# CHAPTER 1

# INTRODUCTION

## 1.1 Background of Study

A pump is a device used to move fluids, such as gases, liquids or slurries. Pump works by displace fluid and causing flow. When the flow is resisted or blocked, it will cause a pressure raise in the flow. There are many types of pump and they are classified on the basis of applications they serve, the materials from which they are constructed, the liquid they handle, and even their orientation in space as shown in Figure 1.

Pumps may be divided into two major categories [1]:

a) Dynamic

- Energy is continuously added
- Increase the fluid velocities
- Subdivided into several varieties of centrifugal and other special-effect pumps

b) Displacement

- Energy is added periodically
- Direct increase in pressure
- Divided into reciprocating and rotary types, depending on the nature of movement of the pressure-producing members

The study will focused on centrifugal pump since disc pump is basically a centrifugal type. Pumps are commonly rated by flow rate, horsepower, outlet pressure and inlet suction. Performance of a pump is characterized by its net head, H (change in Bernoulli head between inlet and outlet of the pump).

Net head is then proportional to the useful power actually delivered to the fluid which is called water horsepower. All pumps will suffer from irreversible losses. This is due to friction, internal leakage, flow separation on blade surfaces, turbulent dissipation, and etc. Thus, the mechanical energy supplied to the pump is usually larger than water horsepower. Brake horsepower (BHP) is the external power supplied to the pump.

There are numerous applications where a pump is required. For example axial pump is used in sewage movement, flood control and other application that required high volumetric mass transfer and centrifugal pump is used in irrigation, water supply, gasoline supply, slurry transfer, and any application that required high head pressure. However, the centrifugal pump has some problems when used in this application where cavitation and wear may occur.

Multidisc pump is basically a disc pump or in certain area known as drag pump. It is called Multidisc pump because it has multiple disc act as an impeller in order to increase the flowrate.

Slurry can be a mixture of virtually any liquid combined with some solid particles. The combination of the type, size, shape and quantity of the particles together with the nature of transporting liquid determine the exact characteristics and flow properties of the slurry [2].

For this project, multidisc pump is used to transfer slurry with higher volumetric flowrate. This is because of slurry properties which are abrasive, contaminated and viscous where the multidisc pump is efficient.

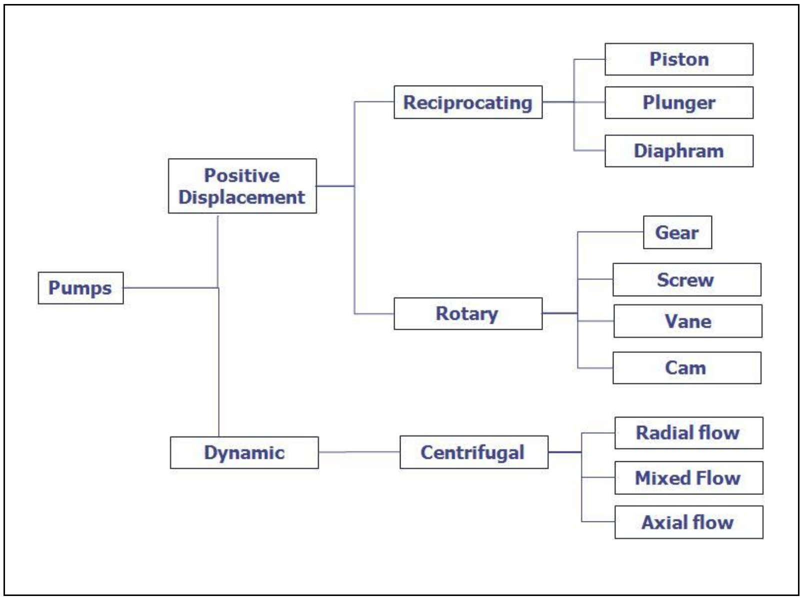



Figure 1: Various Types of Pump

### 1.2 Problem Statement

Pumps normally use impeller or vanes to push or move the fluid. The problem occurs when the fluid is contaminated, or mixed with other solid particles such as slurry. The pump would experience vibration. The impeller or vanes can scatter because of the impact with the solid particle. One way to overcome this is to use flat disc impeller pump. However, the efficiency and volumetric flow rate is low when using the flat disc. Thus, the parallel pump concept is used to increase the flow rate of the pump by multiplying the number of disc impeller. The project aims to design flat disc impeller with increased in volumetric flow rate by introducing Multidisc Pump.

## 1.3 Objectives

The objectives of this project are:

1. To design a Multidisc Pump to transfer contaminated or high solid contaminated fluid with high volumetric flow rate.
2. To determine the efficiency of the Multidisc pump based on simulation with known flow rate and head.

## 1.4 Scope of Study

The study will be divided into two parts:

2 parts:

Part 1-

- Investigation or research in pump focusing on centrifugal pump by searching in the internet, journals or books
- Calculation process
- Design process includes the usage of CAD modeling software such as CATIA

Part 2-

- Continuing Design and Modeling
- Meshing Process by using GAMBIT
- Flow analysis by using FLUENT software
- Finalize design

# CHAPTER 2

# LITERATURE OF REVIEW

## 2.1 Centrifugal pump

Based on Yunus A. Cengel [3], a centrifugal pump is a rotating machine and used an impeller to increase a pressure of a fluid. Static fluid pressure is increased by conversion of the rotational kinetic energy, usually from an electric motor or turbine. The kinetic energy form the impeller rotation is transferred to the fluid which is sucked from the impeller eye and is forced outward through the impeller vanes to the outlet. Fluid kinetic energy is then converted to static pressure due to the fluid experienced the resistance as it moves to the volute section in the outlet. Typically the volute shape of the pump casing which increasing in volume, or the diffuser vanes which serve to slow the fluid, converting to kinetic energy in to flow work are responsible for the energy conversion. The conversion of the energy results in an increased pressure on the downstream side of the pump, causing flow. Advantages when using centrifugal pump is it can produce high head pressure and can discharge a large amount of fluid.

The disadvantages of centrifugal pumps are:

- Cavitations
- Wear of the Impeller
- Corrosion inside the pump caused by the fluid properties
- Overheating due to low flow
- Leakage along rotating shaft
- Cannot run dry or in zero flow at long time

## 2.2 Disc Pump

Disc pump is one type of centrifugal pump but instead of having an impeller with vanes, disc pump rotate a disc or several disc in the same shaft. From Max I. Gruth patent title "Rotary Disc Pump" [4], a rotary disc pump comprises an outer housing with an inner cylindrical rotor chamber having an inlet at one end and outlet at it outer periphery. It also comprises at least 2 parallel spaced discs connect together for rotation about their center axis. The plain disc pump is suitable for pumping both fragile and severely abrasive materials, highly viscous fluid, and fluids with a high solid content where all of these fluids can cause damage to close-fit impellers and vanes on traditional vanes or bladed pumps.

Based on John Capello [5], disc pump minimize the contact between the pump and the product being pumped is suited to these types of applications. The working principle of this type of pump is when a fluid enters the pump from center of the disc its molecules adhere to the surfaces of these discs, providing a boundary layer. As the disc rotate, the molecule of fluid adhered to the disc will experience centrifugal force. This force will pushed the fluid to the edge of the disc and finally thrown out form the disc surface to the outlet entranced. This force also pushed the fluid through the pump in a smooth, pulsation-free flow. The fluid moves parallel to the discs, with the boundary layer creating a molecular buffer between the disc surfaces and the fluid.

The advantages of disc pump are [5]:

- Able to pass high solids
- Clog resistant (Max uptime)
- Non-impingement (Longer pump life)
- No damage to delicates (Higher yields)
- Pulsation-free (reduced wear on pump, piping)
- Run dry, dead-head discharge, starved suction
- Minimal radial loads
- Laminar flow

Donald S. Durand [6] cited that reducing the spacing between the discs greatly increases the pump efficiency. The spacing between discs is very important and must be varied accordance with the viscosity of the fluid being pumped. However, based on Max I. Gruth on his patent title “Rotary Slurry Disc Pump” [7], the spacing between discs allows handling of fluids carrying solids, entrained air or gas, stringy materials with little or no risk of clogging.

However, the disc pump also has a few disadvantages such as low head and volumetric flow rate because it has no vanes. Besides, disc pump is less efficient than a similar sized centrifugal pump in non-viscous applications. Figure 2 below shows the performance curve of a disc pump running at 1160 RPM and 13.45 inch impeller size.

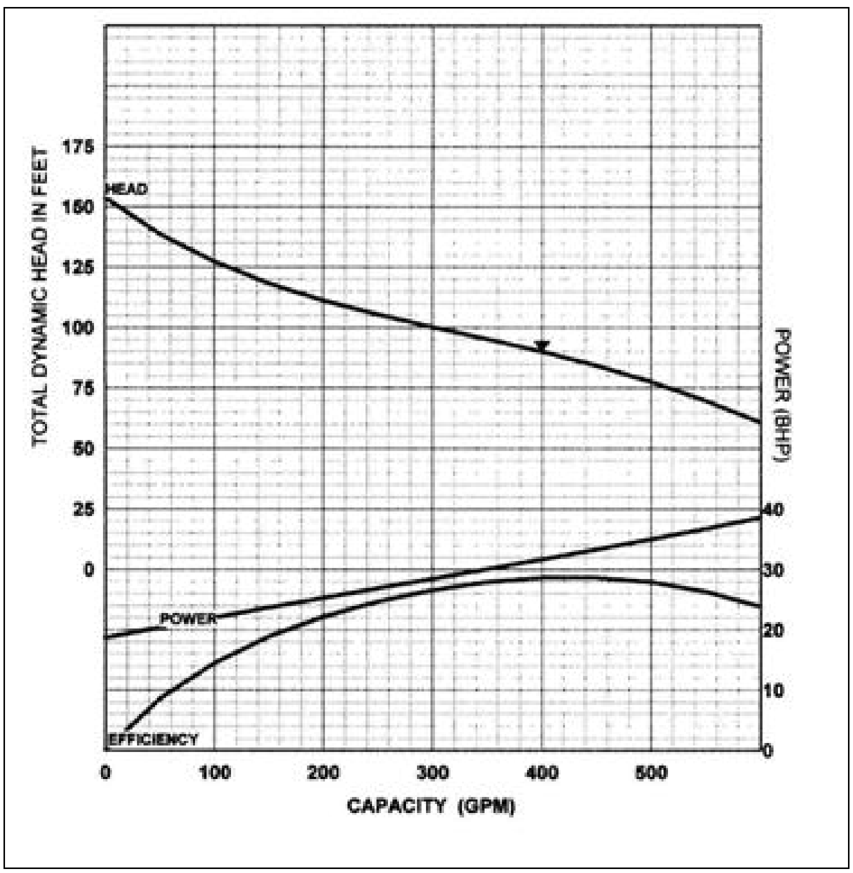



Figure 2: Performance curve of a 13.45 inch impeller disc pump running at 1160 RPM [5]

## 2.3 Slurry Pumping

Based on Warman [2], slurry is a mixture of some solid particles and liquids combined together. Generally, there are 2 groups of slurry; settling and non-settling types. There are many type of pumps used to pump slurries from positive displacement and special effect types such as Venturi eductors but the common type used is centrifugal pump. Important centrifugal slurry pump factors need to be considered is impeller size and design and its ease of maintenance. Many other important considerations are also required.

The type of impeller for slurry pump is usually plain or Francis vane. Some advantage of the Francis vane profile are the higher efficiency, improved suction performance and slightly better wear life in certain types of slurry because the incidence angle to the fluid is more effective .

However, according to Max I. Gurth [5], a rotary pump having a plain discs impeller also can be used to pump highly abrasive slurries with very little wear. Any number of discs can be used as the impeller of the pump. The spacing for the discs is preferably less than one half inch even for large diameter pumps. For fine particle abrasive material, this spacing should be as close as from 0.25mm to 0.5mm. With the pump comprising a plain disc impeller with a considerably unobstructed passage between the inlet and outlet of the pump, the slurries and fragile particle can be carried along in the fluid stream without impact with the portions of pump assembly.

### 2.3.1 Centrifugal Slurry Pump Design and Calculation

For this project, the Multidisc pump will be compared with the previous design centrifugal pump. Therefore, the working slurry pump specifications and requirements are extracted from Warman [2]. Below is the design requirement and specification for the slurry centrifugal pump:

For slurry with;

- Specific gravity of solids, $SG_s = 2.65$
- Specific gravity of mixture, $SG_l = 1.23$
- Average particle size d50 = 211 microns (0.211mm)
- Concentration of solids $C_w$ = 30% by weight, viscosity = 5.57cP
- Static discharge head ($Z_d$) = 20m = 65.6ft
- Suction head ($Z_s$) = 1m (positive) = 3.28ft
- Length of pipeline = 100m = 328ft
- Valves and fittings = 5 x 90° long radius bends

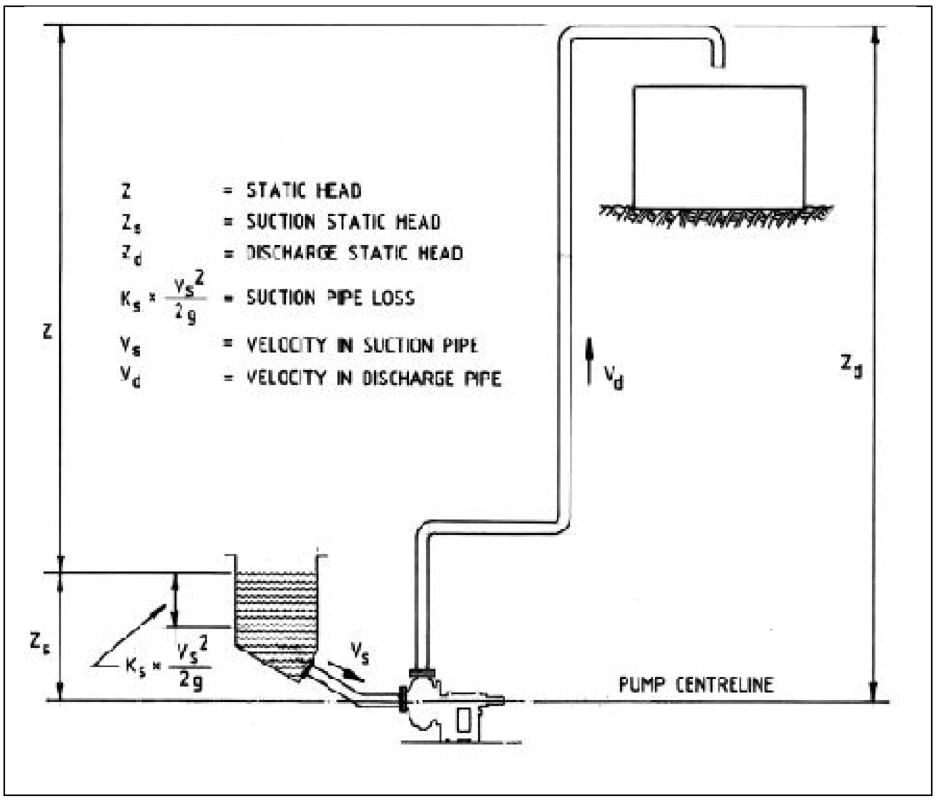



Figure 3: Typical Pump Application [2]

Pump size, speed and recommended size of delivery pipeline are determined as follows:

- Capacity: 49L/s = 775.2GPM
- Pipe diameter: 150mm = 6in
- Total Dynamic Head, TDM: 25.4m = 83.312ft
- RPM: 1130
- Impeller type: 5 vane closed rubber
- Efficiency: 66%
- Power Input: 30kW (40.23hp) motor

The pump curve for this pump is shown as Figure 4.

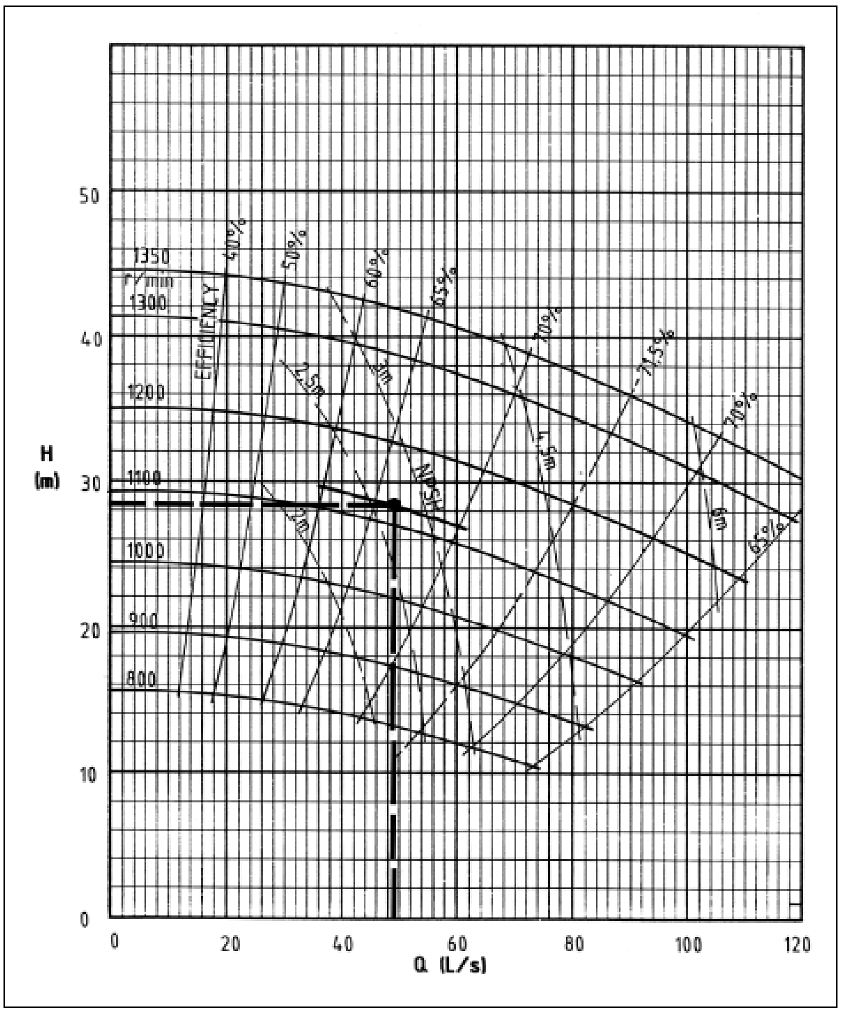



Figure 4: Performance Chart of a Warman Slurry Pump [2]

## 2.4 Standards

The standards used by the author for this report are:

1. ANSI: American National Standards Institute. A term often used in connection with the classification of flanges, ANSI class 150, 300, etc.
2. ANSI B73.1: This is a standard that applies to the construction of end-suction pumps. It is the intent of this standard that pumps of all sources of supply shall be dimensionally interchangeable with respect to mounting dimensions, size and location of suction and discharge nozzles, input shafts, base plates, and foundation bolts.
3. ASME: American Society of Mechanical Engineers. The Boiler pressure power piping code B31.3 is a code that is often used in connection with the term ASME, the maximum pressure safely allowable can be calculated using this code.
4. ASME B16.5: for the pressure rating of ANSI class flanges.

# CHAPTER 3

# METHODOLOGY

## 3.1 Governing Equations

Head is defined as the height at which a pump can displace a liquid to. Head is also a form of energy. In pump systems there are 4 different types of head: elevation head or static head, pressure head, velocity head and friction head loss. It is also know as a specific energy or energy per unit weight of fluid, the unit of head is expressed in feet or meters. The static head corresponding to any specific pressure is dependent upon the weight of the liquid according to the following formula:

$$\text{Head (ft)} = \frac{2.31 \times \text{Pressure (psi)}}{\text{Specific gravity}} \tag{1}$$

2.31 = conversion factor

Relationship between the head and velocity developed in pump is expressed by,

$$H = \frac{V^2}{2g} \tag{2}$$

H = Total head developed (ft)
V = velocity of the impeller (ft/sec)
g = 32.2 ft/sec$^2$

The approximate head of any centrifugal pump can be predicted by calculating the velocity of the impeller tip. In case of diameter of the impeller is given, the impeller tip velocity can be calculated based on the following equation,

$$U_1 = (RPM \times D_o) / 229 \qquad (3)$$

$U_1$= Impeller tip velocity (ft/sec)
$D_o$ = Impeller outside diameter (in)
RPM = Angular velocity in revolution per minute
229 = conversion factor

Therefore, to calculate the outside impeller diameter ($D_o$), rearrange equation (2) and (3) gives;

$$D_o = \frac{229 \times (8.025 \times \sqrt{H})}{RPM} \qquad (4)$$

Flow Velocity is the velocity of the fluid leaving the pump or entering the pump (suction eye velocity. This can be calculated based on the following calculation:

$$Cm_1 = \frac{0.4085 \times Q}{D_i^2} \qquad (5)$$

$Cm_1$ = Suction eye velocity (ft/s)
Q = Flowrate (GPM)
$D_i$ = Impeller Inside Diameter (in)
0.4085 = conversion factor

Capacity is define as how much the pump can transfer or move in certain amount of time. Capacity usually expressed in Gallons per minute (GPM) or cubic meter per seconds ($m^3/s$). Liquid are essentially incompressible hence, there are direct relationship between the capacity in pipe and the velocity of flow,

$$Q = 449 \times A \times V \qquad (6)$$

Q = Capacity (GPM)
V =Velocity of flow (ft/sec)
A = Pipe cross section area ($ft^2$)
449 = conversion factor

Pump output or hydraulic horsepower, $P_{out}$ (hp) is the liquid horsepower delivered by the pump,

$$P_{out} = \frac{Q \times TDH \times \text{Specific Gravity}}{3960} \qquad (7)$$

$P_{out}$ = Hydraulic horsepower (hp)

3960 = conversion factor

The input power usually is electrical power ($P_{in}$)

$$P_{in} = \omega T = \frac{2\pi NT}{33000} \qquad (8)$$

$P_{in}$ = Power input (hp)
T = Torque (lbf.ft)
N = Angular velocity (revolution per minute)
33000 = conversion factor

Pump efficiency is the ratio between power output and power input,

$$\text{Pump efficiency, } \eta = \frac{P_{out}}{P_{in}} \qquad (9)$$

## 3.2 Flow Chart

Figure 5 below shows the execution flow chart of the project.

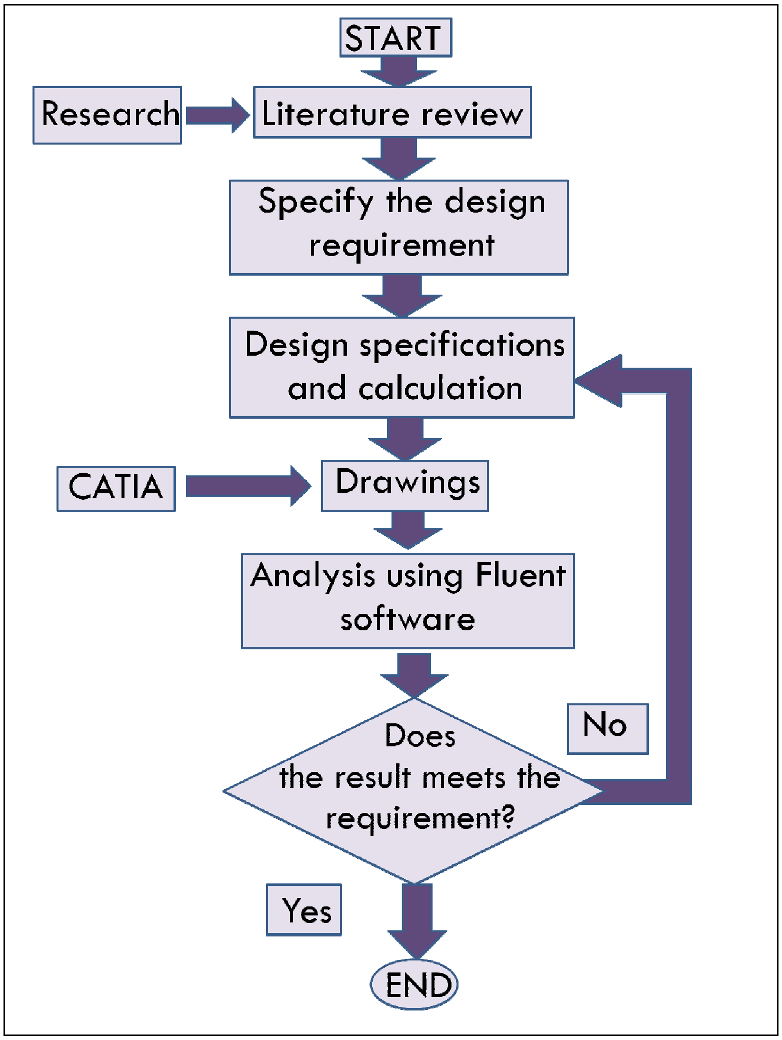



Figure 5: Execution Flow Chart of the Project

## 3.3 Research Methodology

Further investigations of the impeller designs and sizing calculation, flow analysis and simulation need to be done. A thorough search will be made through the internet, journals and from the libraries to collect all required information. Then, the author needs to consult a few lecturers to get a brief idea on how to continue researching on the topic.

The author needs to understand the theory from the research done. The author has listed which part to be highlighted in this project study:

- Research for existed slurry pump to be replaced with Multidisc Pump
- Research for the spacing between impeller needed to increase efficiency
- Design calculation for the impeller diameter
- Research on simulating flow using a software
- Suitable motor to be used with Multidisc Pump

### 3.4 Project Activities

Before designing the Multidisc Pump, the first thing to do is develop a design concept. Then, the design is chosen based on standard. After that, a few calculations need to be done in order to determine the inside and outside diameter of the impeller. The specification such as suction and discharge head need to be verifies first in order to calculate the impeller inside and outside diameter. The author makes a comparison between the slurry pump and the existing centrifugal pump in the same application in order to specify the data. The detail design specification and requirement is then finalized. Various proposed Multidisc Pump is designed using CATIA software. The design is varied form the number of impeller and the spacing between the impeller. Table 1 shows the variations in the design of the disc impeller.

The design is then meshed by using meshing software. For this project, the meshing software used is GAMBIT. Then, the meshed design is then simulated using FLUENT software and the result is recorded before the final design is chosen based on the result obtain from the simulation**.** A timeline is prepared for completion of this FYP by the author based on the academic schedule, FYP guideline for students and supervisor requirements.

| Variable | Design | |
|---|---|---|
| No. of Impeller | 1 | 3 |
| Spacing Between 3 Discs Impeller | 0.2 inches | 0.5 inches |

Table 1: The Variations in the Design of the Disc Impeller.

### 3.5 Concept Design

Figure 6 below shows the design concept of the Multidisc pump.

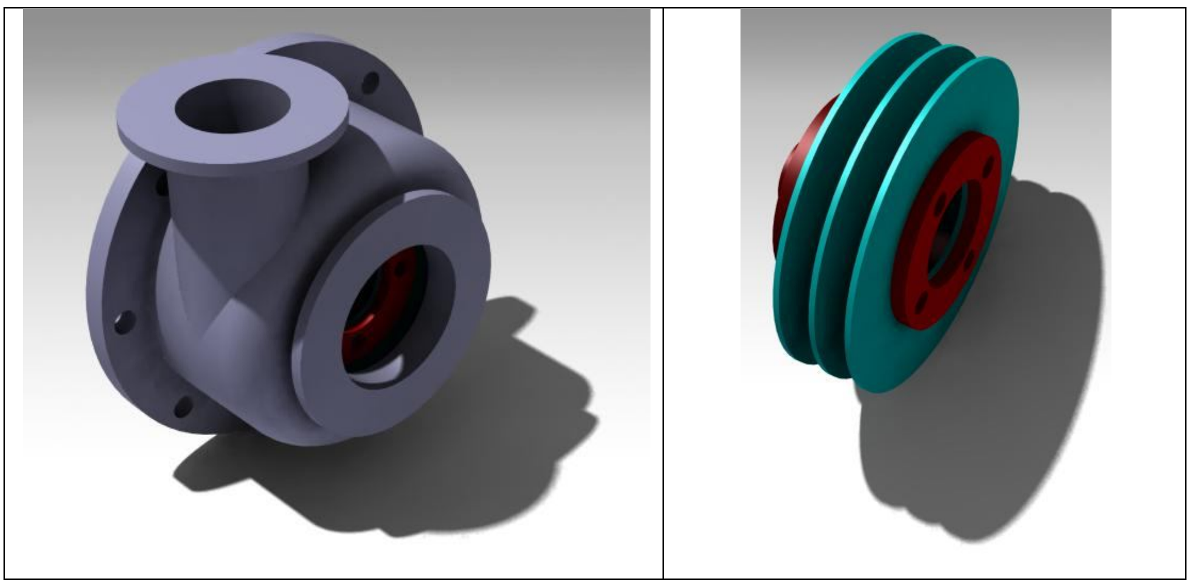


Figure 6: The Design Concept of the Multidisc Pump

## 3.6 Tool Required

### 3.6.1 Computer

- Used as medium to install software in order to design Multidisc Pump
- Report writing and research medium

### 3.6.2 Software

1. CATIA

   Software used to draw a design of Multidisc Pump before it can be analyzed and simulated in other software. The design will be saved as IGES format.

2. GAMBIT

   Software used to mesh the design before it can be simulated in the simulation software. The meshed design will be saved as MSH format.

3. FLUENT

   Software used to simulate the flow in the Multidisc Pump and to observe the velocity and pressure variation in the pump.

# CHAPTER 4

# RESULT & DISCUSSION

## 4.1 Design Requirement

The design requirements for the Multidisc pump are the same as the previous centrifugal pump as cited in literature review with the same application. Therefore the design requirements for Multidisc pump are as below:

- Application: Slurry pumping (sand and water)
- Pumping capacity , Q = 775.2 GPM
- Total dynamic head on the pump, H = 83.3ft
- Equivalent water head, TDH = 93.5ft
- Specific gravity of solids, $SG_s$ = 2.65
- Specific gravity of mixture, $SG_m$ = 1.23
- Slurry density: 1230kg/$m^3$ (76.9lb/$ft^3$)
- Average particle size, $d_p$ = 211micron
- Concentration of solids, $C_w$ = 30% by weight
- Static discharge head , $Z_d$ = 65.62ft
- Suction head, $Z_s$ = 3.28ft
- Length of pipeline = 328.08ft
- Valves and fittings = 5 x 90° long radius bends
- Pipe inlet diameter = 3.7in

Total dynamic head calculation is shown in Appendix 4. The design requirements are taken from previous designed centrifugal pump for the same application.

## 4.2 Calculations

### 4.2.1 Impeller diameter

Outside impeller diameter can be calculated by using equation (4);

H = 83.3ft, N = 1500RPM,

Therefore, $D_o = \frac{229(8.025\sqrt{83.3})}{1500}$ = **11.2in**

From equation (3), when $D_o$ = 11.2in

Impeller tip velocity, $U_1$= (11.2 x 1500) / 229 = **73.4ft/s**

From suction eye velocity curve in Figure 7, when $Z_s$ = 3.28, $U_1$= 73.4ft/s, suction eye velocity, $Cm_1$= **23.2ft/s**.

Therefore, inside impeller diameter can be calculated by using equation (5);

When Q = 775.32GPM, $Cm_1$= 23.2ft/s

$D_i = \sqrt{(0.4085 \times 775.2 / 23)}$

= **3.7 in**

Therefore, the impeller design must have 11.2inch outside diameter and 3.7inch inside diameter. The thickness is set to be 0.2inch. From calculation, the inlet flowrate is 23.2ft/s. Spacing between the impeller is set to be as lower as possible but it will allow the fluid particle to pass through. For this project, the fluid particle size is only 211microns, thus the spacing is sufficient to give maximum efficiency.

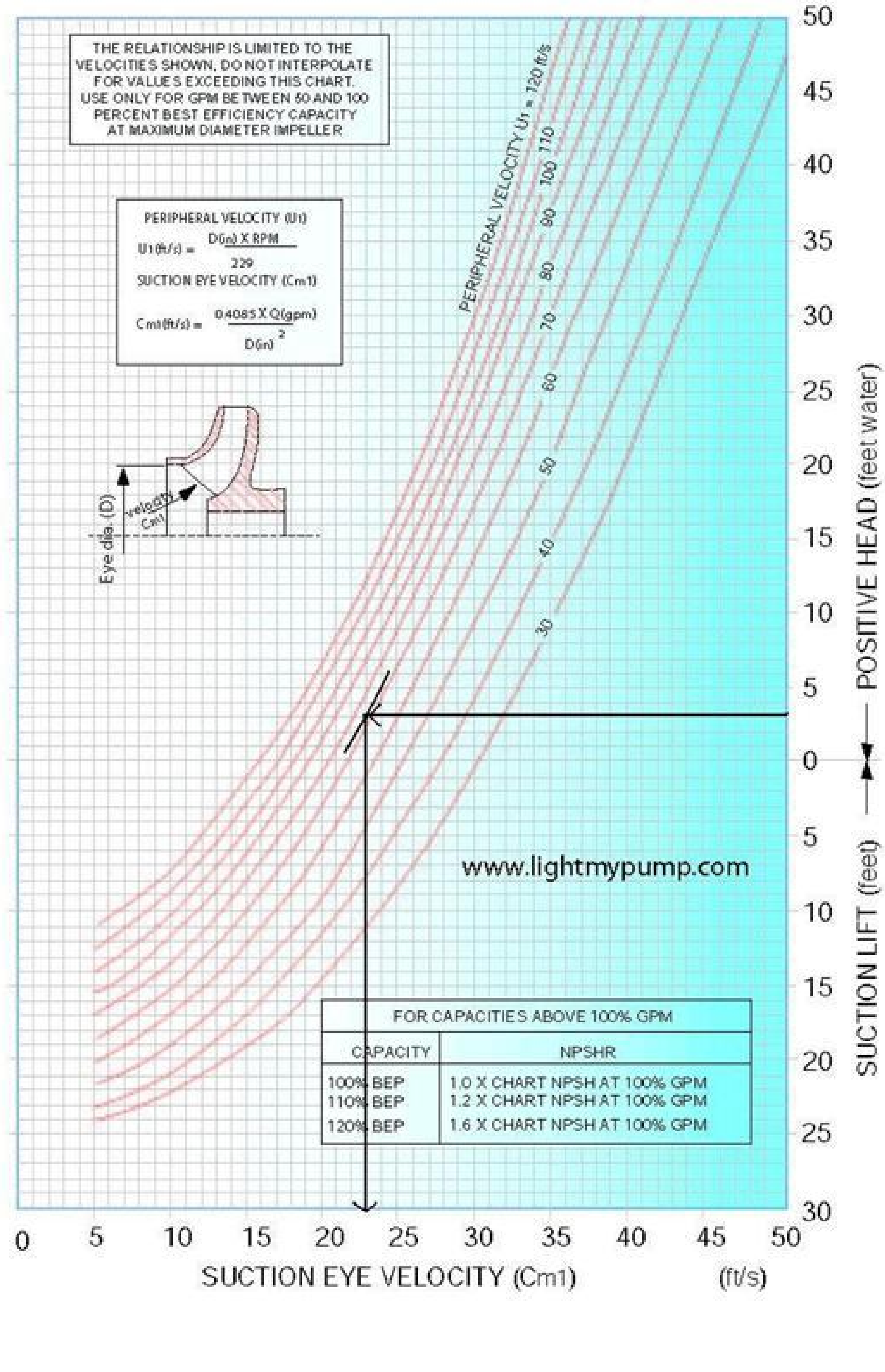



Figure 7: Suction Eye Velocity Value

### 4.3 Multidisc Pump Design Specifications

From the calculations, the design specifications for Multidisc Pump have been carried out. The impeller thickness is assumed to be at 0.2 inch to avoid vibrations and the casing is the normal volute casing. The impeller designed must follow the specification listed as below:

- Impeller outside diameter: 11.2in
- Impeller inside diameter: 3.7in
- Casing inlet diameter: 3.7in
- Casing outlet diameter: 2.8in
- Impeller thickness: 0.2in

### 4.4 Impeller Design Drawings

The impeller is designed according to the specifications. It is shown in Figure 8-11

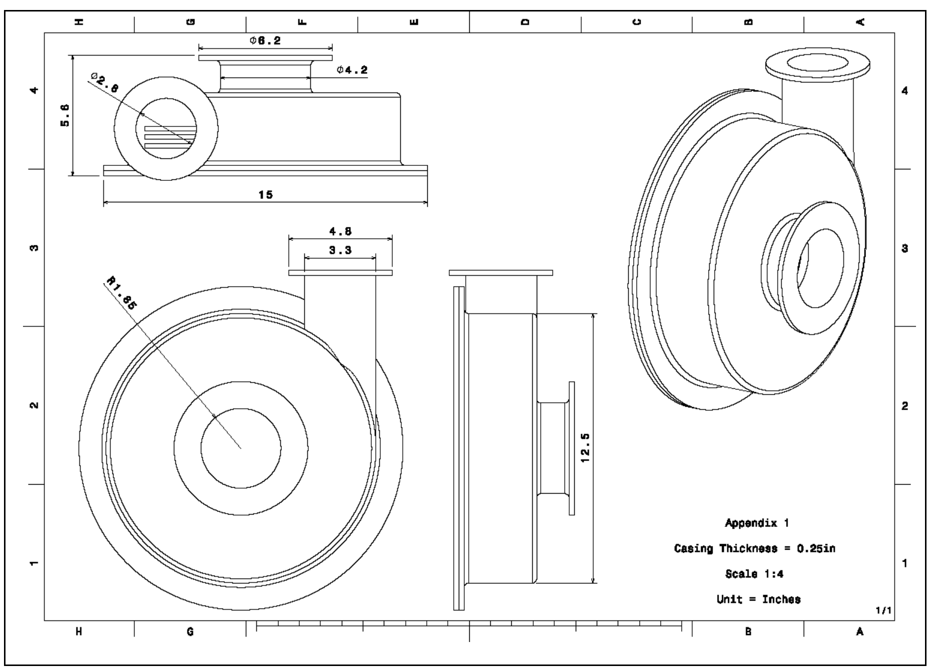



Figure 8: Overall Design of Multidisc Pump

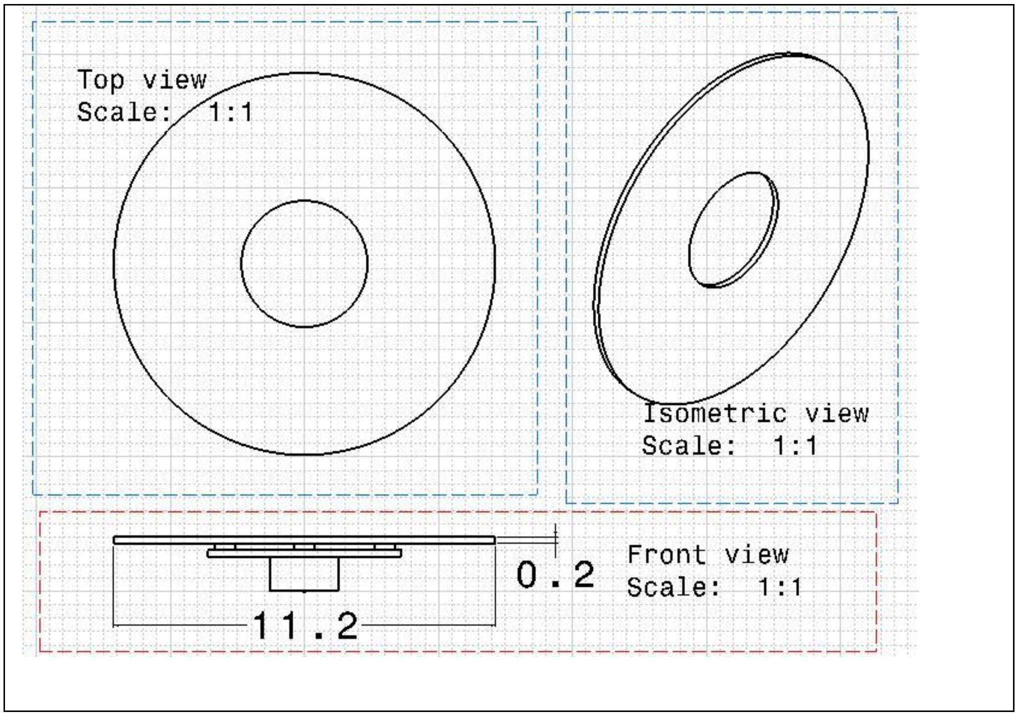



Figure 9: One Disc Impeller Design

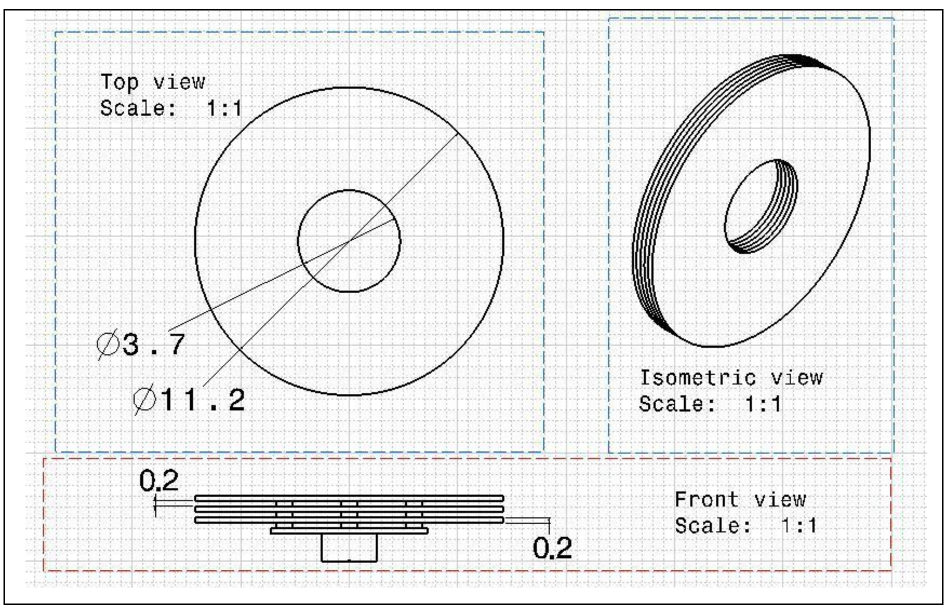



Figure 10: Three Discs Impeller Design with 0.2inch Spacing

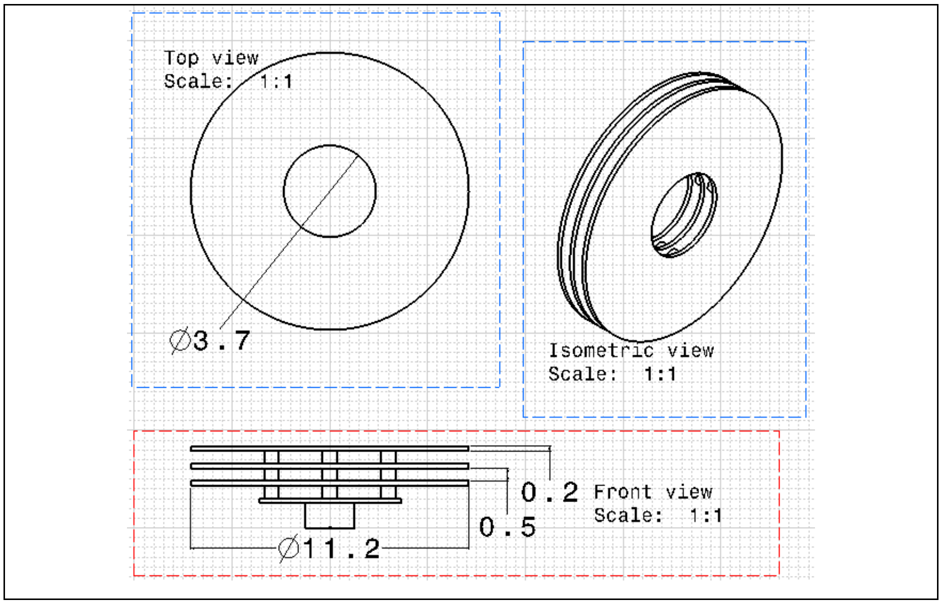



Figure 11: Three Discs Impeller Design with 0.5inch Spacing

The impeller and pump casing detail drawings are shown in the Appendix 1 and 2. The concept designs are shown as below:

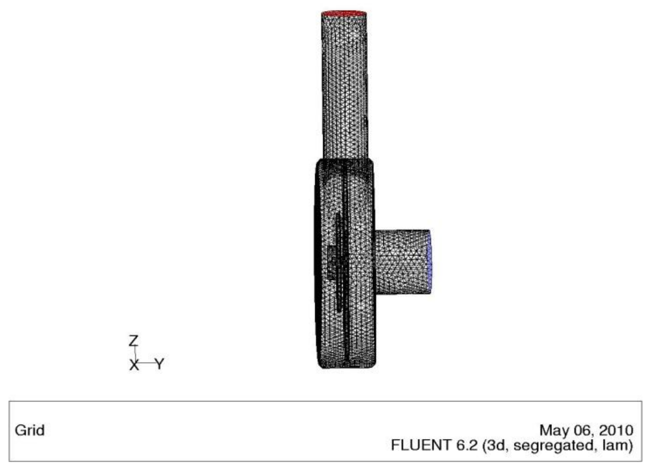
**4.5 Meshing**

The design is then meshed before the simulation process. The meshing software used is GAMBIT. The results are in Figure 12.

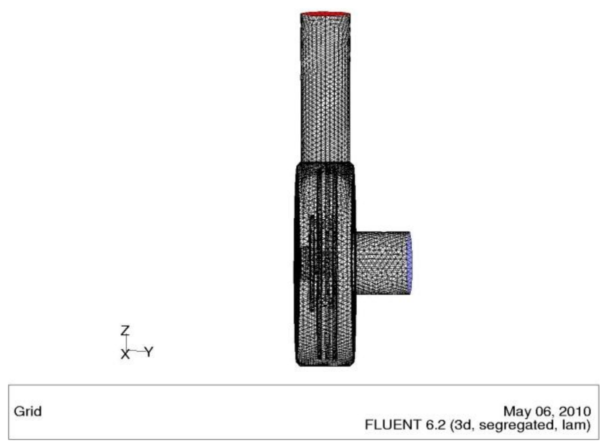
The meshing input parameters:

Mesh Volume;

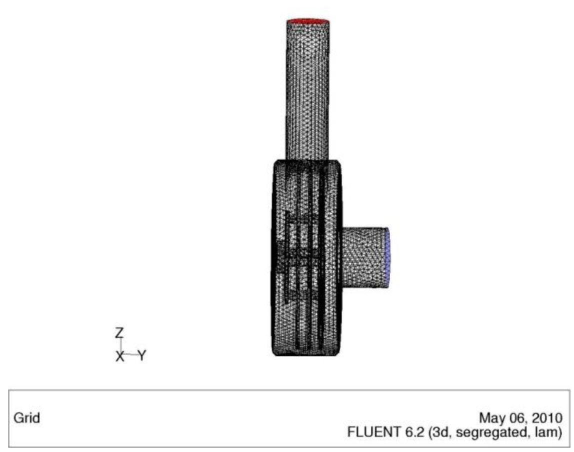
- Elements: Tetrahedral/Hybrid
- Type: T Grid
- Interval size: 0.3mm
- Boundary type : Velocity Inlet, Outflow and Wall




One Impeller Disc




3 Impeller Discs with 0.2 inch Spacing




3 Impeller Discs with 0.2 inch Spacing

Figure 12: Meshed Multidisc Pump.

## 4.6 Simulation

The design is then simulated by using FLUENT software. For the simulation, the input required is the flow velocity at the pump inlet (23.2ft/s) along with the angular velocity of the impeller (1500RPM). The outcome is shown in Figure 13-15

### 4.6.1 Fluid Velocity at the Impeller Tip

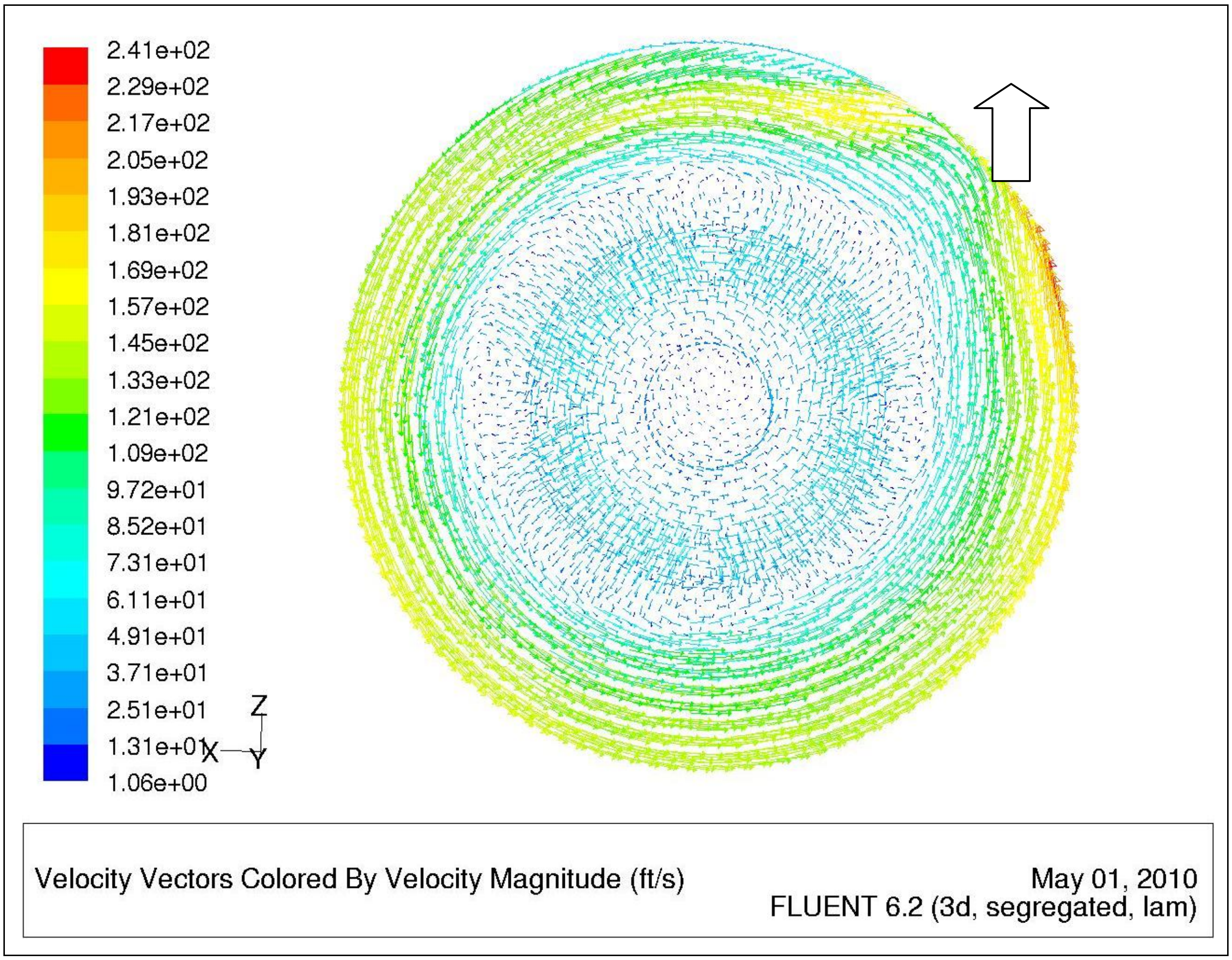



Figure 13: Fluid Velocity at the Impeller Tip (1 Impeller)

Fluids enter the pump inlet and rotate to the impeller tip before exit to the outlet. Fluid velocity is higher at the impeller tip and maximum at the outlet.

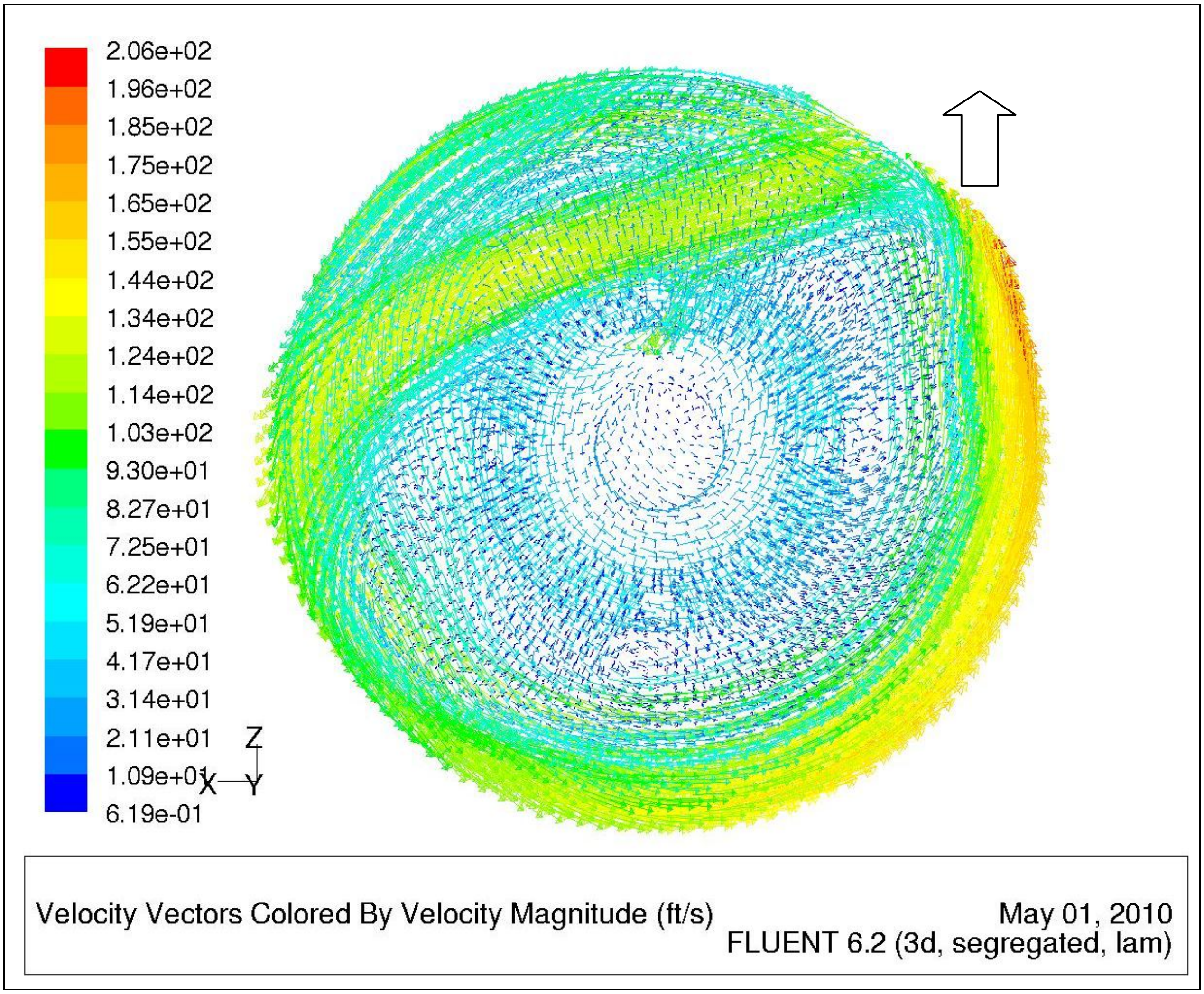



Figure 14: Fluid Velocity at the Impeller Tip (3 Impellers with 0.2 inch Spacing)

From the observation, fluids enter the pump inlet and rotate to the impeller tip before exit to the outlet. The maximum fluid velocity is at the outlet. There are also recirculations of fluids observed. This is maybe due to small pump outlet.

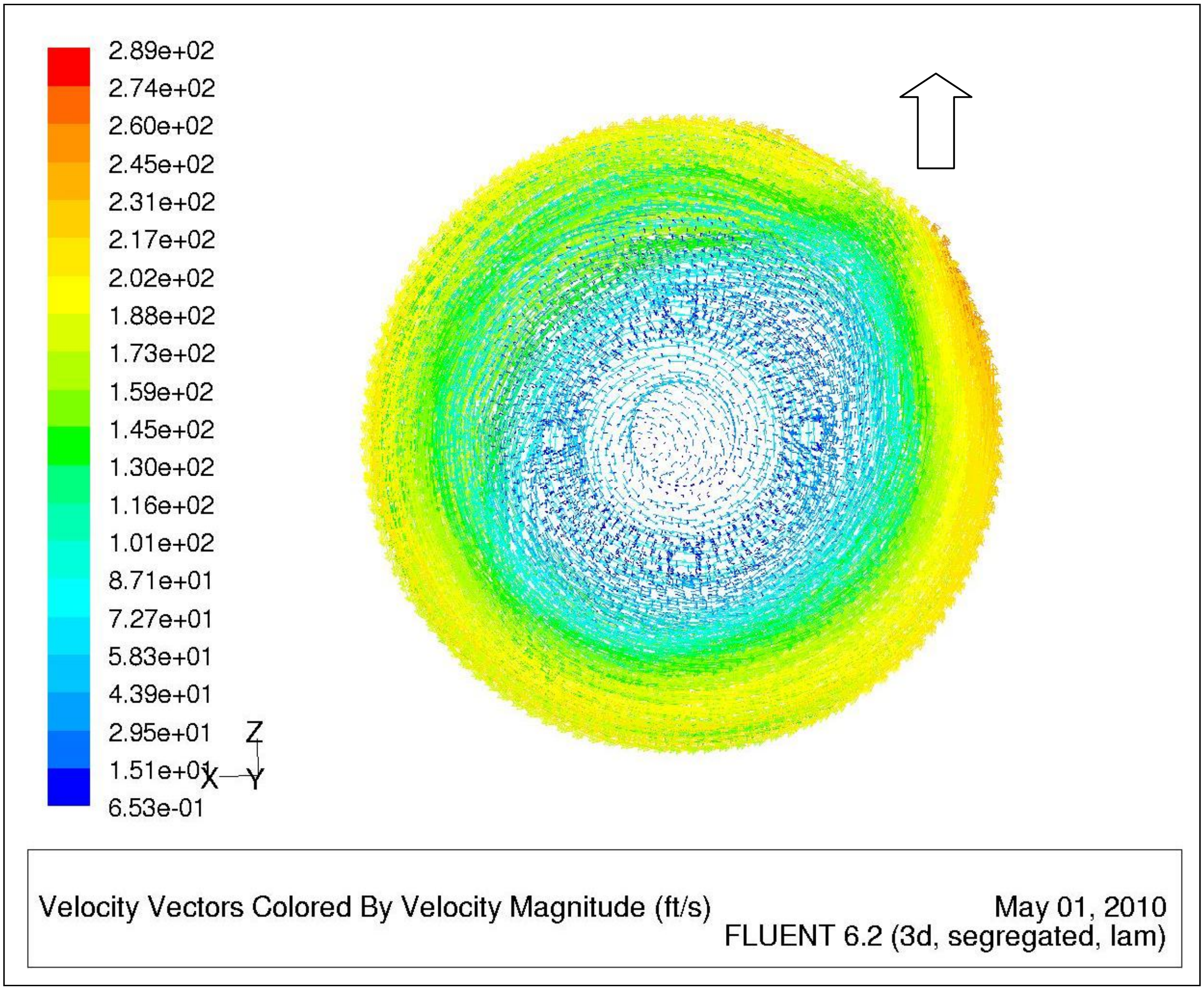



Figure 15: Fluid Velocity at the Impeller Tip (3 Impellers with 0.5 inch Spacing)

Fluids enter the pump inlet and rotate to the impeller tip before exit to the outlet. The maximum velocity observed is at the pump outlet. There is also recirculation of fluid in this design.

### 4.6.2 Fluid Velocity at the Pump Outlet

Figure 16-18 shows the velocity of fluid at the pump outlet.

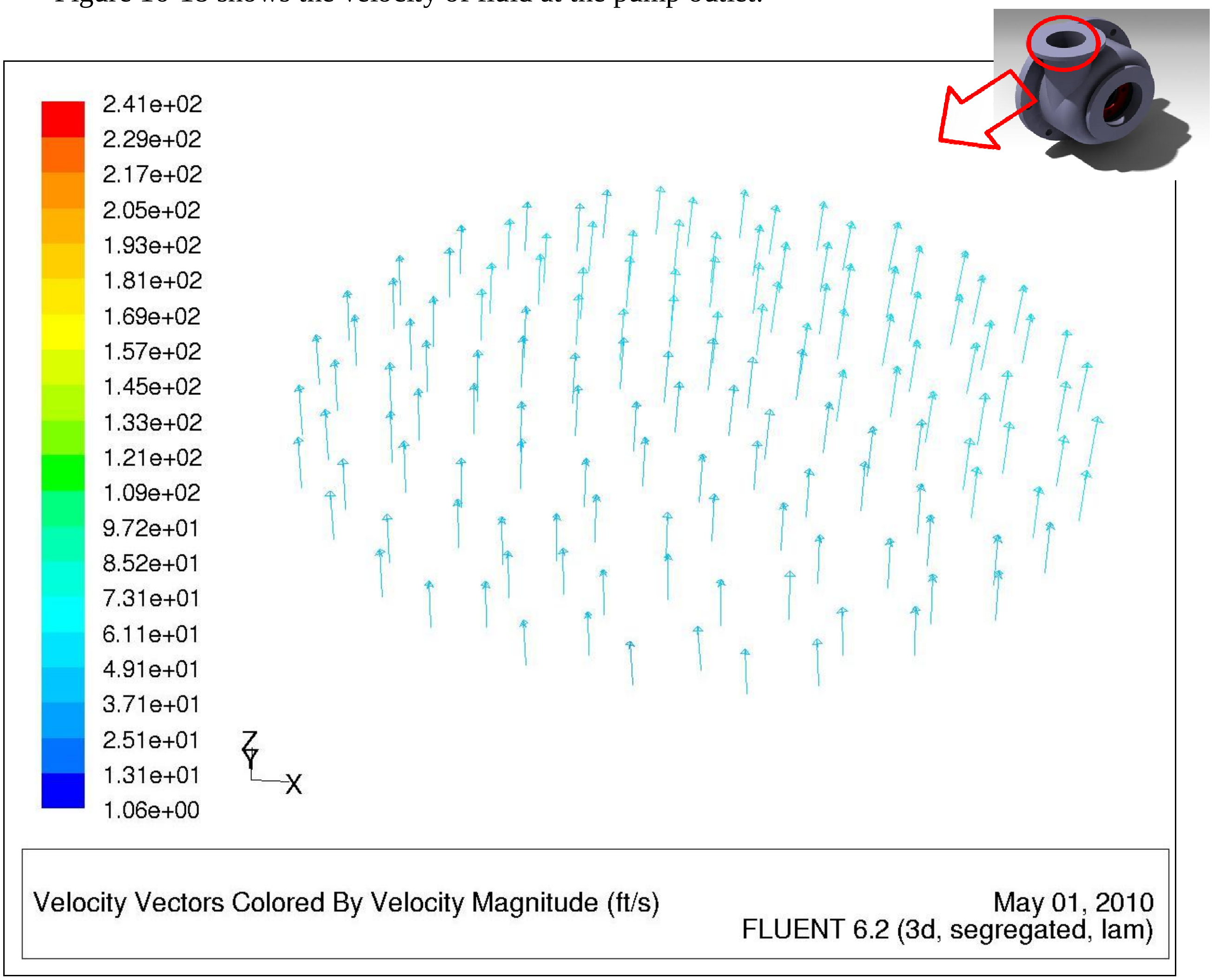



Figure 16: Fluid Velocity at the Pump Outlet (1 Impeller)

The arrow shows the velocity of the fluids at the outlet of the pump. The average fluid velocity is 49.1ft/s

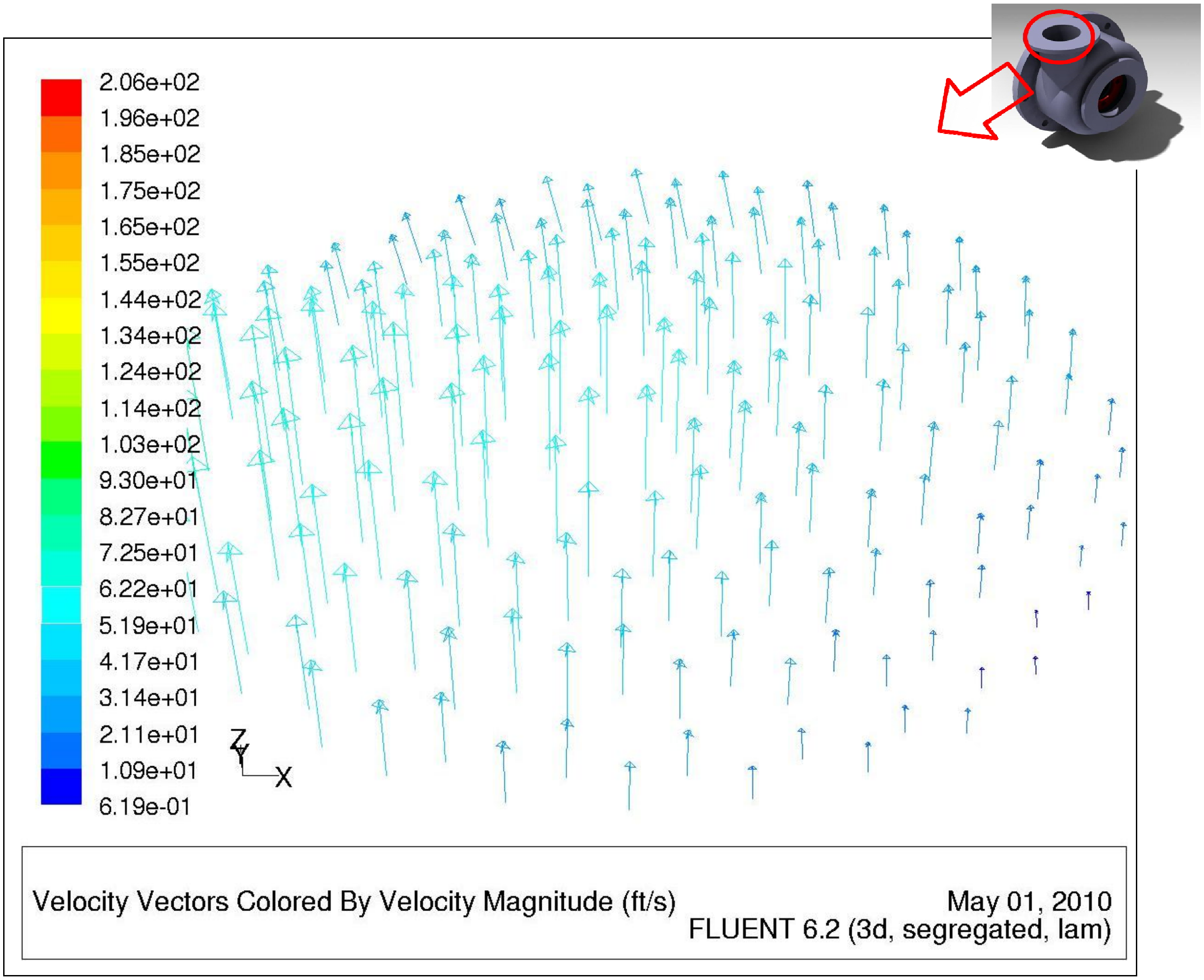



Figure 17: Fluid Velocity at the Pump Outlet (3 Impellers with 0.2 inch Spacing)

The arrow shows the velocity of the fluids at the outlet of the pump. The average fluid velocity is 67.3ft/s

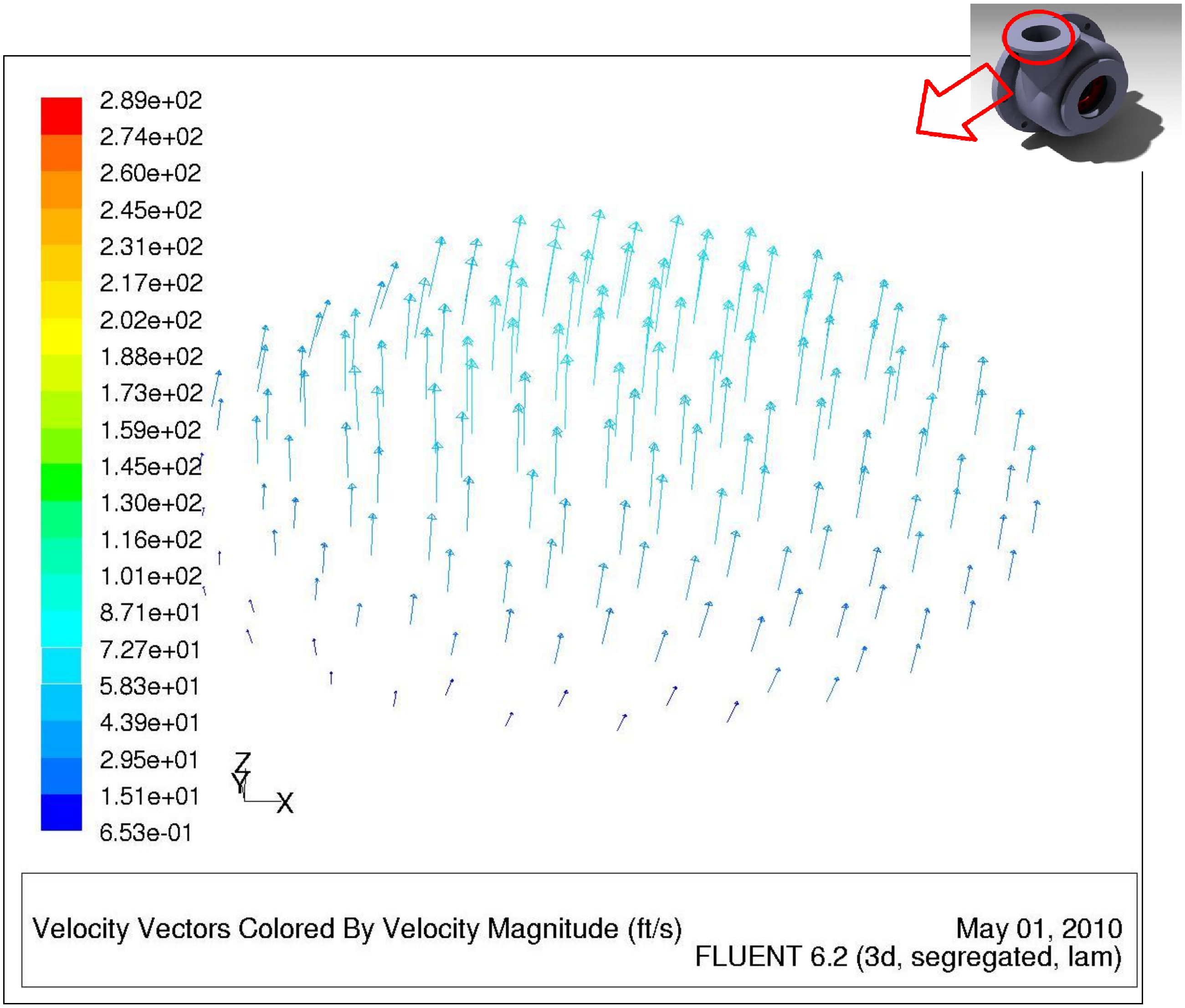



Figure 18: Fluid Velocity at the Pump Outlet (3 Impellers with 0.5 inch Spacing)

The arrow shows the velocity of the fluids at the outlet of the pump. The flow is turbulent and swirling at the outlet. The average fluid velocity is 51.1ft/s.

### 4.6.3 Flow Velocity Distribution in the Pump

Figure 19-21 shows the flow distribution in the pump.

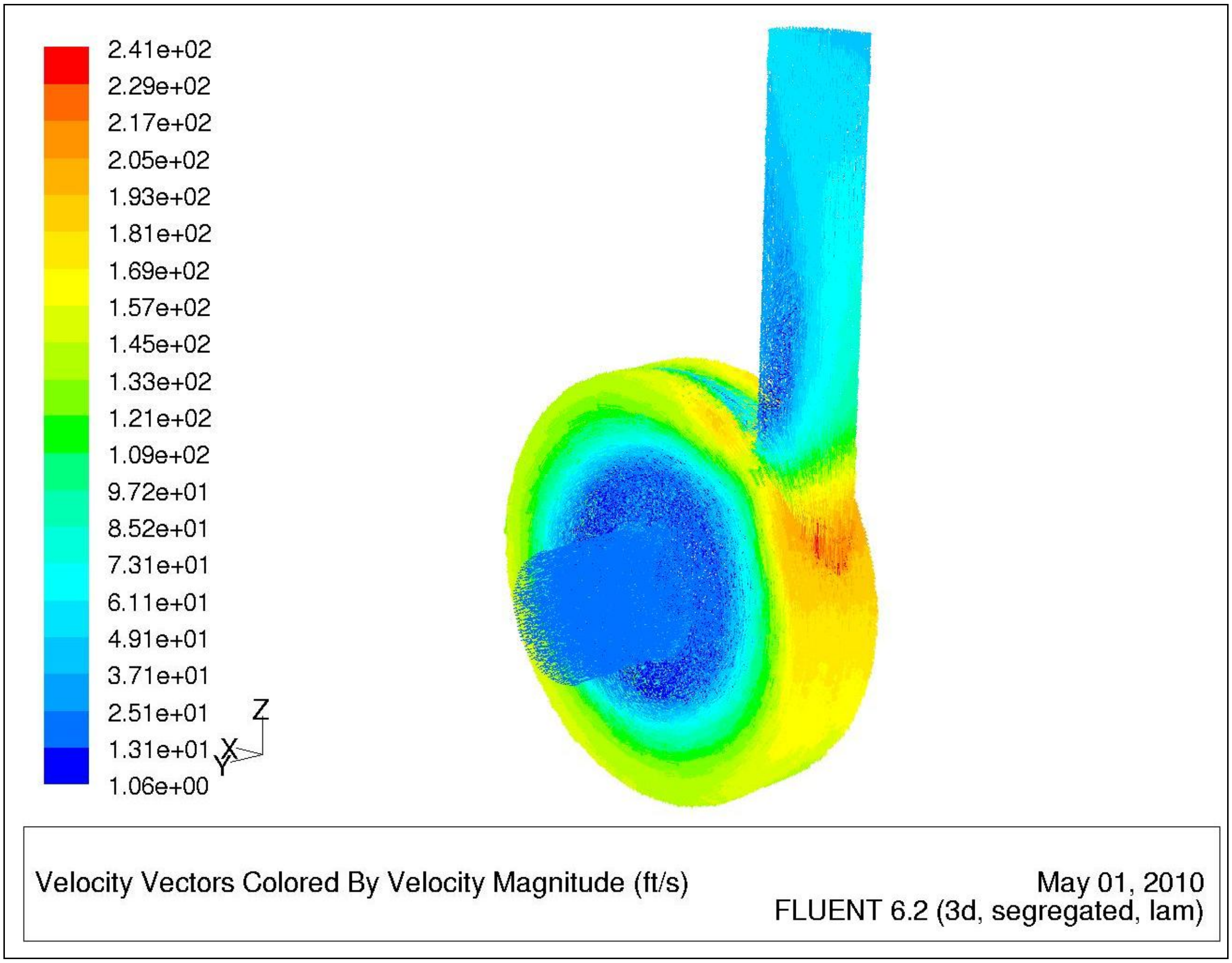



Figure 19: Flow Velocity Distribution (1 Impeller)

Fluids enter the pump inlet before travel to the outlet. At the outlet entrance, the velocity is high because of the cross section area become smaller.

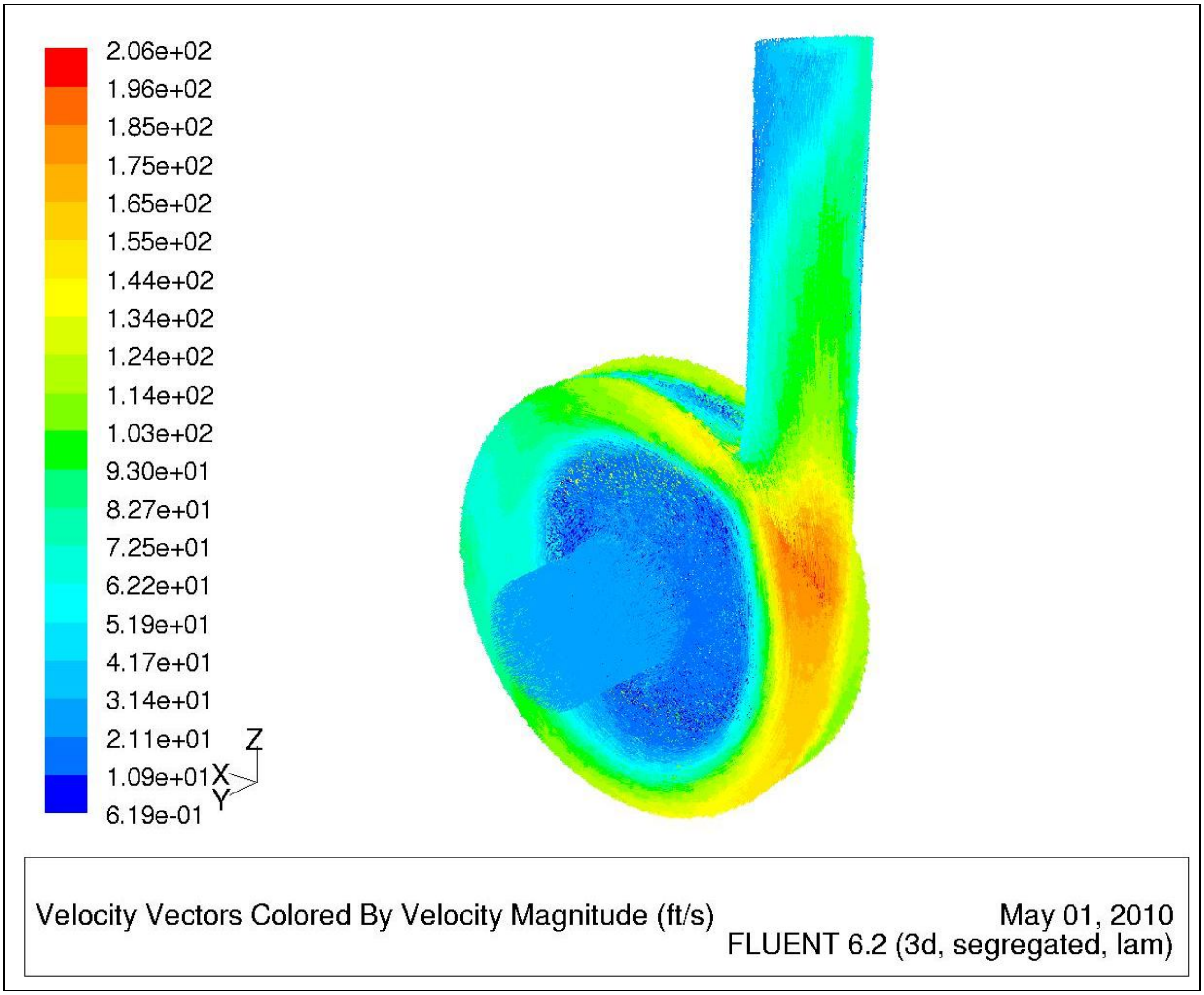



Figure 20: Flow Velocity Distribution (3 Impellers with 0.2 inch Spacing)

Fluids enter the pump inlet before travel to the outlet. At the outlet entrance, the velocity is high because of the cross section area become smaller.

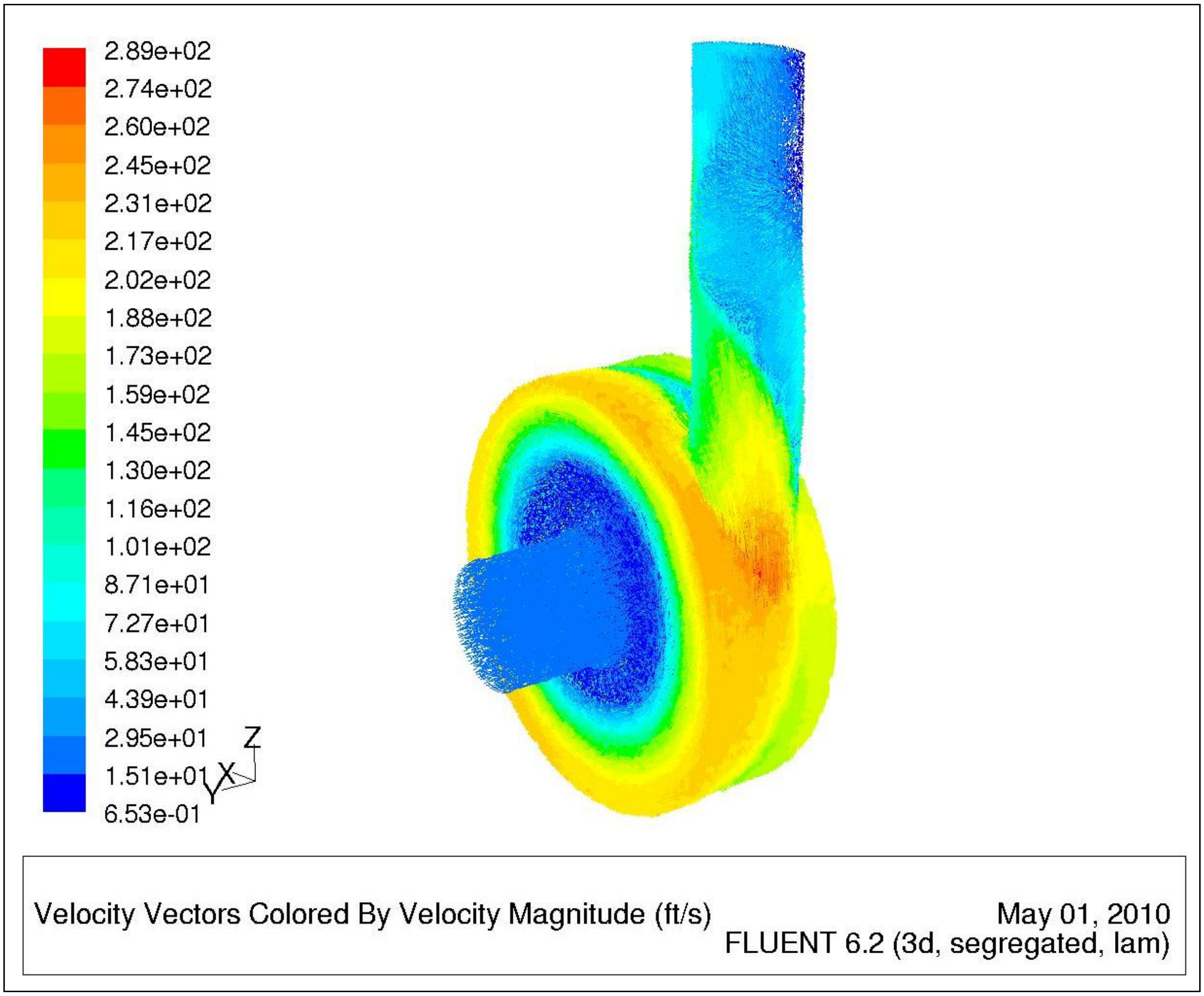



Figure 21: Flow Velocity Distribution (3 Impellers with 0.5 inch Spacing)

Fluids enter the pump inlet before travel to the outlet. At the outlet entrance, the velocity is high because of the cross section area become smaller. The fluid at the outlet is also swirling. This is due to the turbulence flow at the outlet.

### 4.6.4 Discharge Flowrate Calculation

Discharge flowrate can be calculated with the data obtain from the simulation. The example calculation is as below:

For 3 impellers with 0.2 inch spacing, the average discharge velocity = 67.3ft/s and the cross sectional area of the discharge pipe = 0.0426ft$^2$. Therefore from equation (6),

Q = 449 x 67.3 x 0.0426

= **1287.3 GPM**

The discharge flowrate for various designs of Multidisc pump is shown in Table 2.

| Impeller | Average Discharge Velocity (ft/s) | Discharge Flowrate (GPM) |
|---|---|---|
| 1 | 49.1 | 939.2 |
| 3 (0.2 spacing) | 67.3 | 1287.3 |
| 3 (0.5 spacing) | 51.1 | 977.4 |

Table 2: Discharge Flowrate Various Designs of Multidisc Pump

From the simulation, when the impeller is one, the discharge flowrate is 939.2GPM and when the impeller is three, the flowrate is higher. But for 3 impellers, when the spacing between discs is lower, the discharge flowrate is higher. This is because of the working principle of disc type of pump which fluid is transfer by a boundary layer and centrifugal force effect. When the space between discs is lower, the boundary effect is higher. Therefore, the discharge velocity for three impellers with 0.2inch spacing is higher than three impellers with 0.5inch impeller.

### 4.6.5 Power Input, Power Output and Efficiency Calculations

From the simulation, the discharge head can be calculated from equation (2). Table 3 below shows the discharge head for various pump impeller designs.

| Impeller | Average Discharge velocity (ft/s) | Discharge head (ft) |
|---|---|---|
| 1 | 49.1 | 37.4 |
| 3 (0.2 spacing) | 67.3 | 70.3 |
| 3 (0.5 spacing) | 51.1 | 40.5 |

Table 3: Discharge Head for Various Design of Multidisc Pump

The discharge head is lower compared to the previous design centrifugal slurry pump. This is due to the plane disc impeller which has no vanes where the circulation of fluid may occur

- 3 impellers with 0.2 inch spacing power calculation example are shown as below:

When the average discharge velocity is 67.3 ft/s, by using equation (2), the discharge head for the pump is;

$H = 67.3^2 / (2 \times 32.2) = \mathbf{70.3ft}$

From equation (7), pump output or hydraulic horsepower, $P_{out}$ (hp) is the liquid horsepower delivered by the pump;

Q = 1287.3GPM, H =70.3ft

$$P_{out} = \frac{1287.3 \times 70.3 \times 1.23}{3960} = \mathbf{28.1hp}$$

The power input for the pump will be the same as the previous design centrifugal slurry pump since the motor used is the same. Therefore the power input is **30kW (40.23hp)**

From equation (9), the pump efficiency;

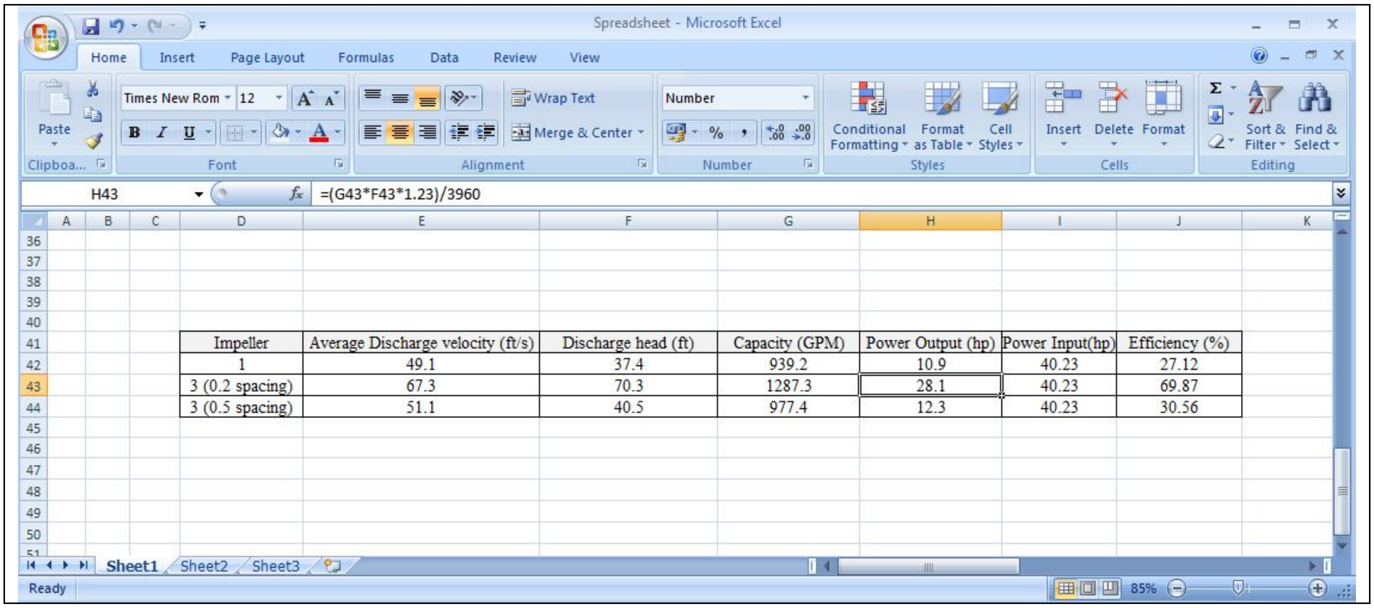
$\eta = 28.1 / 40.23$

$= 0.6985 \approx$ **69.87%**

The result for the other impeller is shown in the Table 4 below and Figure 22 shows the spreadsheet used for calculation.

| Impeller | Average Discharge velocity (ft/s) | Discharge head (ft) | Capacity (GPM) | Power Output (hp) | Power Input(hp) | Efficiency (%) |
|---|---|---|---|---|---|---|
| 1 | 49.1 | 37.4 | 939.2 | 10.9 | 40.23 | 27.12 |
| 3 (0.2 spacing) | 67.3 | 70.3 | 1287.3 | 28.1 | 40.23 | 69.87 |
| 3 (0.5 spacing) | 51.1 | 40.5 | 977.4 | 12.3 | 40.23 | 30.56 |

Table 4: The Efficiency of Various Impeller Design



| Impeller | Average Discharge velocity (ft/s) | Discharge head (ft) | Capacity (GPM) | Power Output (hp) | Power Input(hp) | Efficiency (%) |
|---|---|---|---|---|---|---|
| 1 | 49.1 | 37.4 | 939.2 | 10.9 | 40.23 | 27.12 |
| 3 (0.2 spacing) | 67.3 | 70.3 | 1287.3 | 28.1 | 40.23 | 69.87 |
| 3 (0.5 spacing) | 51.1 | 40.5 | 977.4 | 12.3 | 40.23 | 30.56 |

Figure 22: Spreadsheet Software Used in Calculation

It is known that the efficiency of pump with no vanes will be lower [5]. Therefore, to increase the efficiency of those pumps, the impeller should be multiplied. Based on the results, the efficiency for three discs impeller is higher compared to only one disc impeller. But, the efficiency for three discs impeller with smaller spacing between discs is much more higher compared to three discs impeller with higher spacing between discs. First of all, this is because of the working principle of disc pump as cited before.

In addition, when there is more spacing such as one disc impeller and three discs impeller with 0.5inch spacing, there will be more circulation of fluid in the pump. This is also a common problem in centrifugal pump with small number of vanes.

From the results, when the capacity of the pump is higher, the discharge pressure which is the head will be lower. This trend can be seen from the entire centrifugal pump where the head versus capacity graph has a negative slope.

There have been several errors during calculation and simulation. For example, there are pressure losses in piping system and fittings. Therefore the actual value for head is lower.

Viscosity correction factor should be applied to calculation since the fluid pumped having higher viscosity compared to water.

There are several assumptions made for this project. They are:

- No irreversible losses in the system
- The pump is working on the sea level
- Net positive suction head available is higher required. Therefore no cavitations.
- The vortex breaker not occur at the suction inlet
- The temperature of water is same before and after leaving the pump**.**

From the results, 3 discs impeller is more efficient compared to only 1 disc impeller. For the 3 discs impeller, it is more efficient when the spacing between discs is smaller. The highest efficiency pump is the 3 discs impellers with 0.2inch spacing pump. Therefore, the design with 3 discs impeller with 0.2inch spacing is chosen to be more suitable for this project.

# CHAPTER 5

# CONCLUSION & RECOMMENDATION

## 5.1 Conclusion

Disc pump has numerous advantages when pumping liquid with high amount of solid content and air entrained, in addition to pump high viscous fluid such as slurry.

From the results and simulation, the pump with three discs impeller is more efficient. Therefore, when the number of disc impeller in the pump is higher, the efficiency expected also will be higher. For the three discs impeller, when the spacing is smaller it is more efficient. Therefore the final design should have 3 discs impeller and smaller spacing between discs.

In conclusion, the objective of this project which is to design a Multidisc Pump with higher volumetric flowrate is proved since the discharge volumetric flowrate for Multidisc pump is higher compared to centrifugal pump in the same application. Moreover, the efficiency of the Multidisc Pump also can be determined which satisfy the second objective for this project.

## 5.2 Recommendation

The recommendation for this project is to improve in the calculation part. They are many assumptions have been made. Thus the accuracy will be lower. For the design, the Author should consider each corner of the pump such as the volute shape in order to increase the efficiency for the pump. Next in the simulation part, the error can be reduced while using the software but this will consume more time.

Stress analysis can be applied to the pump design to determine the crucial part with high stress level. Lastly, working model should be made to confirm the simulation and the results.

# CHAPTER 6

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