) as shown in Figures 4.1 and 4.2.

The fully-developed turbulent flow Nusselt number suggested by Dittus-Boelter (1930) for smooth pipe is as follows:

$$Nu = 0.023 Re^{0.8} Pr^{0.4} \qquad [0.7 \leq Pr \leq 160 \text{ and } Re > 10000] \tag{4.1}$$

It is clear from Figure 4.1 that, the difference between present results and Dittus-Boelter's formula is nearly 25%, which may cause from measurement errors. The empirical correlation for the tested empty pipe is found as

$$Nu = 0.066 Re^{0.6} \tag{4.2}$$

where the regression coefficient (R^2) is found as 99%.

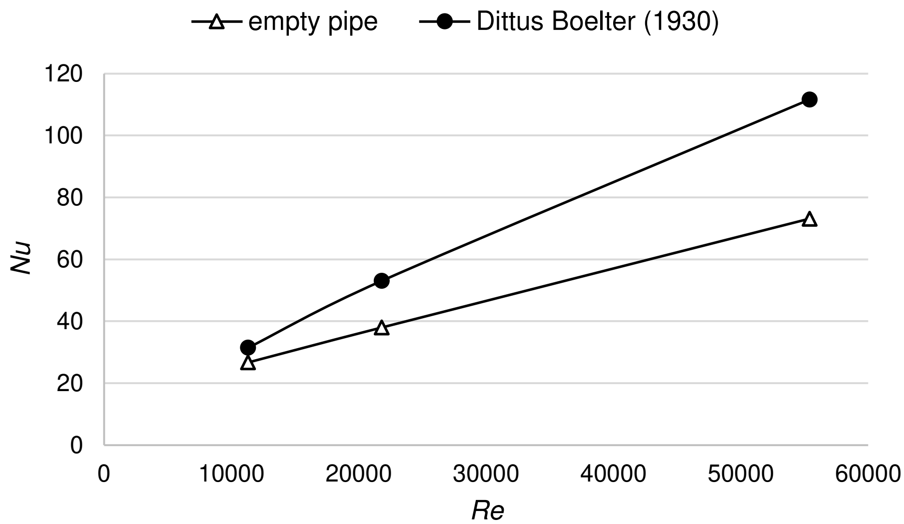


Figure 4.1. Nusselt number for empty pipe (without tabulator and vibration)

The second stage of the verification study is conducted for the dimensionless pressure loss, namely friction factor. Figure 4.2 presents this comparison graphically. The values obtained using the empirical correlation of Blaisus (1908) is compared with the values found in the experiments for the verification of the friction factor.

$$f = 0.316Re^{-0.25} \tag{4.3}$$

As is seen, the experimental study shows significant difference (~33%) from the compared empirical formula of Blasius. Both for the heat transfer and pressure loss measurements the difference of the measured values and theoretical values may be due to errors arising from the experimental assembly and the measuring tools. Further, the equation above is applied permanently to smooth pipes, whereas a rough copper pipe jointed by welding has been used in the experimental study. It is suggested that the welding joints, that’s to say the roughness is an important factor on the difference of the results.

The new-derived empirical correlation for the tested empty pipe is found as

$$f = 0.1327Re^{-0.13} \tag{4.4}$$

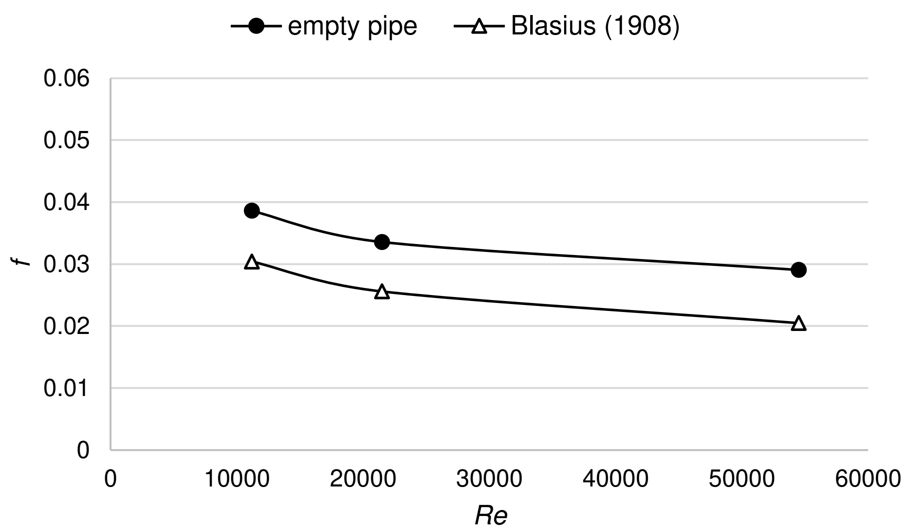


Figure 4.2. Friction factor for empty pipe (without tabulator and vibration)

4.2 Tests with Turbulators-without Vibration

As was previously mentioned three types of turbulators are used in this thesis. Figure 4.3 presents Nusselt number versus Reynolds number variations, for the turbulator types namely; *Type 1*, *Type 2* and *Type 3* with comparison of empty pipe. Similarly, Figure 4.4 shows the friction factor versus Reynolds number for all types of turbulators with the comparison to empty pipe.

Each turbulator has a different angle and different distance between triangles and each triangle has a different size as shown in Figure 3.3(a) and 3.3(b). Careful inspection of the curves in Figures 4.3 indicates that the highest heat transfer occurs for the *Type 3* whilst the lowest occurs for *Type 1*. The augmentation of the heat transfer for *Type 1*, *Type 2* and *Type 3* comparatively to the empty pipe is respectively 5%, 21% and 75%.

As it was pointed in the introduction section, the turbulator is a device that turns the laminar flow into the turbulent flow. Using turbulators in the concentric pipes cause a separation of the boundary layer in the turbulent flow. This separation results in enhancement of the heat transfer as expected. Furthermore, the boundary layer gets thinner as Reynolds number increases in the turbulent flows. Thus, Nusselt number increases with increasing Reynolds number, as is shown in Figures 4.3.

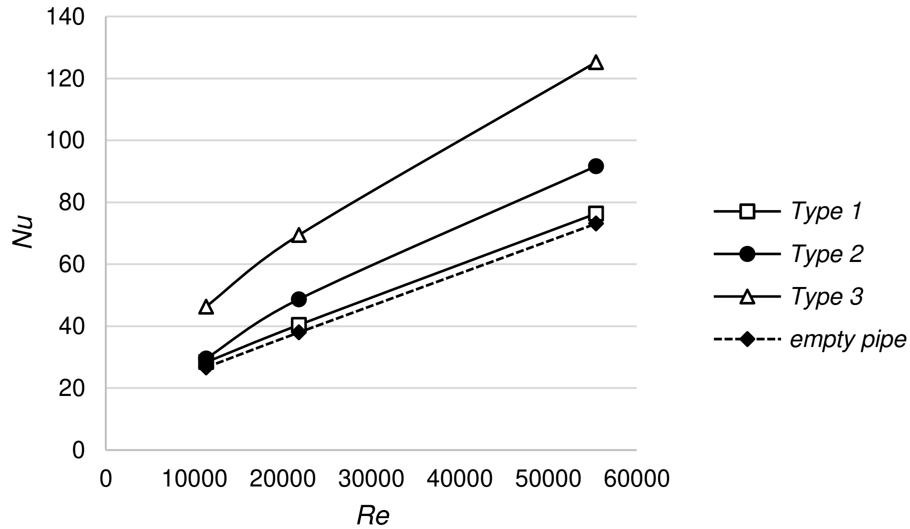


Figure 4.3. Nusselt number for the turbulators heat exchangers without vibration

From many of the previous studies we conclude that using turbulator is a key factor to increase the heat transfer, whatever the geometric shape of the turbulator is. Since the ratio of turbulated-Nusselt number to the empty pipe-Nusselt number gives the enhancement times, it will also be useful to give those values. The turbulated-Nusselt number is 1.05, 1.2 and 1.7 times of empty-pipe Nusselt number for *Type 1*, *Type 2* and *Type 3* respectively.

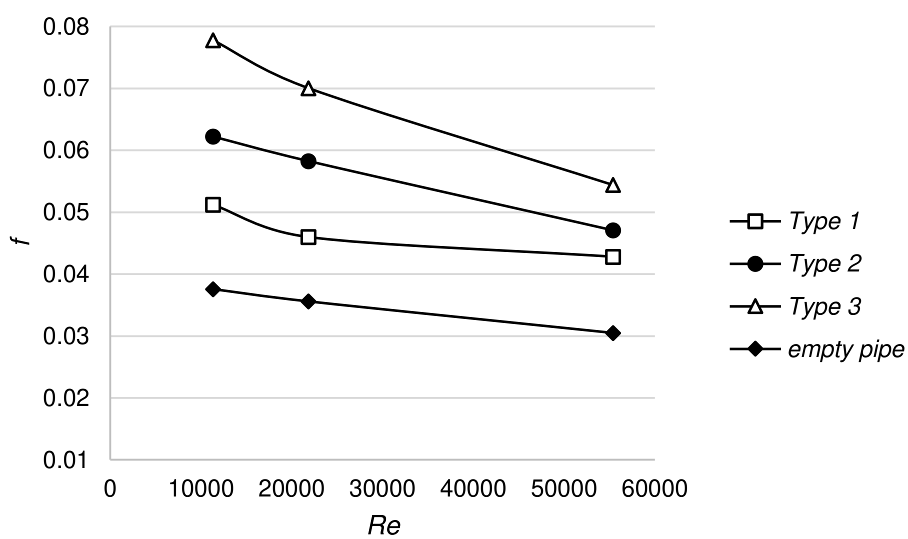


Figure 4.4. Friction factor for the turbulated heat exchangers without vibration

Figures 4.4 shows the variation between the *Re* number and friction factor. It is evident from the figures that the friction factor was decreases with increases of *Re* number. In Figure 4.4 the friction factor has its highest value in the case of turbulator *Type 3*, whilst it has the lowest value for the case of *Type 1*. As it is clear that Nusselt number is also highest for *Type 3* and lowest for *Type 1*. Hence, it is expected that increasing heat transfer results in enhancement of friction factor meanwhile. The enhancement percentages of the friction factor for the *Type 1*, *Type 2* and *Type 3* comparatively to the empty pipe are respectively, 35%, 61% and 95%. In other words, the friction factor is 1.3, 1.6 and 1.9 times of the friction factor in the empty tube.

The new derived empirical correlations and the regression coefficients for the trend of the *Nu-Re* and *f-Re* for *Type 1*, *Type 2* and *Type 3* are obtained and presented in Table 4.1.

Table 4.1. New empirical correlations found for turbulated cases

Types of turbulators	New empirical correlation	R^2 (%)
Type 1	$Nu = 0.079Re^{0.63}$	99
	$f = 0.1416Re^{-0.11}$	95
Type 2	$Nu = 0.0393Re^{0.71}$	99
	$f = 0.338Re^{-0.18}$	96
Type 3	$Nu = 0.1327Re^{0.63}$	99
	$f = 0.666Re^{-0.23}$	98

4.3 Tests with Turbulators and Vibration

4.3.1. Heat Transfer

In this section the attention will be turned to the heat transfer and pressure loss results of the heat exchanger with turbulators under mechanical vibration effects. The comparison of the with-vibration and without-vibration cases will also be performed at the end of this chapter.

The Nusselt number extracted from Eq. (3.6) versus Reynolds extracted from Eq. (3.1) are plotted to show the heat transfer enhancement of the turbulated and vibrated heat exchanger system. Each graph presents the comparison of the three types of turbulators. Figure 4.5 to 4.13 are plotted to show the effect of especially the frequency and vibration type. Figure 4.14 and Figure 4.15 are generated to show the effect of amplitude on the result.

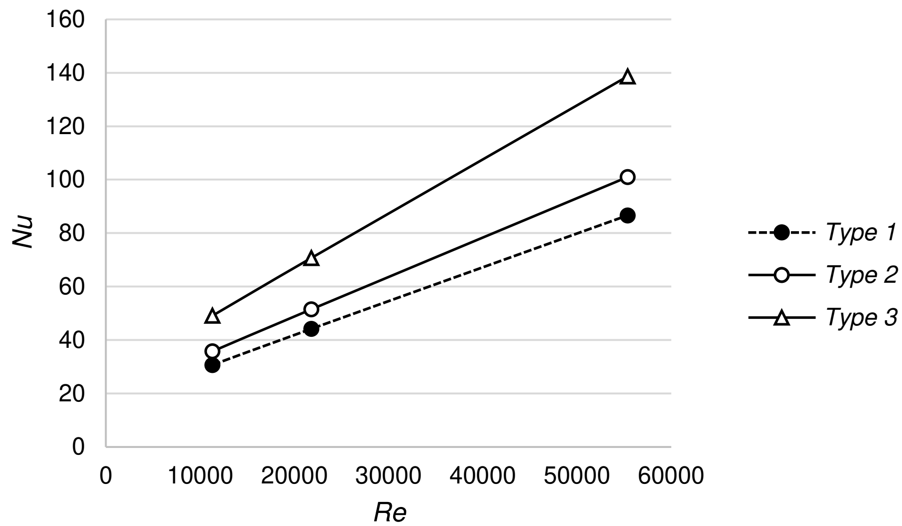


Figure 4.5. Variation of *Nu* with respect to *Re* for the case of F = 100 Hz, a = 100 m/s^2, *sine* signal

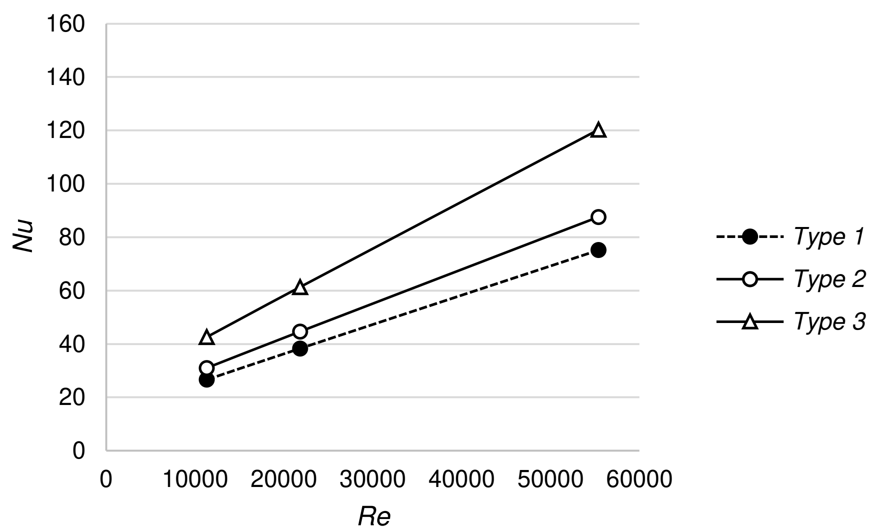


Figure 4.6. Variation of *Nu* with respect to *Re* for the case of F = 300 Hz, a = 100 m/s^2, *sine* signal

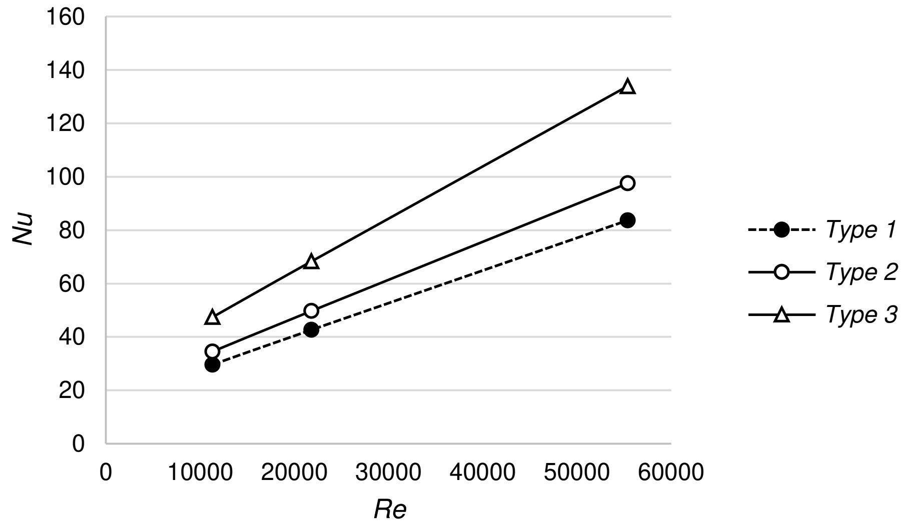


Figure 4.7. Variation of *Nu* with respect to *Re* for the case of F = 600 Hz, a = 100 m/s^2, *sine* signal

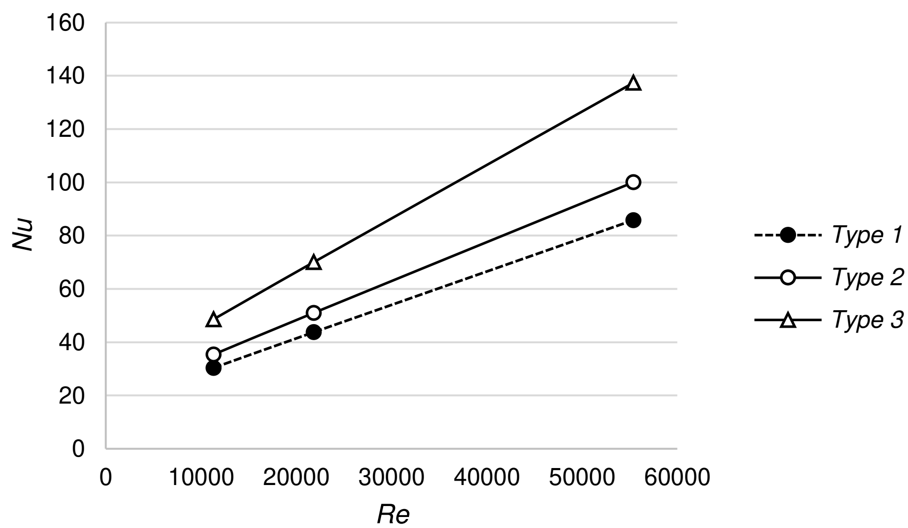


Figure 4.8. Variation of *Nu* with respect to *Re* for the case of F = 100 Hz, a = 100 m/s^2, *chirp* signal

As is generally known, the heat transfer enhancement techniques essentially reduce the thermal resistance in a conventional heat exchanger by promoting higher convective heat transfer coefficient with or without surface area increases. Generally heat transfer enhancement techniques can be classified broadly as passive and active techniques. Passive techniques, without requiring direct input of external power, use surface or geometrical modifications, or incorporate an insert, material, or additional device to increase heat transfer. In the case of active techniques, the addition of external power essentially facilitates the desired flow modification and the concomitant improvement in the rate of heat transfer.

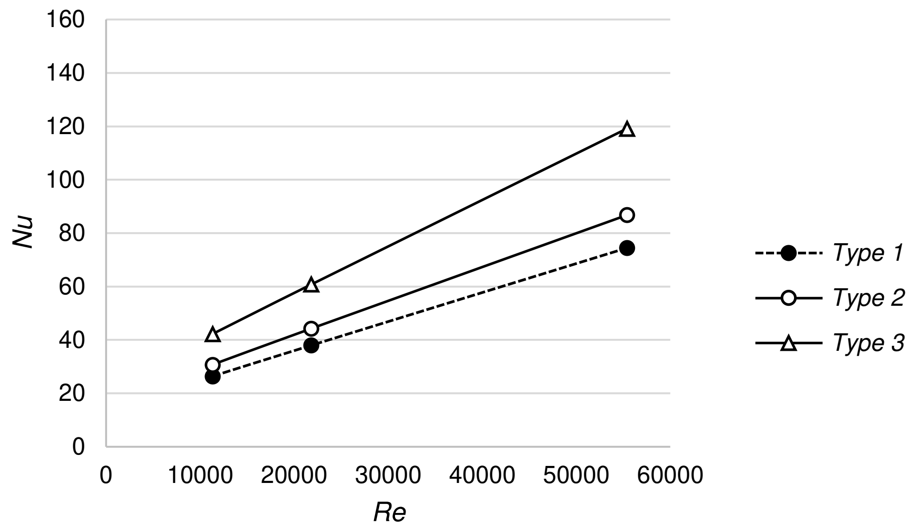


Figure 4.9. Variation of *Nu* with respect to *Re* for the case of F = 300 Hz, a = 100 m/s^2, *chirp* signal

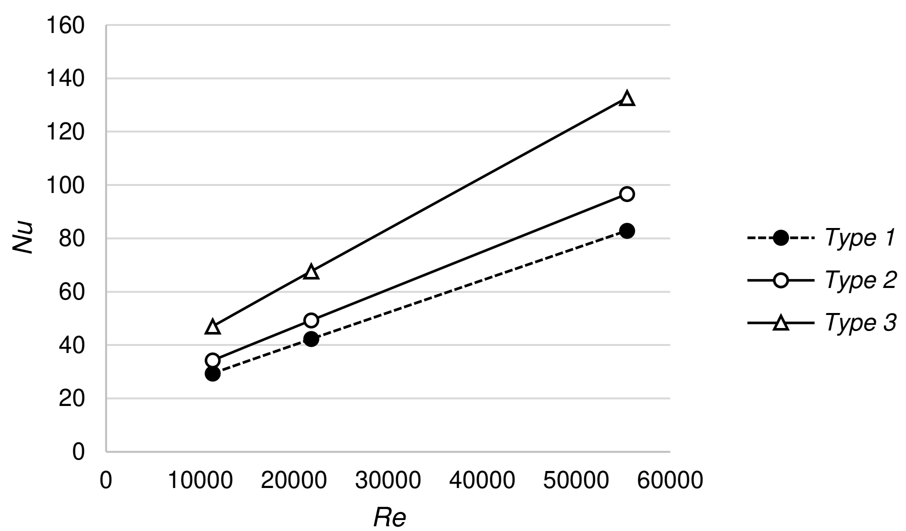


Figure 4.10. Variation of *Nu* with respect to *Re* for the case of F = 600 Hz, a = 100 m/s^2, *chirp* signal

The vibration is one of the active techniques which have been considered by researchers for heat transfer enhancement. All of the heat transfer results have shown that the vibration gives rise to turbulence intensity in boundary layers and increases the heat transfer rate. Furthermore, the mechanical vibration applied to the setup increases the fluid mixing which leads to increase in the heat transfer coefficient. The literature reviews have shown that the vibrations can augment overall heat transfer coefficient on 10–257% (Hosseini et al., 2018). In general, the heat transfer rate increases approximately linearly with the vibration amplitude.

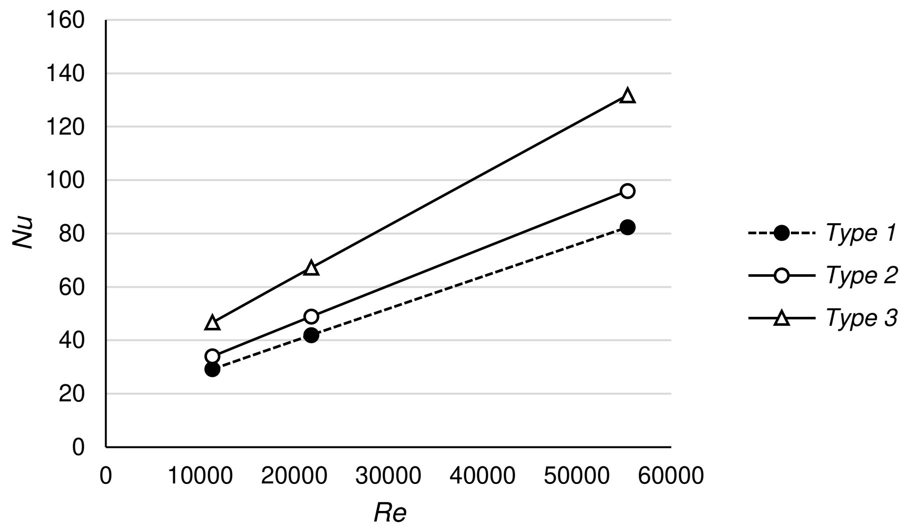


Figure 4.11. Variation of *Nu* with respect to *Re* for the case of F = 100 Hz, a = 100 m/s^2, *random noise* signal

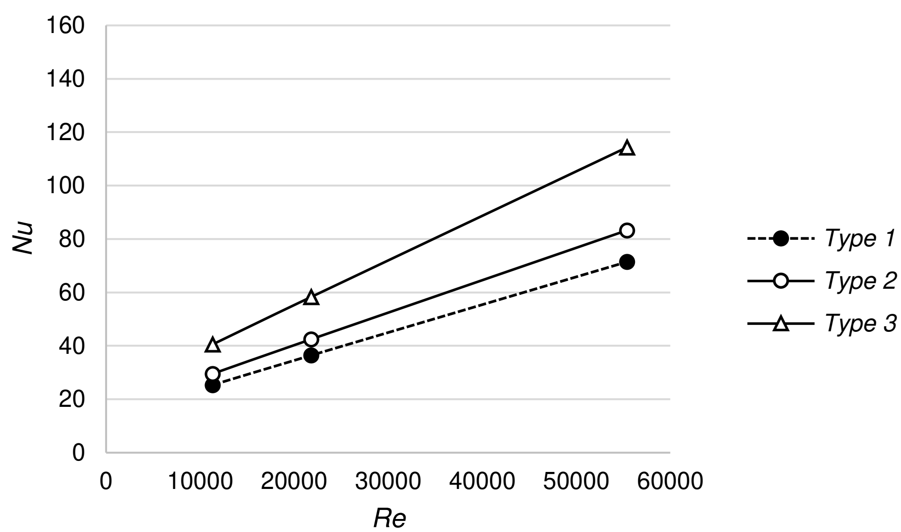


Figure 4.12. Variation of *Nu* with respect to *Re* for the case of F = 300 Hz, a = 100 m/s^2, *random noise* signal

Figures 4.5 - 4.13 show the variation of Nusselt number as a function of Reynolds number under the same amplitude (a = 100 m/s^2) but various frequency and signal type conditions. As can be seen, for all cases the increase in vibration frequency level from 100 Hz to 300 Hz, decreases the heat transfer, but increment of the frequency from 300 Hz to 600 Hz results in increase of heat transfer again. For example in Figure 4.5, the Nusselt number for *Type 3* is found 49, 70 and 138 respectively at Reynolds number values of ~10,000, ~20,000 and ~50,000 when the frequency is 100 Hz.

In Figure 4.6, namely when the frequency is 300 Hz, the Nusselt number for *Type 3* is found 42, 61 and 120 respectively for the Reynolds values of ~10,000, ~20,000 and ~50,000. However, in Figure 4.7, meaning when the frequency is 600 Hz, the Nusselt numbers are 47, 68 and 133 respectively for the Reynolds number values of ~10,000, ~20,000 and ~50,000.

The type of the vibration seems to have a very little effect on heat transfer. In addition to that, the amplitude appears to have no considerable effect on heat transfer either. The type of the turbulator is the important parameter that affects the heat transfer. The turbulator coded as *Type 3* gives the highest heat transfer results, whilst *Type 1* gives the lowest heat transfer results, regardless of the vibration type or the values of frequency and amplitude. The effect of the Reynolds number seems to be very high from all tested cases. Increasing Reynolds number from ~10,000 to ~50,000, results in nearly 80% enhancement of Nusselt number for all cases.

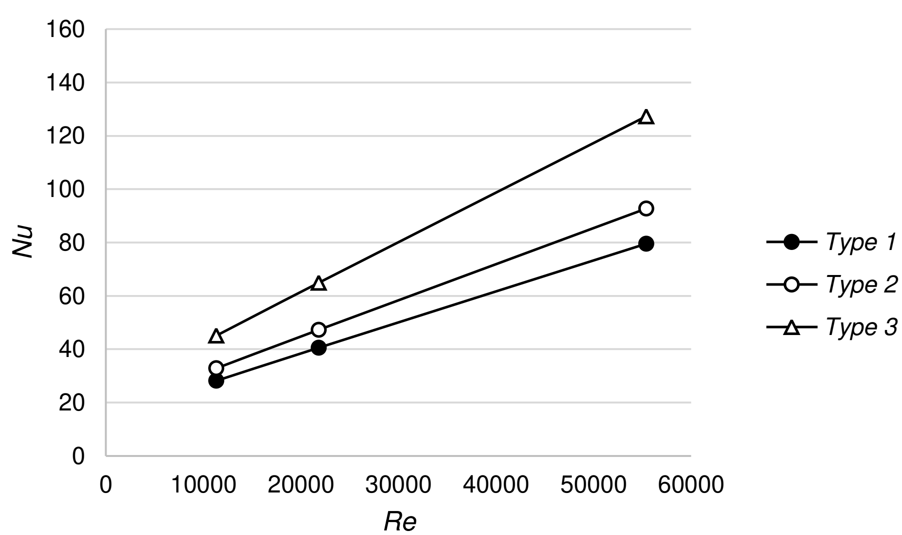


Figure 4.13. Variation of *Nu* with respect to *Re* for the case of F = 600 Hz, a = 100 m/s^2, *random noise* signal

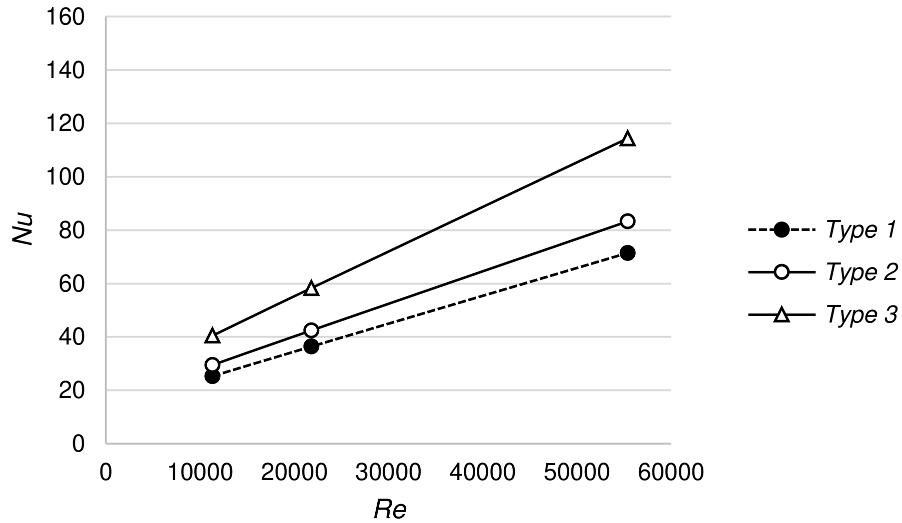


Figure 4.14. Variation of *Nu* with respect to *Re* for the case of F = 300 Hz, a = 300 m/s^2, *sine* signal

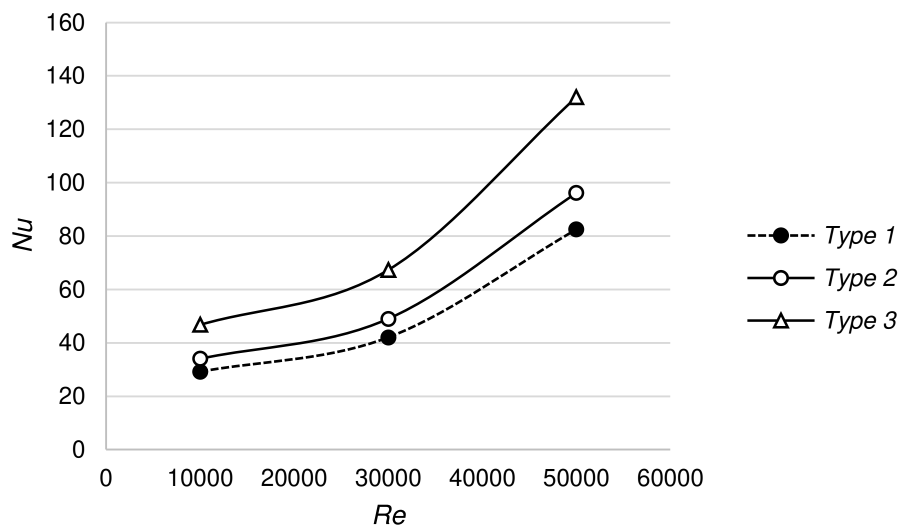


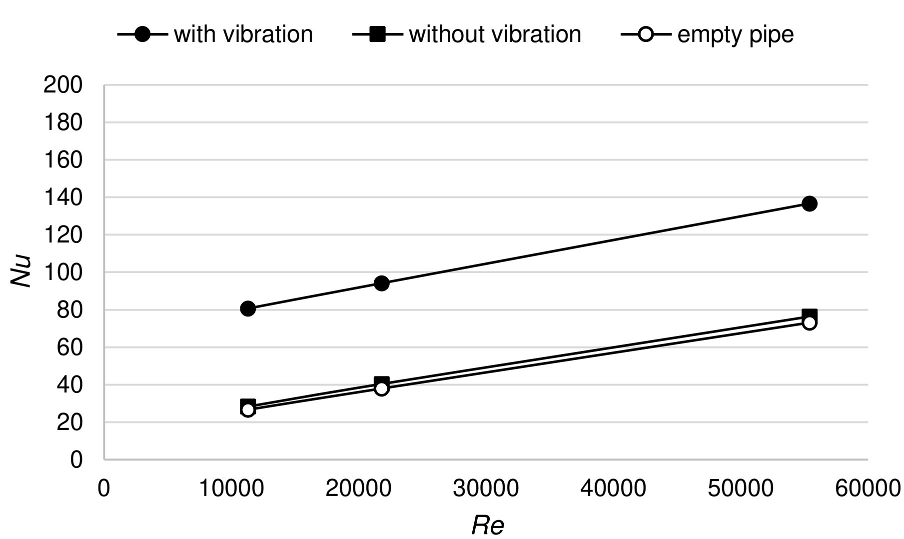
Figure 4.15. Variation of *Nu* with respect to *Re* for the case of F = 600 Hz, a= 600 m/s^2, *sine* signal

The effects of the design parameters on the heat transfer can be seen in Table 4.2. This table is generated from the *Taguchi* analysis. It is important, since the contribution of the parameters on the result can be seen from this analysis. As it is clear from Table 4.2, the effect of Reynolds number (which was already coded as A in the *Taguchi* design analysis) is 80%, which is also seen in the figures. The effect of turbulator type is 16% and the effect of frequency is 1.49 %.

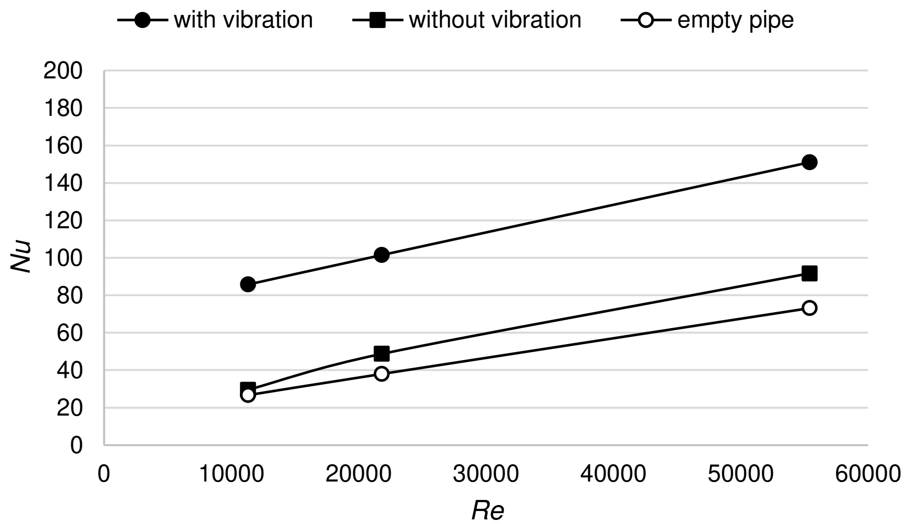
Table 4-2 Effects of the design parameters on the Nusselt number

	Degree of Freedom	Sum of squares	Variance	F Value	% Contribution	Table of F Value	99%
A	2	376.662	188.331	773.9524	80.35984372	6.22623528	OK
B	2	7.490374	3.745187	15.39097	1.496154266	6.22623528	OK
C	2	0.906529	0.453264	1.862705	0.089690947	6.22623528	not effective
D	2	78.17071	39.08536	160.6226	16.5951272	6.22623528	OK
E	2	0.990558	0.495279	2.035365	0.107641473	6.22623528	not effective
Error	16	3.893387	0.243337	0	1.35154239	0	
Sum		468.1136					

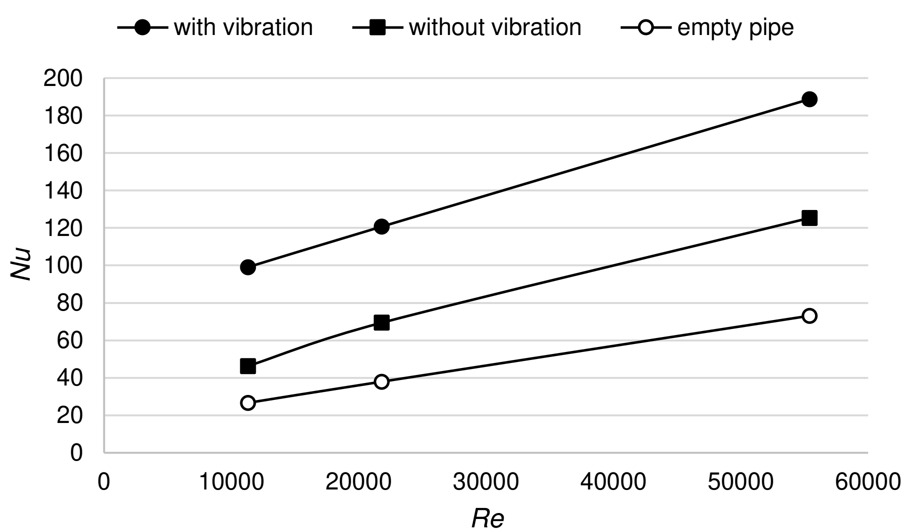
The effects of vibrationn on heat transfer can be understood better, by plotting the with-vibration and without-vibration cases together. Figures 4.16 and 4.17 are plotted to show this comparison. Figure 4.16 shows the case; F = 100 Hz, a = 100 m/s^2, *sine* signal. Figure 4.17 shows the comparison under the conditions of F = 300 Hz, a = 100 m/s^2, *chip* signal. Each subfigures indicated by **(a)**, **(b)** and **(c)** show the types of the turbulators, respectively as *Type 1*, *Type 2* and *Type 3*. The empty pipe results are also added for better evealuation.



a) *Type 1*



b) *Type 2*

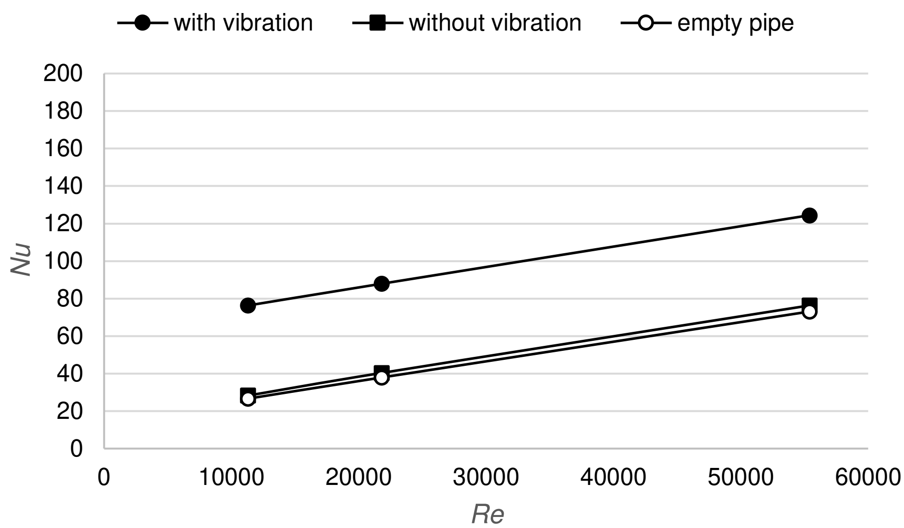


c) *Type 3*

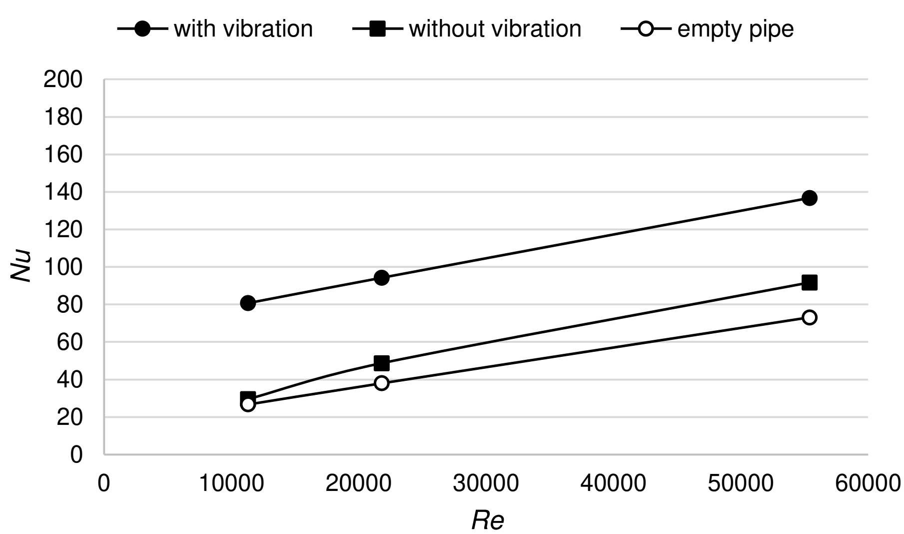
Figure 4.16. Comparison of vibrated and non-vibrated *Nu* for the case of F = 100 Hz, a = 100 m/s^2, *sine* signal

According to the Figure 4.16, the heat transfer with vibration seems to enhance comparatively to the case of non-vibration up to 132%, 121% and 79%, respectively for *Type 1*, *Type 2* and *Type 3*.

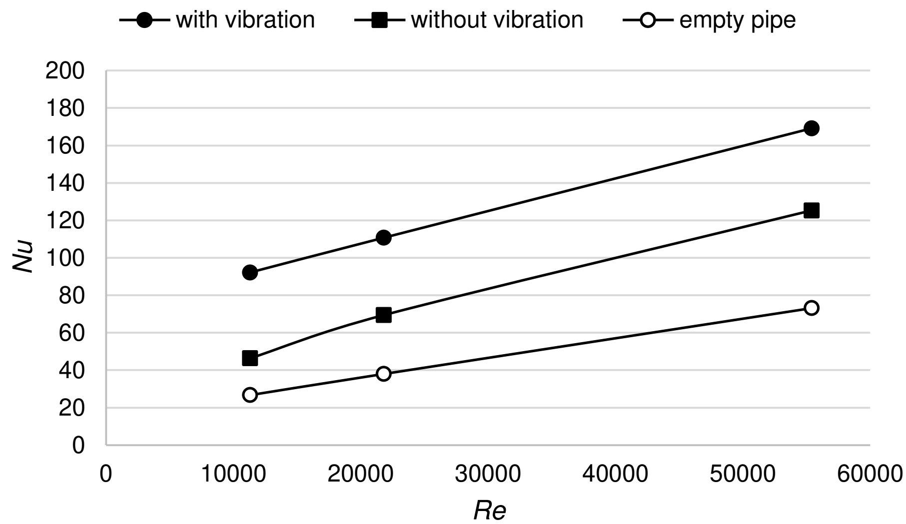
For Figure 4.17., the heat transfer enhances with the percentages of 116%, 105% and 64% respectively for *Type 1*, *Type 2* and *Type 3*. Similar comparison can be applied to many cases. In general it is evident that heat transfer enhances with vibration effects regardless of the types of the wave, frequency or amplitude. In other words, it can be said that the vibration itself has a great effect on heat transfer but the variation of the vibration parameters; such as vibration type, frequency or amplitude has no great impetus on the results.



a) *Type 1*



b) *Type 2*



c) *Type 3*

Figure 4.17. Comparison of vibrated and non-vibrated *Nu* for the case of F = 300 Hz, a = 100 m/s^2, *chirp* signal

4.3.2. Pressure Loss

Now, it is time about to focus on pressure loss by means of friction factor. For all tested cases, the use of vibration results in a substantial increase in friction factor. The friction factor will be shown by some summarizing graphs. Figure 4.18 to 4.26 are plotted to show the friction factor variations with respect to Reynolds number. Figures 4.18, 4.19 and 4.20 show the friction factor for the cases under conditions of a fixed amplitude a = 100 m/s^2, and fixed vibration type; *sine*, but various frequencies, respectively as F = 100, 300, and 600 Hz.

The friction factor increases in all cases, however, is much higher for the case of using *Type 3* turbulator than that of the smooth tube. These increments can be attributed to dissipation of the dynamic pressure of the fluid due to higher surface area and flow blockage of the turbulator along the tube wall.

Friction factor tends to decrease sharply with the increase of Reynolds number in the range of 10,000~20,000. Above this value the decrement rate nearly keeps constant or becomes less.

As is shown in Figures 4.18 to 4.20 the highest friction factor occurs for *Type 3* and the lowest friction factor occurs for *Type 1*. This result was also seen for the heat transfer. As it is obvious from the *Nu-Re* graphs highest heat transfer occurs for the case of *Type 3*. Since enhancement of heat transfer causes some pressure loss, this result was already expected.

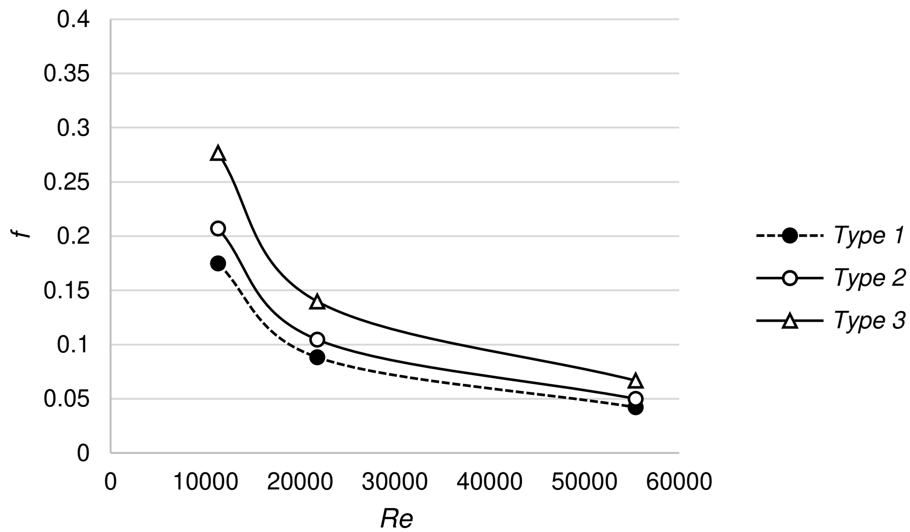


Figure 4.18. Variation of *f* with respect to *Re* for the case of F = 100 Hz, a = 100 m/s^2, *sine* signal

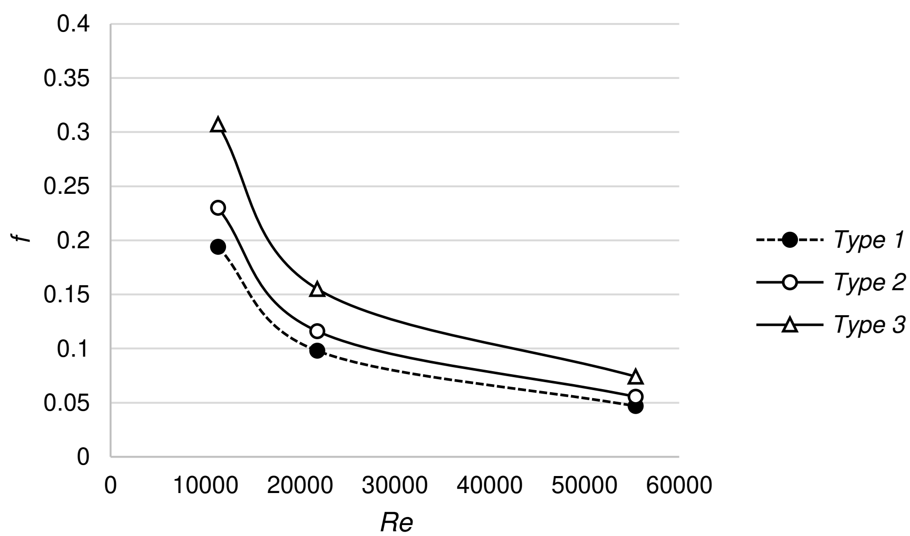


Figure 4.19. Variation of *f* with respect to *Re* for the case of F = 300 Hz, a = 100 m/s^2, *sine* signal

The effect of design parameters about the vibration can be seen more clearly in the pressure loss graphs than it was seen in heat transfer graphs. Increasing the frequency from 100

Hz to 600 Hz cause a decrement in friction factor. The difference of the friction factor at F = 100 Hz and F =600 Hz is nearly 6%, 3% and 1.5%, respectively for Re values of 10,000, 20,000 and 50,000.

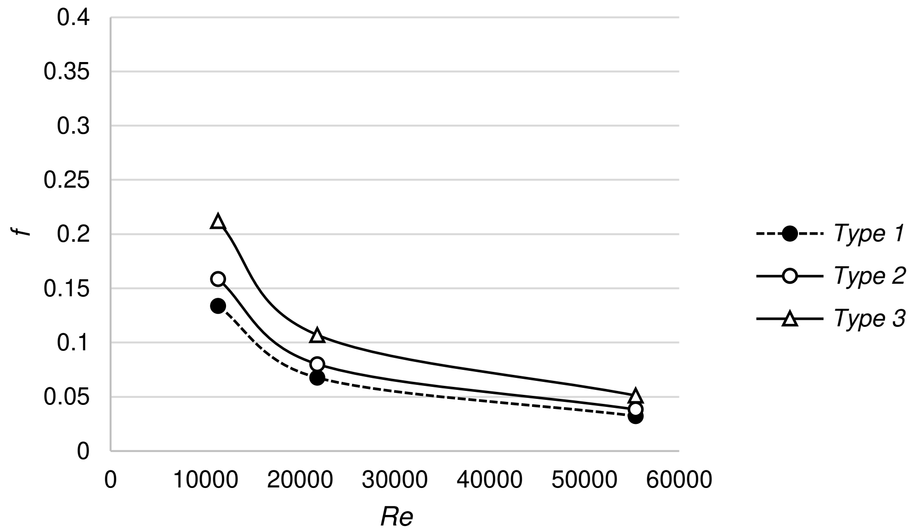


Figure 4.20. Variation of f with respect to Re for the case of F = 600 Hz, a = 100 m/s^2, *sine* signal

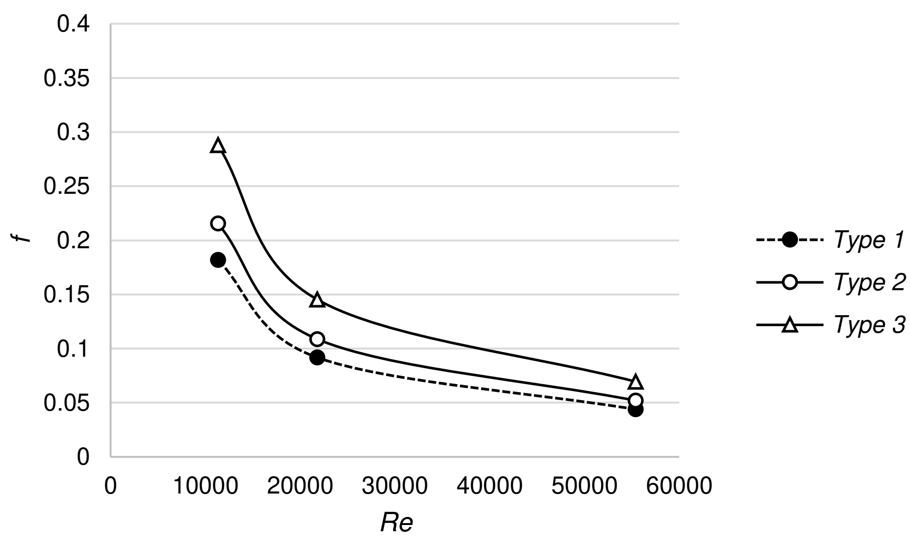


Figure 4.21. Variation of f with respect to Re for the case of F = 100 Hz, a = 100 m/s^2, *chirp* signal

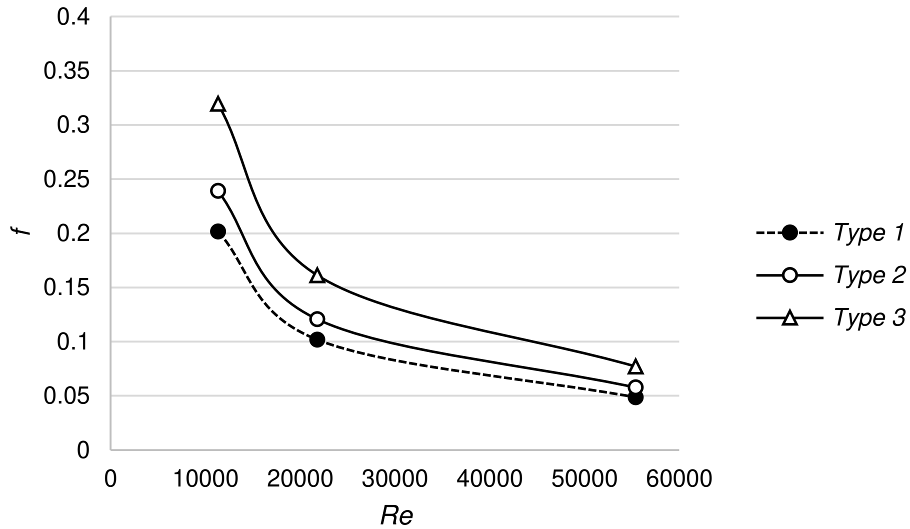


Figure 4.22. Variation of *f* with respect to *Re* for the case of F = 300 Hz, a = 100 m/s^2, *chirp* signal

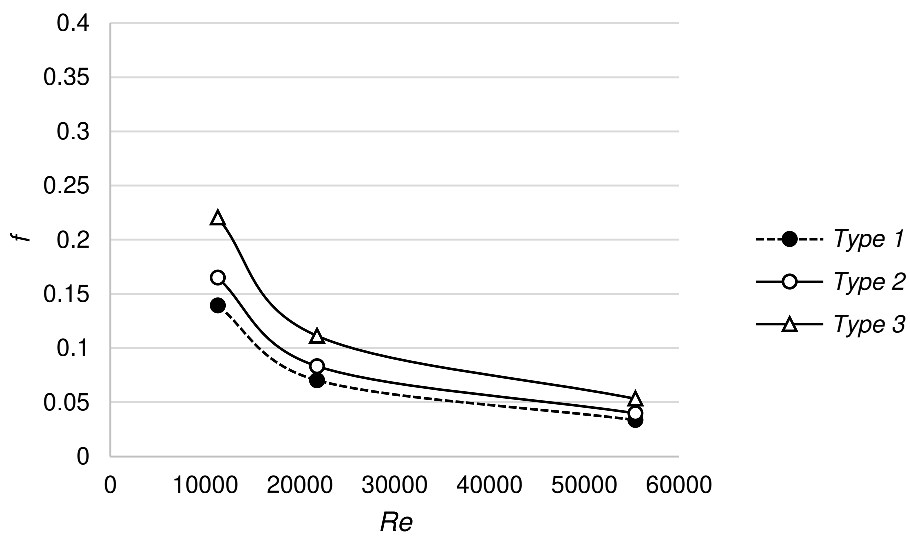


Figure 4.23. Variation of *f* with respect to *Re* for the case of F = 600 Hz, a = 100 m/s^2, *chirp* signal

The changing the vibration type has a little effect on friction factor. In Figures 4.21, 4.22 and 4.23 the vibration type is chirp. It is seen that all of the values are a little higher than the cases of sine type vibration. For example, the friction factor is 3.9% higher for chirp for the case of *Type 3* turbulator regardless of the value of *Re*.

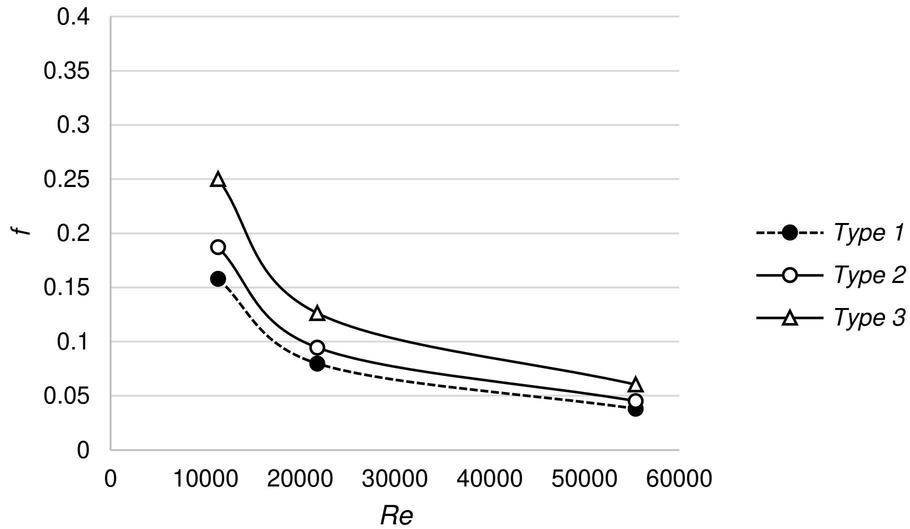


Figure 4.24. Variation of *f* with respect to *Re* for the case of *F* = 100 Hz, *a* = 100 m/s^2, *random* signal

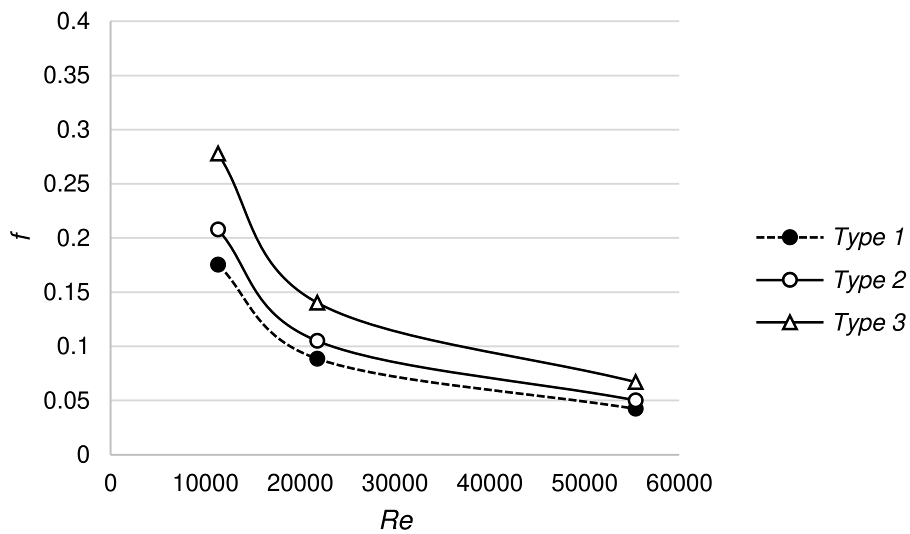


Figure 4.25. Variation of *f* with respect to *Re* for the case of *F* = 300 Hz, *a* = 100 m/s^2, *random* signal

Applying random noise signals seems results in decreasing of friction factor. Comparatively to the *sine* signals the friction factor decreases 10% for the case of random noise (for example *Type 3*). From Figures 4.24, 4.25 and 4.26, it is seen that all of the values are lower than the cases of *sine* type vibration.

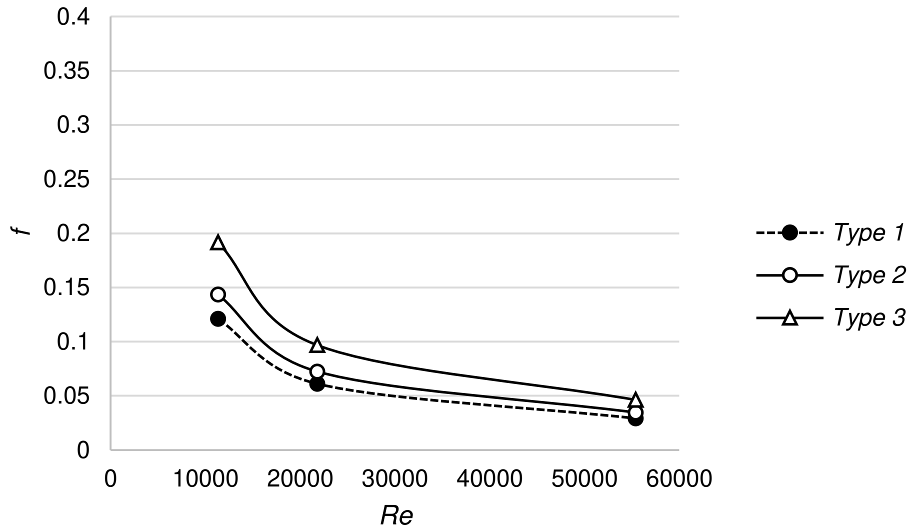


Figure 4.26 Variation of *f* with respect to *Re* for the case of *F* = 600 Hz, *a* = 100 m/s^2, *random* signal

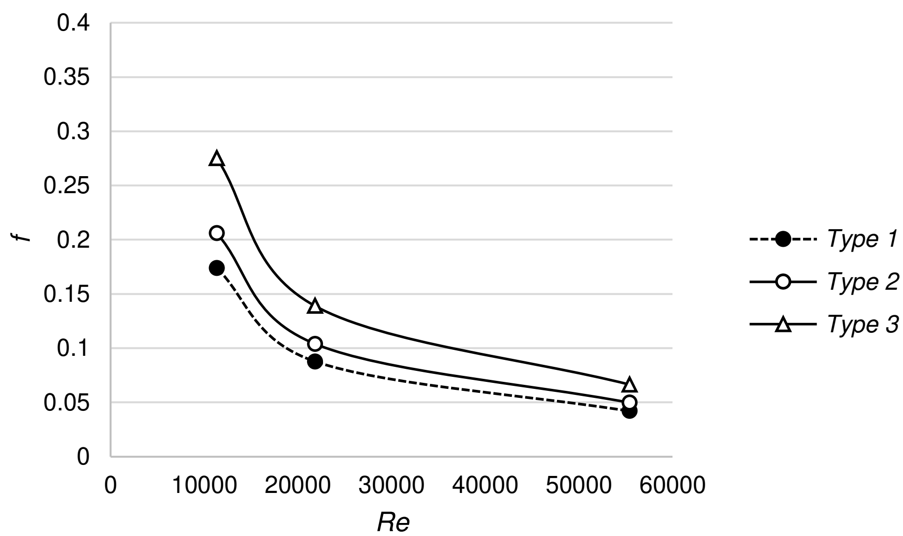


Figure 4.27. Variation of *f* with respect to *Re* for the case of *F* = 300 Hz, *a* = 300 m/s^2, *sine* signal

The effect of variation of amplitude seems very weak in all friction factor results, as previously seen in heat transfer results. The friction factor decreases with increasing Reynolds number as expected.

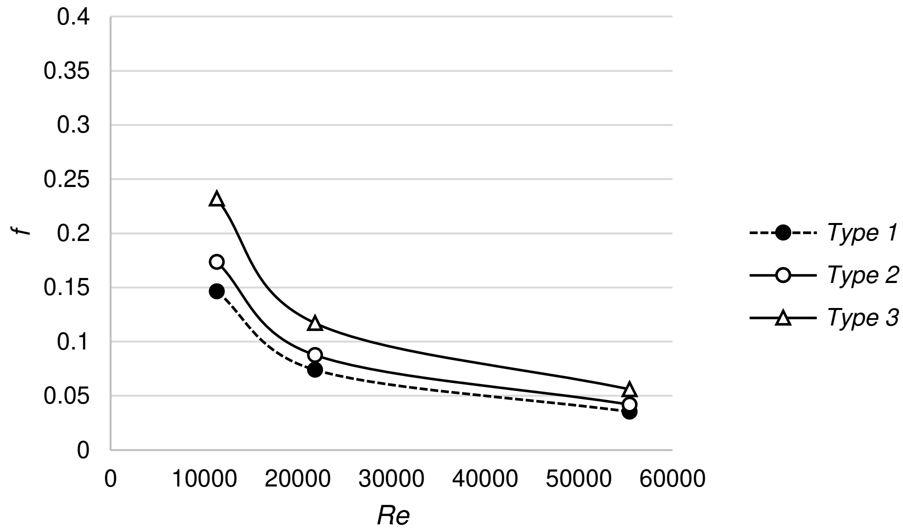
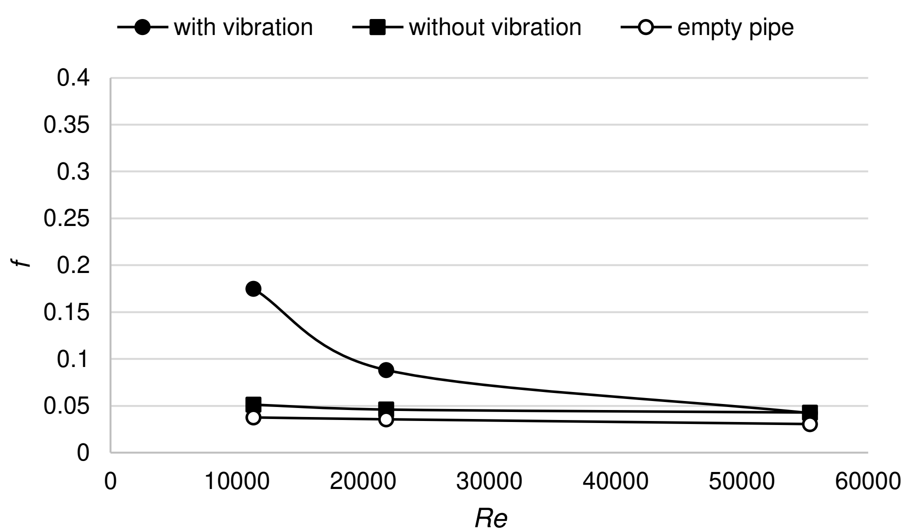
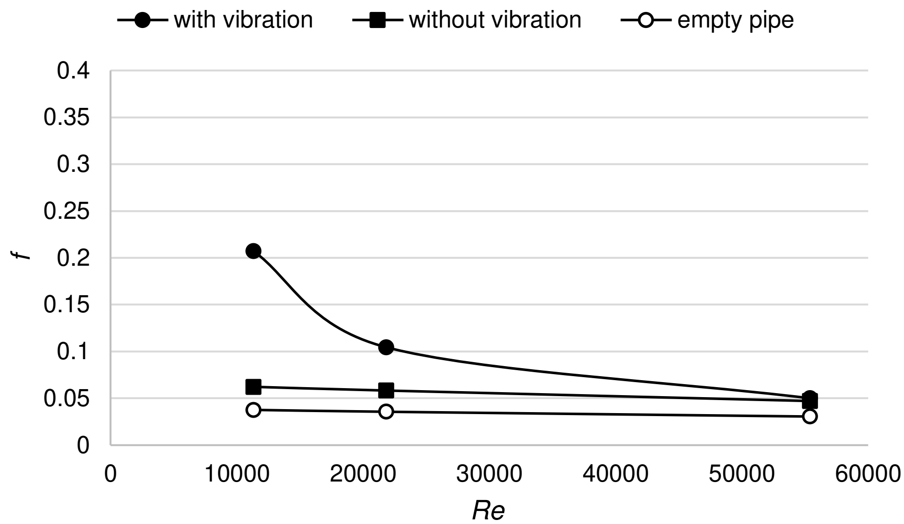


Figure 4.28. Variation of *f* with respect to *Re* for the case of F = 600 Hz, a = 600 m/s^2, *sine* signal

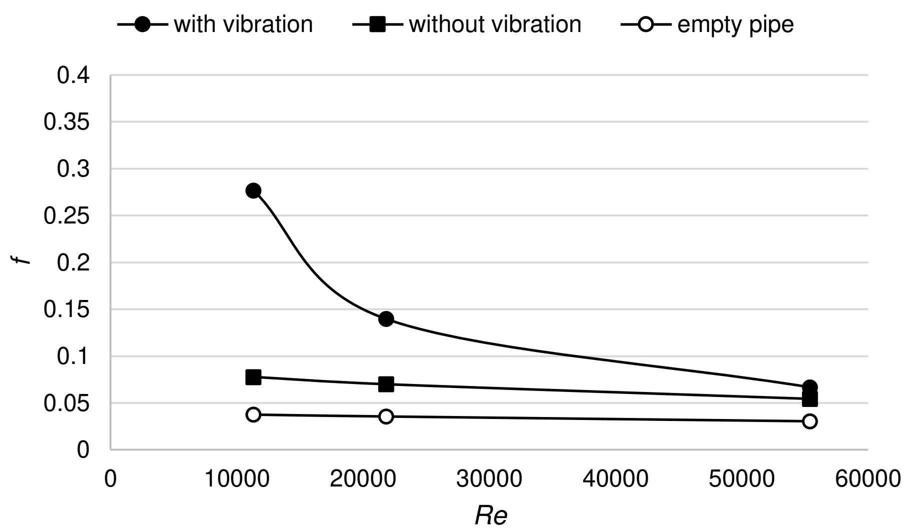
To evaluate the results, it will be better to plot some comparison graphs including the vibrated and non-vibrated cases together. Figures 4.29 and 4.30 are plotted to show this comparison. According to both graphs, in the vibrated cases the pressure loss increases regardless of the type of turbulators.



a) *Type 1*



b) *Type 2*

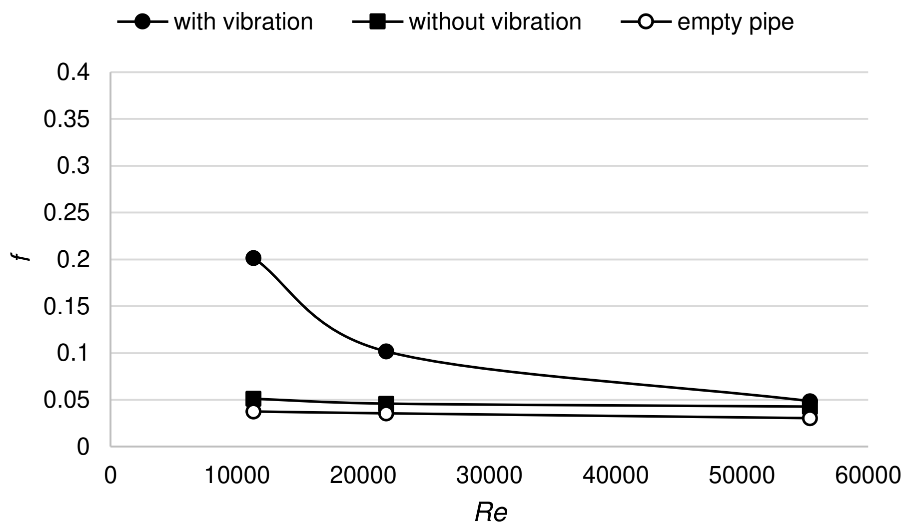


c) *Type 3*

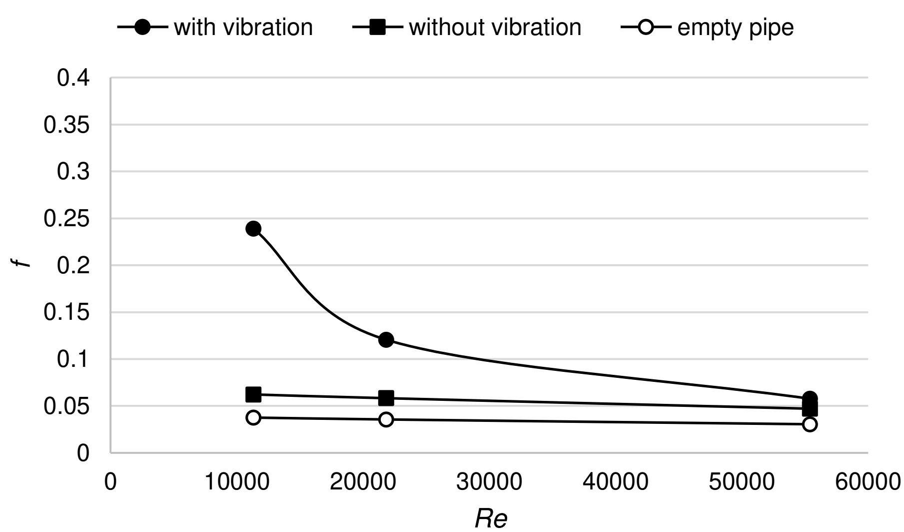
Figure 4.29. Comparison of vibrated and non-vibrated f for the case of F = 100 Hz, a = 100 m/s^2, *sine* signal

It should be noticed that, the highest friction appears at the lowest Reynolds number for all cases. Not only the highest value of friction factor is found for *Type 3*, the difference between the vibrated and non-vibrated cases are also highest for *Type 3*.

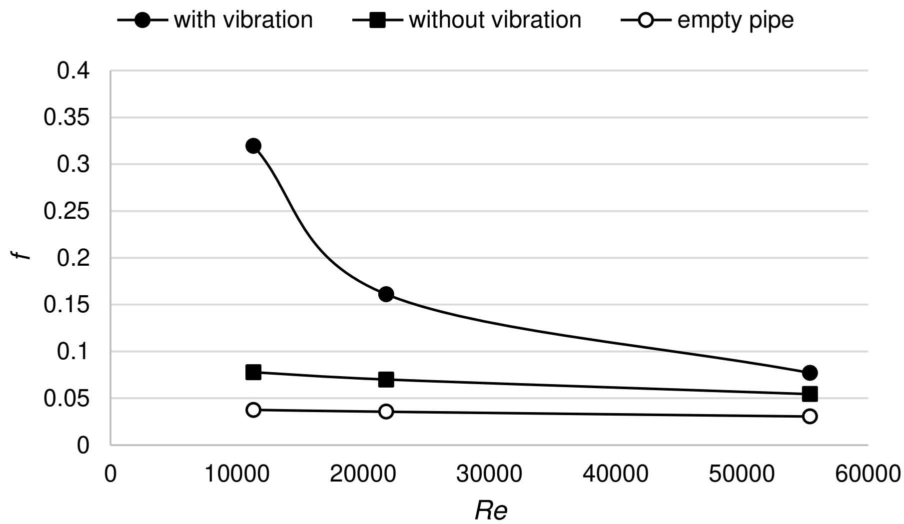
It is also worthy to note that, there has been a sharp decrease at the lowest Reynolds number. For example in the range of *Re* ~10,000 to ~20,000 the friction factor has a really sharp decrease at all cases. For *Re* > 50,000 the friction factor diminishes more slightly. It can also said that the trend of the friction factor curves is nearly same for all cases.



a) *Type 1*



b) *Type 2*



c) *Type 3*

Figure 4.30. Comparison of vibrated and non-vibrated f for the case of F = 300 Hz, a = 100 m/s^2, *chirp* signal

5. CONCLUSIONS AND RECOMMENDATIONS

In the present work, the effect of imposing mechanical vibrations on a heat exchanger with turbulator in it, is investigated experimentally. Imposing the vibrations increases the heat transfer regardless of the type of vibration, frequency or amplitude. The following remarks explain the major conclusions of the study:

- ✓ The heat transfer increases by increasing the Reynolds number.
- ✓ The heat transfer increases by changing the turbulator type especially by enhancing the heat transfer surface area.
- ✓ Although applying vibration to the tests let to enhance the heat transfer considerably, the type of the vibration has no meaningful effect on heat transfer or friction factor.
- ✓ The increase of frequency from 100 Hz to 300 Hz results in a decrease of heat transfer, but increase of the frequency from 300 Hz to 600 Hz results in increase of heat transfer again.
- ✓ The amplitude has weak effect on both the heat transfer and friction factor.

The effect of vibration on heat transfer and pressure loss is found by means of application of the vibration to the turbulator inside the heat exchanger. For comparison and verification it would be better to apply the vibration to the surface of the heat exchanger, since most of the relevant articles in literature had done it in that way. It is suggested that a mechanical vibration can be applied to the surface and then the results can be comparable.

A detailed uncertainty analysis will help to find out the effects of errors especially the vibration of the fan and other equipment on results. Then the effect of mechanical vibration itself can be evaluated better.

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