DESIGN CONSIDERATIONS AND PERFORMANCE EVALUATION OF A SURGE TANK FOR DIAPHRAGM PUMP OPERATION

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APPROVAL OF THE THESIS

DESIGN CONSIDERATIONS AND PERFORMANCE EVALUATION OF A SURGE TANK FOR DIAPHRAGM PUMP OPERATION

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ABSTRACT

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This thesis is performed to evaluate the design consideration and performance characteristics of a surge tank for a diaphragm pump operation and to evaluate the proper volume and inlet area of surge tank in order to reduce the pulsations of the discharge pressure.

An experimental set up is constructed for a three diaphragm positive displacement pump and the experiments are conducted afterwards. The surge tanks having different volumes and the surge tank inlet area configurations are tested in order to achieve the minimum peak to peak pulsations.

Experiments showed that among the different sizes of the surge tanks, the minimum peak to peak pulsations are achieved with the largest volume which is the original surge tank of the test pump used by the pump manufacturer. This result is supported by the literature which states that with greater surge tank size the magnitude of pulsations can be diminished more.

Regarding the surge tank inlet area design; among the eight different adaptors a proper inlet area value is concluded having the minimum peak to peak pulsations also smaller than the original configuration.

Keywords: pressure pulsation dampening, diaphragm pump, surge tank sizing, surge tank testing.

ÖZ

DİYAFRAMLI BİR POMPANIN HAVA KABI TASARIM KRİTERLERİVE PERFORMANSININ ARAŞTIRILMASI

Özdemir, Şahika
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Bu tez çalışması üç diyaframları pozitif deplasmanlı bir pompanın çıkış basıncındaki salınımları azaltmak için, kullanılan hava kabının tasarım ve performansının incelenmesi ve hava kabı için en uygun boyutlandırma ve giriş alanı tayini için yapılmıştır.

Bu amaçla tasarlanan ve kurulan deney düzeneğinde bahsedilen üç diyaframlı pompanın performans özellikleri test edilmiş daha sonra da hava kabı deneyleri gerçekleştirilmiştir. Farklı boyutlara sahip üç hava kabı ve hava kabı girişi için tasarlanmış farklı alan büyüklüklerine sahip adaptörler ile deneyler yapılmış, en düşük salınımların gözlenmesi amaçlanmıştır.

Deney sonuçlarına göre, üç farklı büyüklükteki hava kaplarından, en büyük hacme sahip olan hava kabında en düşük basınç salınımları gözlenmiştir. Bu hava kabı da pompa üreticisinin bu pompa ile endüstride kullandığı hava kabıdır. Bu sonuç literatürde bu konu ile ilgili kaynaklardaki bilgilerle de örtüşmektedir. Çünkü kaynaklarda hava kabının boyutları büyüdükçe çıkış basıncındaki salınımların da azalacağı belirtilmektedir.

Hava kabı giriş alanı ile ilgili olarak yapılan deneylerde sekiz farklı adaptör ve alan büyüklükleri ile yapılan testlerin sonuçlarında kabının mevcut giriş alanından daha küçük giriş alanı ile pompa çıkışındaki basınç salınımlarının azaltılması mümkün olmuştur.

Anahtar Kelimeler: basınç salınım sönümlenmesi, diyafram pompa, hava kabı boyutlandırması, hava kabı testleri.

For My Parents

and

Beloved Husband

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Special thanks to my family for showing me the value of education, support and patience throughout my life. I inscribe my graduation to them.

My heartfelt thanks go to my husband, for his encouragement and light for the last and hardest times of this thesis. Without him in my life I could not make it.

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LIST OF SYMBOLS AND ABBREVIATIONS

C: pump capacity per one revolution
d,D: pump piston or plunger diameter
dv: liquid volume that the search tank must store
E: maximum error
f_{motor}: Frequency of the motor
f_{pump}: Frequency of the pump
K: pump constant
LOD: Level of dampening
n, N: Pump speed
n, N: Number of data
p: average fluid pressure, psi
p_1: minimum pressure in the line
p_2: maximum pressure in the line
p_c: precharge pressure, psi
p_d: Pump discharge pressure
P_{elec}: Electrical power
p_{exit}: Exit pressure of the pump
P_{hyd}: Hydraulic power
p_{in}: Inlet pressure of the pump
p_{max}: Maximum exit pressure of the pump
p_{min}: Minimum exit pressure of the pump
$p_{average}$: Average exit pressure of the pump
Q: Volumetric flow rate f the pump
Q_{piston}: Volumetric flow rate of the single piston
r: Area ratio for the surge tank inlet area adaptors
$r_{belt\&pulley}$: Belt and pulley reduction ratio
s: Stroke of the pump
t: Time for tank to be filled in seconds
V: Voltage value read from the Labjack

V_{surge}: Volume of the surge tank of the pump

V_{tank}: Volume of measurement tank

% pulsation: peak-to-peak pulsation percentage

$\bar{x}$: Mean of the data

x: First parameter

y: Second parameter

z: Gaussian distribution value

Δp: Allowed pressure pulsation, psi

Δp: Pressure difference between inlet and exit of the tank

α: Significance level

η_{pump}: Pump efficiency

η_{system}: System efficiency

ρ_{xy}: Correlation coefficient

σ_x: Standard deviation of the data

q_{surge}: amount of flow entering the surge tank

q_{pump}: volumetric flow rate of the pump

q_{mean}: mean volumetric flow rate of the pump

r: radius of the crank

A Area of the piston

α: crank angle

ω: rotational speed

l: crank length

ΔH_1 amount of water leaving the surge tank

ΔH_2 amount of water enters the surge tank

H_1 amount of water in the surge tank at time t_1

H_2 amount of water in the surge tank at time t_2

ΔHo: the air volume in the surge tank

s_{max}: maximum stroke length

s_{min}: minimum stroke length

Δs: delta of the stroke length

p_2: exit pressure of the pump at time t_2

p_1: exit pressure of the pump at time t_1

CHAPTER 1

INTRODUCTION

The flow pulsations and shaking forces caused by power cut off, valve closure or the nature of the flow may lead to hazardous effects on the pumping systems such as; piping vibration, fatigue failure of the pumping systems' components and valve damage. Also, they cause mechanical and foundation failure [1]. There are several examples for this situation. An oversized piping system is designed to overcome the higher pressure drop induced by the pulsative flow. Also, pump efficiency is decreasing due to requirement of more power to overcome the higher internal pressure peaks, and noise is induced throughout the piping system. Piston rods are exposed to higher stresses because of unbalanced or dynamic forces, [2].

In order to solve the problems caused by flow pulsation, the systems are designed such that the pulsations are taken in account and their amplitudes are minimized. There are several methods to reduce the pulsations in the systems such as; using controllable vanes, one sided vanes and **"surge tanks"** which is a commonly used solution and is indeed one of the focuses of this thesis study [3].

In order to use these solutions efficiently, they must be designed and applied uniquely to the systems according to the characteristics of the flow and the pump parameters. Hence this study is performed to evaluate the design and performance of a surge tank for a diaphragm pump operation, and to evaluate the proper inlet area for surge tank in order to dampen the pulsations in the pumping system.

The theory of the thesis such as; pump types, especially reciprocating and diaphragm pumps and the surge tanks, and also the aim of the thesis are mentioned in this chapter. The literature survey, experimental set-up and instrumentation, experimental results and discussion and the conclusion is explained in the following sections.

1.1 POSITIVE DISPLACEMENT PUMPS

In positive displacement pumps, the diaphragm or piston is moved forward and backward by the reciprocating motion of the plunger and with the help of the valves the fluid is sucked when the diaphragm or piston moves backward and discharged when it is pushed forward again by the plunger (Figure 1). The positive displacement pumps are generally equipped with relief valves which are used for the safety of the pump [4]. When the pump discharge is blocked during the operation, or in case of any unwanted incidence such as stuffed filters, the relief valve opens and the flow is diverted through it, not allowing the flow to over pressurize the discharge of the pump. The positive displacement pumps usually operate with laminar internal flow regimes

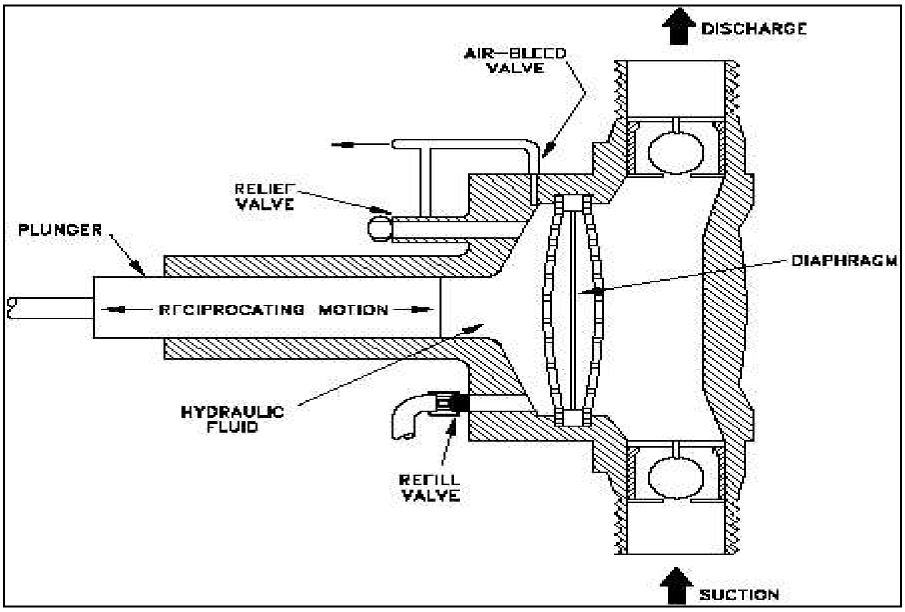

Figure 1 Positive Displacement Pump Detail Drawing [5]

In positive displacement pumps a definite amount of liquid is pumped at any resistance or head of the system at each cycle of the pump [5]. The only parameter that affects the amount of fluid is the rotational speed of the pump. However this is valid for the pure ideal cases. In real conditions there are backflow or leakage in the system and as the resistances increases the volumetric flow rate of the pump decreases due to this leakage. The characteristic curve for a positive displacement pump explaining this condition is shown in Figure 2.

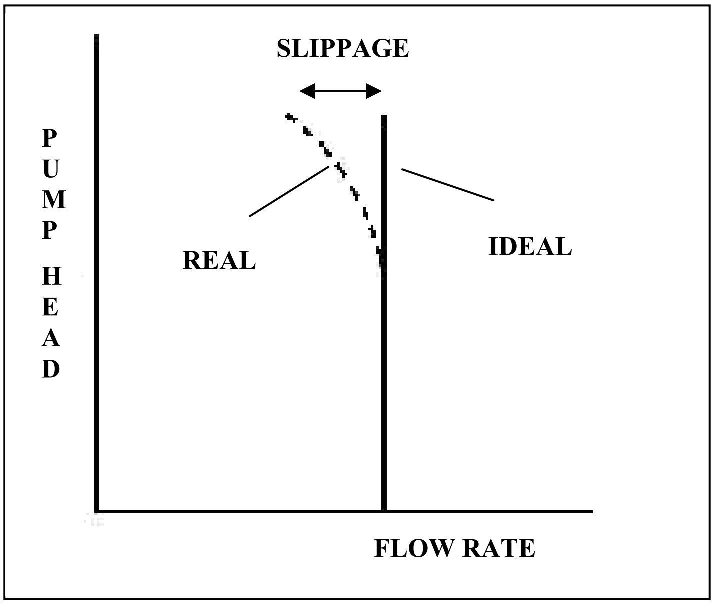

Figure 2 Positive Displacement Pump Characteristics [5]

In Figure 2 the dashed line shows the actual positive displacement pump performance and the solid line shows the ideal case. As the discharge pressure of the pump increases, some amount of liquid leaks from the discharge of the pump back to the pump suction, reducing the effective flow rate of the pump. The rate at which the liquid leaks from the pump discharge is called slippage [5] [6].

The positive displacement pumps are more suitable than others, when high pressure and low flow rates are required for the application. According to Parker [7] oil pipelines are used for transportation of the fluids having different viscosities and require constant flow at various discharge pressure. Therefore they use positive displacement pumps. Because other types may not be able to produce a high enough pressure to clear the line in case of a shut down and restart of the system. Since, with the cooling of the liquid viscosity increases and the pumping need higher pressures than the normal. Similarly, the fuel systems require constant amount of liquid although the pressure may increase or decrease in case of nozzle deformations. These types of requirements can only be fulfilled with positive displacement pumps which supply a steady source of power to the fuel systems.

1.1.1 RECIPROCATING PUMPS

The three basic types of positive displacement pumps are [8]:

- Reciprocating pumps
- Rotary pumps
- Special-purpose pumps

This study is focused on the reciprocating pumps especially on diaphragm pumps, which is a subgroup of reciprocating pumps. A reciprocating pump has two one-way valves, and a chamber that is filled and displaced between the two valves by a piston or membrane moving in a reciprocating motion [8]. There are different classifications of reciprocating pumps according to mechanisms or numbers of cylinder [9]. A general classification can be presented as

- Piston pumps
- Diaphragm pumps
- Bellow pumps

Another classification is done according to the number of the cylinder. In **simplex, single acting pumps**, there is one cylinder creating the flow with a single stroke in a single rotation of the crank of the piston. There is a periodic sine wave pressure in the flow which leads to pressure pulsations and shaking forces in the discharge. A

single acting duplex (or **duplex**) **pump** has two diaphragms and in a single rotation of the crank the pump has two stokes created by the two pistons of the pump. The pressure waves in the discharge are again sine waves but a **triplex pump** is the one having three compression and three strokes in one cycle and having three cylinders and **quadruplex,** having four cylinders.

In single acting pumps there is one intake and one compression in one stroke of the piston. In a double acting pump, differently from single acting, there are two intakes and two compressions in a single stroke of the piston. Bhabani and McAllister [10] give the details of the flow characteristics of these types and the behaviors are indicated.

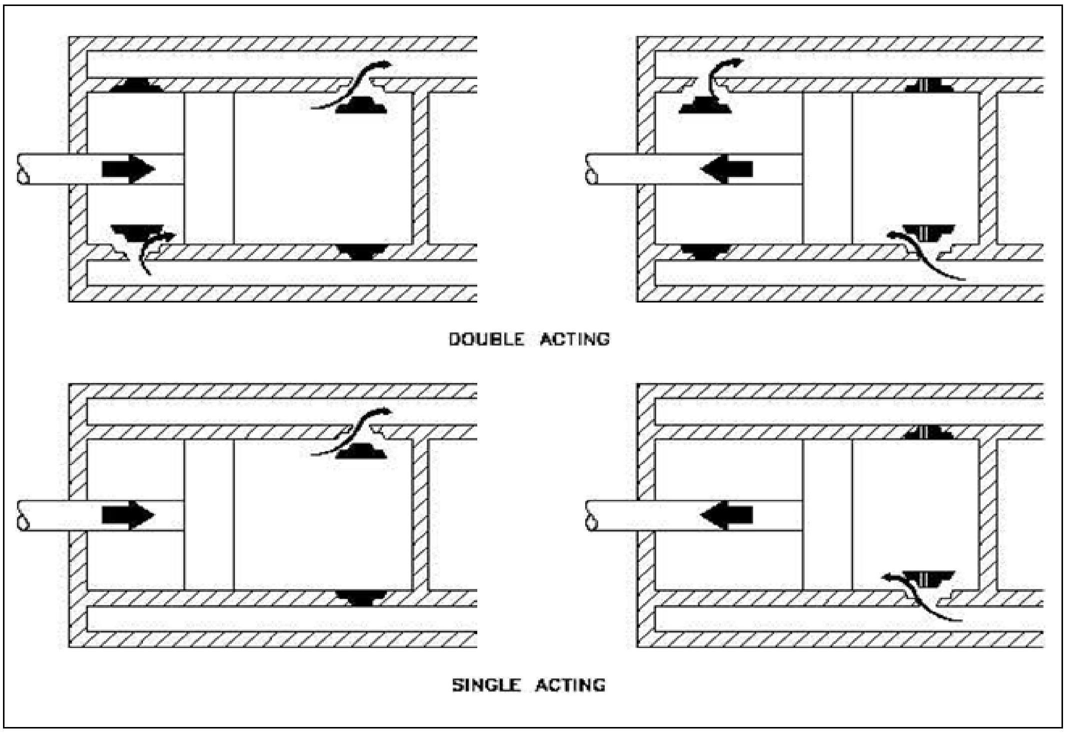

Figure 3 Single Acting and Double Acting Pumps [11]

1.1.1.1 DIAPHRAGM PUMPS

Diaphragm pump is a kind of positive displacement pump, where a crank attaches to the diaphragm and a hydraulic fluid moves the diaphragm up and down (or sideways), leading the piston move in a reciprocating motion up and down. The piston is the liquid end of the pump. The movement of the piston regulates the amount of fluid pumped [4]. A duplex diaphragm pump and its details are indicated in Figure 4.

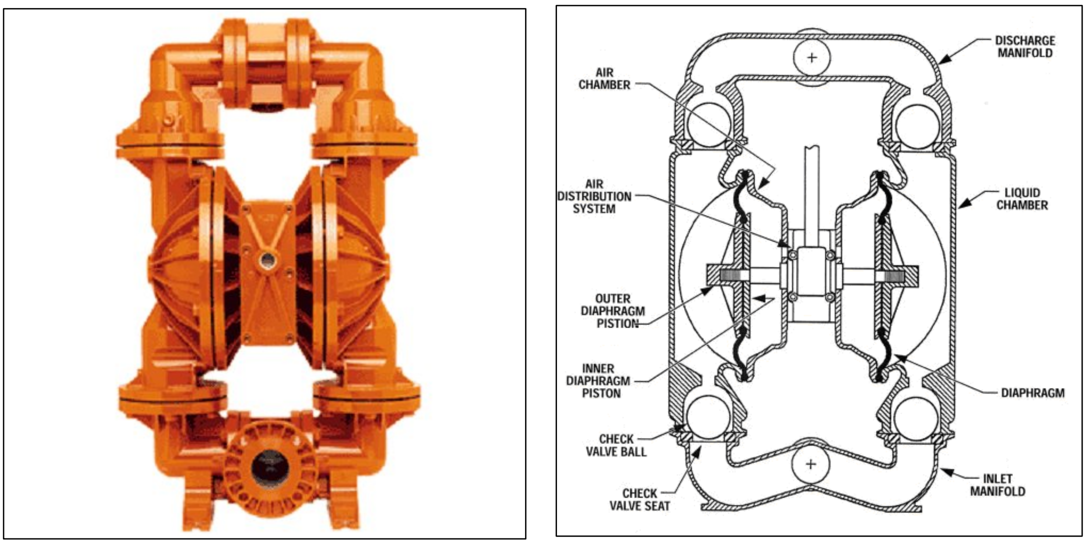

Figure 4 Diaphragm Pump [12]

In a diaphragm pump, pressurized air or mechanically driven piston attached to the diaphragm which pushes the diaphragm across the chamber and fluid on the other side of the diaphragm is forced out. The diaphragm in the opposite chamber is pulled towards the centre by the connecting rod. This creates suction of liquid in chamber, when the diaphragm plate reaches the centre of the pump it pushes across the pilot valve rod diverting a pulse of air to the air valve. This moves across and diverts air to the opposite side of the pump reversing the operation. It also opens the air chamber to the exhaust. Figure 5 indicates the diaphragm pump used in the experiments.

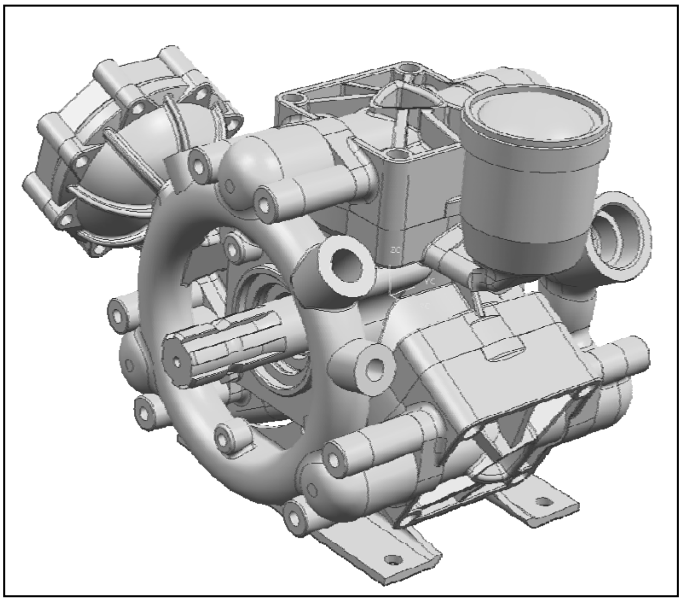

Figure 5 3D Model of the Test Pump

Diaphragm pumps offer smooth flow, reliable operation, and the ability to pump various viscous, chemically aggressive, abrasive and impure liquids. They are used in many industries such as mining, petro-chemical, pulp and paper. In addition, human heart can be classified as a diaphragm pump.

The liquid end of the diaphragm pump consists of a flexible membrane which moves or pulsates in a cylinder. The movement of the membrane and a piston are quite similar but the stroke of the membrane is smaller because of its geometry and it cannot move up and down or sideways like a piston [4]. The periphery of the membrane is fixed as indicated in Figure 6.

A schematic section of drawing of a diaphragm pump is indicated in Figure 6 presenting the elements,: (1)housing, (2)valves, (3)head cover, (4)diaphragm clamping disc, (5)diaphragm (membrane), (6)diaphragm supporting rod, (7)connecting rod, (8)eccentric bushing.

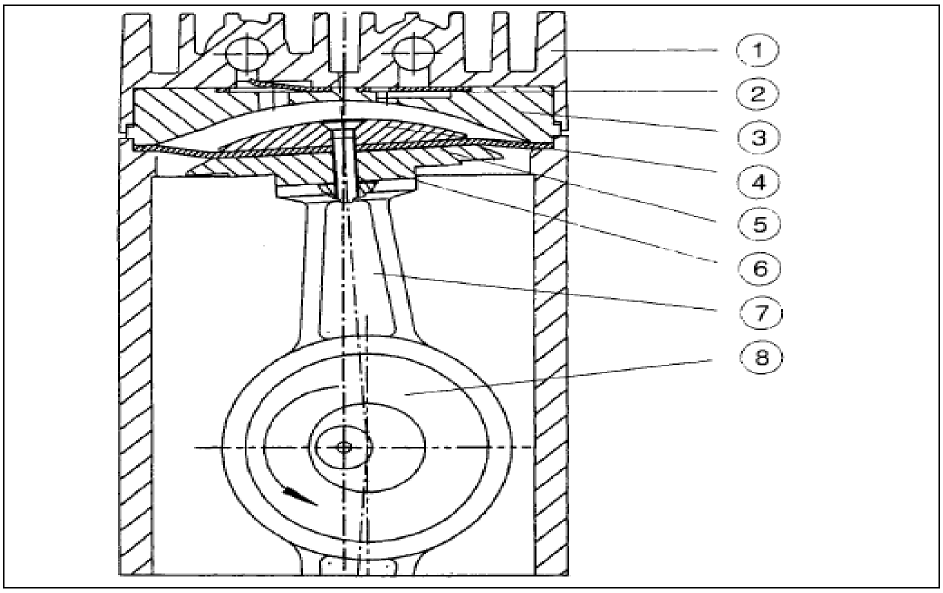

Figure 6 Section Drawing of Diaphragm Pump [13]

1.2 PULSATION DAMPENING AND SURGE TANKS

Because the pistons of the diaphragm pumps have reciprocating movement creating pulsation, the damages of the pumping system due to shaking forces and hydraulic shock in the long term usages are inevitable. These damages may come up as unsteady flow, wear and fatigue, cavitation or noise and vibration [14].

In order to not to face such problems in the pumping systems, some precautions must be taken such as; using equipments having pulsation reducing properties. There are many methods to avoid these pulsations but using surge tanks, controllable vanes, one sided vanes are the most extended used solutions [15]. **Surge tanks,** which can be also found as “pulsation dampener” or “air chamber” in the literature, reduce the pulsations in the flow, and thus enhance the performance and reliability of fluid handling equipment in industrial, chemical transfer, and precision metering applications [14].

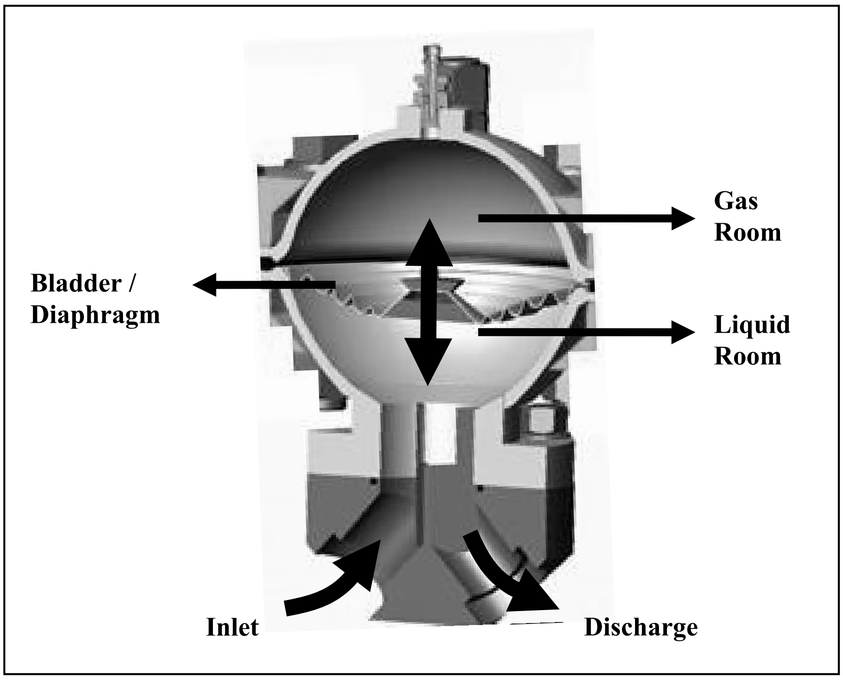

Figure 7 A Typical Surge Tank [16]

As indicated in Figure 7 surge tanks are small pressurized tanks containing both fluid and air together separated by a membrane or diaphragm, and usually connected to discharge of the pumps. The lower part of the surge tank open to the flow allowing the inlet and discharge of the flow and the upper part is for the gas (usually nitrogen or air) resisting the flow with the help of the precharge pressure and the pressure increase due to compression because of the fluid flow. The diaphragm between the gas and fluid chambers serve as a barrier and rapid pressure response times for pressure control pump pulsation and shock-dampening applications are facilitated by the light weight of the bladder [17].

The bladder or the diaphragm moves up and down with the flow pulsations whose frequency is related to the rotational speed of the crank. With the upward movement, the volume of the gas room decreases, increasing the pressure and resisting the pressure pulsations in the liquid room caused by the flow. When the flow in the line starts to fluctuate, fluid fills into the surge tank decreasing the amplitudes of the pulsations with the pre-pressurized air on the surge tank.

The rate of the pressure change in the surge tank is directly related with the pre-pressure value of the surge tank, the flow rate of the fluid, and the thermodynamics of the air inside the surge tank [15]. In order to use the surge tanks efficiently it is recommended to install them as near as the pistons or diaphragms to minimize the length that the pulsated flow follows [3]. The other and the most valuable parameter for the efficiency of the surge tank is its design. For the high speed pumps or long delivery pipelines or against high heads the surge tanks must be designed larger and the inlet area which is evaluated having a significant pulsation dampening effect must be taken into consideration [3].

1.2.1 SURGE TANK TYPES

There are many different pulsation sources in the pumping systems caused by different reasons and therefore there are many types of surge tanks. The classification of them can be made according to their dampening fluid, their bladder type, style or working principle as stated below [18].

Table 1 Classification Criteria for Surge Tank Types

Classification Criteria	**Types**
Types of fluids	Gas, liquid or dual where both gas and liquid is used.
Bladder types:	Confined, unconfined, foam or none
Style:	Appendage, flow thru, diverter
Working principle:	Energy absorbing, acoustic or reactive.

The surge tank that is used in the experiments is gas type with a free bladder inside having the energy absorbing property. There are several types of dampeners which are indicated in Figure 8.

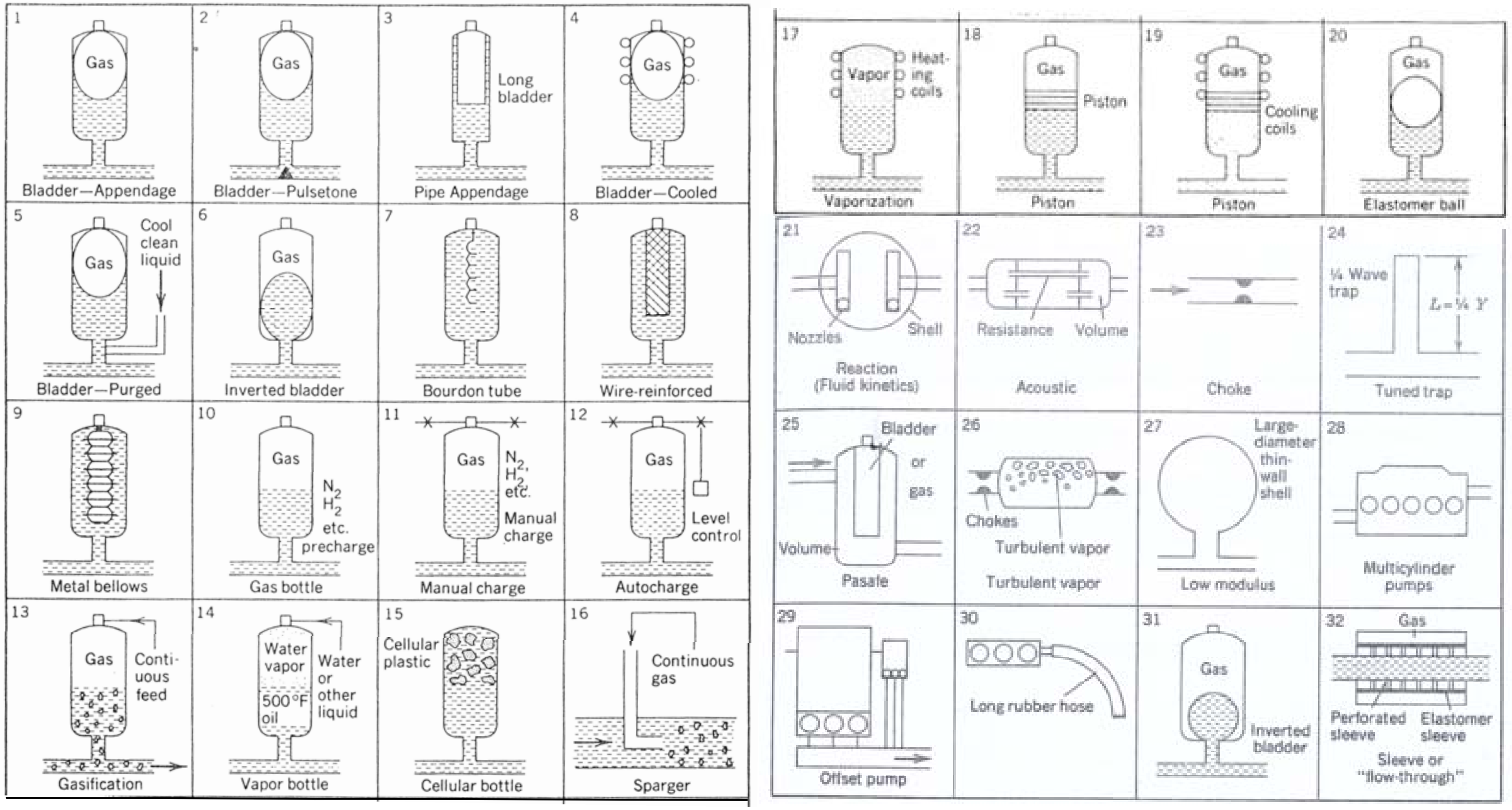

Figure 8 Types of Surge Tanks [18]

CHAPTER 2

LITERATURE SURVEY AND AIM OF THE STUDY

In the previous chapter the theory of the pump classification and pulsation dampening and surge tank properties are explained. In this chapter the literature survey is presented. The prior work on surge tank sizing, pump experiments and inlet area configurations on surge tanks is presented. Although there are plenty of sources in surge tank sizing, there is very little reference on surge tank inlet area and effects on pressure pulsations. Similarly there is not a significant study to be taken as reference on experiments of a diaphragm pump. With the lack of information in the literature the motivation, necessity and the aim of this thesis study is explained. The details are given in the following sections.

2.1 SURGE TANK SIZING

The selection of an installation location, the size and the inlet area at the entrance of the surge tanks are the important parameter in the design stages. The effective use of the surge tanks and eliminating the problems on the piping systems due to the flow pulsations depend on the selection of these parameters.

Hence, in order to use the surge tanks efficiently and to avoid the damaging effects of the pulsations on the piping systems, the installation location, the size and the inlet area of the surge tanks must be designed and applied precisely.

In 1933 Mead [3] suggested, the triplex pumps under low lifts may be provided with surge tanks of a capacity equal to a singe displacement of the plunger while for single cylinder and double acting pumps the chamber should be six to eight times of the plunger displacement. Therefore, the triplex pump used in this thesis study, which has 90 l/min volumetric flow rate with 540 rpm a single displacement of the piston, is:

$$Q_{piston} = \frac{90 \cdot 1000}{540 \cdot 3} = 55.5 cm^3 \tag{2.1}$$

$$Q_{piston} = V_{surge} = 55.5 cm^3 \tag{2.2}$$

Where Q_{piston} is volumetric flow rate of the single piston and V_{surge} is the volume of the surge tank. Hence according to Mead, the surge tank volume of 55.5 cm^3 is enough for the triplex pump having 90 l/min volumetric flow rate with 540 rpm. Though, the size of the surge tanks should be used depending upon the conditions of working and the greater irregularities of the flow leads to grater surge tank capacities. It can be concluded that with greater surge tank size the magnitude of pulsations can be diminished.

By 1964, Özgür [19] introduced an analysis for positive displacement pumps for the duplex pumps. From this study, the volume of a surge tank for a simplex and a duplex pump supplying a definite volumetric flow rate and a rotational speed are to be calculated with a simplified theory. In this study his approach is extended to triplex pumps and the theory is redeveloped. It is assumed that the volumetric flow rate keeps constant and the all the pulsation is assumed to be reduced by the surge tank. In order to calculate the required surge tank a 1% pulsation is allowed and the corresponding surge tank volume for the test pump is calculated as in the following steps:

The mean volumetric flow rate of a triplex pump is

$$q_{mean} = \frac{n}{60} \cdot 2r \cdot A = \frac{2rA}{2\pi/\omega} \tag{2.3}$$

If these two flow rates are compared it is realized that the flow rate is π times greater than the mean flow rate when $\alpha = \pi$. This much irregularity on the compressed volume of the flow leads to big oscillations and vibration on the pump. Therefore using a surge tank and/or increasing the piston number are preferred to avoid this effect.

In Figure 9 the flow rate diagram of a triplex (three piston/diaphragm) pump is presented.

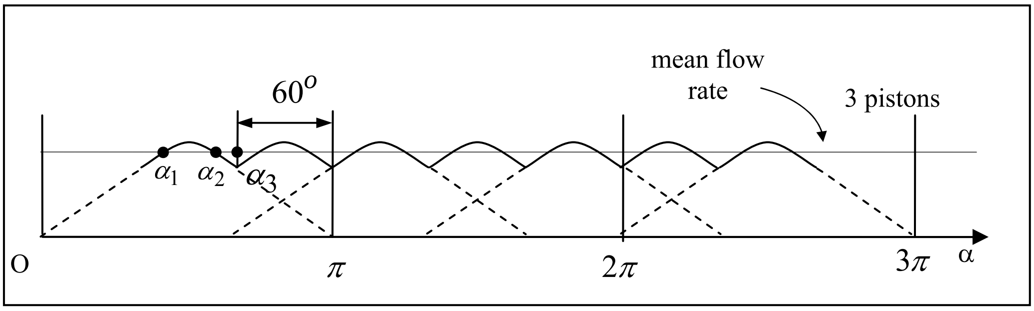

Figure 9 Flow Rate Diagram of Triplex Pump

As indicated in Figure 9 the irregularity decreases and the flow rate fluctuation period decreases to 60o.

In the pumping systems, in order to prevent the excess pressures and pressure oscillations the device which is called "*surge tank*" is used. With the help of a surge tank on the discharge side relives the flow strokes and pumps smoothly and at a constant velocity.

For simplicity, the flow rate of the pump is

$$q_{pump} = r \cdot \omega \cdot A \cdot \sin\alpha \tag{2.4}$$

For a triplex pump, the mean flow rate is

$$q_{mean} = \frac{3 \cdot r \cdot \omega \cdot A \cdot}{\pi} = 0.995 \cdot \omega \cdot r \cdot A \tag{2.5}$$

Therefore the difference between the mean volumetric flow rate and the pump flow rate is always compensated by the surge tank. Then the amount of flow entering the surge tank at any instant is $q_{surge} = q_{pump} - q_{mean}$

For the beginning, the air volume in the surge tank is H_o. From point "O" to α_1 at time t_1, $q_{pump} \leq q_{mean}$, therefore the flow is leaving the surge tank. If the amount of water leaving the surge tank is ΔH_1

$$\Delta H_1 = -\int_0^{t_1} q_{surge} \cdot dt = \int_0^{t_1} q_{mean} - q_{pump} \cdot dt \tag{2.6}$$

$$\Delta H_1 = \int_0^{t_1} \left(\frac{3 \cdot \omega \cdot r \cdot A}{\pi} - \omega \cdot r \cdot A \sin \alpha \right) \cdot dt = r \cdot A \int_0^{t_1} \left(\frac{3}{\pi} - \sin \alpha \right) \cdot d\omega t \tag{2.7}$$

$$\Delta H_1 = r \cdot A \int_0^{\alpha_1} \left(\frac{3}{\pi} - \sin \alpha \right) \cdot d\alpha \tag{2.8}$$

$$\Delta H_1 = r \cdot A \left(\frac{3\alpha_1}{\pi} + \cos \alpha_1 - 1 \right) \tag{2.7}$$

If the mean volumetric flow rate and the pump flow rate are equal to each other at the points α_1 and α_2

$$r \cdot \omega \cdot A \cdot \sin \alpha = \frac{3}{\pi} \omega \cdot r \cdot A \tag{2.8}$$

$\alpha_1 = 1.269$ and $\alpha_2 = 1.87$ in radians.

$$\Delta H_1 = r \cdot A \cdot (1.211 + .297 - 1) = 0.509 \cdot r \cdot A \tag{2.9}$$

From point α_1 to α_2 (from time t_1 to t_2) the amount of water entering the surge tank is ΔH_1

$$\Delta H_2 = r \cdot A \int_{\alpha_1}^{\alpha_2} \left(\sin\alpha - \frac{3}{\pi} \right) \cdot d\alpha = -r \cdot A \left[\cos\alpha + \frac{3}{\pi} \right]_{\alpha_1}^{\alpha_2} \tag{2.10}$$

$$\Delta H_2 = -r \cdot A \left[\left(\cos\alpha_2 + \frac{3\alpha_2}{\pi} \right) - \left(\cos\alpha_1 + \frac{3\alpha_1}{\pi} \right) \right] \tag{2.11}$$

$$\Delta H_2 = -r \cdot A \left[\cos\alpha_2 - \cos\alpha_1 + \frac{3}{\pi}(\alpha_2 - \alpha_1) \right] = -r \cdot A [-0.592 - 0.5739] \tag{2.12}$$

$$\Delta H_2 = 0.018 \cdot r \cdot A \tag{2.13}$$

Where,

$$s\max(\alpha = \pi) = r + \sqrt{l^2} \tag{2.14}$$

$$s\min(\alpha = 0) = -r + \sqrt{l^2} \tag{2.15}$$

$$s = \Delta s = 2r \tag{2.16}$$

Therefore, $\Delta H_2 = 0.009 \cdot s \cdot A$ (2.17)

This means ΔH_2 amount of water enters the surge tank and the water level increases from level "I" to "II". In the next stem between the points α_2 to α_3 ΔH_1 amount of water leaves the surge tank and the water level in the surge tank decreases to level "I". Therefore the level in the surge tank differs in between the levels "I" and "II". The change in volume of the air is equal to the change in volume of the water.

Therefore ΔH_2 shows the difference of the surge tank air volume during the operation.

To calculate the required surge tank volume this air (or the water) volume difference is crucial. According to Özgür [19] to keep the discharge volumetric flow rate as constant the surge tank pressure must be constant and to satisfy this requirement the surge tank size must be infinitesimally large. However practically the pressure is allowed to change 1% and the surge tank volume is calculated consequently:

$$\frac{p_2 - p_1}{\frac{p_2 + p_1}{2}} = \frac{1}{100} \tag{2.18}$$

If the air in the surge tank is assumed as isothermal:

$P_1 \cdot H_1 = P_2 \cdot H_2 = C$, where C is a constant number

If these are substituted in the previous equation:

$$\frac{H_2 - H_1}{\frac{H_2 + H_1}{2}} = \frac{H_1 - H_2}{V_{surge}} = \frac{1}{100} \tag{2.19}$$

$\Delta H_2 = H_1 - H_2$

where, $A = \dfrac{Q \cdot 60}{s \cdot n}$ (2.20)

and $s = 2r$ as discussed before.

Hence, $V_{surge} = (H_1 - H_2) \cdot 100$ (2.21)

If the equation is substituted:

$$V_{surge} = 0.009 \cdot s \cdot \frac{Q \cdot 60}{s \cdot n} \cdot 100 = 54 \cdot \frac{Q}{n} \tag{2.22}$$

$$V_{surge} = \frac{54 \cdot 0.0015}{540} \cdot 10^6 \tag{2.23}$$

$$V_{surge} = 150\text{cm}^3 \tag{2.24}$$

This design solution can be adapted as it can be used for surge tank sizing of the pump which is used in the experiments. Although it is a diaphragm pump, as soon as the flow characteristics and the number of the diaphragms are the same, the given design solution and the result are applicable to this pump.

The critical point of way that this design solution is adapted to the test pump is, the piston area term in equation (2.20) which is affecting the result directly. In the test pump the area of the piston of the connected diaphragm does not lead the way to find the volume in the pump. On the other hand the volumetric flow rate of the pump is supplied by manufacturer of the pump. With this information a virtual piston area can be calculated as if this pump is a piston pump and the surge tank volume required to suppress the pulsations of the pump having this much flow rate can be calculated. Only factor affecting the surge tank size is the amplitude of the flow pulsations (the number of pistons accordingly) and the flow rate of the pump

Where, V_{surge} is volume of the surge tank, s is the stroke length of the piston, Q is the volumetric flow rate of the pump and n is the rotation speed of the pump. The calculation procedure is introduced in Appendix A in detail. If the same procedure is applied to the pumps having different number of pistons, the equations become:

$$V_{surge} = 3300 \cdot \frac{Q}{n} \text{ for one piston/diaphragm pump} \tag{2.25}$$

$V_{surge} = 1260 \cdot \frac{Q}{n}$ for two pistons/diaphragm pump having 180^o angle between the crank angles. (2.26)

$V_{surge} = 240 \cdot \frac{Q}{n}$ for four pistons/diaphragm pump having 90^o angle between the crank angles. (2.27)

The detailed calculation is given in Appendix A

In 1995 Miller [18] was the one who considered the reducing of pressure pulsations can be achieved not only with proper surge tank size but also with the correct surge tank gas precharge pressure. The precharge pressure is the gas filled into the gas surge tank, prior the operation. The most desirable precharge pressure offered by Miller is 60% to 70% of the average working pressure of the pump or the limit of the bladder allows. For the gas over liquid type surge tanks which is the type that is used for the test pump the precharge can be 100% of the working pressure. It is indicated in the literature that the precharge pressures below 25% of the average working pressure should be avoided, to extend the bladder life. Also low precharge pressures decreases the efficiency of the surge tanks However the manufacturer of the test pump used in this thesis study advised a precharge pressure of about 10% of the average working pressure. Moreover the surge tank cannot maintain higher pressures. In case of higher precharge pressures there is air leakage from the surge tank.

According to the design calculations of Miller with the suggested 10% precharge pressure of the test pump, the surge tank size is calculated as,

$$V_{surge} = K \cdot s \cdot D^2\, p_d / p_c \tag{2.28}$$

$$V_{surge} = 0.1 \cdot 0.394 \cdot 3.7^2\, 40/5 = 4.315\text{gal} \tag{2.29}$$

$$V_{surge} = 16340\text{cm}^3 \tag{2.30}$$

Where, K is the pump constant indicated in Table 2. s is the stroke of the pump, D is piston or plunger diameter, P_d is pump discharge pressure, P_c is the precharge pressure and V_{surge} is the volume of the surge tank. It can be concluded that the procedure which Miller suggests is not applicable for the surge tanks having 60% to 70% precharge pressure. Since the calculated surge tank size is too large for a 90 liters triplex diaphragm pump.

Table 2 Pump Type Constants [18]

Pump Type	Numb Cyls	Crank Angle	A(1)	B(2)	Total(3) Percent	K(4)	Max Press Percent(5)
Simplex SA	1	360	0.58	1	158	0.684	250
Simplex DA	1	180	0.29	1	129	1.368	166
Duplex SA	2	180	0.29	1	129	0.558	166
Duplex DA	2	90	0.24	0.22	46	0.199	93
Triplex SA	3	120	0.06	0.17	23	0.100	43
Triplex SA (0)	3	120	0.062	0.17	23	0.100	44
Triplex SA (30)	3	120	0.061	0.12	18	0.078	35
Triplex SA (60)	3	120	0.05	0.09	14	0.061	27
Triplex DA	3	120	0.06	0.17	23	0.200	43
Quadruplex SA	4	90	0.11	0.22	33	0.143	62
Quadruplex DA	4	90	0.11	0.22	33	0.286	62
Quintuplex SA	5	72	0.02	0.05	7	0.030	14
Quintuplex DA	5	72	0.02	0.05	7	0.076	14
Sextuplex SA	6	60	0.06	0.17	23	0.100	43
Sextuplex DA	6	60	0.06	0.17	23	0.200	43
Septuplex SA	7	51.4	0.012	0.026	3.8	0.016	8
Septuplex DA	7	51.4	0.012	0.026	3.8	0.032	8
Octuplex SA	8	45	0.026	0.052	8	0.035	15
Octuplex DA	8	45	0.026	0.052	8	0.070	15
Nonuplex SA	9	40	0.006	0.002	2.1	0.009	2
Nonuplex DA	9	40	0.006	0.002	2.1	0.018	2

On the other hand, Arnold and Stewart's [20] approach is similar to the other author's that, they advise a precharge pressure of 60-70% of average fluid pressure. The surge tank volume is suggested as;

$$V_{surge} = \frac{Ksd^2p^2}{100(\Delta p)p_c} \tag{2.31}$$

Where

p_c=precharge pressure (psi)

K = pump constant

s = pump stroke (in)

d= pump piston or plunger diameter (in)

p=average fluid pressure (psi)

Δp = allowed pressure pulsation (psi)

Therefore according to the properties of the flow and the pump the surge tank volume is calculated as;

$$V_{surge} = 521.3 cm^3 \tag{2.31}$$

Botteler and Wende [21] have a different approach but still they advise an 80% of precharge pressure. According to their formulation but with the test pump's pressure values the surge tank size would be;

$$V_{surge} = \frac{PumpDsiplacement}{\%precharge \cdot LOD \cdot Constant} \tag{2.32}$$

The pump constant is advised as 7 for triplex pumps. Therefore the equation (2.32) becomes;

$$V_{surge} = \frac{stroke \cdot D^2 \pi/4}{0.122 \cdot 0.315 \cdot 7} = 258.4 cm^3 \tag{2.33}$$

According to the reference [22] the surge tank volume is related with the liquid volume that the surge tank must store. Also maximum and minimum exit pressures affect the surge tank size. Therefore the liquid volume that the surge tank must store can be found as

$$V_{surge} = \frac{p_2 \cdot dv}{0.8 \cdot \%precharge \cdot (p_2 - p_1)} \qquad (2.34)$$

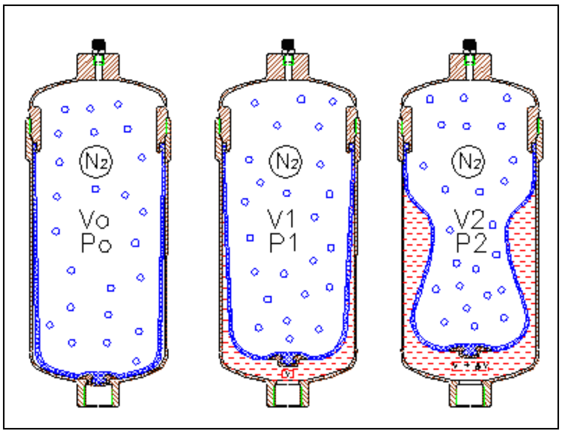

Figure 10 Surge Tank Representation [22]

And for triplex pumps

$$dv = C/18 \qquad (2.35)$$

Where

dv= liquid volume that the surge tank must store

C= the pump capacity per one revolution

P_2=maximum pressure in the line

P_1=minimum pressure in the line

Therefore

$$dv = \frac{77.37/540}{18} \cdot 10^3 = 7.96\text{cm}^3 \tag{2.36}$$

$$V_{surge} = \frac{46 \cdot 7.96}{0.8 \cdot 0.12 \cdot (46-33)} = 292.8\text{cm}^3 \tag{2.37}$$

With the experiment results gathered for the proper surge tank inlet area configuration.

Table 3 Results of Surge Tank Sizing Calculations

Calculation According to Reference	**Surge Tank Volume**
Mead [3]	56 cm^3
Özgür [19]	150 cm^3
Miller [18]	16340 cm^3
Arnold and Stewart [35]	521 cm^3
Boteler, Wende and Jennings [21]	258 cm^3
Hidracar	293 cm^3
Pump Manufacturer	415 cm^3

According to the calculations gathered from the above references the Miller's theory is far from the applicability since the resulting surge tank is very large. All the other theories seem applicable to the test pump. In order to create a vision to the solution of the problem, the two different sized surge tanks are manufactured. a 200cm^3 and a 80 cm^3 tank is manufactured and tested. The results are compared with the original surge tank of the pump which is 415cm^3.

2.2 EFFECT OF SURGE TANK INLET AREA ON PULSATIONS

Reviewing the literature about surge tank sizing the experimental study on pumping systems is searched to find similar studies as reference. In 1947 Becthold [24] introduces his study on gas compression systems and nature of pulsations to be expected from various types of compressors, as well as the effects which are produced by different types of piping and other equipment which may be associated with the compressor. In 2009 Almasi [25] had performed a study on pulsation and shaking force in reciprocating compressor systems. However these studies are about systems using gas as the working fluid and the authors did not introduce a sight on surge tank performance or surge tank inlet area for triplex diaphragm pumps

In spite of the several computational studies on surge tank design in the literature there are a few experimental studies on pumps regarding the surge tank performance and design studies. Besides, there is no considerable study on effect of surge tank inlet area on pulsations which is considered as a significant parameter.

2.3 AIM OF THE THESIS

Each of the previous researches has a different approach in the design and performance evaluation of surge tanks for diaphragm pumps. It is found that the experimental studies generally deal with surge tanks on gas compression systems. The lack of information in the literature about the nature of the pressure pulsations of a triplex diaphragm pump with different surge tank sizes and different inlet cross sectional areas and the experimental studies of the effect of them on the exit pressure pulsations has been a weak point on surge tank performance studies.

Based on the knowledge taken from the previous studies, the aim of this thesis study is to design and evaluate the performance of a surge tank for a triplex diaphragm pump in order to achieve maximum reduction of exit pressure fluctuations with the proper design of the proper surge tank size (volume) and optimization of the surge tank cross sectional area. Two surge tanks having different volumes are manufactured and performances are investigated and compared with the manufacturer's design. Moreover the effect of inlet cross sectional area of the surge tank on exit pressure pulsations is investigated and a proper design solution is searched both for the inlet area and the volume of the surge tank.

The results of this study will hopefully lead to the broader knowledge of the surge tank design and performance evaluation for a triplex diaphragm pump and the insight of the effects of surge tank sizing and inlet cross sectional area on the exit pressure pulsations.

CHAPTER 3

EXPERIMENTAL SETUP AND INSTRUMENTATION

Several experiments are conducted in this research in order to investigate the effect of surge tanks and its parameter on the discharge pressure fluctuations. An experimental set up is designed and constructed to perform these tests. Furthermore the characteristics of the pump are analyzed before surge tank design experiments. After the pump characteristics is obtained and the experiment set up is modified to perform the surge tank tests.

3.1 EXPERIMENTAL SETUPS

This experimental study is classified into two parts. First one is done to determine the pump performance characteristics and second one is done to evaluate the effect of surge tank parameters for discharge pressure fluctuations of the diaphragm pump. For these purposes, an experimental setup is constructed. A diaphragm pump used in the experiments is a positive displacement pump of type triplex pump having a surge tank on discharge side. The technical properties of this positive displacement pump are listed in Table 4.

In the following sections the experimental procedures, apparatus are explained in detail. Specifications of equipments as well as the test procedure are also notified.

Table 4 Technical Specification of the Test Pump

TEST PUMP		
Piston Diameter	[mm]	63
Eccentricity	[mm]	10
Discharge	[l/min]	90
Pressure	[kg/cm2]	50
Speed	[rpm]	540
Power	[hp/kW]	12 / 9
Number of Pistons		3
Membrane Dia.	[mm]	120
Useful Membrane Dia.	[mm]	94
Efficiency	[%]	76
Oil Capacity	[l]	1,5
Size	[cm]	35 x 42 x 35
Mass	[kg]	35

3.1.1 EXPERIMENTAL SETUP FOR OBTAINING PUMP PERFORMANCE

In the preliminary study, an initial set-up is constructed and initial experiments are performed in order to examine the characteristics of the diaphragm pump. Some of the structural parts are designed and manufactured for instrumentation of the pump for the data collecting and system stability. Firstly a clutch is designed and installed to the pump shaft to engage it to the rotating shaft of the electric motor. In order to avoid the vibration of the pump during working, an aluminum bench is manufactured. Also an adaptor part is designed and installed on top of the surge tank in order to install the pressure transmitter and the gage to the surge tank. The technical drawing of the adaptor and details accordingly is given in Appendix B.

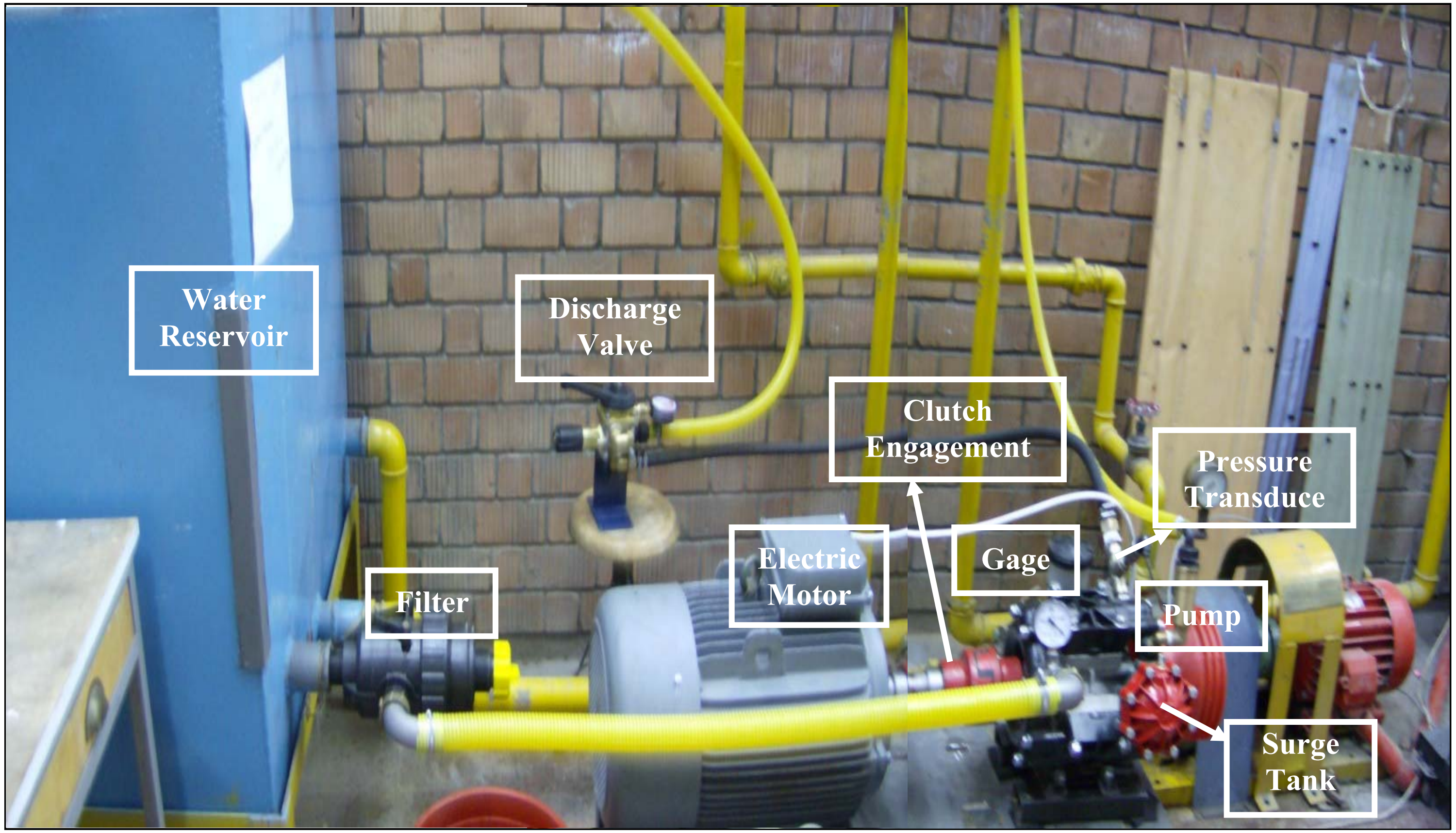

Figure 11 Experimental Setup for Pump Performance Tests

As indicated in Figure 11 the pump which sucks the fluid from the water tank and discharges to the container is driven by the electric motor. The exit pressure is controlled by the discharge valve. A filter is mounted at the suction side of the pump to avoid unwanted particles inside the pipeline. The inlet, surge tank and exit pressures are measured by pressure transducers and the gages at the same time. Before the experiments, the calibrations of the gages are checked with the sensors which are calibrated with a dead weight tester before. Dead weight tester is a hydraulic pressure calibration device. It is a piston-cylinder type measuring device. As primary standards, it is the most accurate instrument for the calibration of electronic or mechanical pressure measuring instruments [26]. The calibration curves for the pressure sensors are presented in Figure 12 and Figure 13.

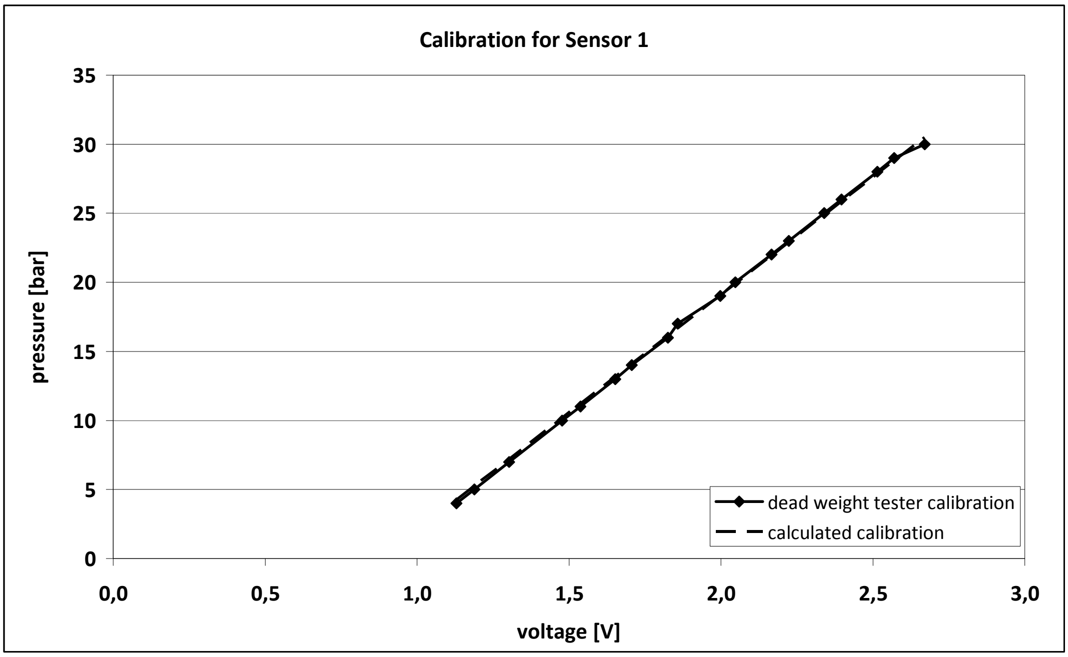

Figure 12 Calibration Curve for Pressure Sensor 1

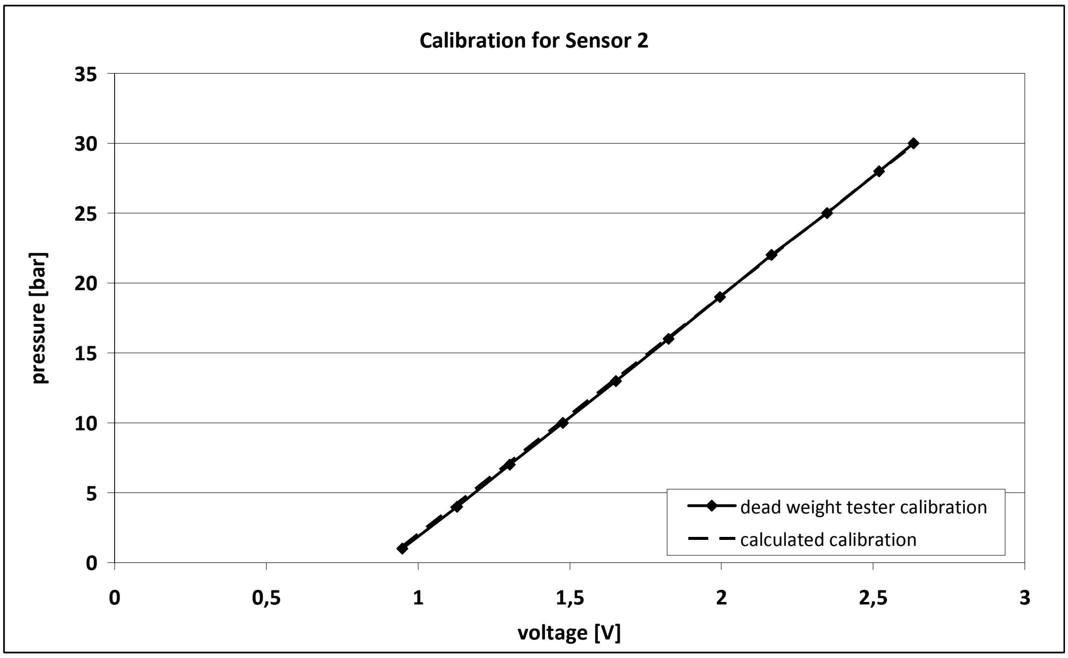

Figure 13 Calibration Curve for Pressure Sensor 2

The calculated pressure values and dead weight tester results are just identical for the two sensors as indicated in Figure 12 and Figure 13. Therefore the conversion of voltage data read from the pressure sensor can be converted to pressure value as in the following formula which is actually an interpolation according to property of the sensor and the resistance used to convert the ampere output to voltage data.

$$p_{exit} = \frac{60}{(4{,}4-0{,}8)} \cdot (V-0{,}8) \tag{3.1}$$

In the experiments, the volumetric flow rate, the electric and hydraulic power consumption of the system, the inlet and discharge pressures for different rotational speeds and discharge pressures are measured. With the help of these data, the flow characteristic of the pump and efficiency is determined. The results of these data are compared with the test results sent from pump manufacturer which they are performed in their facility.

In the following sections the experimental procedures, apparatus are explained in detail. Specifications of equipments as well as the test procedure are also notified

3.1.1.1 INSTRUMENTATION FOR PUMP PERFORMANCE TEST SETUP

The instrumentation used in the experimental set-up are indicated this section. The figures of the instruments are presented in set-up representation diagrams and the specifications are detailed in Appendix B.

Pump system is filled with water. Pressures at the inlet and exit of the pump are measured by Bourdon gages. At the pump inlet, the flow and also the pressure value read from the gage fluctuates because of the pulsations of the diaphragm pump. Due to the fluctuations average values are recorded. Also, two pressure transmitters are used to measure the pressure in the surge tank and at the pump discharge. This value is also compared with the gage display at the discharge side. The technical specification of the transmitters is given in appendix B The pressure transmitter is connected to a DC power supply in order to activate the pressure sensors. The output signals from the sensors are connected to a data logger to collect the data and record. In these experiments, Elimko data logger is used to gather pressure transmitter data for pump performance tests. The pressure transmitter is connected to the data logger and data logger collects the data coming from the transmitters and sends them to the computer. The data logger software interface page is indicated in

Figure 14. In Appendix B, it is possible to find the technical details of this data acquisition device.

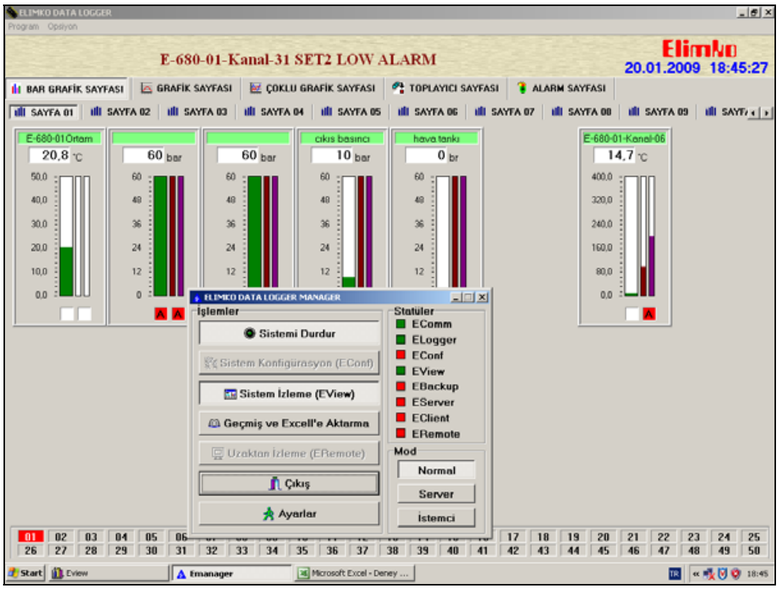

Figure 14 Datalogger Interface of Computer

A computer is used to monitor the collected data. For the preliminary experiments, an electric motor having 7.5kW power is used. However the available power to run the pump could not be obtained. Therefore, another electric motor with 18.5kW whose specifications is presented in Appendix B is used in the experiments. In other words, the electric motor is replaced with a more powerful one and the coupling is modified accordingly.

Siemens frequency converter is used to adjust the electric motor's rotational speed. A tachometer is used to measure the rotational speed of the pump. A voltmeter and clamp meter is used to measure the electric voltage and current input to the electric motor. In these tests, the volumetric flow rate is measured by a using calibrated tank method. 30 liter container is used for this purpose. By using chronometer, the time passed to be filled for the tank is measured and then it is converted to the volumetric flow rate.

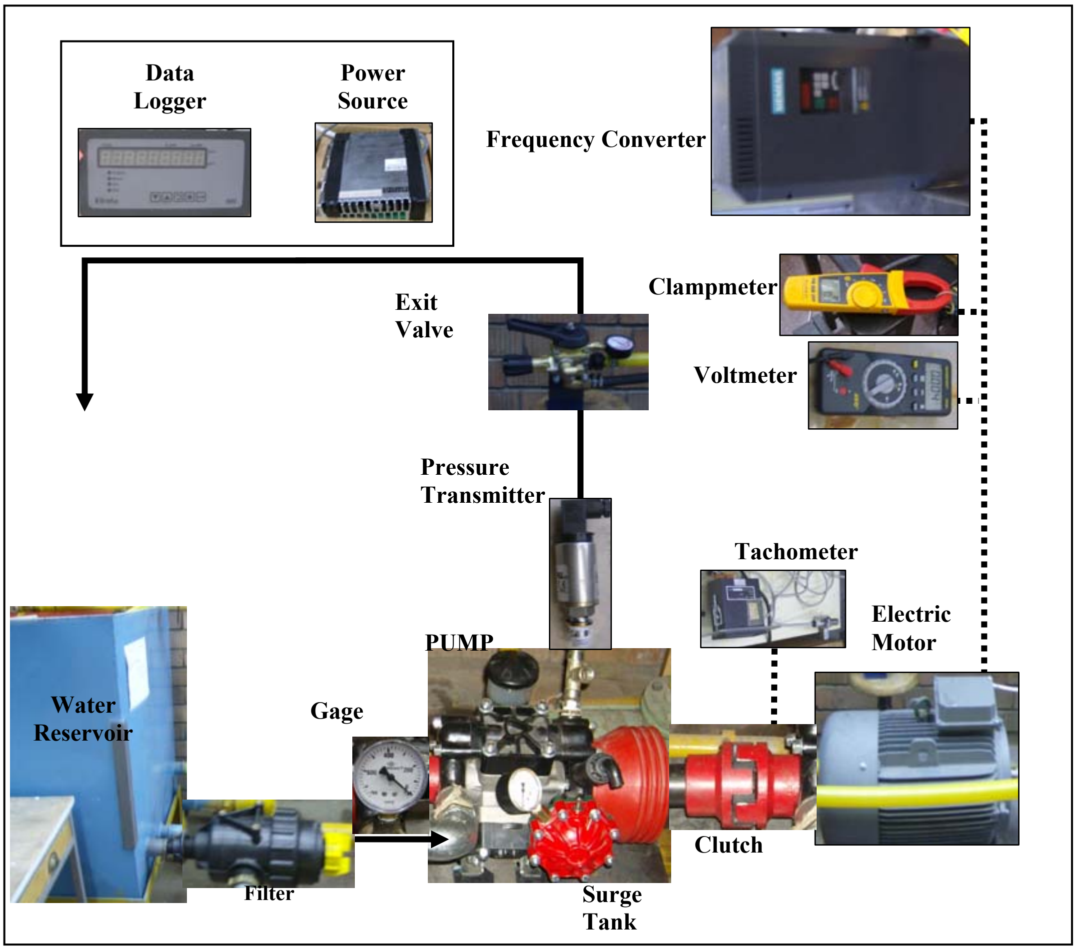

Figure 15 Schematic Representation of Instrumentation for Pump Performance Tests

3.1.2 EXPERIMENTAL SETUP FOR SURGE TANK PERFOMANCE TESTS

The surge tank tests are performed in order to achieve the aim of this thesis: to evaluate the design and performance of a surge tank for a triplex diaphragm pump in order to achieve maximum reduction of exit pressure fluctuations, with the proper design of the surge tank size optimization of the surge tank cross sectional area.

However, after the preliminary experiments are finished and the results are gathered, it is concluded that the set-up needs to be improved. The main working principle of the set-up remained the same but some extra elements are added to the set-up. It is necessary to improve the system components in order to evaluate the surge tanks parameters. In the previous experimental setup, the electric motor could not run the pump at higher rotational speed. The coupling used to connect the electric motor and the pump is replaced with a belt and pulley mechanism in order to work the motor at the normal speed. This modification is performed to eliminate the unknown efficiency decrease due to low rotational speed of the pump. After the modification the only parameter changing the efficiency of the pump became the ratio of the power of the electric motor to its rated power.

To collect the data from the surge tank some mechanical modifications are completed. An adaptor is designed and manufactured to connect the gage and the pressure transducer to the surge tank to collect surge tank pressure change data during the experiments. However after some tests, it is noticed that the surge tank pressure is not changing although the exit pressure is increased to 40 bars. It is realized that there is air leakage from the surge tank during the experiments. Since this issue affects the exit pressure pulsations, the adaptor and the pressure transducer is disconnected from the surge tank, although this means to sacrifice the surge tank pressure data.

In the preliminary experiments the Elimko data logger is used but it was impossible with the data logger to collect time history data. Therefore the succeeding experiments are done with Labjack U12 which is able to read 300 Hz data. We recorded the data with the computer.

The efficiency of the system is unable to determine because of the inaccurate reading of the clamp meter and voltmeter. Moreover in order to determine the electric motor efficiency, the motor needs to be run at specified conditions. However with using the clutch between the pump and the electric motor, the motor have to be run at 540 rpm instead of 1460 rpm which is normal working speed of the motor. Because of that a belt and pulley is installed in between the shafts of the electric motor and the pump. In short, a belt and pulley connection provides the electric motor operated at rated speed. With the help of belt and pulley, the speed of the motor can be decreased by 1/ 2,72.

Furthermore in order to determine the electric power of the system a voltmeter and a clamp meter is connected to the electric motor. However after the efficiency calculations it is realized that the electric power calculated with the data from the voltmeter and clamp meter is not correct. This is because the devices in the market such as the ones used in the laboratory can have some inaccuracy at the frequency range that we study. The system used in this thesis works at 27 Hz which will be calculated in the following chapters. Therefore a current transformer system is added to the system to measure the electric current accurately. Also to improve the ease of replacement of the pump there is an adaptor shaft is installed between the pump shaft and electric motor shaft.

When the pressure pulsations are analyzed, it is realized that the pulsations are quite high than expected after a couple of measurements for the surge tank tests,. The setup is improved such that

- The pressure transducers are connected to a battery for power in order to disconnect them from the DC power source. If the electric motor and the transducers have the same power source, the signals could be interfered.
- The electric motor is connected to ground by means of cables.
- Screened cables are used to connect the pressure transducers to the data acquisition system in order to avoid the interference.

There are great advantages of the improvements such as determining the pressure fluctuation amplitude and the pump efficiency with measuring the electric power correctly. However there are disadvantages of these improvements such as increasing the complexity of the set-up and also the friction losses emanated from extra links. The final set-up is indicated in Figure 16.

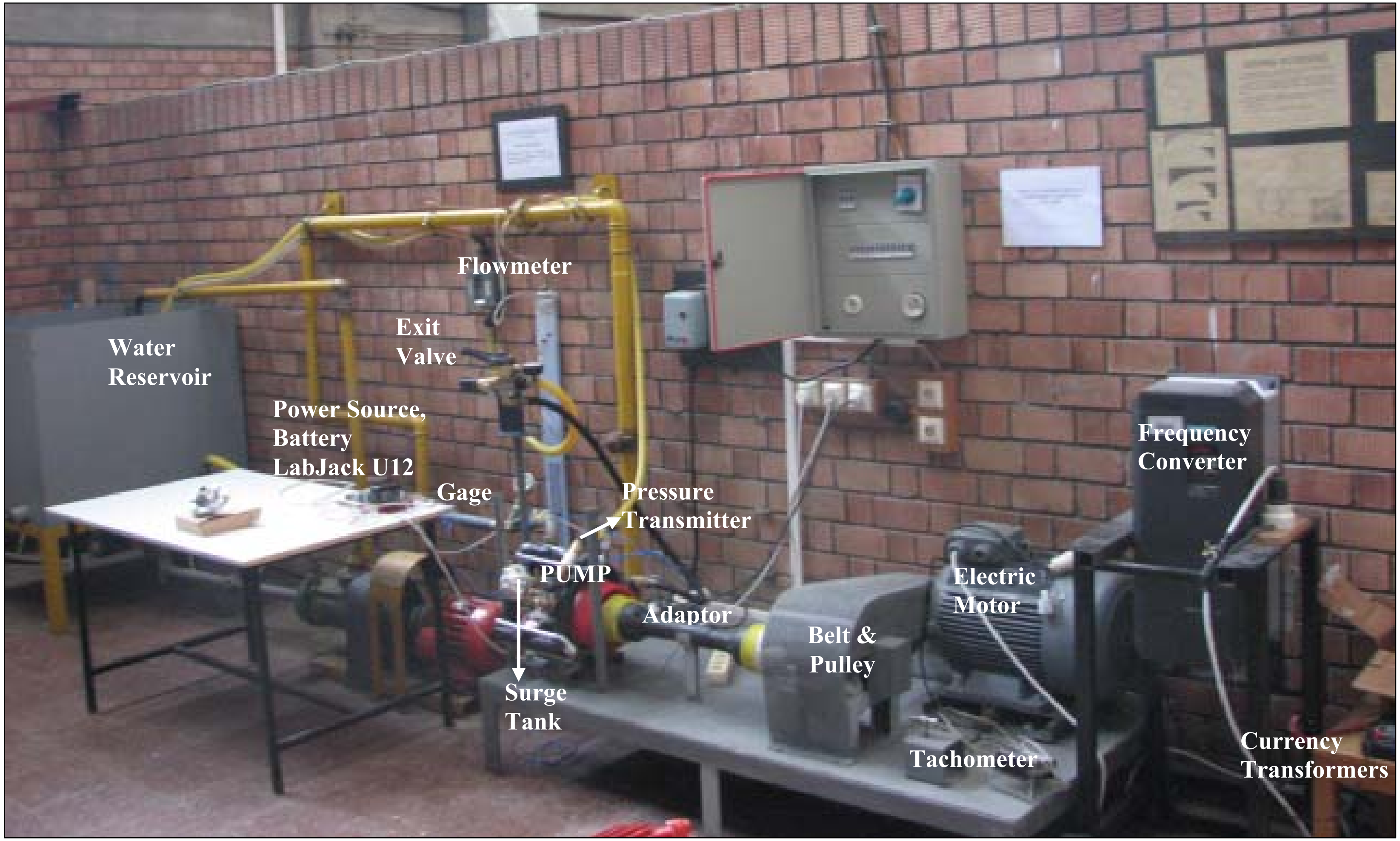

Figure 16 Experimental Setup for Surge Tank Performance Tests

3.1.2.1 INSTRUMENTATION FOR SURGE TANK PEROFORMANCE EVALUATION TESTS

The instrumentation used in the experimental set-up are indicated this section. Same Bourdon gages are used to measure the pressure at the inlet and discharge of the pump. Also same analog pressure transmitter is used to measure the pressure in the surge tank and at the pump discharge. This value is also compared with the gage display at the discharge side. The pressure transmitter is connected to a power source and to a data logger to collect the data and record. Instead of Elimko brand data logger the pressure time history of the data is recorded with LabJack U12 data acquisition device for surge tank performance tests. The data from pressure transmitters are gathered and uploaded to the computer. Pressure time histories are visualized. LabJack collects data at a rate controlled by its own crystal, and stores that data in an onboard buffer until read by the PC. The data is transferred from the LabJack buffer to the PC buffer, simultaneous with data collection, allowing the data to be streamed to disk continuously [27]. The user interface page is as presented in Figure 17. The technical specification of this device can be found in Appendix B.

The data collected with LabJack U12 are saved in ".txt" format and then imported to Microsoft Excel to ".xls" format. All the data acquisition is performed in Excel with the help of data analysis tools, Fast Fourier Transforms (FFT) and to speed up Visual Basic codes are implemented.

The first step in data processing is to convert sensor output data to pressure values. The pressure sensor expresses the pressure value in terms of amperes and this output is needed to be converted to into bars. On the other hand the Labjack U12 instrument can collect the voltage data. In order to satisfy these needs the sensor signal cable is connected to 220Ω resistance.

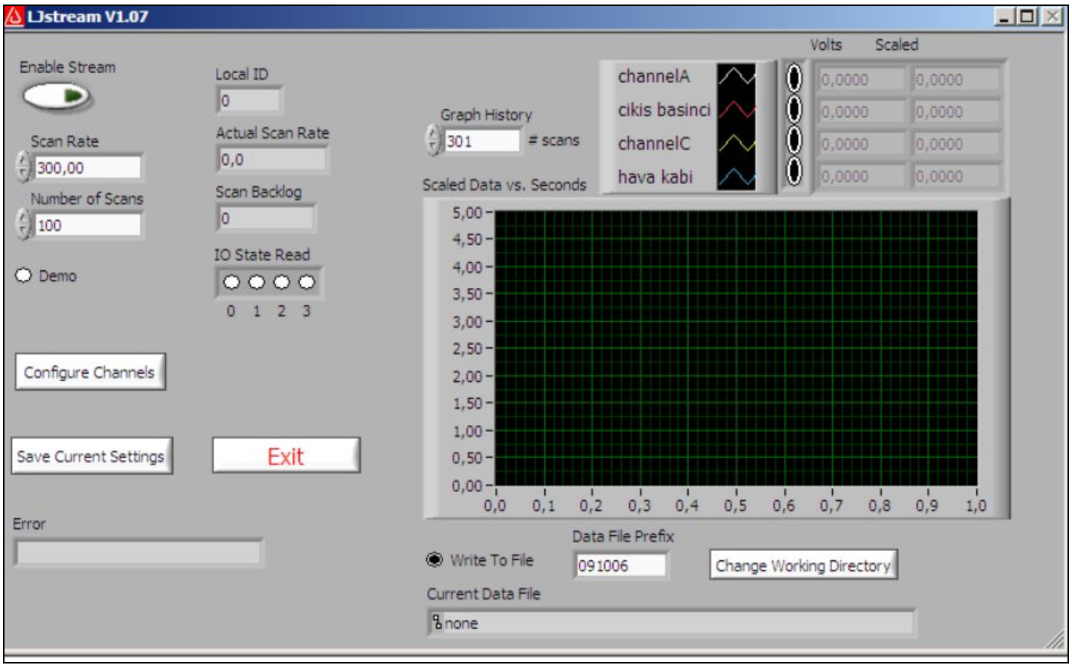

Figure 17 LabJack12 User Interface

A computer is used to monitor the collected data. The pressure transmitter is connected to the data logger and data logger sends them to the computer for monitoring. Same frequency converter is used to adjust the electric motor's rotational speed. A Racine Flow Meter [28] used to measure volumetric flow rate is a digital variable area flow meter. The features are specified in Appendix B. Again, a tachometer is used to measure the rotational speed of the pump. Current transformers are an indispensable tool to aid in the measurement of AC current. They provide a means of scaling a large primary current into a smaller, manageable secondary current for measurement and instrumentation [29] . The figures of the instruments are presented in set-up representation diagrams and the specifications are detailed in Appendix B.

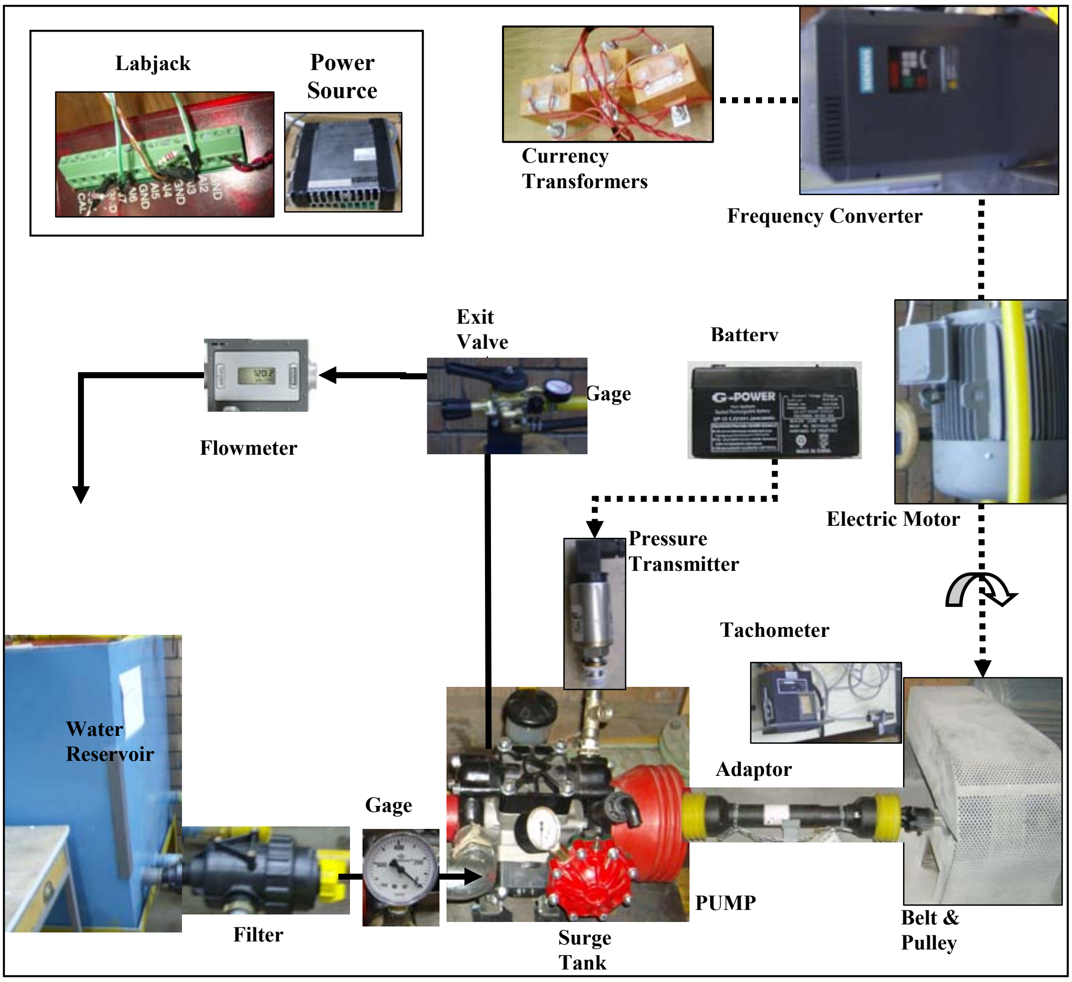

Figure 18 Schematic Representation of Instrumentation for Surge Tank Performance Tests

3.2 EXPERIMENTAL PROCEDURE

In this section the experimental procedures for the pump performance and the surge tank tests are explained in detail

3.2.1 PUMP PERFORMANCE TESTS

The purpose of the experiments in the pump performance tests is to analyze the characteristics of the pump and to verify the results taken from the set-up. In this scope a set of experiments is performed and the results are compared with the data supplied by the pump manufacturer. Following parameters are measured in the duration of experiment.

a) Rotational Speed of the pump from a tachometer
b) Pump Inlet pressure for Bourdon gage
c) Pump Discharge pressure from Bourdon gage and pressure transmitter
d) Electric Voltage Input from voltmeter
e) Electric Current Input from clamp meter
f) Time passed to fill the calibrated tank.

To evaluate the pump performance in the preliminary study it is needed to evaluate electrical power. However as mentioned before due to some issues and incompatible instrumentation that cannot be possible.

In order to calculate the flow rate the calibrated tank is used. The time for the tank to be filled is measured and the volumetric flow rate is calculated as follows:

$$Q = \frac{V_{\tan k}}{t} \cdot [l/s] \tag{3.2}$$

Where t represents the time for 30 l tank to be filled in seconds and V_{tank} is the volume of the measurement tank (30 liters). The volumetric flow rate, hydraulic power is calculated as:

$$p_{hyd} = \Delta p \cdot Q \tag{3.3}$$

$$\Delta p = \cdot p_{disch\,arg\,e} - p_{inlet} \tag{3.4}$$

where Δp is pressure difference between inlet and exit of the pump and Q is the volumetric flow rate.

The pressure values are read from data logger which does not collect time history of the data. It only gives the instantaneous value read from the transducers. Therefore the pulsations were not possible to evaluate and investigate. The pump properties are determined and the drawbacks of the experiment set are realized with the help of the preliminary pump properties experiments

3.2.2 SURGE TANK PEROFRMANCE TESTS

After getting the characteristic behavior of the pump, the deficiencies of the experimental set up is eliminated. The main purpose of the experiments in the scope of this thesis study is to find a proper area ratio for the surge tank of the 3 diaphragm positive displacement pump having a 90lt /min volumetric flow rate and a rotational speed of 540 rpm.

For this purpose the existing set-up is modified and a final set-up is obtained which is indicated in Figure 18 and the following parameters are measured with the corresponding equipment explained above.

a) Pump inlet pressure
b) Pump exit pressure
c) Pump rotational speed
d) Electric power
e) Pump volumetric flow rate

Moreover surge tank pressure is measured in couple of experiments. But after some runs the adaptor part started to slip some air. Therefore the adaptor part with the pressure transducer is disconnected and the original surge tank is used.

The flow frequency is calculated as follows:

$$f_{pump} = \frac{540}{60} \cdot 3 = 27 \text{ Hz} \tag{3.5}$$

The electric motor frequency is:

$$f_{motor} = \frac{540}{60} \cdot r_{belt\&pulley} = 24.5 \text{ Hz} \tag{3.6}$$

In order to evaluate the minimum pulsations and to determine a proper design for both surge tank size and the inlet area adaptor, the maximum, minimum and the average pressure values are recorded and peak-to-peak pulsation is calculated as [19]:

$$\%pulsation = \frac{p_{max} - p_{min}}{p_{average}} \tag{3.7}$$

The main design consideration is the peak to peak pulsation percentages at 40 bars since the normal working pressure of the pump is around this value. To evaluate the data frequency spectrum to investigate that if the pulsation peaks are at the pump working frequency, the data is processed. The processing of the collected data is done by Microsoft Office Excel program. The time histories and their frequency spectrum are evaluated. The data is converted to frequency domain by the FFT converter of Office Excel and with the help of the manual by Klingenberg [30]. The peaks of frequency spectrum of the pressure data are achieved at 27 Hz as expected .The regarding results are presented in Chapter 4.

Variation of pressure with time is only collected in these experiments Rotational speed or the volumetric flow rate history in time is not being able to be collected. Only a single data for rotational speed and the volumetric flow rate could be recorded for each run. It could be a further study to improve the instrumentation and visualize the flow rate change with the pressure pulsations and the rotational speed accuracy during the runs.

Various test runs are performed to simulate the different conditions to analyze the behavior of the pump and come out with optimization concept. The parameters which kept constant in the runs are;

- Rotational speed of the pump
- Surge tank initial pressure
- Surge tank volume

The test matrix is presented in section 3.2.3.

The rotational speed of the pump is adjusted by the frequency converter. Similarly the surge tank pressure is intended to be kept constant during the runs. However there may be some leakage on the valve. Hence a set of experiments is intended to be performed with a higher pre-pressure of the surge tank. However the valve on top of the surge tank did not let pressurizing it higher than 5 bars and this test is cancelled. It could be a further study to improve the instrumentation and see the effect of surge tank pre-pressure o the pulsations of exit pressures. Moreover after some tests the adaptor connecting the pressure transducer to the surge tank was out of order and it is disconnected sacrificing the surge tank pressure time history.

A valve installed to the discharge side of the pump. The size and inlet cross sectional of the surge tank are analyzed to obtain the effects on the pressure fluctuations at the discharge of the pump. Before finding the proper area ratio of the inlet area, different surge tanks having different sizes are tested. The exit pressure is varied from 0 bars to 40 bars with 10 bars increment. Although the maximum exit pressure of the pump is 50 bars. The tests are performed at 40 bars maximum because of the safety issues in the fluid laboratory and the excess vibration of the set up. It could be a further study to improve the instrumentation such as using a more stable bench which pump and the electric motor is installed or using any suitable damper for the pump.

Three surge tanks are tested with different volumes whose figures are indicated in Table 5. The experiments that are performed to find the proper inlet cross sectional area using the eight adaptor configurations are performed with the largest surge tank which is suggested by the manufacturer. The other two surge tanks are manufactured according prior work of Chapter 2 but about 5% larger. The properties of the surge tanks are presented in Table 5. After the experiments are conducted to evaluate the surge tank size leading the minimum pulsations the experiments are performed to evaluate the proper adaptor configuration. There are eight adaptor configurations whose properties are indicated in Table 6 and detail drawings are indicated in Appendix B.

Table 5 Surge Tanks Used in the Experiments

ID	Volume	Figure
Surge Tank 1	415 cm^3	
Surge Tank 2	200 cm^3	
Surge Tank 3	80 cm^3	

Table 6 Surge Tank Inlet Area Configurations

Adaptor no	Number of holes	Hole diameter	Area Ratio
1	Surge tank is disconnected.		0
2	7	2	0,086
3	5	3	0,139
4	3	4	0,148
5	7	3	0,194
6	5	4	0,247
7	7	4	0,346
8	1	18	1

The area ratios of the adaptors are calculated rationalizing the area of the adaptor to the maximum allowable inlet area. The ratio is calculated as:

$$\text{AreaRatio} = \frac{\text{Open} \cdot \text{area.} \cdot \text{of} \cdot \text{the} \cdot \text{adaptor}}{\text{Maximum} \cdot \text{available} \cdot \text{area} \cdot \text{of} \cdot \text{the} \cdot \text{adaptor}} \qquad (3.8)$$

The efficiency of the system is calculated as follows:

$$\eta_{\text{system}} = \frac{P_{\text{hyd}}}{\text{Pelec}} \qquad (3.9)$$

Where,

P_{hyd} =Hydraulic power

P_{elec} =Electrical Power

η = Efficiency

Furthermore the efficiency of the pump is calculated with the following information

- The efficiency of the belt and the pulley system is 0,93 [31]
- The efficiency of the frequency converter is 0,97 [32]

- The efficiency of the electric motor is 0,91 at the rated power [33]

In order to calculate the efficiency and the loss of efficiency of the electric motor the power percentage is calculated and efficiency loss percentage is calculated for every test run with the help of Figure 19 in Motor Challenge Factsheet [34]. In every measurement the electric power is measured from current transformers and rationalized with the motor rated power which is 18,5 kW as the specifications are given in Appendix B.

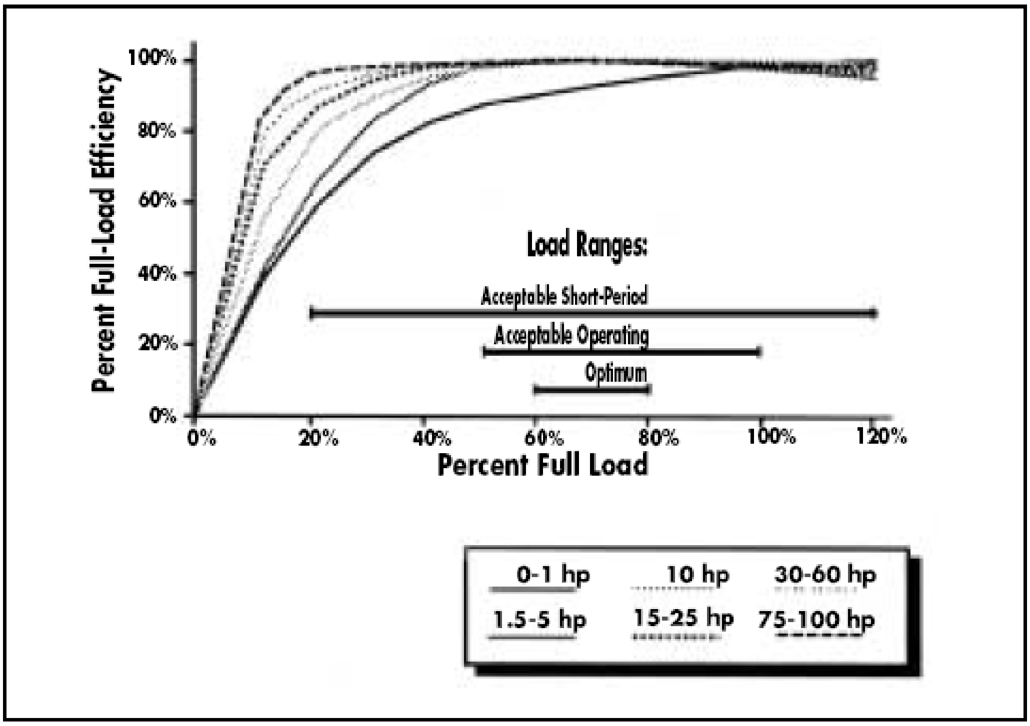

Figure 19 Motor Part Load Efficiency [34]

All the comparison between the results is done according to the peak to peak pressures and the % pulsations. The exit pressure pulsations are compared for different exit pressures and for different surge tank cross sectional areas.

3.2.3 TEST MATRIX

The eight configurations of surge tank inlet area adaptors are used whose detail is indicated in previous sections. After this experiment the two more surge tanks are tested with a single inlet area adaptor. The test matrix is presented in Table 7. For all 8 configurations four exit pressure values; 0, 20, 30 and 40 bars are tested

Table 7 Test Matrix

Test No		P_{exit} [bar]
1	Surge tank volume 80 cm^3 & r=0.14	5
2		20
3		30
4		40
5	Surge tank volume 200 cm^3 & r=0.14	5
6		20
7		30
8		40
9	Surge tank volume 415 cm^3 & r=0.14	5
10		20
11		30
12		40
13	Surge tank volume 415 cm^3 & r=0.09	5
14		20
15		30
16		40
17	Surge tank volume 415 cm^3 & r=0.14	5
18		20
19		30
20		40
21	Surge tank volume 415 cm^3 & r=0.15	5
22		20
23		30
24		40
25	Surge tank volume 415 cm^3 & r=0.19	5
26		20
27		30
28		40
29	Surge tank volume 415 cm^3 & r=0.25	5
30		20
31		30
32		40

Table 7 Test Matrix (Continued)

33	Surge tank volume 415 cm^3 & r=0.35	5
34		20
35		30
36		40
37	Surge tank volume 415 cm^3 & r=1	5
38		20
39		30
40		40

CHAPTER 4

RESULTS AND DISCUSSION

In this section of the thesis study, the outcomes of the experiments and the results are presented and the remarkable conclusions are reached. The effect of surge tank size and inlet cross sectional area is investigated to evaluate the proper configuration dampening the pulsations in the system.

4.1 PUMP PERFORMANCE TESTS

In the pump performance tests, pump exit and inlet pressures, surge tank pressure, rotational speed of the pump and flow rate of the system are recorded for different rotational speeds and different exit pressures. With the measurements, the volumetric flow rate and the hydraulic power consumption of the system are calculated. The test results are presented in Table 8. This study is done to determine the performance of the pump and verify the test set-up reliability. There are more than one data for each test points. However, the average values are tabulated in Table 8.

Table 8 Pump Performance Test Results

N [rpm]	t [s]	p_{exit} [bar]	p_{in} [mm Hg]	Q [l/min]	Δp [Pa]	P_{hyd} [W]
400	28,7	1	-100	63	1,13E+02	118
400	30,6	10	-170	59	1,02E+03	1004
400	31,4	20	-200	57	2,03E+03	1936
500	23,0	1	-90	78	1,12E+02	146
500	24,3	10	-50	74	1,01E+03	1243
500	25,0	20	-60	72	2,01E+03	2413
540	21,4	1	-90	84	1,12E+02	157
540	22,5	5	-50	80	5,07E+02	675
540	22,4	10	-100	80	1,01E+03	1357
540	23,2	15	-50	78	1,51E+03	1951
540	22,8	20	-100	79	2,01E+03	2649
540	23,4	25	-60	77	2,51E+03	3216
540	23,6	30	-60	76	2,96E+03	3771
540	23,2	40	-100	78	4,17E+06	5428
550	21,5	1	-100	84	1,13E+02	158
550	22,0	10	-100	82	1,01E+03	1382
550	23,0	20	-80	78	2,01E+03	2627

According to the data indicated above the pump performance properties are presented in the following paragraphs.

Volumetric flow rate change with pump speed for different exit pressures is presented in Figure 20. According to the figure, as the pump speed increases the volumetric flow rate also increases up to the normal working speed of the pump which is 540 rpm. Higher than this value the volumetric flow rate starts to decrease slightly which is caused by the fluid leakage from the pump.

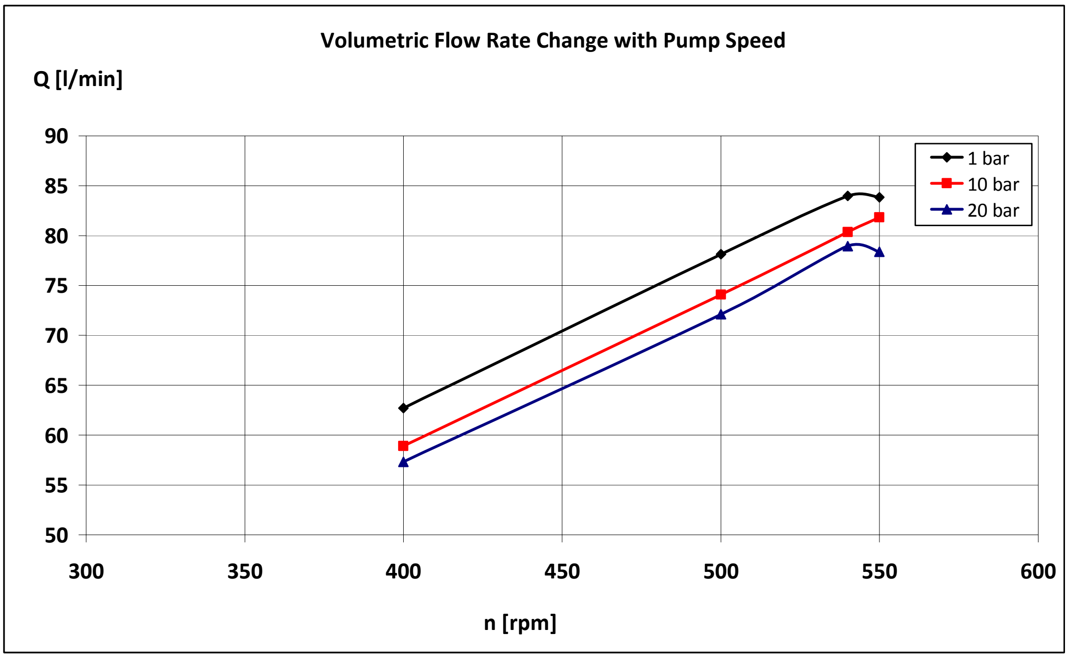

Figure 20 Volumetric Flow Rate vs. Pump Speed

Volumetric flow rate change with exit pressure for different pump speeds is presented in Figure 21. According to the figure, as the exit pressure increases the volumetric flow rate is supposed to keep constant where an ideal diaphragm pump does. However due to the increase in exit pressure there is some leakage in the pump for the real cases. As a result, the volumetric flow rate slightly decreases as the exit pressure increases.

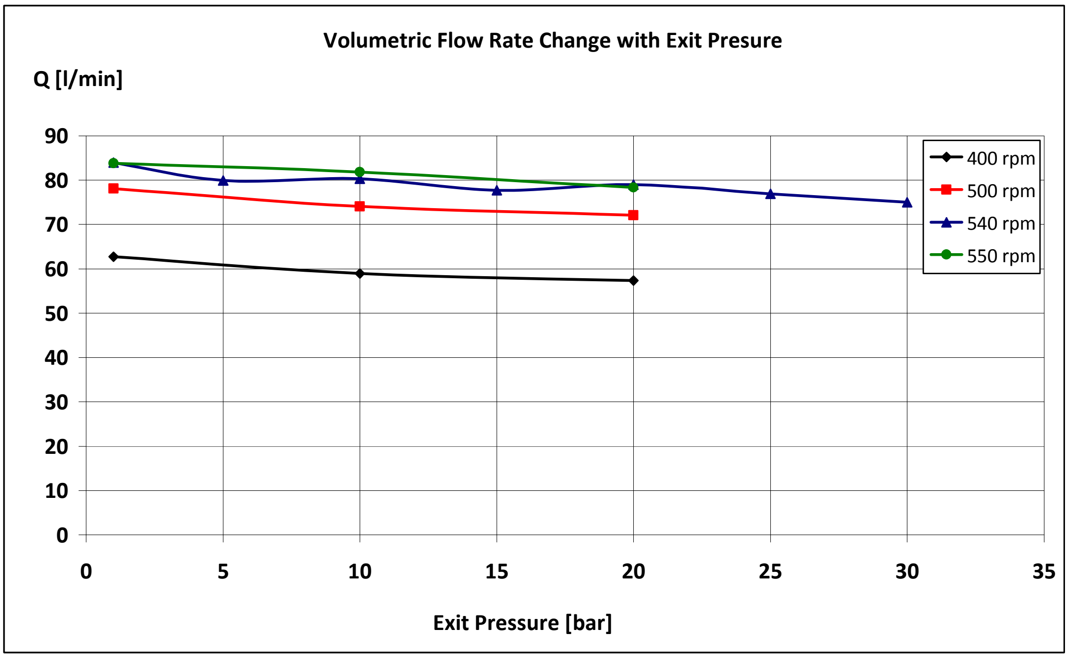

Figure 21 Volumetric Flow Rate vs. Exit pressure

The hydraulic power change with change in pump speed for different exit pressures is presented in Figure 22. According to the figure with the increase in pump speed the hydraulic power increases. At the same pump speed as the exit pressure increases the hydraulic power increases too. This result is expected since the hydraulic power relates the exit pressure. As presented in Figure 21 the volumetric flow rate is directly proportional to pump speed therefore hydraulic power is directly proportional to pump speed too.

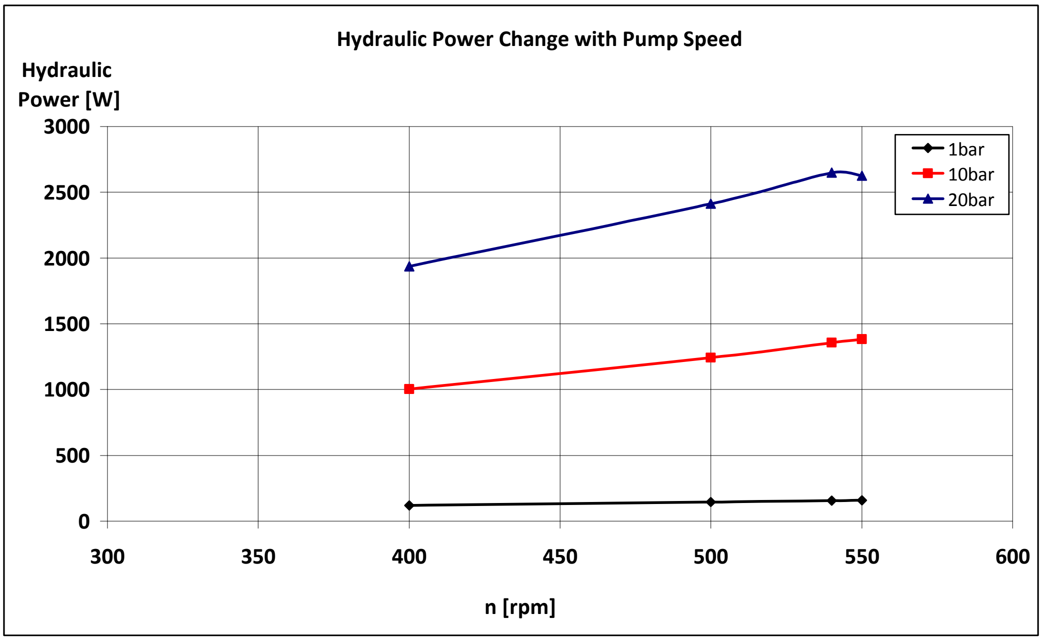

Figure 22 Hydraulic Power vs. Pump speed

The hydraulic power change with change in pump exit pressure for different pump speeds is presented in Figure 23. According to the figure, with the increase in exit pressure hydraulic power increases too.

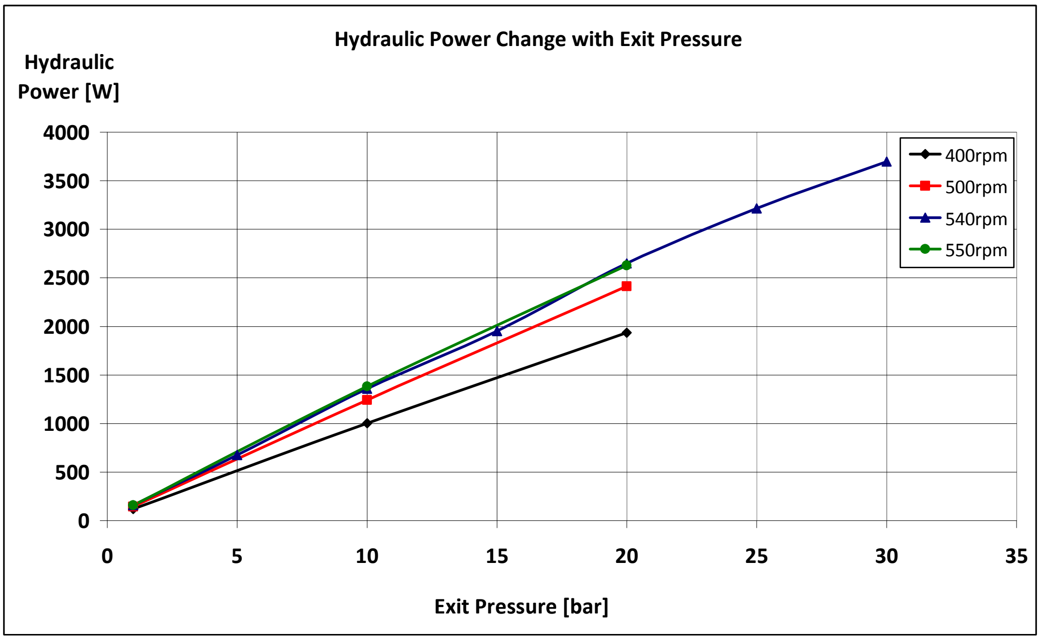

Figure 23 Hydraulic Power vs. Exit Pressure

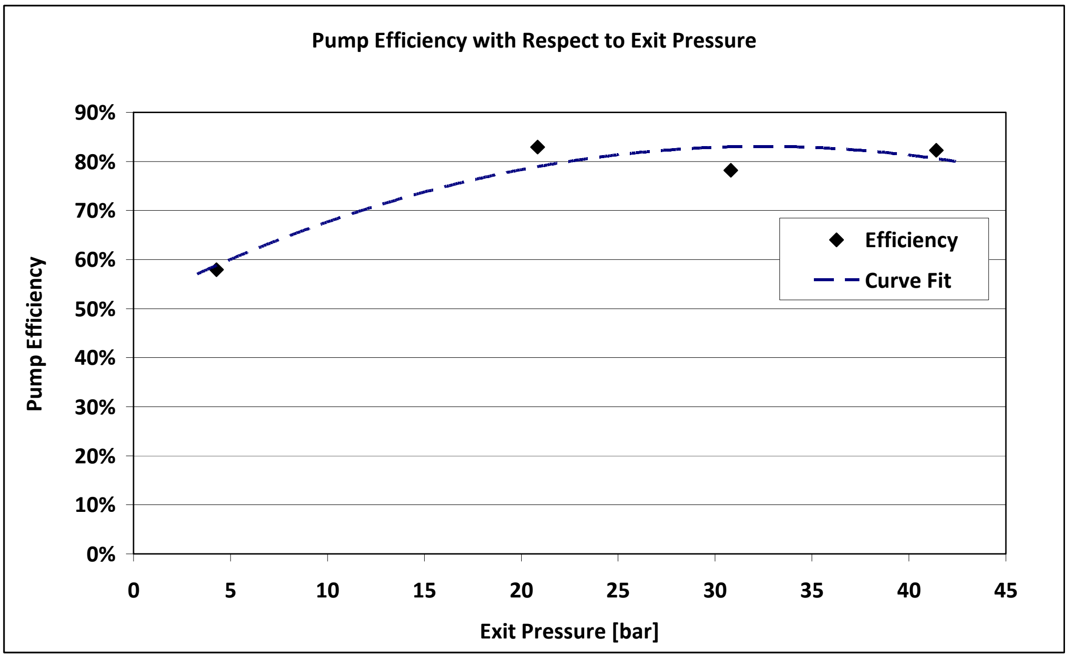

Figure 24 Pump Efficiency vs.Exit Pressure

The pump efficiency behaviors with the change in exit pressure and the volumetric flow rate are indicated in Figure 24 and Figure25. According to the efficiency figures the best efficiency is achieved at 40 bar exit pressure and as the volumetric flow rate increases the efficiency of the pump decreases. The results support each other since, as the exit pressure increases, the volumetric flow rate of the pump decreases due to the leakage of the pump.

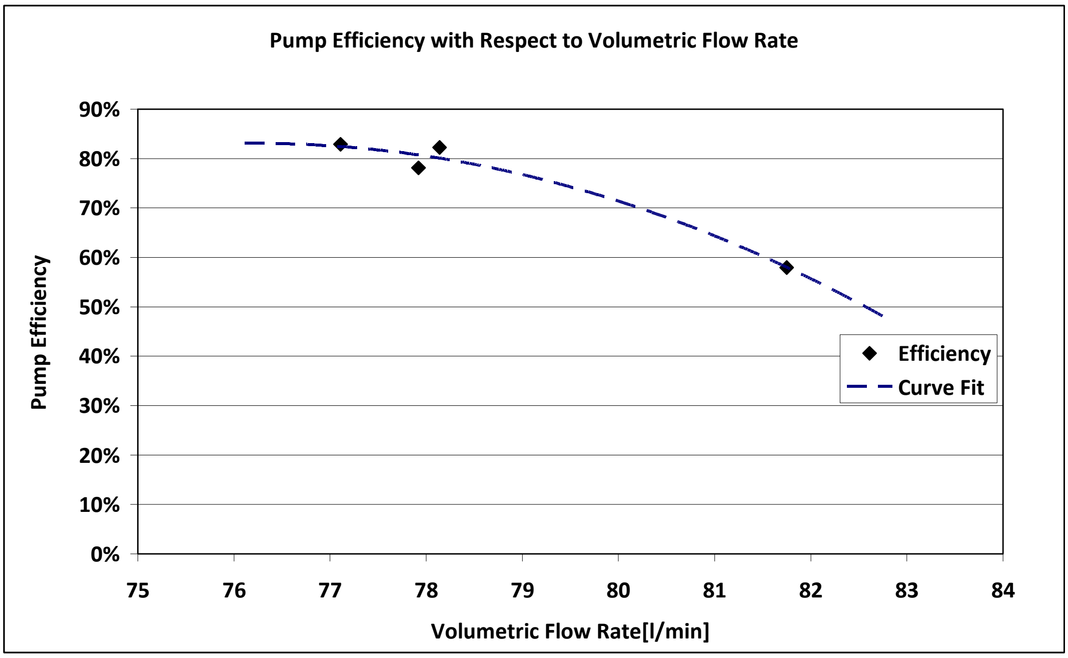

Figure 25 Pump Efficiency vs.Volumetric Flow Rate

The preliminary pump performance tests are finalized and the results are presented. According to the data and the figures presented the results are compatible with the nature of the diaphragm pumps. Therefore it can be concluded that the test set up is proper and the further experiments could be performed with the set-up.

4.2 SURGE TANK PEROFRMANCE TESTS

After pump performance tests, the experiments for the main focus of this thesis study surge tank performance tests are conducted. In the surge tank tests the effect of the surge tank size and the effect of inlet area of the surge tank on the exit pressure pulsations are investigated. The proper designs for the surge tank size and the surge tank inlet area configuration is evaluated. The exit pressure time history is recorded and the evaluations are done with the time history data.

The pressure sensors are used to measure the pressure of flow on the discharge side of the pump and the pulsations in the surge tank. With the data gathered from the sensors the relation of the exit pressure and the surge tank pressure is interrogated with linear correlation analysis and Pearson analysis functions of Microsoft Excel and also with the hand solution of correlation coefficient ρ_{xy}. In every test run the sample correlation function which is supposed to be in the interval [-1, 1] comes out to be 1 or -1 as expected as indicated in Figure 26. The figure is a sample figure of configuration 1 for 30 bars data.

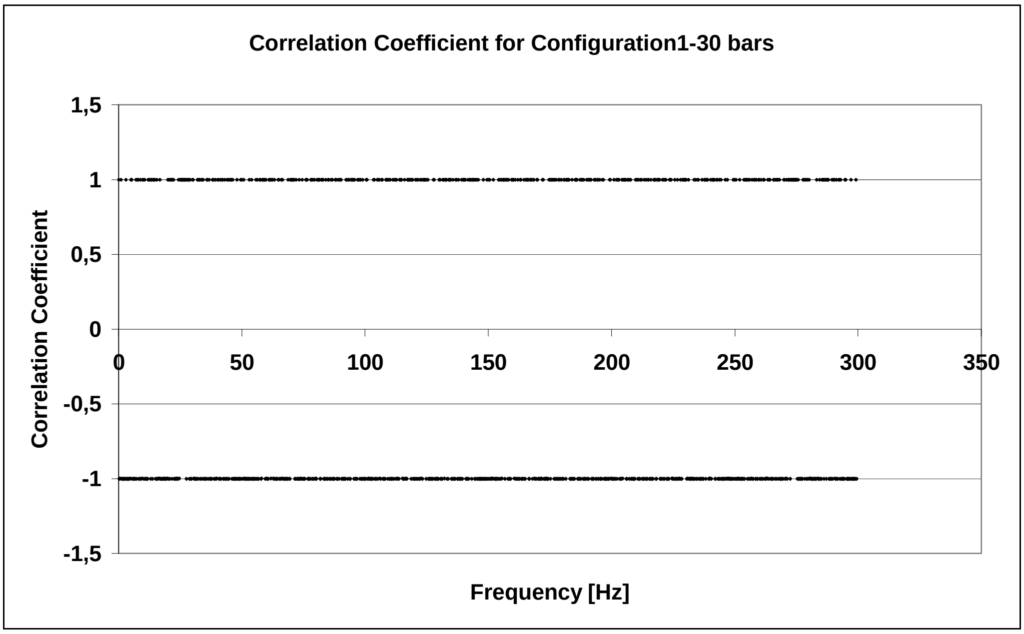

Figure 26 Correlation Coefficient

When the sample correlation function is ±1 the variables have a perfect linear dependence. A nonlinear relationship and data scatter due to measurement error or imperfect correlation of variables will force ρ_{xy} towards zero. For N number of data;

$$\rho_{xy} = \frac{\sum_{1}^{N}\left[(x_i - \bar{x})\cdot(y_i - \bar{y})\right]}{\left[\sum_{1}^{N}(x_i - \bar{x})^2 \cdot \sum_{1}^{N}(y_i - \bar{y})^2\right]^{1/2}} \qquad i = 1,\ldots N \qquad (4.1)$$

$$\bar{x} = \frac{\sum_{1}^{N} x_i}{N} \qquad i = 1,\ldots N \qquad (4.2)$$

$$\bar{y} = \frac{\sum_{1}^{N} y_i}{N} \qquad i = 1,\ldots N \qquad (4.3)$$

It can be concluded that the pulsations of the flow are directly related with the surge tank pulsations for the whole frequency spectrum of the flow.

In order to identify the peak frequency of the pulsations, the pressure time history of the data is converted to frequency domain. A frequency spectrum is calculated for a time range between 5 to 8.41 seconds which is 1024 data points as recommended in [30].This range was chosen randomly however a time lag is considered for the system to reach steady state after changing the exit pressure.

In Figure 27 the time history of a sample data is indicated for a 40 bars exit pressure. The frequency of this data is evaluated to identify the frequency of the maximum pulsation frequency in the system. As presented in Figure 28 the spectrum analysis of the data shows that the pulsation peaks are at 27 Hz and at 270Hz. This is expected for a 540 rpm 3 diaphragm pump flow since our system frequency is also 27 Hz as explained in Chapter 3.

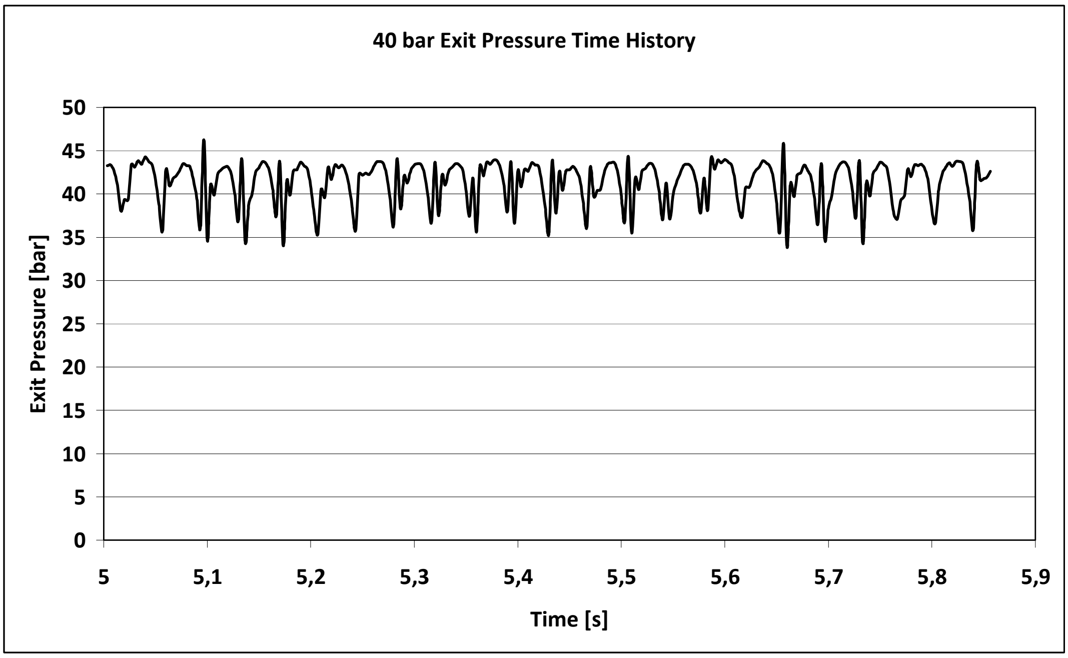

Figure 27 Pressure Time History at 40 bar Exit Pressure with adaptor having area ratio of 0.14

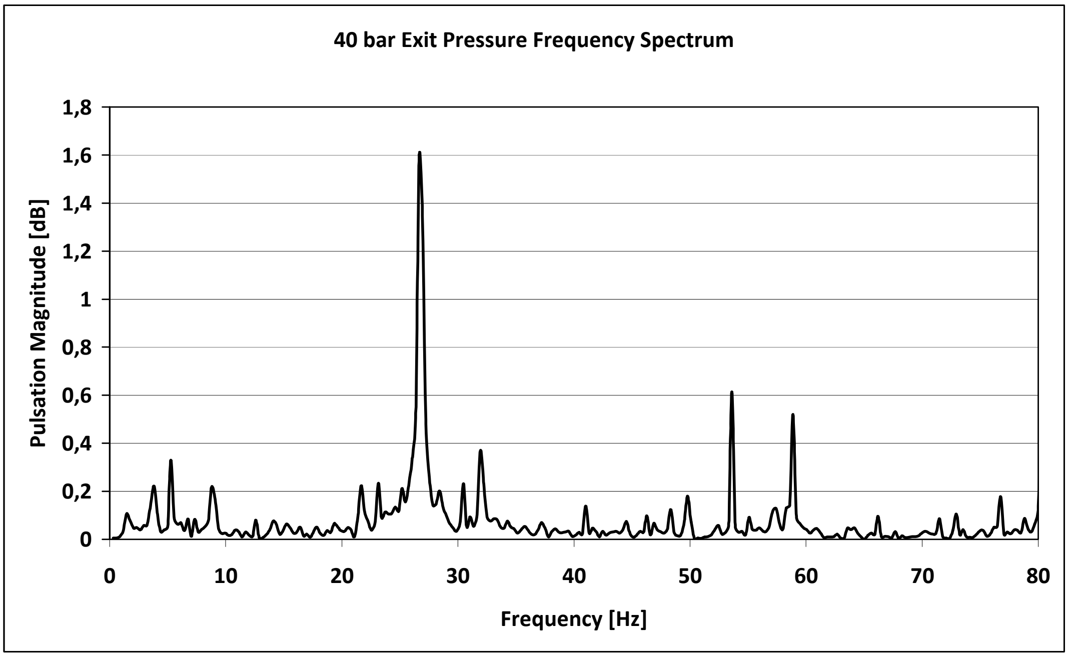

Figure 28 Pressure Frequency Spectrum at 40 bar Exit Pressure with adaptor having area ratio of 0.14

4.2.1 EFFECT OF THE SURGE TANK SIZE

Experiments are performed with surge tanks having different volumes. Six surge tank sizing design procedures are presented in section 2.1. In order to analyze the effect of surge tank size on the exit pulsations for the pump having three diaphragms and having 90l/min volumetric flow rate, and to find a proper surge tank size, a 80cm3, 200cm3 and 415cm3 surge tanks are tested. The original, manufacturer's surge tank's section view is presented in Figure 29. The results are compared with the data where the pump is run without the surge tank. The surge tank is disassembled from the system and the pump worked with no surge tank.

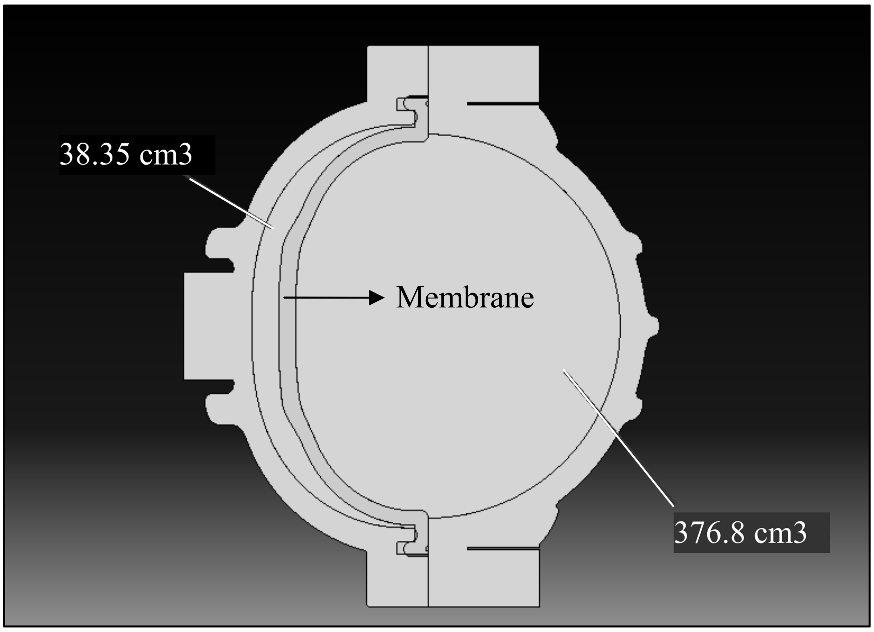

Figure 29 Section View of the Surge Tank of the Test Pump

All the surge tanks are tested with the adaptor part having the area ratio of 0,14. This is the proper area ratio creating the minimum pulsations. This result will be detailed in the succeeding sections. In Table 9 the results are tabulated.

Table 9 Experimental Data for Different Surge Tank Sizes

Surge Tank volume [cm^3]	N [rpm]	Q [l/min]	p_{exit} [bar]	p_{in} [mmHg]	ΔP[Pa]	P_{elec} [W]	P_{hyd} [W]	η_{system}	η_{Pump}	p_{exit} Max [bar]	p_{exit} Min [bar]	%pulsation
0	540	89,73	4,5	-180	4,71E+05	2100	704	34%	59%	12,9	0,1	285%
0	540	81,13	20,9	-180	2,11E+06	4740	2858	60%	83%	39,3	9,0	145%
0	540	80,61	31,2	-180	3,15E+06	6180	4226	68%	84%	51,8	15,3	117%
0	540	79,18	41,5	-180	4,17E+06	7930	5504	69%	85%	61,3	23,6	91%
80	540	86,67	4,4	-200	4,71E+05	1870	680	36%	64%	13,0	0,3	287%
80	540	79,7	20,6	-200	2,09E+06	4470	2772	62%	85%	24,7	14,5	50%
80	540	80,71	30,7	-200	3,10E+06	6220	4172	67%	83%	36,6	21,6	49%
80	540	80,03	41,2	-200	4,15E+06	8080	5537	69%	84%	50,3	28,7	52%
200	540	83,65	4,3	-200	4,59E+05	1880	640	34%	60%	8,1	1,0	166%
200	540	78,28	20,2	-200	2,05E+06	4410	2669	61%	83%	25,6	15,0	52%
200	540	79,51	30,8	-200	3,11E+06	6380	4121	65%	80%	35,4	23,0	40%
200	540	79,87	41,0	-200	4,13E+06	8040	5499	68%	84%	46,6	30,8	39%
415	540	82,06	4,4	-180	4,61E+05	1950	631	32%	57%	10,6	1,3	211%
415	540	77,3	21,0	-180	2,12E+06	4490	2731	61%	83%	26,2	16,6	46%
415	540	78,26	30,8	-180	3,10E+06	6170	4046	66%	81%	36,6	24,7	39%
415	540	77,37	41,3	-180	4,16E+06	8120	5362	66%	81%	46,3	33,4	31%

According to the experimental data above the percentage of pressure pulsation are calculated as indicated in equation (3.7)

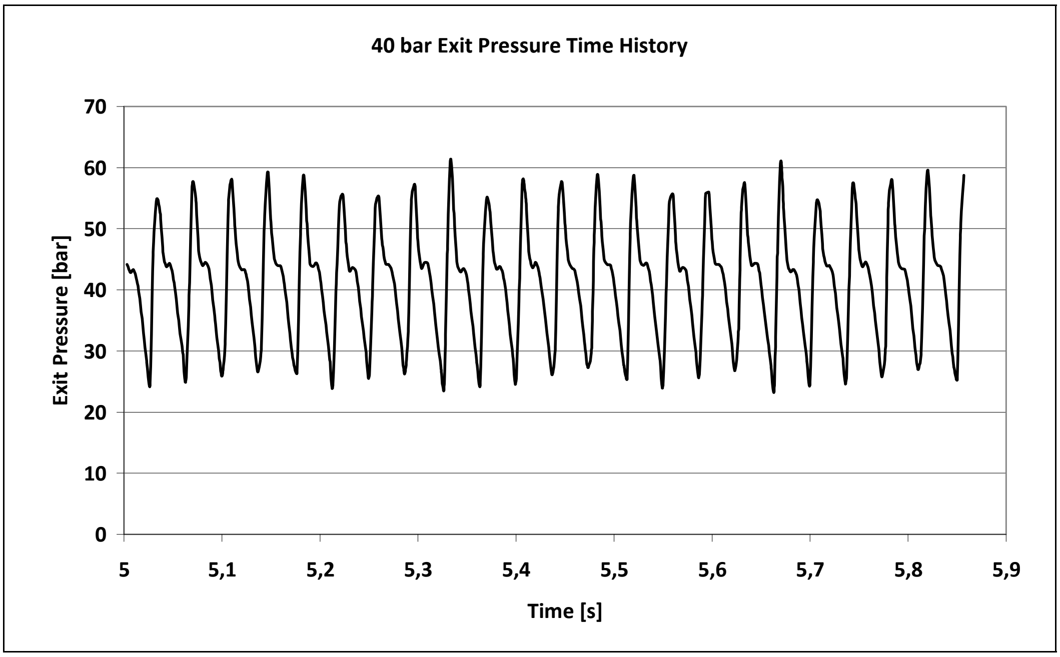

Figure 30 Pressure Time History at 40 bar Exit Pressure without surge tank on the pump

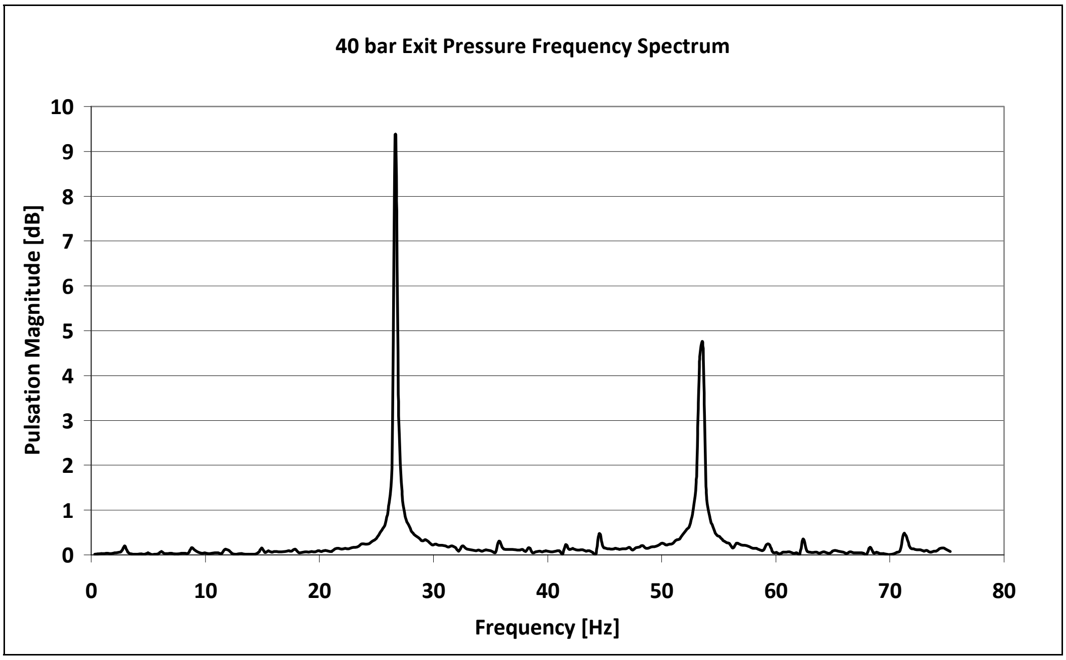

Figure 31 Pressure Frequency Spectrum at 40 bar Exit Pressure without surge tank on the pump

As indicated in Figure 30 and Figure 31 when there is no surge tank on the pump the pressure pulsations are quite different from the configurations with surge tank expressed in Figure 27 and Figure 28. The pressure pulsations are much higher when the surge tank is disconnected from the system. Form the figures it is obviously seen how effective surge tank is in pulsation reducing. The exit pressure pulsation changes with surge tank size are explained in following figures in detail.

In Figure 32 the exit pressure pulsation percentages are presented. As mentioned several times before, the main purpose of this study is to find a proper surge tank size and inlet area creating the minimum exit pressure pulsation. What can be concluded from the figure is; the minimum pulsations in the system are achieved in 40 bars where the best efficiency is achieved. Therefore to create the minimum pressure change in the flow this pump should be run at 40 bars. On the other hand, only scattered data could be achieved for 5 bars for all the surge tank sizes. The reason of the scattered data is unknown. However a curve is fitted to a polynomial and presented in Figure 32.

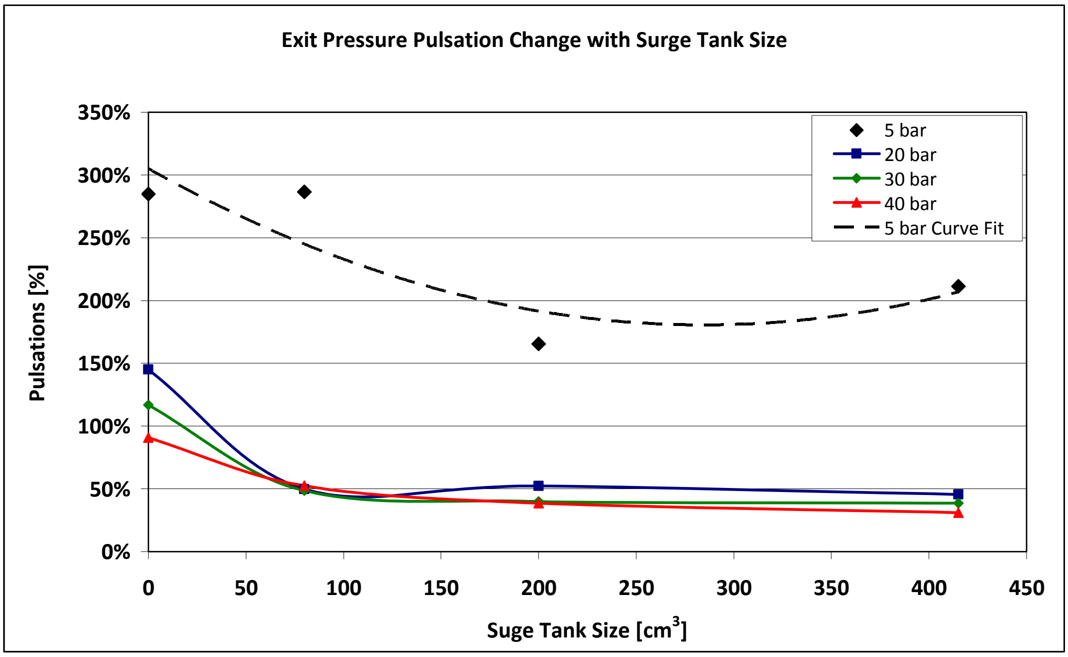

Figure 32 Exit Pressure Pulsation Change with Surge Tank Size

If Figure 32 is considered closely one should be focus on Figure 33. The figure indicates a detail view on pulsations at 40 bars. The minimum pulsation is achieved with the surge tank having the maximum volume. The pressure peak-to peak pulsation percentage is 31%. Therefore it can be concluded that to achieve the minimum pressure fluctuations in the system the pump should be run with the biggest surge tank having 415 cm^3. It is obvious that if there were tanks having higher volume than 415cm^3 the pulsations would be lower. This can be a further study to do the test with larger surge tanks. This outcome agrees the theory which Mead [3] introduced as with greater surge tank size the magnitude of pulsations can be diminished more. With increasing the surge tank size the pulsations can be decreased more.

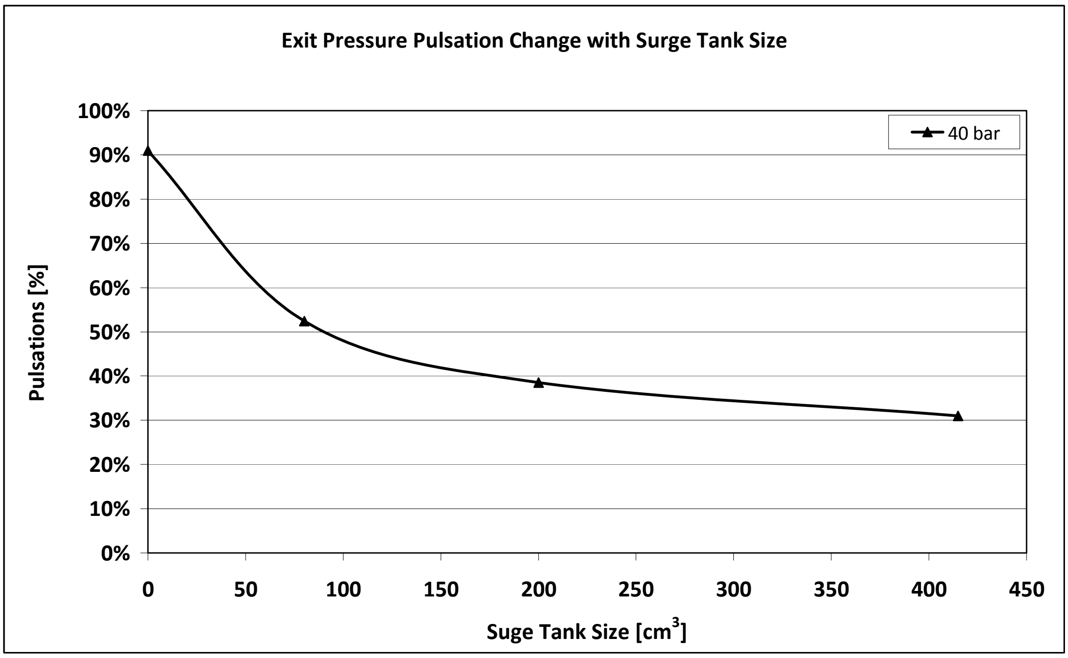

Figure 33 Exit Pressure Pulsation Change with Surge Tank Size at 40 bar

4.2.2 EFFECT OF THE INLET AREA OF SURGE TANK

In the previous section the effect of the surge tank size on the exit pressure pulsations is explained and the results are presented. To analyze the effect of the surge tank inlet area the surge tank giving the best results are used in the next experiments. The largest tank having 415 cm^3 is used in the following experiments having the minimum pulsations values. In this section the experimental results are presented at Table 10. Among the inlet area configurations a proper configuration creating the minimum pulsations is searched

The normal working pressure of the pump is 40 bars. Therefore the concentration to find the proper configuration is focused on the 40 bars exit pressure data.

The sizes and the areas of the adaptors noted in Chapter 3. Therefore, eight inlet area configurations of surge tank are tested. Among the eight, the one creating minimum pressure pulsation at 40 bars is concluded as the proper solution for this pump and surge tank size.

Table 10 Experimental Data for Different Surge Tank Inlet Areas

Adaptor Area Ratio	N [rpm]	p_{exit} [bar]	p_{in} [mmHg]	Q [l/min]	Δp[Pa]	P_{elec} [W]	P_{hyd} [W]	η_{system}	η_{Pump}	p_{exit} max	p_{exit} min	%pulsation
0	540	4,5	-180	89,73	4,71E+05	2100	704	34%	58%	12,9	0,1	285%
0	540	20,9	-180	81,13	2,11E+06	4740	2858	60%	82%	39,3	9,0	145%
0	540	31,2	-180	80,61	3,15E+06	6180	4226	68%	83%	51,8	15,3	117%
0	540	41,5	-180	79,18	4,17E+06	7930	5504	69%	85%	61,3	23,6	91%
0,086	540	4,4	-200	83,12	4,70E+05	1880	652	35%	60%	8,8	1,1	174%
0,086	540	21,1	-200	79,31	2,13E+06	4630	2818	61%	82%	25,8	13,0	61%
0,086	540	31,0	-200	79,89	3,12E+06	6130	4160	68%	83%	35,8	21,4	46%
0,086	540	41,3	-200	79,3	4,16E+06	8080	5497	68%	83%	45,7	30,7	36%
0,139	540	4,4	-180	82,06	4,61E+05	1950	631	32%	58%	10,6	1,3	211%
0,139	540	21,0	-180	77,3	2,12E+06	4490	2731	61%	83%	26,2	16,6	46%
0,139	540	30,8	-180	78,26	3,10E+06	6170	4046	66%	78%	36,6	24,7	39%
0,139	540	41,3	-180	77,37	4,16E+06	8120	5362	66%	82%	46,3	33,4	31%
0,148	540	4,4	-200	83,38	4,70E+05	1880	653	35%	56%	8,4	1,3	160%
0,148	540	20,8	-200	77,91	2,11E+06	4560	2738	60%	82%	26,8	16,2	51%
0,148	540	30,9	-200	79,01	3,12E+06	6250	4105	66%	80%	37,4	24,0	43%
0,148	540	40,6	-200	79,4	4,08E+06	7840	5404	69%	80%	46,0	32,2	34%
0,194	540	4,4	-180	82,71	4,60E+05	1910	634	33%	60%	9,1	2,0	162%
0,194	540	20,7	-180	77,96	2,09E+06	4500	2722	61%	81%	25,9	15,6	49%
0,194	540	31,1	-180	77,85	3,13E+06	6280	4065	65%	80%	36,0	23,6	40%
0,194	540	41,2	-180	77,96	4,14E+06	7800	5384	69%	84%	45,3	31,6	33%
0,247	540	4,3	-200	81,75	4,55E+05	1860	619	33%	58%	10,5	0,6	229%
0,247	540	20,8	-200	77,11	2,11E+06	4430	2714	61%	82%	28,1	14,9	63%

Table 10 Continued

Adaptor Area Ratio	N [rpm]	p_{exit} [bar]	p_{in} [mmHg]	Q [l/min]	Δp[Pa]	P_{elec} [W]	P_{hyd} [W]	η_{system}	η_{Pump}	p_{exit} max	p_{exit} min	%pulsation	N [rpm]
0,247	540	30,8	-200	77,92	3,11E+06	6290	4036	64%	79%	37,9	22,7	49%	
0,247	540	41,4	-200	78,14	0,0013	4,17E+06	8040	5428	68%	79%	47,0	31,9	37%
0,346	540	4,3	-200	84,1	0,0014	4,56E+05	1940	639	33%	84%	10,0	1,1	207%
0,346	540	19,4	-200	78,6	0,0013	1,96E+06	4400	2570	58%	79%	27,7	6,0	112%
0,346	540	30,7	-200	78,72	0,0013	3,09E+06	6340	4056	64%	78%	37,7	22,5	50%
0,346	540	40,7	-200	79,75	0,0013	4,10E+06	8000	5445	68%	83%	46,5	30,7	39%
1	540	4,4	-181	83,51	0,0014	4,62E+05	1950	644	33%	57%	9,7	1,8	180%
1	540	20,6	-180	78,38	0,0013	2,09E+06	4490	2726	61%	82%	26,4	14,9	56%
1	540	31,6	-180	79,44	0,0013	3,18E+06	6170	4216	68%	83%	37,8	23,0	47%
1	540	41,1	-180	78,79	0,0013	4,14E+06	8120	5433	67%	82%	45,8	30,8	37%

According to the result tabulated in Table 10 the following conclusions can be reached.

In Figure 34 the exit pressure pulsation percentages are presented. From this figure, the minimum pulsations in the system are achieved at 40 bars where the best efficiency is achieved. Also 40 bars is the normal working pressure of the pump itself. Therefore to create the minimum pressure change in the flow this pump should be run at 40 bars. On the other hand, only scattered data could be achieved for 5 bars for all the surge tank sizes. The reason of the scattered data is unknown. However a curve is fitted to a polynomial and presented in Figure 34.

Figure 34 shows that, if the surge tank is disconnected from the pump the pressure pulsations become extensively high. For smaller exit pressures the pressure pulsations are higher for the configuration having no surge tank. This result shows the necessity of the surge tank in this pump application. When a surge tank with inlet adaptor ratio of 0.09 is connected to the pump the pressure pulsations starts to decrease apparently. When the surge tank inlet area ratio is increased more than 0.14 there is not much change in pressure pulsations.

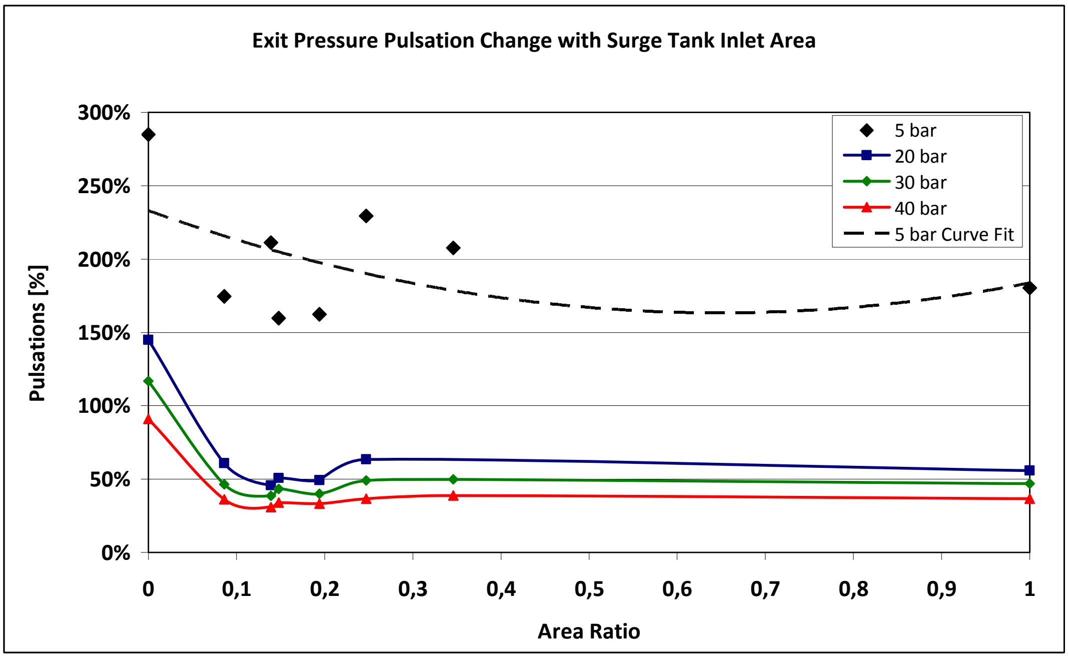

Figure 34 Exit Pressure Pulsations vs. Surge Tank Inlet Area

If Figure 34 is considered closely one should be focus on Figure 35. The figure indicates a detail view on pulsations at 40 bars. The minimum pulsation is achieved with the adaptor having area ratio of 0.14. The pressure peak-to peak pulsation percentage is 31%. Therefore it can be concluded that to achieve the minimum pressure fluctuations in the system the ratio of the surge tank inlet area to the maximum available inlet area should be 0.14.

The manufacturer is better to change the design of the inlet area of the surge tank to achieve smaller pulsations. Because the manufacturers original surge tank has an inlet area ratio of 0.25 creating 37% exit pressure pulsation. With this thesis study and the concluded results it is possible to decrease the exit pressure pulsations by 6%.

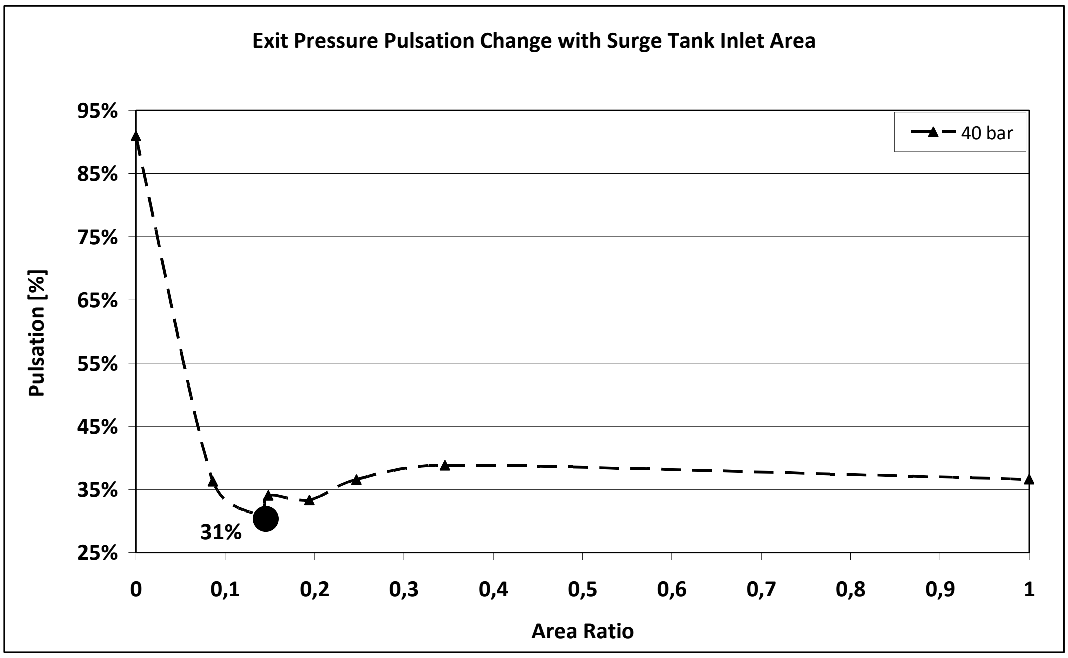

Figure 35 Exit Pressure Pulsations vs. Surge Tank Inlet Area at 40 bar

One may conclude that instead of creating an inlet area with a ratio to the allowable, it is acceptable to open the inlet area to the maximum value. This can be correct if only the pulsation effects are considered. Because if the surge tank inlet area is machined to the maximum allowable diameter the peak-to-peak pulsation is 37%. The effort of machining should be considered against the pulsation reduction. However as soon as the surge tank diaphragm edge is very close to the surge tank inlet if the connector combining the surge tank to the pump is machined thoroughly the sharp edges may deform and even tear the diaphragm. However, with the adaptor having the area ratio smaller than 1 has a supporting face for the diaphragm avoiding deforming or tearing. Therefore the proper solution both mechanically and for the minimum peak-to-peak pulsation is the adaptor having area ratio of 0.14.

4.3 UNCERTAINTY ANALYSIS

The uncertainty analysis of the experiment is performed to facilitate determine the errors in the system and confidence of the results. In order to perform the analysis 30 measurements are recorded for Surge Tank 1 adaptor no 3 and the average exit and surge tank pressures are evaluated for every test.

Table 11 presents the data and the maximum errors for the most common confidence level of 95% [35].

In order to find the maximum error in an experiment the significance level must be determined. For the confidence levels, the corresponding significance levels are as follows:

$$z = \begin{Bmatrix} 1{,}645 & \text{for} & 90\%\text{confidence} & \text{level} \\ 1{,}96 & \text{for} & 95\%\text{confidence} & \text{level} \\ 2{,}58 & \text{for} & 99\%\text{confidence} & \text{level} \end{Bmatrix} \tag{4.1}$$

$$E = z_{\alpha/2} \cdot \frac{\sigma_x}{\sqrt{n}} \tag{4.2}$$

Where,

E: Maximum error
z: Gaussian distribution value
α: significance level
σ_x: standard deviation of the data
n: number of data
$\bar{x}$: mean of the data

For the data in Table 11 the error of the measured exit pressure value for the 95% confidence level is:

$$E = 1.96 \cdot \frac{0.0267}{\sqrt{30}} = 0.010 \qquad (4.3)$$

Table 11 Uncertainty Analysis Data and Results

	P_{in} [mm Hg]	Q [l/min]	N [rpm]	V [Volt]	I [A]	P_{exit} [bars]	$P_{surge\ tank}$ [bars]
Mean	-140	81,26	542,6	275,92	12,	44,597	44,387
Standard Deviation	0	0,177	0,504	0,14	0,072	0,0267	0,0241
E	0	0,063	0,180	0,051	0,026	0,010	0,009

Since all the error values are small enough, it can be concluded that the tests are accurate and the experiment results are truthful.

CHAPTER 5

CONCLUDING REMARKS

In the previous chapters the introduction information on the diaphragm pumps, surge tanks and types are introduced, the corresponding literature survey and the aim of this thesis in presented. The aim of this thesis study is to evaluate the design and performance of a surge tank for a triplex diaphragm pump in order to achieve maximum reduction of exit pressure fluctuations, with the proper surge tank size and the surge tank cross sectional area. On behalf of this aim and experimental set up is constructed and related experiments are conducted, in order to conclude to a proper design of the surge tank. Corresponding results and discussion is also explained n detail.

In this chapter concluding remarks on the whole thesis study is presented.

The surge tank design study is performed with six different methods of different authors and three different surge tanks which are size representative of calculations are used on the pump and experiments are conducted. Among the surge tanks are having $80cm^3$, $200cm^3$ and $415cm^3$ the minimum peak to peak pulsations are achieved with the largest surge tank having $415cm^3$ volume which is the original surge tank of the test pump supplied by the pump manufacturer. The result; achieving the minimum peak to peak pulsations is supported by Mead [3] who introduced the concept; as with greater surge tank size the magnitude of pulsations can be diminished more. According to this if larger surge tank is used in the experiments peak to peak pressure pulsations of the pump will decrease. Conducting the experiments with a number of larger surge tanks can be a further study.

As far as the surge tanks having almost the same volume with the results of the design calculations of literature survey, the pulsation percentages are far from what is stated in the literature. The closest result is achieved with the method of Arnold and Stewart78[19] resulting a surge tank size of 521cm$^{3.}$ This is about 25% larger than what the manufacturer request. On the other hand Arnold and Stewart advise a precharge pressure of 60-70% of average fluid pressure. However in the original system the precharge pressure is 12% only. Conducting the experiments with a sure tank design capable of 60-70% precharge pressure can be a further study.

Although there are many computational studies on surge tank design in the literature there are a few experimental studies on pumps regarding the surge tank performance and design studies. Besides, there is no considerable study on effect of surge tank inlet area on pulsations which is considered as a significant parameter. The cause of this can be; since this pumping and surge tank issues are part of industrial subjects the companies or the institutes studying on this subject may not want to share their knowhow which is their financial source.

In order to evaluate the design and performance of the surge tank and experimental study is planned and conducted. An experimental set up is designed and constructed to perform the surge tank tests. The pump performance is also analyzed before surge tank design experiments. A baseline study is performed, the experimental set up verified before the surge tank tests. The convenient instrumentation is assembled to the set-up and the data acquisition devices are used. In this period couple of issues are encountered and solved with the help of different resources. The details are given in Chapter 3. With the help of set up construction and the solve the problems encountered, both the mechanical engineering department gained a diaphragm pump and surge tank experimental set-up for further studies and the master of science student preparing this report had a considerable mechanical and electrical and electronically experience.

As indicated in chapter 3 and 4 before the surge tank pressure transducer was out of order there was data gathered from surge tank and discharge of the pump. These data is used to evaluate the dependence of the surge tank and the exit pressure pulsations. The exit pressure and the surge tank pressure is interrogated with linear correlation analysis and Pearson analysis functions. The results shoed that the correlation coefficient was always ±1 showing the variables have a perfect linear dependence as expected. Because the surge tank and the ext pressure transducers are connected to the discharge side but only to different locations.

In order to analyze the set-up uncertainty an uncertainty analysis is performed and presented in Chapter 4. 95% confidence level is considered and the error is maximum 0.027% which is anyway very little and can be neglected.

Finally, the aim of this thesis study is achieved that a proper surge tank inlet area can be concluded. There are eight different configurations with the adaptors having different areas are used in the experiments. These adaptors are connected to the surge tank inlet to crate different resistances before the flow. With the different adaptors connected to the surge tank inlet the experimentations are conducted and the data are recorded. According to the results among the eight area configurations the minimum peak to peak pulsations are achieved with the adaptor having the area ratio of 0.14. The corresponding peak to peak pulsation is 31%. The original inlet area gave a pulsation value of 37%. It can be concluded that the manufacturer is better to change the design of the inlet area of the surge tank to achieve smaller pulsations.

As indicated in Table 10 with the adaptor having no special configuration, no resistance against the flow and machined thoroughly had the same peak to peak pulsation value with the manufacturer's original design. This configuration needs no mechanical effort of creating an area with precise drilling operations. One may conclude that instead of creating an inlet area with machining; it is acceptable to sacrifice the 6% peak to peak pressure pulsation difference and use the adaptor machined thoroughly having the area ratio of 1.This can be logical if the pulsation effects are considered independently

However the surge tank diaphragm's bottom end is very close to the surge tank inlet. Therefore the adaptor having the area ratio of 1 having the sharp edges may deform and even tear the diaphragm. On the other hand any adaptor having the area ratio smaller than 1 has a supporting face for the diaphragm avoiding deforming or tearing. Therefore the proper solution for this pump and surge tank configuration is the adaptor having area ratio of 0.14 both mechanically and regarding the pressure pulsation reduction purpose.

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APPENDIX A

SURGE TANK SIZING CALCULATIONS

In this appendix section the surge tank sizing calculation approach of Özgür which is only held for duplex pumps [19] is extended for triplex simplex and quadruplex pumps. The calculation is explained in this section in detail.

FOR SIMPLEX PUMP

For a simplex (single piston) pump, the mean flow rate is

$$q_{mean} = \frac{r \cdot \omega \cdot A \cdot}{\pi} = 0.318 \cdot \omega \cdot r \cdot A \quad \text{(A.1)}$$

In

Figure 36 the flow rate graph of a simplex pump is given.

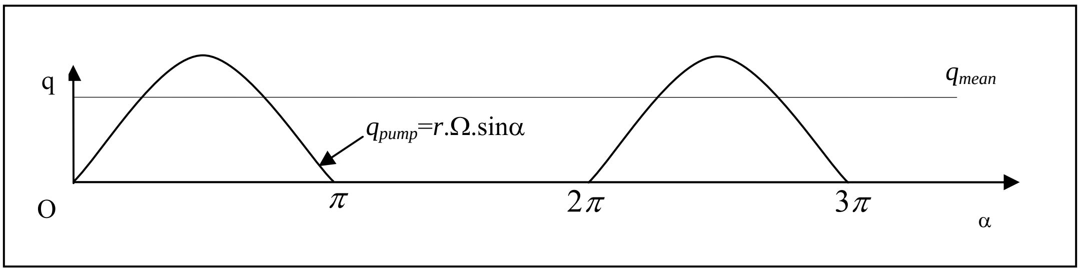

Figure 36 Flow Rate Diagram of Simplex Pump

As indicated in Figure 36 for single piston or diaphragm pumps the half period is for discharge and for rest half the piston/ diaphragm is inactive.

Therefore the amount of flow entering the surge tank at any instant is:

$$q_{surge} = q_{pump} - q_{mean} \tag{A.2}$$

For the beginning, the air volume in the surge tank is H_o. From point "O" to α_1 at time t_1 $q_p \leq q_{mean}$, therefore the flow is leaving the surge tank. If the amount of water leaving the surge tank is ΔH_1

$$\Delta H_1 = -\int_0^{t_1} q_{surge} \cdot dt = \int_0^{t_1} q_{mean} - q_{pump} \cdot dt$$

(A.27)

$$\Delta H_1 = \int_0^{t_1} \left(\frac{\omega \cdot r \cdot A}{\pi} - \omega \cdot r \cdot A \sin\alpha \right) \cdot dt = r \cdot A \int_0^{t_1} \left(\frac{1}{\pi} - \sin\alpha \right) \cdot d\omega t = r \cdot A \int_0^{\alpha_1} \left(\frac{1}{\pi} - \sin\alpha \right) \cdot d\alpha \tag{A.3}$$

$$\Delta H_1 = r \cdot A \left(\frac{\alpha_1}{\pi} + \cos\alpha_1 - 1 \right) \tag{A.4}$$

If the mean volumetric flow rate and the pump flow rate are equal to each other at the points α_1 and α_2

$$r \cdot \omega \cdot A \cdot \sin\alpha = \frac{1}{\pi} \omega \cdot r \cdot A \tag{A.5}$$

$\alpha_1 = 0.324$ and $\alpha_2 = 2.818$ in radians.

$$\Delta H_1 = r \cdot A \cdot (0.103 + 0.948 - 1) = 0.0509 \cdot r \cdot A \tag{A.6}$$

From point α_1 to α_2 (from time t_1 to t_2) the amount of water entering the surge tank is ΔH_1

$$\Delta H_2 = r \cdot A \int_{\alpha_1}^{\alpha_2} \left(\sin\alpha - \frac{1}{\pi} \right) \cdot d\alpha = -r \cdot A \left[\cos\alpha + \frac{1}{\pi} \right]_{\alpha_1}^{\alpha_2} \tag{A.7}$$

$$\Delta H_2 = -r \cdot A \left[\left(\cos\alpha_2 + \frac{\alpha_2}{\pi} \right) - \left(\cos\alpha_1 + \frac{\alpha_1}{\pi} \right) \right] \tag{A.8}$$

$$\Delta H_2 = -r \cdot A \left[\cos\alpha_2 - \cos\alpha_1 + \frac{1}{\pi}(\alpha_2 - \alpha_1) \right] = -r \cdot A[-1.896 + 0.794] \tag{A.9}$$

$$\Delta H_2 = 1.102 \cdot r \cdot A \tag{A.10}$$

where,

$$s\max(\alpha = \pi) = r + \sqrt{l^2} \tag{A.11}$$

$$s\min(\alpha = 0) = -r + \sqrt{l^2} \tag{A.12}$$

$$s = \Delta s = 2r \tag{A.13}$$

Therefore, $\Delta H_2 = 0.551 \cdot s \cdot A$ (A.14)

$$\frac{p_2 - p_1}{\frac{p_2 + p_1}{2}} = \frac{1}{100} \tag{A.15}$$

If the air in the surge tank is assumed as isothermal:

$p_1 \cdot H_1 = p_2 \cdot H_2 = C$, where C is a constant number

If these are substituted in the previous equation:

$$\frac{H_2 - H_1}{\frac{H_2 + H_1}{2}} = \frac{H_1 - H_2}{V_{surge}} = \frac{1}{100} \tag{A.16}$$

$$\Delta H_2 = H_1 - H_2 \tag{A.17}$$

where $Q = \frac{n}{60} 2 \cdot r \cdot A$ (A.18)

and, $A = \frac{Q \cdot 60}{s \cdot n}$ (A.19)

where $s = 2r$ as discussed before.

Hence $V_{surge} = (H_1 - H_2) \cdot 100$ (A.20)

If the equation is substituted:

$$V_{surge} = 0.551 \cdot s \cdot \frac{Q \cdot 60}{s \cdot n} \cdot 100 = 3306 \cdot \frac{Q}{n} = 9167 cm^3 \tag{A.21}$$

which is a quite large surge tank for a pump having this much flow rate. The volume of the surge tank is extensively larger than the one for a triplex pump.

FOR DUPLEX PUMP

For a duplex pump, the mean flow rate is

$$q_{mean} = \frac{2 \cdot r \cdot \omega \cdot A \cdot}{\pi} = 0.637 \cdot \omega \cdot r \cdot A \tag{A.22}$$

In

Figure 37 the flow rate graph of a duplex (two piston) pump is given.

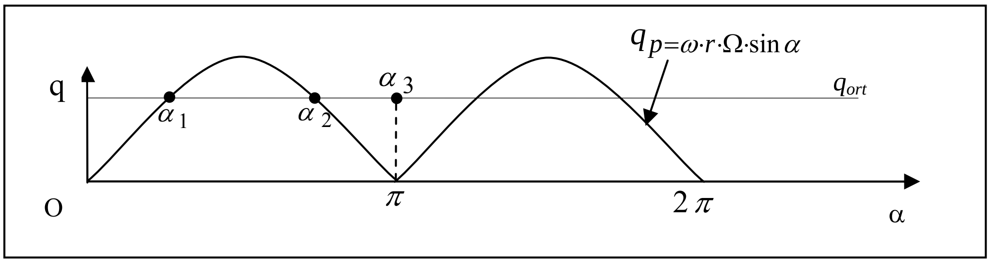

Figure 37 Flow Rate Diagram of Duplex Pump

As indicated in

Figure 37 the irregularity decreases and the flow rate fluctuation period decreases to π.

Similarly for duplex pumps

$$q_{surge} = q_{pump} - q_{mean} \tag{A.23}$$

$$\Delta H_1 = -\int_0^{t_1} q_{surge} \cdot dt = \int_0^{t_1} q_{mean} - q_{pump} \cdot dt \tag{A.24}$$

$$\Delta H_1 = \int_0^{t_1} \left(\frac{2 \cdot \omega \cdot r \cdot A}{\pi} - \omega \cdot r \cdot A \sin\alpha \right) \cdot dt = r \cdot A \int_0^{t_1} \left(\frac{2}{\pi} - \sin\alpha \right) \cdot d\omega t = r \cdot A \int_0^{\alpha_1} \left(\frac{2}{\pi} - \sin\alpha \right) \cdot d\alpha \tag{A.25}$$

$$\Delta H_1 = r \cdot A\left(\frac{2\alpha_1}{\pi} + \cos\alpha_1 - 1\right) \tag{A.26}$$

If the mean volumetric flow rate and the pump flow rate are equal to each other at the points α_1 and α_2

$$r \cdot \omega \cdot A \cdot \sin\alpha = \frac{2}{\pi}\omega \cdot r \cdot A \tag{A.27}$$

$\alpha_1 = 0.69$ and $\alpha_2 = 2.45$ in radians.

$$\Delta H_1 = r \cdot A \cdot (0.439 + 0.771 - 1) = 0.210 \cdot r \cdot A \tag{A.28}$$

From point α_1 to α_2 (from time t_1 to t_2) the amount of water entering the surge tank is ΔH_1

$$\Delta H_2 = r \cdot A \int_{\alpha_1}^{\alpha_2}\left(\sin\alpha - \frac{2}{\pi}\right) \cdot d\alpha = -r \cdot A\left[\cos\alpha + \frac{2}{\pi}\right]_{\alpha_1}^{\alpha_2} \tag{A.29}$$

$$\Delta H_2 = -r \cdot A\left[\left(\cos\alpha_2 + \frac{2\alpha_2}{\pi}\right) - \left(\cos\alpha_1 + \frac{2\alpha_1}{\pi}\right)\right] \tag{A.30}$$

$$\Delta H_2 = -r \cdot A\left[\cos\alpha_2 - \cos\alpha_1 + \frac{2}{\pi}(\alpha_2 - \alpha_1)\right] = -r \cdot A[-1.541 + 1.120] \tag{A.31}$$

$$\Delta H_2 = 0.42 \cdot r \cdot A \tag{A.32}$$

$$s\max(\alpha = \pi) = r + \sqrt{l^2} \tag{A.33}$$

$$s\min(\alpha = 0) = -r + \sqrt{l^2} \tag{A.34}$$

$$s = \Delta s = 2r \tag{A.35}$$

Therefore, $$\frac{p_2 - p_1}{\frac{p_2 + p_1}{2}} = \frac{1}{100} \tag{A.36}$$

If the air in the surge tank is assumed as isothermal:

$p_1 \cdot H_1 = p_2 \cdot H_2 = C$, where C is a constant number

If these are substituted in the previous equation:

$$\frac{H_2 - H_1}{\frac{H_2 + H_1}{2}} = \frac{H_1 - H_2}{V_{surge}} = \frac{1}{100} \tag{A.37}$$

$$\Delta H_2 = H_1 - H_2 \tag{A.38}$$

where, $$A = \frac{Q \cdot 60}{s \cdot n} \text{ and} \tag{A.39}$$

$$s = 2r \tag{A.40}$$

Hence, $$V_{surge} = \left(H_1 - H_2\right) \cdot 100 \tag{A.41}$$

$$V_{surge} = 0.21 \cdot s \cdot \frac{Q \cdot 60}{s \cdot n} \cdot 100 = 1261 \cdot \frac{Q}{n} = 3500\text{cm}^3 \tag{A.42}$$

which is a quite large surge tank for a pump having this much flow rate. The volume of the surge tank is extensively larger than the one for a triplex pump.

FOR QUAD PUMP

For a quad pump, the mean flow rate is

$$q_{mean} = \frac{4 \cdot r \cdot \omega \cdot A \cdot}{\pi} = 1.273 \cdot \omega \cdot r \cdot A \qquad \text{(A.43)}$$

In

Figure 38 the flow rate graph of a duplex (two piston) pump is given.

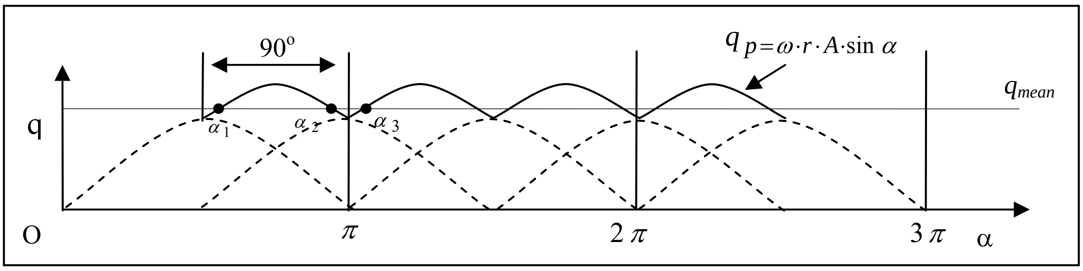

Figure 38 Flow Rate Diagram of Quad Pump

As indicated in

Figure 38 the irregularity decreases and the flow rate fluctuation period is to π/2.

Similarly for quad pump

$$V_{surge} = 0.04 \cdot s \cdot \frac{Q \cdot 60}{s \cdot n} \cdot 100 = 240 \cdot \frac{Q}{n} = 667 cm^3 \qquad \text{(A.44)}$$

Even a quadruplex pump is used the resulting surge tank size s very larger than the one for a triplex pump.

APPENDIX B

TECHNICAL SPECIFICATIONS OF INSTRUMENTS

Table 12 Test Pump Specifications

TEST PUMP		
Piston Diameter	[mm]	63
Eccentricity	[mm]	10
Discharge	[l/min]	90
Pressure	[kg/cm2]	50
Speed	[rpm]	540
Power	[hp/kW]	12 / 9
Number of Pistons		3
Membrane Dia.	[mm]	120
Useful Membrane Dia.	[mm]	94
Efficiency	[%]	76
Oil Capacity	[l]	1,5
Size	[cm]	35 x 42 x 35
Mass	[kg]	35

Table 13 Pressure Transmitter Specifications

Brand	GEMS
Type	2200B G
Pressure Interval	0-60 bars
Current	4-20 mA
Pressure Limit	2 times nominal pressure
Feed	24V DC Rated, 12-36V DC operable
Operating Temperature	-20°C...+80°C compensated
	-40°C...+100°C operable
Connection	G ¼ to BS2779 compatible with ISO228
Shield	IP 65
Response time to pressure	0.5 ms

Table 14 LabJack U12 Properties [27]

Command/Response	50 scans/second (up to 4 AI)
Command/Response	25 scans/second (up to 8 AI)
Stream	1200 samples/second (up to 4 AI)
Burst	8192 samples/second (up to 4 AI)

Table 15 Electric Motor Specifications [36]

Voltage	380 V
Frequency	50 Hz
Current	38 A
Power	18.5 kW
Power Factor	0.87
Speed	1460 rpm

Table 16 Flow Meter Features [28]

Digital display of rate and total flow
10 point linearization
4-20 mA loop-powered
0-5 or 0-10 V DC outputs

APPENDIX C

TECHNICAL DRAWINGS

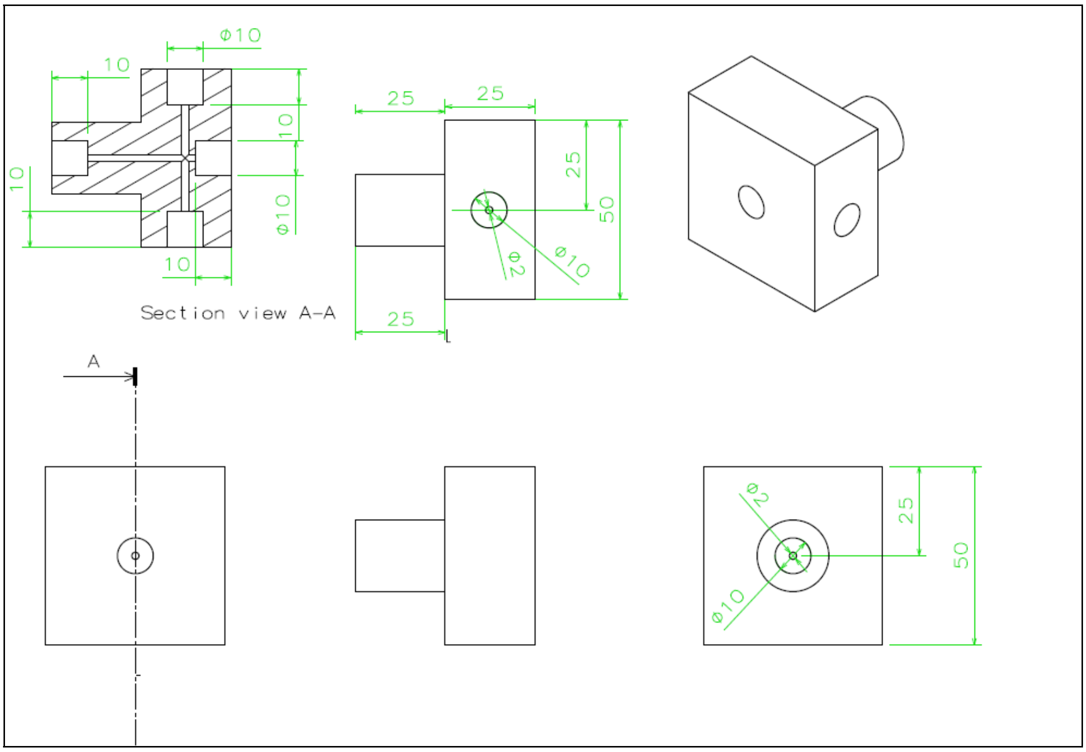

Figure 39 Technical Drawing of Pressure Transducer Adaptor

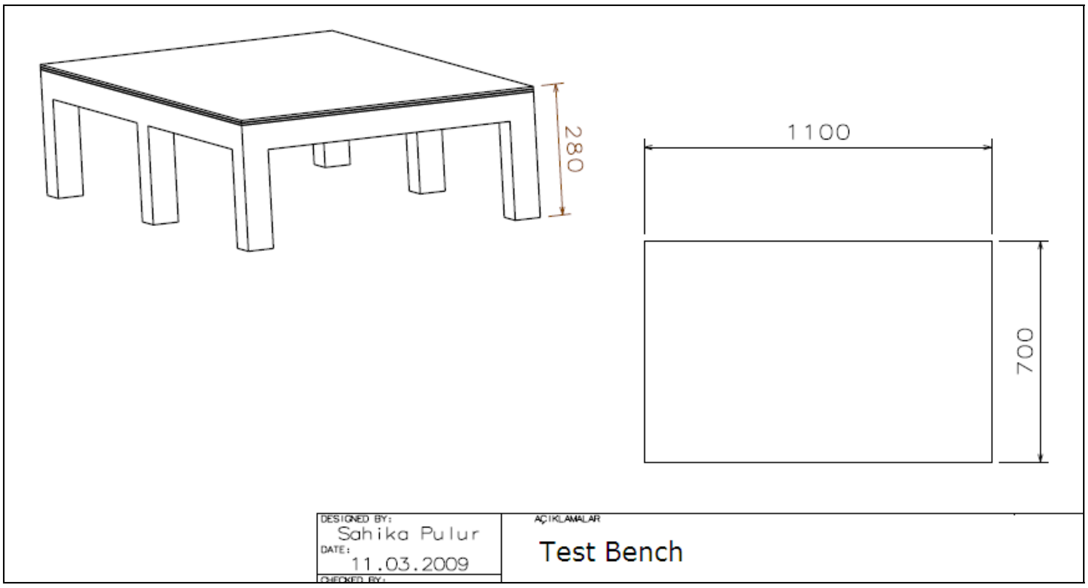

Figure 40 Technical Drawing of Test Bench

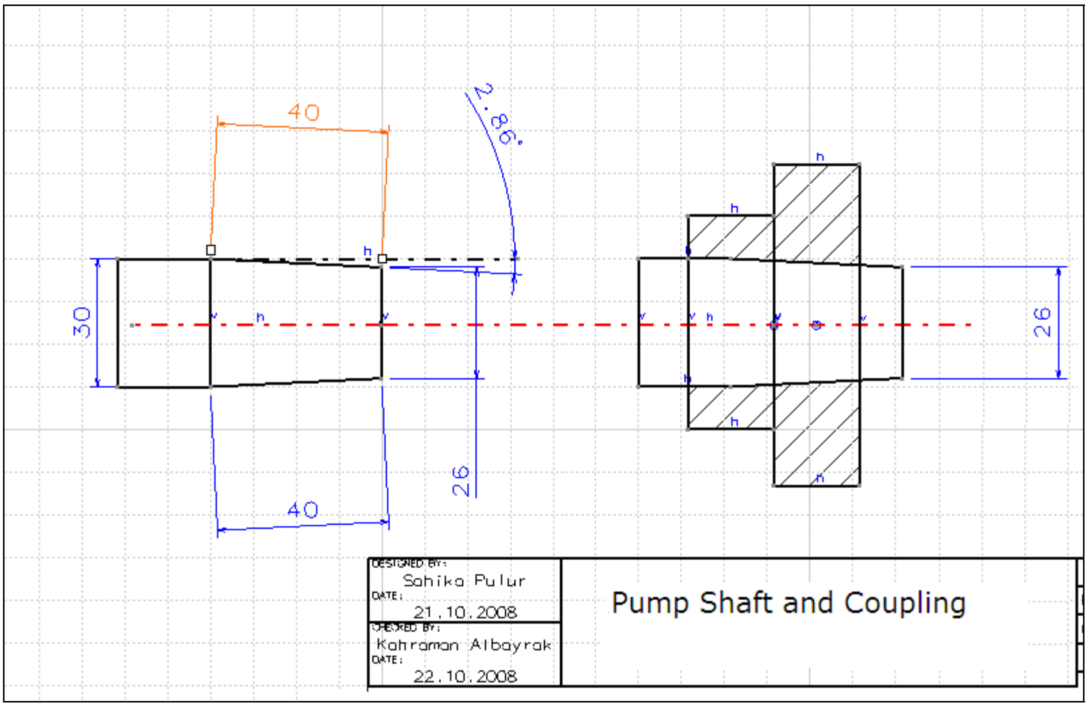

Figure 41 Technical Drawing of Coupling Pump Shaft

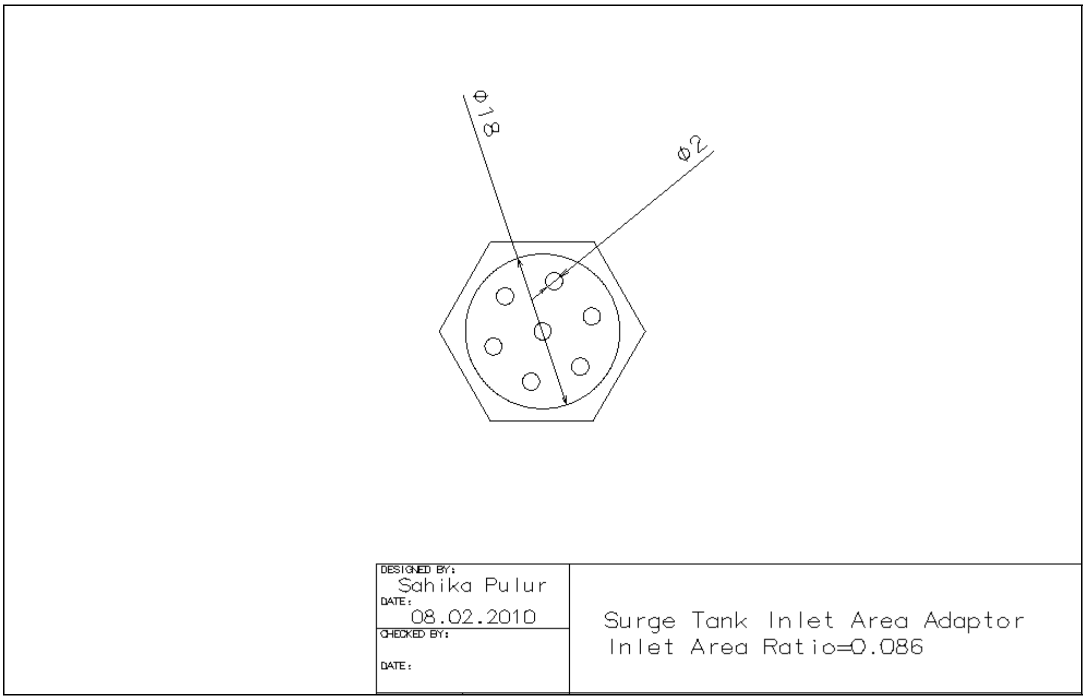

Figure 42 Technical Drawing of Adaptor with r=0.09

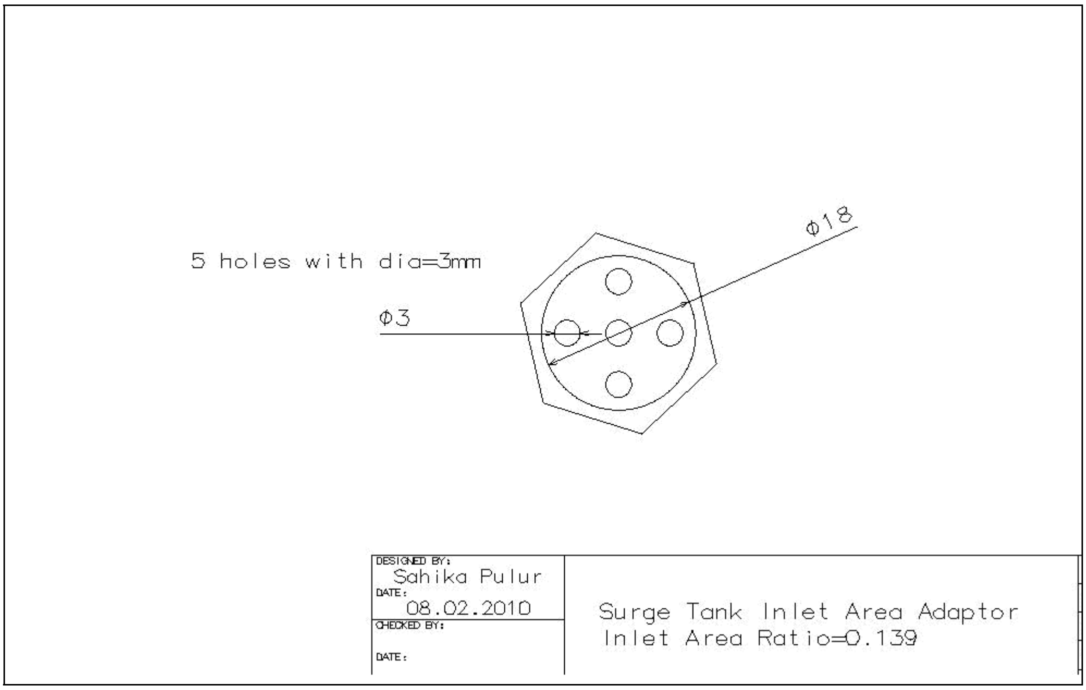

Figure 43 Technical Drawing of Adaptor with r=0.14

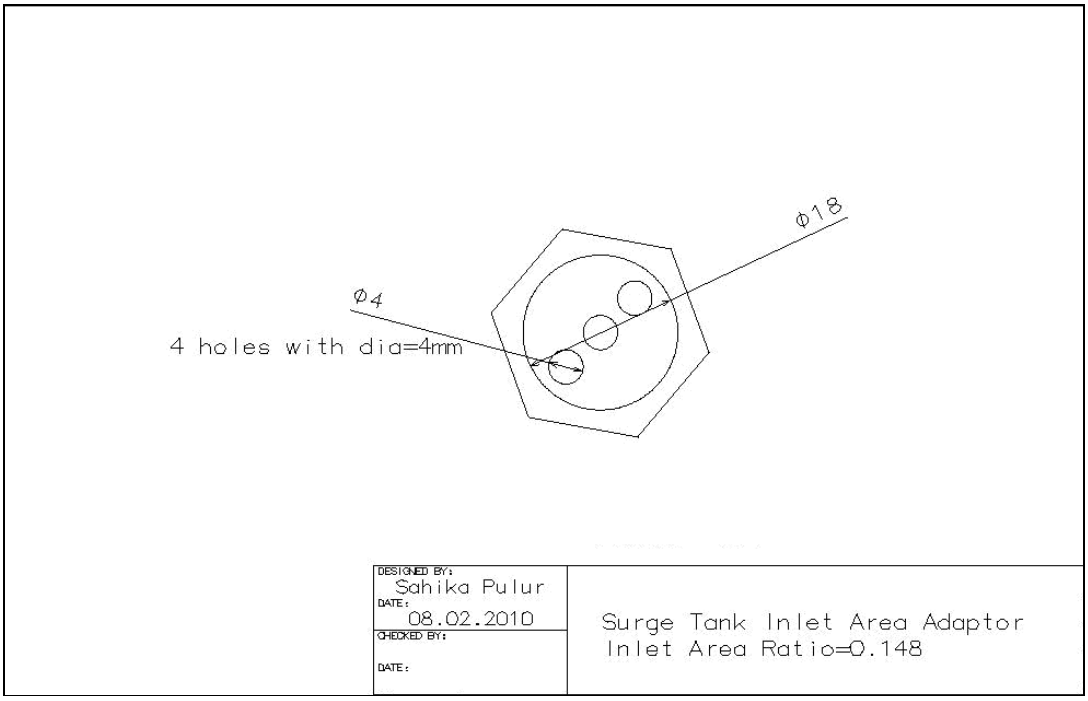

Figure 44 Technical Drawing of Adaptor with r=0.15

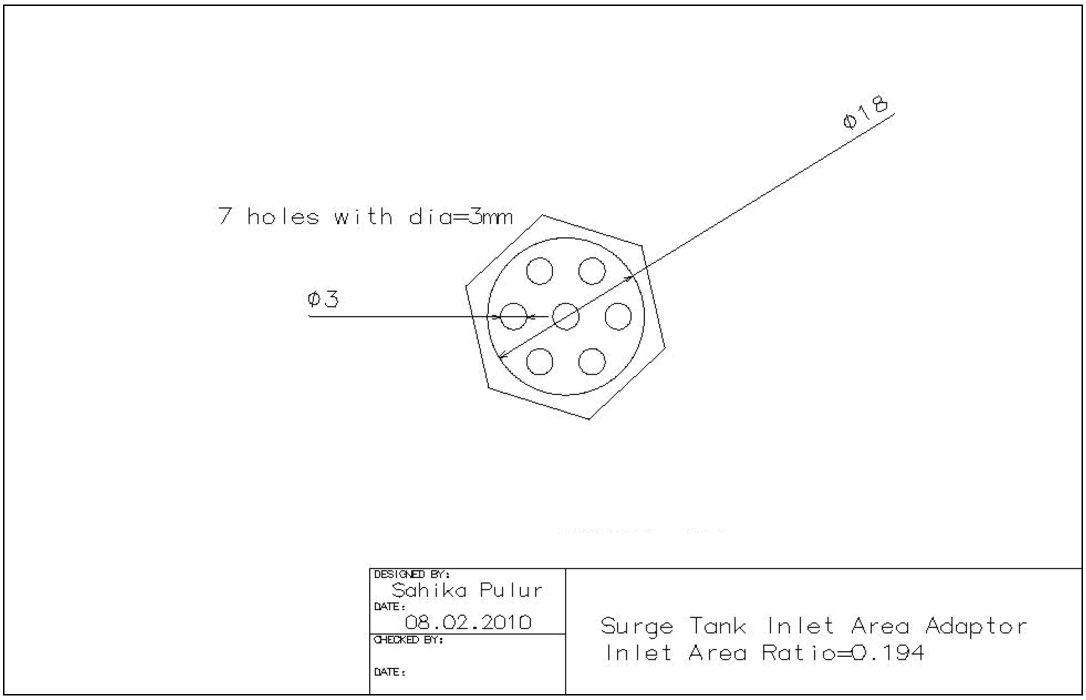

Figure 45 Technical Drawing of Adaptor with r=0.19

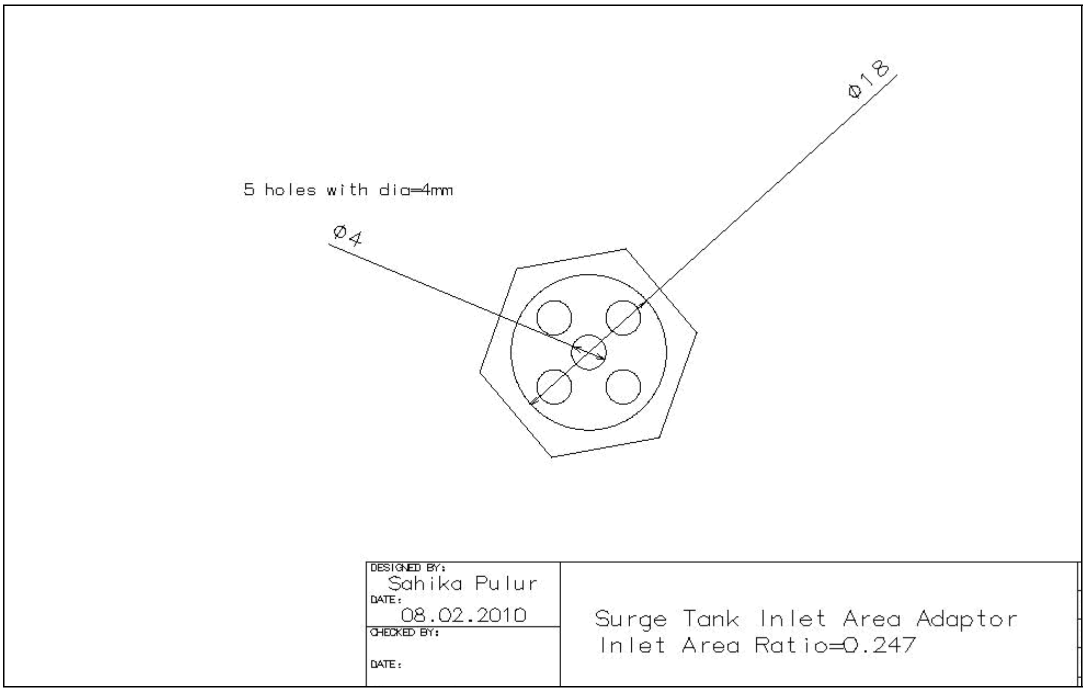

Figure 46 Technical Drawing of Adaptor with r=0.25

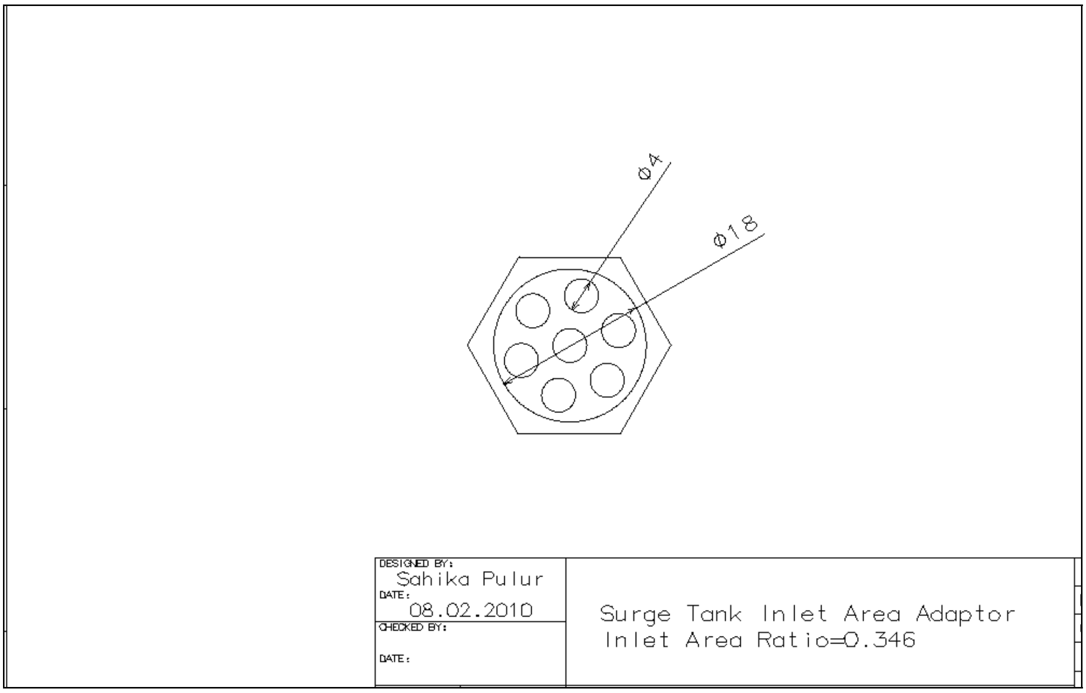

Figure 47 Technical Drawing of Adaptor with r=0.35

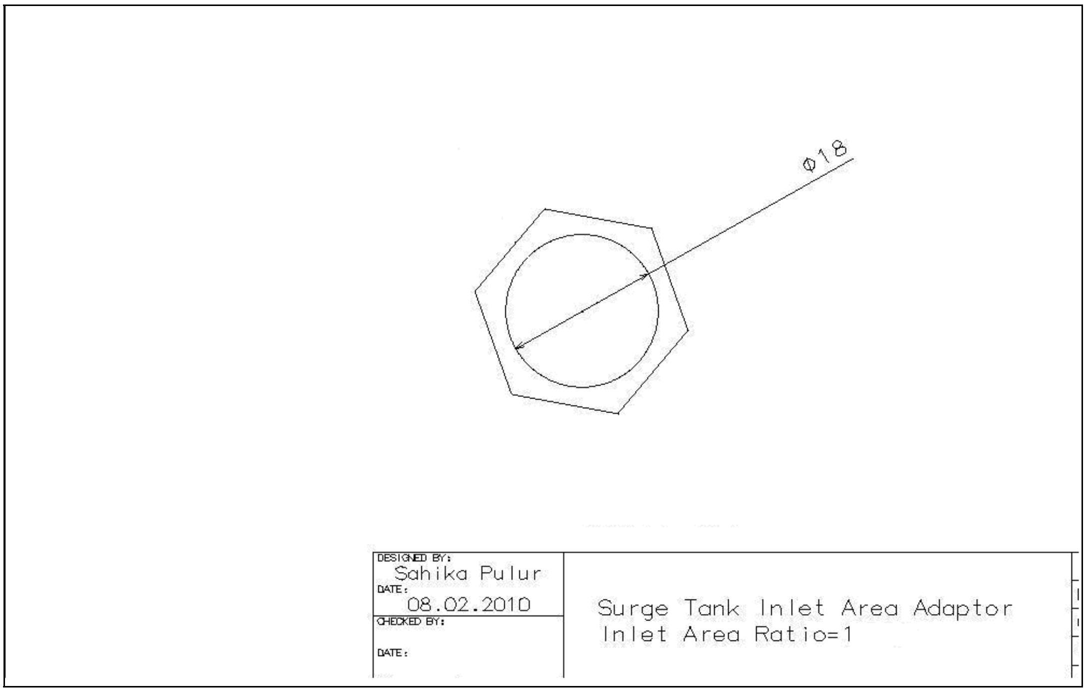

Figure 48 Technical Drawing of Adaptor with r=1