



Faculty of Engineering

**Pump System Optimization through Pumping System Assessment Tool
(PSAT): Case studies in Johor and Sarawak Manufacturing Plants**

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**Master of Engineering
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A thesis submitted

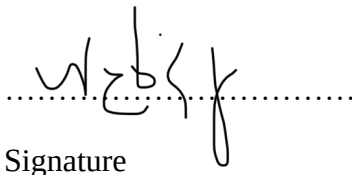
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DECLARATION

I declare that the work in this thesis was carried out in accordance with the regulations of Universiti Malaysia Sarawak. Except where due acknowledgements have been made, the work is that of the author alone. The thesis has not been accepted for any degree and is not concurrently submitted in candidature of any other degree.



Signature

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ABSTRACT

Pumping System Assessment Tools (PSAT) is a tool that can be used to assess the pump system. Targeted pump in Plant A and B had been identified to apply energy savings. Through PSAT, end user can calculate electricity consumption, pump optimization rating and annual savings consumption and savings opportunities that can be done through PSAT. In this research stud, Plant A which is located in Peninsular of Malaysia, is glove manufacturing plant. Meanwhile, Plant B which is in Sarawak is cement manufacturing plant. Targeted pump for both plants has been identified to apply the PSAT. Condition of the pump, lifespan, running hours were taken into considerations before selecting the targeted pump. To optimize pump parameters, further analysis was conducted on the pump using PSAT. Annual savings potential for the existing performance of the system can be compared with the optimal condition that is suggested by the PSAT database. Annual savings potential for Plant A is found to be RM 7,500 meanwhile for Plant B, about RM 16, 500 annual savings potential that can be saved. Through this research study, pump system limitation to collect the data is the research limitation as some parameters were not available during walkthrough energy audit and data collection. The present of PSAT is great tools to be used to assist user to do preliminary assessment for each of the pump system available in the plant. Condition of the pump was varied by increasing the flow rate of the pump which will lead to higher pump efficiency, thus bring saving opportunities to the user.

Keywords: Energy savings opportunity, Pump system, Annual savings, PSAT, Walkthrough energy audit

Pengoptimuman Sistem Pam melalui PSAT: Kajian kes di Kilang Pembuatan Johor dan Sarawak

ABSTRAK

Alat Penilaian Sistem Pam (PSAT) telah digunakan untuk menentukan peluang penjimatan yang boleh dilakukan melalui sistem pam di kilang pembuatan. Pam ialah salah satu pengguna elektrik tertinggi di loji, pam sasaran di Loji A dan B telah dikenal pasti untuk menjimatkan tenaga. Melalui PSAT, pengguna boleh mengira penggunaan elektrik, penarafan pengoptimuman pam dan penggunaan penjimatan tahunan dan peluang penjimatan. Dalam kajian ini, Loji A yang terletak di Semenanjung Malaysia adalah kilang pembuatan sarung tangan. Manakala, Loji B yang berada di Sarawak ialah sebelum memilih pam yang disasarkan. Potensi penjimatan tahunan untuk prestasi sedia ada sistem boleh dibandingkan dengan keadaan optimum yang dicadangkan oleh pangkalan data PSAT. Anggaran penjimatan bagi Loji A didapati RM 7,500 manakala Loji B, kira-kira RM 16,500 potensi penjimatan tahunan yang boleh diperolehi kerana terdapat beberapa parameter tidak tersedia semasa audit tenaga panduan dan pengumpulan data. PSAT adalah alat yang boleh digunakan untuk membantu pengguna membuat penilaian awal bagi setiap sistem pam yang terdapat di loji.

Kata kunci: *Peluang penjimatan tenaga, Sistem pam, Penjimatan tahunan, PSAT, Audit tenaga*

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LIST OF ABBREVIATIONS

ac	Alternating current
ANSI	American National Standard Institute
BEP	Best efficiency point
BEMS	Building Energy Management System
CO	Carbon Monoxide
CO ₂	Carbon Dioxide
cP	Centipoise
dc	Direct current
EMS	Energy Management System
FLA	Full Load Amps
HL	Head Losses
H _s	Static Head
HVAC	Heating, Ventilation and Air-conditioning
I/C/F/EMS	Industrial, Company, and Factor Energy Management System
NO _x	Nitrogen Oxide
NPV	Net positive suction head
N _s	Specific Speed
N _{ss}	Suction Specific Speed
PSAT	Pumping System Assessment Tool
SDGs	Sustainable Development Goals
SO ₂	Sulphur Dioxide
UNIDO	United Nations Industrial Development Organization

CHAPTER 1

INTRODUCTION

1.1 Study Background

Energy is a main need for different purposes in industrial facilities around the world. For economic competitiveness and employment, energy has become a crucial factor in current faster economic growth. With the recent global population, with rapidly increasing energy needs, this scenario must be addressed to avoid shortage of energy in the future. In addition, it is important to raise awareness of its adverse impact on the environment. In rational use of energy concept, main aims are reduction of how the energy has been used and optimum usage of the limited resources.

This project was conducted in two manufacturing plants, Plant A is an Examination Nitril Powder Free Glove Plant which is in the southern coast of Malaysia, Johor and Plant B is producing cement product which is in Sarawak. These two manufacturing plants are consuming energy the whole year, running all machines 24 hours in 365 days a year. Thus, energy savings is crucial to survive between production and profit.

In this thesis, both factories have been chosen. An energy audit and collection of data throughout the whole plant were conducted to see any potential machine for energy savings opportunities. Survey, questionnaire to the person in charge of the machines has been conducted to understand further how the machines are operated.

The United Nations Industrial Development Organization (UNIDO) has provided free tools called PSAT which are Pumping System Assessment Tools to be used for manufacturers or organizations. The pump system operations can be analysed through

PSAT. Potential savings in energy and cost can be calculated with the data obtained from Hydraulic Institute Standard and the motor performance can be obtained from Motor Master. Based on field-measured data, the application is allowing users to calculate savings opportunities in their pumping system. Annual energy usage and their costs for existing and optimal equipment also can be provided through PSAT.

Many benefits can be obtained by using PSAT, one of them is the efficiency of the system can be established and potential in energy savings can be quantified. Through these, our economics can be examined, and the effect of energy can be observed at different operating cases.

Benefits of PSAT include establishing system efficiency, quantifying potential energy savings, examining the economics and impacts of energy at different operating scenarios. Although PSAT will not provide users how to improve the systems, it does allow users to identify the pumping systems for further investigation. In pump systems, there are few symptoms that can be observed by using PSAT pre-screening, for example throttle-valve control for the system, damage or cavitation noise in the system, and continuous operation to support the process. It is challenging to stop the production to meet the production demand as some of the pump is working non-stop and no preventive maintenance being taken.

In most industries, pumps are used to by transferring fluids for processes. It also provides motive forces in hydraulic systems, which pump used to provide cooling to all machineries.

In the glove manufacturing industry, which is Plant A in this study, the pump is very important for the plant. Pumps are working to supply chilled water into all tanks in the production area. Pumps are also used to supply back used water to the chiller system.

Cooling process is important to produce the glove as it may cause defects in the production goods. In the glove industry, the temperature of the tank is crucial as the whole process depends on the temperature given by the tank. Plant B which is CMS Cement Plant Industries Sdn.Bhd, targeted pump is located at water cooling pump house. Pump is running 24 hours in for the whole year and is being used to provide chilled water to all of the tanks in production.

Energy walkabout through the plant has been conducted to assess the pump type, specification, function, and details in both plants. Each specification, process and user needs has been studied to collect all the data needed by the PSAT.

PSAT is a tool that can help to monitor and analyse the pump performance across the plant despite the type of pump. As a result, efficiency of pump and motor, annual energy usage, cost for both existing and optimal condition, annual savings and rating pump optimization can be estimated.

1.2 Problem Statement

In plant manufacturing, high electricity consumption is due to inefficient machinery usage, improper planning, and preventive maintenance. Lack of proper machinery maintenance causes reduction of their lifespan. Improper planning of the machine usage also causes high consumption of electricity. There are many various ways to reduce electricity consumption and one of them is through pump system optimization. By improving efficiency of the pump and to extend the lifespan of the pump, it is important to achieve optimal performance for the pump. Preventive maintenance and monitoring the condition of the pump is crucial to mitigate any potential impact that might cause a drop in pump performance. As an extension of our research, we are motivated to apply a pumping system

assessment tool (PSAT) to targeted pumps to obtain reduction in energy consumption in manufacturing plants. Reduce energy consumption in industrial plant will contribute to achieve one of the Sustainable Development Goals (SDGs) for UNIDO which is climate action. UNIDO has promotes to reduce carbon dioxide emission in industrial sector and other greenhouse gas emissions. This is including sustainable energy solutions, uptake of resource-efficient technologies and practices and cleaner production in industrial processes. Thus, the study to reduce energy consumption through pump system optimization is important as a part in climate action.

1.3 Research Hypothesis

In this study, PSAT tool application will be giving a positive impact to the pump system as energy opportunities can be obtained. If the machineries are being used in optimum condition, the energy consumption will be reduced. If the pump system is not running in its optimum condition, the lifespan of the pump will also be decreased and consuming power. Total amount of greenhouses gases, primarily carbon dioxide, CO₂ and other carbon compounds was emitted direct or indirectly by individual, organization and also mostly by any industrial activities. In return, this will cause global warming potentials. There is statistically significant reduction in carbon footprint resulting from energy-saving initiative.

1.4 Objectives of Research

The objectives of this study are:

- i. To analyse and evaluate performance of selected two (2) pumps in Johor and Sarawak by using Pumping System Assessment Tool (PSAT).

- ii. To investigate the impact of varied flow rate on the pump performance.
- iii. To propose system optimization for energy savings opportunities for the pumps

1.5 Scope of Research

To comply with the objectives, there are several considerations and constraints that need to be considered. They are:

- i. Pump was selected based on condition of all pumps available in the plant. Only two (2) pumps were selected in this research after assessment had been done. These pumps were selected as both pumps are the oldest pump and were running with highest voltage among the other pumps exist in respective plant. Older pumps have been in operation for a long time and are often part of legacy systems. They might not have been replaced or upgraded due to budget constraints or because they were considered reliable. Older pumps may be less energy-efficient due to outdated technology and wear and tear. Identifying and upgrading these pumps can lead to significant energy savings. Replacing all pumps in a plant can be expensive and cost consuming for the end user. Studying and upgrading older pumps can provide a cost-effective way to improve energy efficiency without a complete overhaul. Older pumps may offer the most immediate and substantial energy-saving opportunities since newer equipment is often designed to be more efficient from the start. Since the management are trying to adopt sustainability goals and commitments, thus by upgrading the older pump within the plant, make less efficient equipment aligns with these goals by reducing energy consumption and greenhouse gas emissions.

- ii. Downtime of machineries that may affect the production line. Since the production output demand is important to the operation, it is difficult to have machine downtime. Thus, this study was carried out when the pump is running, and no downtime is allowed.
- iii. The flow rate of the pump needs to be measured manually. During energy walkabout, we observed that some of the flow meter was not installed properly. This has caused some delay to assess the condition of the pump and we need to install portable flow meter to the pump to know the flow rate running.

1.6 Chapter Summary

This thesis is divided into five chapters. Chapter 1 describes the overall intention of the study. In this chapter, background of study, research objectives including scope of research are included. Chapter 2 is the literature review revolving on the study that is gathered from confided thoughts research, publications and reports worldwide. This chapter also discusses about energy savings opportunity, pumping system, the main components of the centrifugal pump, best efficiency point (BEP) in pump system, PSAT software including its layout and overview regarding the tool. Chapter 3 explains about the methodology used in completing this project. The main activities in this project include walk-through energy audit at targeted industry and experimental process at industry. Chapter 4 will be discussing on the results and findings of the project. The result gained from the preliminary data collection, taking parameters of the current process pump and from PSAT Software. Discussion on the results will also be reported in this chapter. In the last chapter, Chapter 5, the conclusion will be made based on the data analysis from Chapter 4. Some recommendations on further works will also be provided in Chapter 5.

CHAPTER 2

LITERATURE REVIEW

2.1 Energy Savings Opportunity

It is about a 33% increase in world marketed energy consumption projected from 2010 to 2030. Total world energy is expected to rise from 82,919 Zettawatt (ZW) in 1980 to 116,614 ZW in 2000, about 198,654 ZW (Abdelaziz et al., 2011) as shown in Figure 2.1.

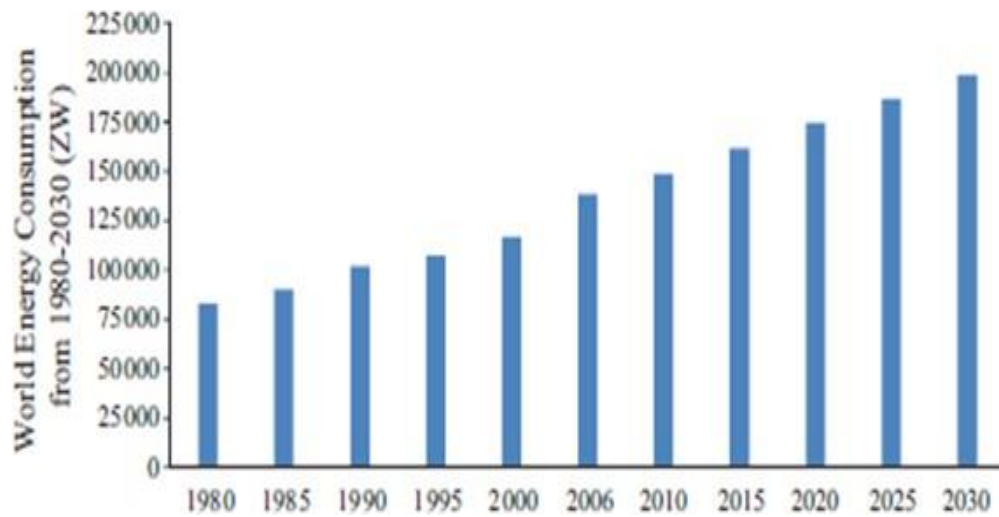


Figure 2.1: World Marketed Energy Consumption from 1980 to 2030 (ZW) (Abdelaziz et al., 2011)

Many industries like manufacturing, agriculture, mining, and construction are consuming energy for bigger range activities like processing and assembly, space conditioning, and lighting (Abdelaziz et al., 2011).

In the next 25 years, worldwide industrial energy consumption is projected to grow from 51,275ZW in 2006 to 71,961ZW in 2030 by an average of 1.4% per year (Abdelaziz et al., 2011) as shown in Figure 2.2.

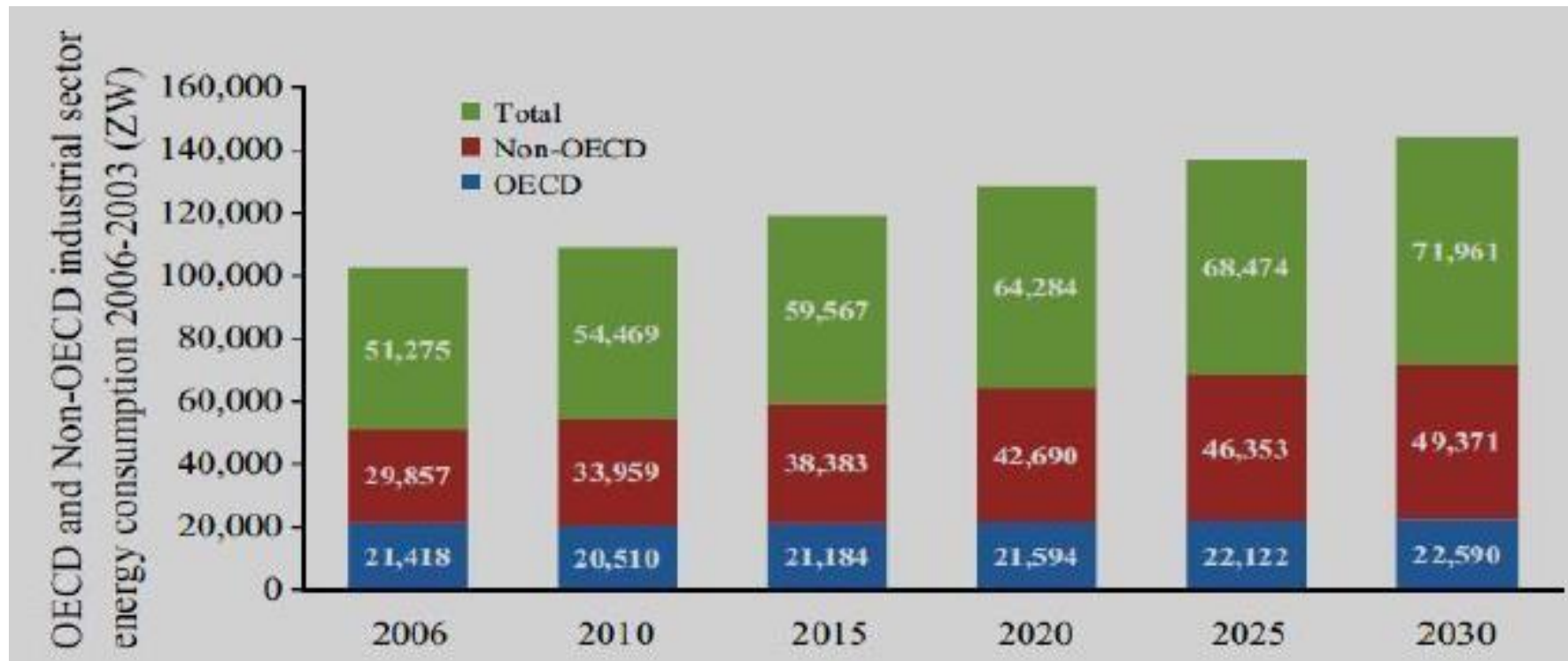


Figure 2.2: OECD and non-OECD industrial sector energy consumption from 2006 to 2030 (Abdelaziz et al., 2011)

From the figure, it can be seen that, from the year 2006 to 2030, shows there is an increasing in energy demand proposed for nations outside the Organization for Economic Cooperation and Development (non-OECD). Greenhouse gas emission, GHG, has been contributed by all various industries. It has been proved in a previous study, 10-30% of GHG emissions can be reduced either with implementation of a few energy savings options with less or zero cost to the industrial sector (Ghaddar & Mezher, 1999). In any building or manufacturing plant, energy savings on major equipment or installed drive can decrease the usage of energy consumption. In 2006, worldwide industrial energy consumption was projected to grow. By an average of 1.4% per year from 51,275 ZW to 71,961 ZW in 2030 for over the next 25 years (Abdelaziz et al., 2011).

In the literature of energy consumption, economic growth and environment are getting attention from the past few decades. The Environmental Kuznets Theory (EKC) is the first theory which reveals a relationship between economic growth and environment. The theory postulates that with the increase in economic growth, the environmental degradation increases till it crosses the peak level (Ang, 2007; Banday et al., 2014; Banday and Ismail, 2017; Saboori et al., 2012). Increase in global temperature is due to increase of consumption of fossil fuels. Previous study output can be seen as being conducted for Asian economies (Qingquan et al., 2020); China (Ahmad et al., 2018); OECD (Ahmad et al., 2020); Pakistan (Chandia et al., 2018); Finland (Kunnas & Myllintaus, 2017); Malaysia (Saboori et al., 2016); BRICS (Danish et al., 2019) and also developing countries (Sapkota & Bastola, 2017). To cope with deteriorating environment, all these G7 nations have been adopting different eco-innovation strategies. The relationship between the economic growth and carbon emission has been testified in few researches (Shafik, 1994; Heil and Selden, 1999; Friedl

and Getzner, 2003; Dinda and Coondoo, 2006; Coondoo and Dinda, 2008; Managi and Jena, 2008).

At rate of three to four percent, the global economic growth keep on rising. In an excessive use of fossil fuel gives impact on CO₂ emission arising thus result in higher risk to environment (Bennet & James, 2017). In the context of the pollution, from an academic perspective, scholars have examined the international trade-CO₂ nexus. In recent years, there are debate from association between international trade and CO₂ emission (Ding et al., 2020). In 1960, percentage of gross domestic product (GDP) is only twenty-three percent, which compared to year 2017 which GDP has reached fifty-eight percent, as reported by the World Bank (2019). The cause of rising international trade to the world might increase economic opportunities, but this is also an impact to the ecological damage for example release of CO₂ emission (Hasanov et al., 2018).

Current amount of energy and emission in China's cement industry has been also analysed for cement production in various levels from year 2011 to year 2030. Potential for cumulative final energy savings of 27.1 to 37.5 Exajoules, EJ was calculated under the best practice scenarios, which also give reductions by 1.2 Gigatonnes of CO₂ for the energy emissions (Ke et al., 2012). Similarly, for the cement industry in Taiwan, under the technical diffusion scenario, around 25% savings for electricity and 9% savings for fuels by year 2035 were estimated in eighteen energy efficient technologies adopted over there (Ke et al., 2012).

Cement industry is observed as the second most energy-intensive user in european country. It is important to understand its promising potential for energy efficiency improvements. In the European market, Levelized Cost of Energy (LCOE) has analysed the energy savings used by modified split pumps and this energy solution has shown promising

results. The calculated LCOE is ranging from 0.440 USD/kWh up to 0.194 USD/kWh depending on the considered hybrid energy solution (Ke et al., 2012). Either petroleum or chemical plants already made a commitment to reduce consumption of energy or usage in terms of thermal, electrical, and mechanical energy.

There are so many ways that can be done to reduce the energy consumption either in plants or in buildings. For example, energy management system (EMS), or building energy management system (BEMS), for building that uses refrigerating, lighting systems, and motors. These are the tools used to study energy saving effects in an industrial plant. Study has conducted and shows that, in a year, about 11.39% to 16.22% savings effects of BEMS can be increased. For effective energy savings, these key EMS functions can be identified (Lee & Cheng, 2016). Negative impact of the trade and economic growth on the environmental quality environment can be reduced by applying cleaner production technologies (Zhang et al., 2017).

In this world, the cement industry has become one of the most intensive industries for energy consumption. It is that estimated at about 2% energy consumption by the cement industry contributes to global primary energy consumption. Processes for cement manufacturing either wet or dry processes cost a large amount in energy cost because high thermal energy is needed for the wet and dry process and electrical energy for operation of motors. Grinding machines, fans system, conveyors of the machine, and equipment used motor driven has caused high energy cost. Cement industry has operated at about 50% of thermal efficiency. High amount of carbon dioxide released by cement manufacturer. Around all man-made CO₂ emissions which are about 20% and 10% of global warming

which comes from human activities are contributed by this cement manufacturing industry (Atmaca & Yumrutas, 2014).

In the US industrial sector, they offered the greatest net positive value (NPV), NPV-positive energy-efficiency opportunity by 3.65 quadrillion BTUS and not only did they consume more energy than others combined. This value is calculated for processes, support systems and buildings themselves. The smallest percentage which is 18 percent is calculated as a percentage of end-use consumption. \$447 billion savings has been obtained throughout this opportunity even though the value of investments is \$113 billion. Around 67 percent was provided by varied industrial processes of the potential energy sector such as motor, buildings, and steam systems (Creyts et al., 2010).

Industries that uses energy, which is like a system that used steam, motors, and building, could provide up to \$164 billion for a \$34 billion investment. Other than standardized industry-specific process equipment, boilers and the like are easier to pursue although these represent a smaller opportunity. Waste-heat recovery, better insulated distribution systems, valve and fitting improvements are all the opportunities that can be done in industry. Factories can become more efficient if their power components like load requirements, pumps and process equipment and systems that use electric motors as their system drive are matching (Creyts et al., 2010).

In Malaysia, about 20% of rubber producing industries are consuming (Saidur and Mekhilef , 2010). Figure 2.3 shows different consumption of energy used in process heating in rubber industries in Malaysia. About 38 billion GJ (energy) of coal, natural gas, and nuclear energy U.S electric utilities converted into 12.3 billion GJ (electricity) of electricity in 2012 as reported by Agency Usei, 2015. In past researches, the importance of eco-

innovation in abating CO₂ emissions has been widely recognized (Alvarez-herranz et al., 2017, Jiao et al., 2018, Mensah et al., 2018).

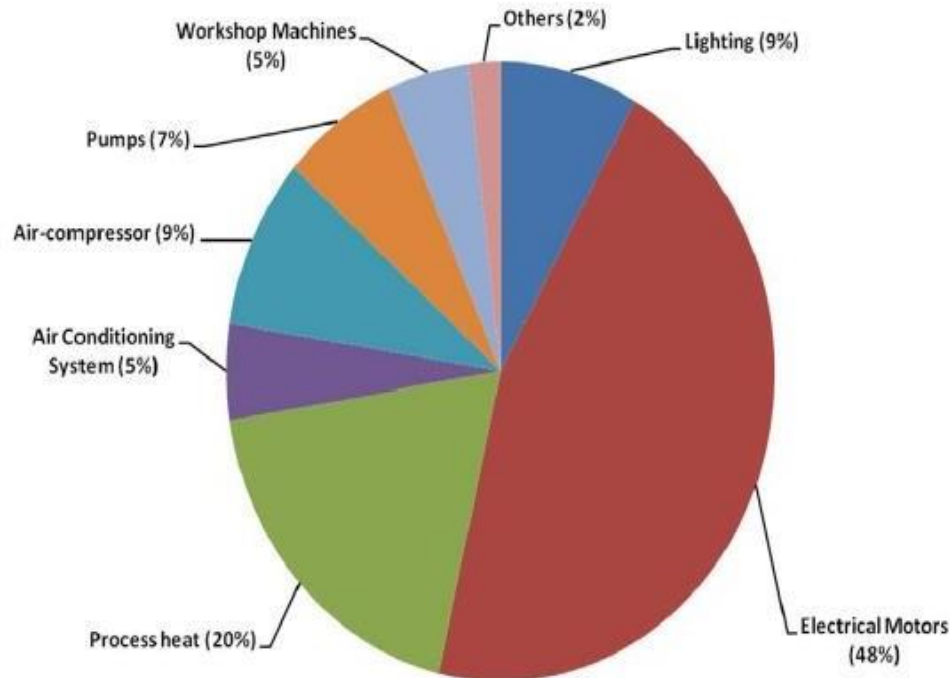


Figure 2.3: Energy used by process heat in rubber industries (Saidur & Mekhilef, 2010)

By enhancing energy efficiency and energy costs, energy productivity can improve environmental quality (Nataly Echevarria Huaman & Xiu Jun, 2014). High economic output at the cost of less energy consumption can be generated at an increase in energy productivity. CO₂ emissions can be controlled by energy productivity through three channels. First channel is in energy productivity which will bring decline in the per unit of energy consumption for production purposes. Second, it will help to lower energy costs and lastly it will lead to decreasing in oil imports, thus lesser emission. Thus, for achieving sustainable development, energy productivity is remained as a top priority (Choi et al., 2017).

In recent years, many studies had been done for the energy efficiency in industry through pump system. A hybrid solar hot water and air source heat pump (HSAHP)

combined heating system, which was mainly composed of solar collectors, an air source heat pump, a water pump and a water tank was established (Long et al., 2021). Azeotropic dividing wall column (ADWC) a thermally coupled was explored for the separation of industrial wastewater to recycle the organic solvent tert-butanol. To improve the released heat duty quality of the condenser achieving the energy saving, heat pump technology was used (Yanget al., 2019).

2.2 Pumping System

Pump is one of the systems that is variously used in the manufacturing industry. Pumps are functioning to provide machine cooling and lubrication, transferring fluids and providing the motive force in hydraulic systems. In industrial systems, pumps represent 27% of the electricity in most manufacturing plants, commercial buildings, and municipalities. All of these are relying on pumping systems for daily operation.

Heating process, ventilation, and air-conditioning (HVAC) systems in commercials during heat transfer can provide water. Meanwhile, in most manufacturing plants, they use pumps for water and wastewater transfer and treatment, and for land drainage. Manufacturing plants range in size from fractions of a horsepower to several thousand horsepower, since they serve such diverse needs. Figure 2.4 shows typical pumping systems components which contain five basic components: pumps, prime movers, piping, valves, and end-use equipment like heat exchangers, tank, and hydraulic equipment.

Motors have been used to drive most pumps, whether they will use direct current (DC) motors or alternating current (AC) motors. AC motors have low cost and high reliability compared to DC motors. In high run applications, operating costs can be significantly reduced by improved motor efficiencies. Motor controller is a subcomponent

of a pump motor. From a low-power circuit, the motor controller will receive signals such as an on-off switch and connect or disconnect the high-power circuits to the primary power supply from the motor. Operating costs can be reduced significantly when motor efficiency is improved.

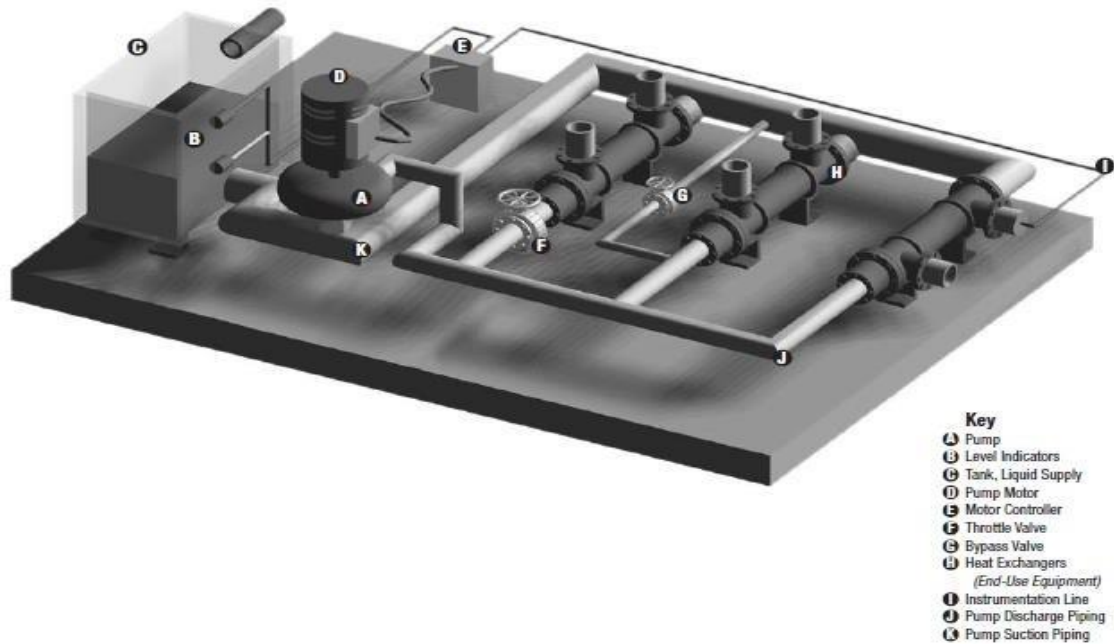


Figure 2.4: Pumping System Components (Stefan et al., 2016)

Another important part of the pump is piping. All critical aspects including dimensions, material type and cost of the piping have been considered. Piping functions as a fluid carrier from the pump to the point of use. Pipe sizing is important as all the three aspects are related. As the pipe diameter gets larger, at a specified flow rate, the flow resistance decreases. In high pressure, when operating the systems, it is easier to route and install small pipes with thinner walls than large-diameter pipes.

Valves are another component in the pump system, which is used to control pump flow. Valves might have different positions either shut or open, and others might be used to

throttle flow. Some factors can be used while selecting the valves, like ease of maintenance, reliability, leakage tendencies, cost and frequency to open and shut the valve while it is being used. Valves also function to isolate equipment or regulate flow. For operating purposes or maintenance, an isolation valve is used to seal off a part of the system. Flow regulating valves will restrict the flow through a system branch which is also called a throttle valve or allow flow around it, which is also called a bypass valve. Flow resistance is controlled by the throttle valve either by increasing or decreasing flow resistance across it. Meanwhile, a bypass valve allows flow by increasing or decreasing the flow resistance across it. A check valve protects equipment from pressurization from the wrong direction by allowing fluids to move in only one direction and it also helps to keep fluids flowing in the right direction. Type of valve used at the discharge of many pumps is called a check valve. This kind of valve is helping to prevent flow reversal when the pump is stopped.

Pumping systems can either operate as a closed-loop or open-loop system. A closed-loop system recirculates fluid around a path with common beginning and end points, meanwhile an open-loop works by transferring fluid from one point to another since an open-loop system has input and output. Normally, closed-loop systems do not have to contend with static head loads, for example a cooling water system, unless at different elevations, when there are vented tanks. The frictional losses of system piping, and equipment are the predominant pump load in the closed-loop systems. Meanwhile, because of elevation and tank pressurization needs, an open-loop system requires pumps to overcome static head requirements, for instance, a mine dewatering system as it uses pumps to move water from the bottom of a mine up to the surface. In such cases, the static head is often the dominant pump load.

System operating cost in pump system is calculated as per below equation, the amount of fluid power that a system consumes is a product of head and flow according to this equation in horsepower:

$$HP = PQ/1714 \quad \text{(Equation 2.1)}$$

where,

HP = horsepower

P = Pressure (Pascal, Pa)

Q = flow rate m³/s

To generate these head and flow conditions, due to pump inefficiencies the motor power is required. In addition, the flow conditions need to be high. Pump efficiency was measured by dividing the fluid power by the power of the pump shaft for directly coupled pump or motor combinations. This is the brake horsepower (bhp) of the motor (Stefan et al. 2006).

One of the energy savings through the pump system can be obtained from pump impeller trimming. The extent of pump oversizing in commercial office buildings in both heating and cooling systems had been assessed (Singh & Mitchell, 1998). Objective of the study is to estimate the potential energy savings. To match the actual system head requirements, the pump's impeller was trimmed. Results of the study reported that pumps in large commercial office buildings are oversized and if the pump impeller is trimmed to the proper size, savings can be obtained. From 14 large commercial buildings, average annual energy savings were close to 28,000 kWh, results of an annual cost savings came up to \$1,220 and with an average trimming cost of \$1,540 per building, which give 1.3 years payback. To conclude, besides short payback, pump trimming is an energy conservation

opportunity as it is easy to implement since no disruption is caused to normal building or system operation.

2.3 Centrifugal Pump Overview

In the pump system, there are two kinds of pumps, which are centrifugal and positive displacement pumps. These two pumps are classified on how energy was added to a fluid. For positive displacement pumps, the fluid is squeezed directly, meanwhile centrifugal pumps are also known as “rotodynamic pumps”. This pump can level up the fluid and the pressure will be converted from kinetic energy. For a centrifugal pump, normally it is used for pumping lower viscosity fluids and for higher flows specifically. Compared with positive displacement pumps, centrifugal pumps normally suffer less wear and only require fewer part replacements. It also can be operated under a broad range of conditions and if preventive measures are in place, the risk of damage is low if there is improper valve positioning. One of the characteristics of a centrifugal pump is having a variable flow of pressure relationship. Less flow will be generated when the system pressure is low. Performance curve that plots the flow rate as a function of head (pressure) can be used to describe a centrifugal pumps’ flow or pressure relationship (Stefan et al., 2006).

Through the impeller’s vane tips, fluid will enter the rotating impeller rapidly along its circumference. The key component of a centrifugal pump is the impeller. Result of the impeller action will increase the fluid’s velocity and its pressure. This will also direct it towards the pump outlet. The impeller consists of a series of curved vanes. Figure 2.5 shows the type of impeller. These are an enclosed impeller, which is normally sandwiched between two discs, an open impeller, and semi-open impeller, which is backed by a single disc. For fluids entrained solids, open and semi-open impellers are preferred.

When selecting the pump type, as each pump has its own function and speciality, most suitable pump was selected during assessment. Centrifugal pump which is also an end suction pumps are a type of centrifugal pump with a single impeller, one inlet and one outlet with the same or different diameters (Moore, 2020). Based on the Second Law of Newton, this pump type is produced mechanical energy from pressure and kinetic energy (Bala, 2011; Agarwal, 2012).

End suction pump can work in turbine mode without modifying its mechanical elements (Ismail et al., 2014). Comparing with rate of pumping, end suction pumps have a BEP that needs a larger fluid flow rate. End suction pump type is also helping to lower total operating cost as this type of pump is equipped with impellers and can work more efficiently (Ismail et al., 2015). During end suction pump operates, axial load is present and the axial load which is present due to force will bring impact on performance of the pump and its efficiency (Pehlivan & Parlak, 2019). According to Natsu (2019) this type of pump is much easier to repair and due to low flow disturbance end suction pump has high efficiency compared to other pump type.



Figure 2.5: Impeller Types: Open, Semi-Enclosed and Enclosed (Sam and Jason, 2022)

Impeller constant and high-speed rotation contributes to the efficiency of the centrifugal pump. In order to maintain a specific flow rate, greater resistance and higher pressure is needed, as the centrifugal pump will become increasingly inefficient if the viscosity feeds are high. Thus, it can be concluded that, centrifugal pump type is suitable for low pressure, high capacity and pumping application of liquids with viscosities between 0,1 and 200 cP. In addition to this, a centrifugal pump cannot provide suction when dry as it must be primed with the pumped fluid. Thus, this pump type is not suitable for any application where the supply is intermittent. There is a case that led to damage and premature failures of the pump, which are slurries such as mud and any high viscosity oils. Therefore, any pump medium that is sensitive to the slurries can be damaged by high speed of centrifugal pumps' impeller.

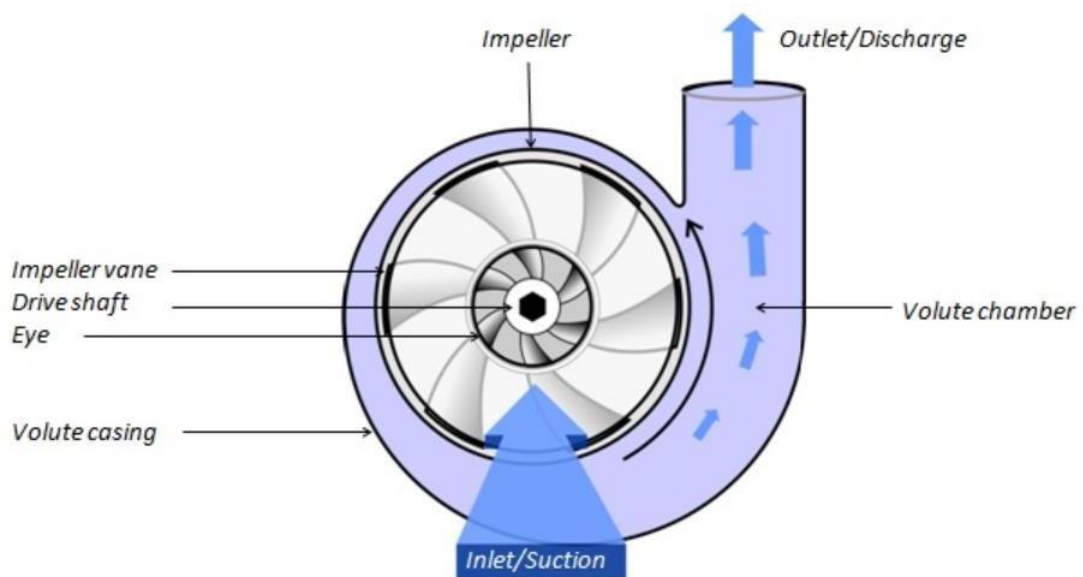


Figure 2.6: Volute Case Design of Centrifugal Pump (Sam and Jason, 2022)

For pump casing in centrifugal pumps, there are two designs basically, which are volute and diffuser design. These designs must be able to transfer the flow of the fluid into

a controlled discharge at pressure. Figure 2.6 shows volute case design for the centrifugal pump. The impeller is offset in a volute casing, which is effectively creating a curved funnel by increasing cross-sectional area towards the pump outlet. This design causes the fluid pressure to increase towards the outlet compared to diffuser case design, where the fluid pressure increases as fluid has expelled between a set of stationary vanes surrounding the impeller as per Figure 2.7.

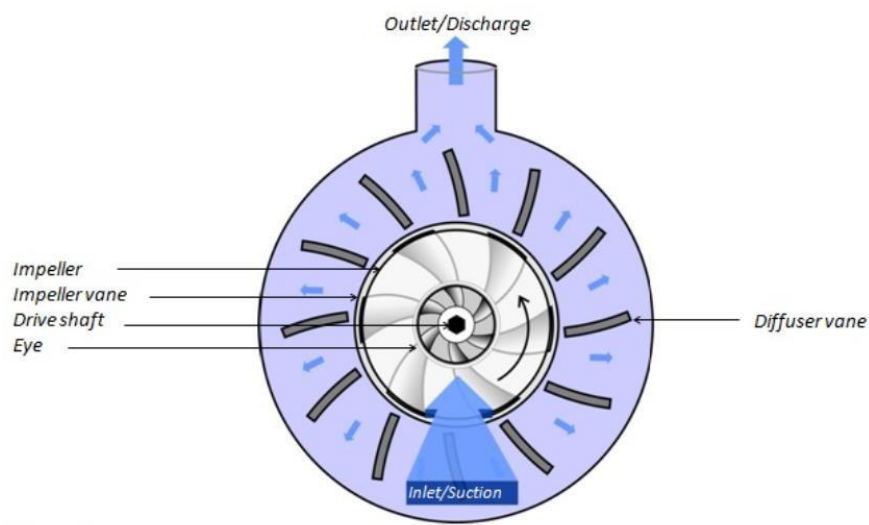


Figure 2.7: Diffuser Case Design of Centrifugal Pump (Sam and Jason, 2022)

In diffuser case design, as fluid has expelled between a set of stationary vanes, pressure of the fluid will increase in the impeller surrounding. Compared with dilute case design, diffuser case design is more efficient as it can be tailored for specific applications. For the volute case design, it is suitable for applications that involve entrained solids or high viscosity fluids, as it is advantageous to avoid the added constrictions of diffuser vanes. It is also resulting in greater wear on the impeller and drive shaft as the design is asymmetry.

There are many energy- efficient improvements opportunities that can be made for the pumping system. For example, proper maintenance, machine monitoring, controls,

proper sizing, impeller trimming, avoiding throttling valves, correct pipe sizing, surface coatings or polishing and improvement of sealing.

2.4 Theoretical of Pump Operation

Relation between pump head or the head added to the system, to the flow rate and its characteristic curve is defined as a pump curve. The pump performance curve provides information on how the pump reacts to variations in operational parameters such as total head and flow rate. It is possible to optimize pump selection and operation for maximum efficiency by studying the pump curve (Dey, 2022). At different flow rates, this curve is indicative of the ability of the pump to add head. Point at which the pump curve crosses the system curve is a system operating point and it represents the static lift and head losses due to friction and minor losses. The operating point can easily be found when these two curves are superimposed as per Figure 2.8.

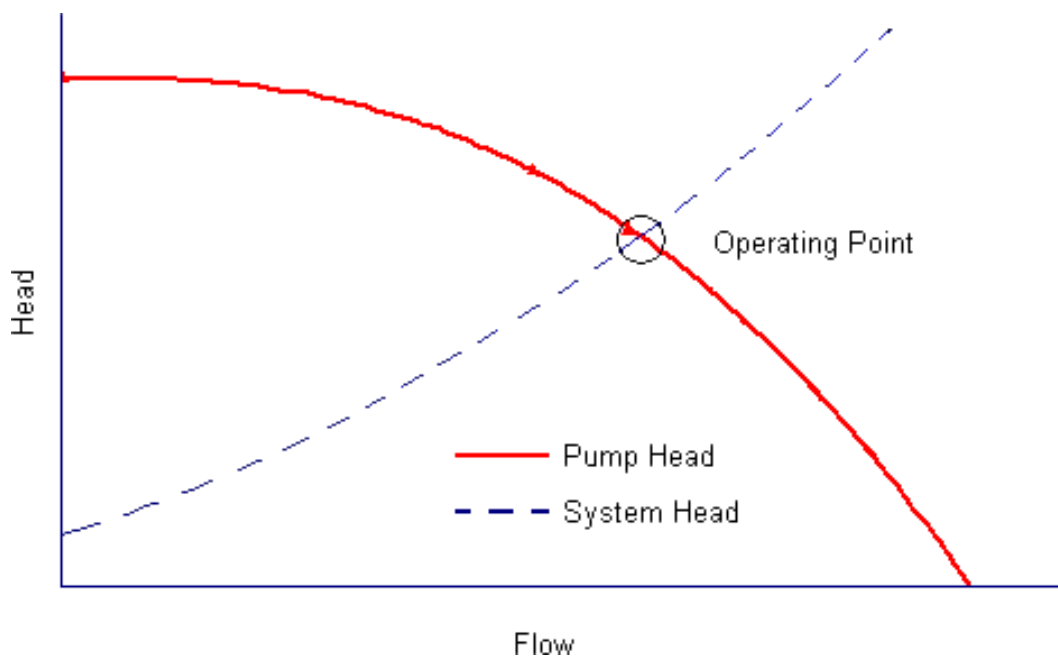


Figure 2.8: System Operating Point (Sam and Jason, 2022)

The static head (H_s) and head losses (H_L) will vary as the water surface elevations and demands throughout the system change. The pump characteristics curve will remain constant as the location of the system curve changes. Thus, result in a shifting operation over time.

When selecting a pump size, the flow rate (Q) and the head (H) of the desired operating point which is characteristic of the curve is important, including the pump size, speed of rotation and also number of stages (KSB Pumps Ltd, 2005).

In affinity laws, any speed and any diameter can be determined as the pump's characteristic curve is fixed for a given motor speed and impeller diameter. Refer Figure 2.9 for the effect of relative speed on the pump curve.

$$\frac{Q_1}{Q_2} = \frac{n_1}{n_2} \quad (\text{Equation 2.2})$$

and

$$\frac{h_1}{h_2} = \frac{n_1}{n_2} \quad (\text{Equation 2.3})$$

Where:

Q = Pump flow rate (m^3/s , cfs)

h = Pump head (m, ft.)

n = Pump speed (rpm)

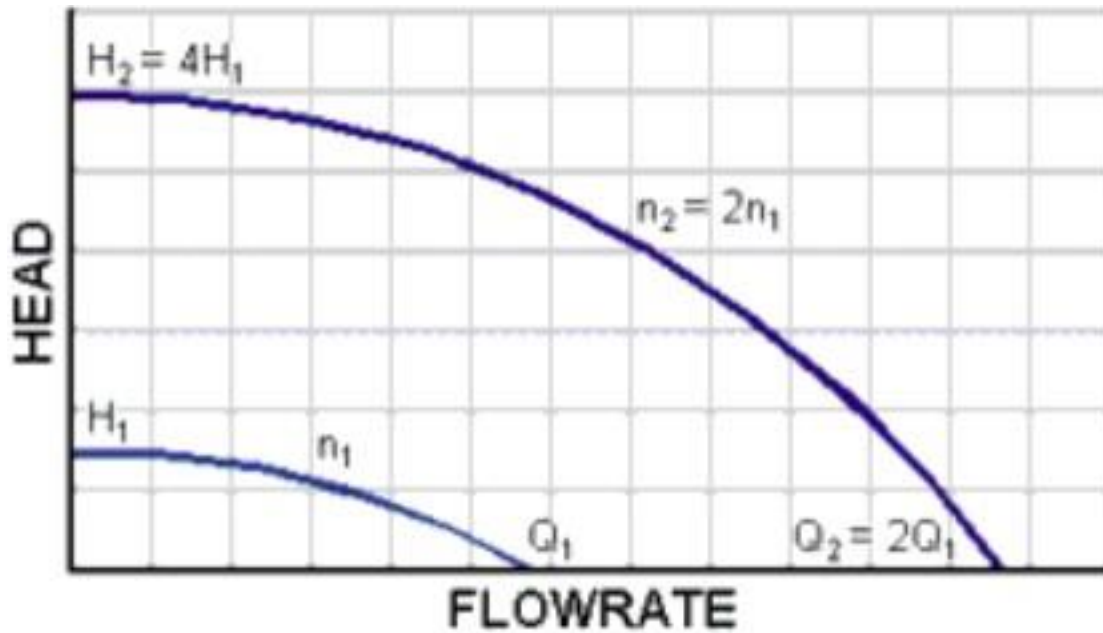


Figure 2.9: Effect of Relative Speed on Pump Curve (Sam and Jason, 2022)

2.5 Best Efficiency Point

Figure 2.10 shows the relation of best efficiency point (BEP) with pump curve. A point along a pump line curve is called a BEP where the pump efficiency is the highest. Good system performance must be operated as close to BEP as possible. Some pump values are derived from BEP, for instance, for most centrifugal pumps, the preferred operating range or POR is defined as 70% to 120%. Specific speed (N_s) is derived coefficient at various pump sized, N_s is allowing a comparison of impeller when its operating data differ which the parameters are flow rate and head at BEP, rotational speed. An optimum impeller design and the corresponding pump characteristic curves can be classified used the n_s . Meanwhile, suction specific speed (N_{ss}) is an indicator of impeller inlet geometry and discharge characteristic. This coefficient is a function proportional to the speed, pump flow rate and Net positive suction head required (NPSHr).

When the pump is operating as close to BEP as possible, it means that the pump is at its most efficient, which means it is using input power as efficiently as possible and can reduce the operating costs. By running close to its BEP, the pump lifespan is increased, and the maintenance cost can be reduced.

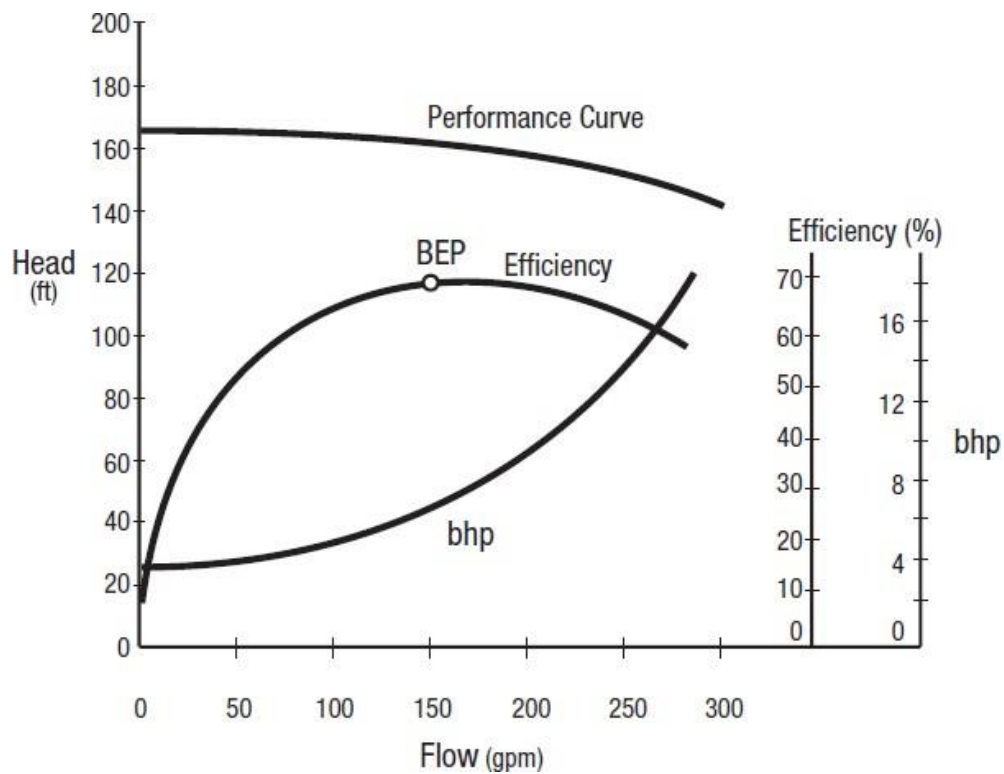


Figure 2.10: Relation of BEP point with Pump Curve (Sam and Jason, 2022)

2.6 Pumping System Assessment Tool (PSAT)

Pumping system assessment tools is an application provided by the United Nations Industrial Development Organization which is designed to improve pumping system efficiency and measuring potential savings opportunities in both Ringgit Malaysia and electrical energy savings. This tool is used to assess the efficiency of pumping system operations. It uses an achievable pump and efficiency algorithm from Hydraulic Institute.

All the data needed for the PSAT are including pump style, system of units, number of stages, pump and motor speed, motor nameplate ratings, operation duty, energy cost rates, flow rate of the system, its pump head which is calculated from user-supplied pressure and line dimensional data and electric power or current or voltage. PSAT can search for the highest motor efficiency available to drive the found pump at that point. In addition, annual energy use and energy costs for existing and optimal equipment can be provided by the tools including pump system potential annual energy savings. McEntyre (2008) research paper had applied PSAT which focusing on achieve energy savings in pumping system operation and through this application, user can identify improvement opportunity in pump system.

According to most research, energy efficiency can be achieved through pump system optimization in industrial plant. We are inspired to propose pump system optimization by using PSAT. This tool will help industrial user to do preliminary assessment for their pump and beneficially will suggest energy savings opportunity for the pump. To see the outcome of the PSAT, evaluation of two (2) targeted pumps in Johor and Sarawak was conducted. This research will be further explained in the upcoming chapters. It is important to address pump optimization rating, total energy consumption and cost savings obtained through PSAT assessment.

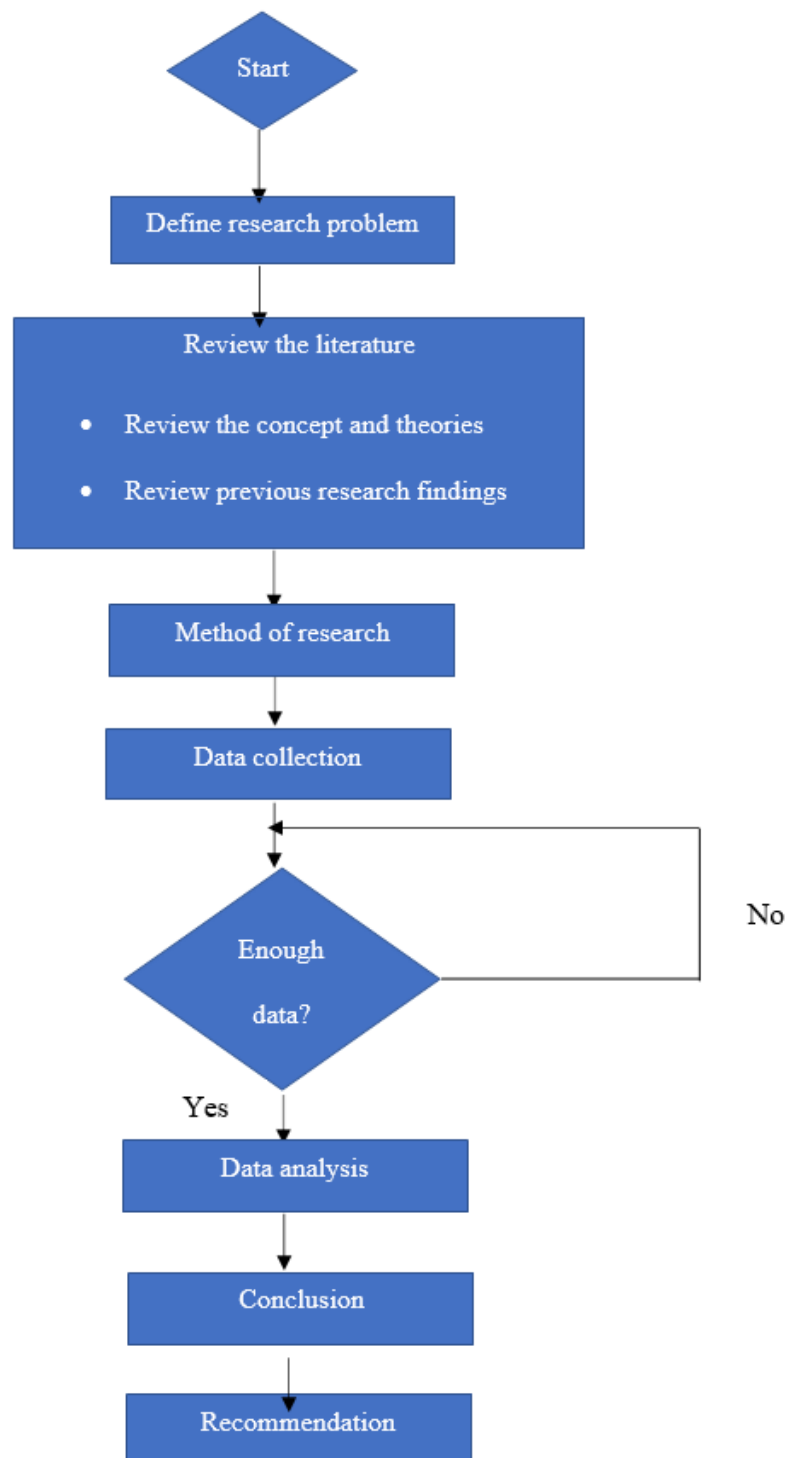
2.7 Chapter Summary

This chapter have reviewed and studied energy savings opportunity, pumping system, centrifugal pump overview, theoretical of pump operation, best efficiency point and pumping assessment tool (PSAT).

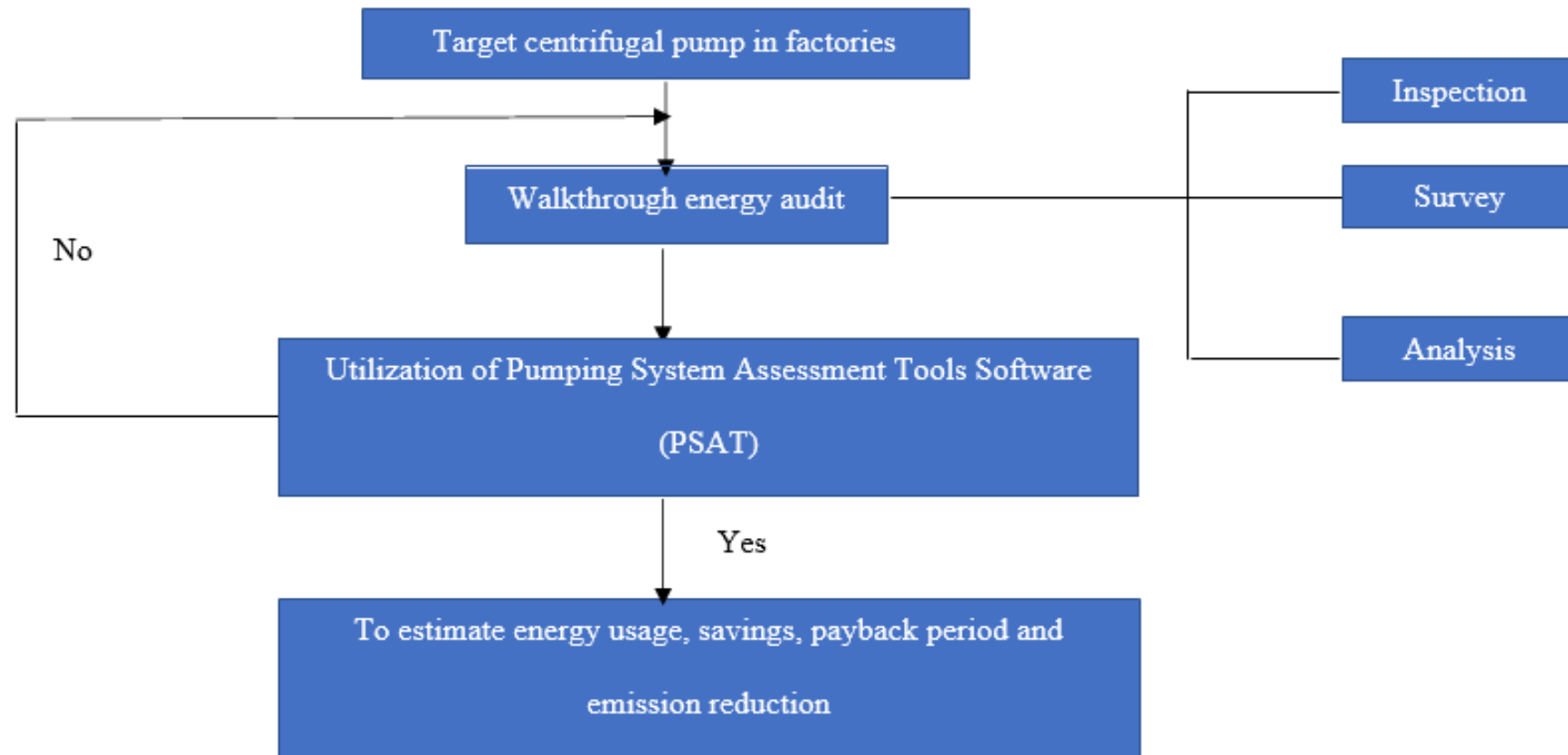
CHAPTER 3

MATERIALS AND METHODS

3.1 Flow Chart of the Methodology



3.2 Workflow of the Experiment



3.3 Walkthrough Energy Audit

Energy audit is an inspection, survey, and analysis of energy flows for energy conservation to reduce the amount of energy input into the system without negatively affecting the output. In the energy management area, energy audit is the key for decision-making. Thus, it can be said that this way is reliable, and the approach is systematic in the industrial sector. In this approach, management or any organization can analyse its energy use and see the area where energy can be reduced or optimized. This also can help to determine wastage of energy use. In this energy audit, each system in the plant can be assessed to analyse main contributors of the plant.

A walkthrough energy audit has been conducted at Plant A, an Examination Nitrile Powder Free Glove Plant, whereas Plant B is a Cement Manufacturing Plant. As PSAT has been used for this research study, each specification of each part of the pump needs to be checked and verified. Several readings were recorded to monitor the consistency of the reading and all the data are used to analyse in PSAT software. Only two (2) pumps were selected in this research after assessment through all pumps available has been done. Condition of the pump, lifespan, running hours were taken into considerations before selecting the targeted pump.

In every energy audit, a walk-through visit and meetings with plant officials and interviewing of site operators are the most important steps to do. Walk through visit, facility utility bills such as electricity gas, production output versus consumption and other operating data will help to become familiar with the building and any waste of energy can be identified through out energy audit (Barma et al., 2017).

To collect the on-site information, the following tools are needed. One is the flow meter, which is needed to measure wet, saturated, and superheated steam and water flow correctly. Clamp-on power meter is important to measure the power consumption, current consumption, load factor and power factor can be measured. This tool can record measurements without disrupting the production process. Other than that, is ultrasonic flow meter which is used to measure flow rate of pump accurately.

3.4 Preliminary Data Collection

Table 3.1 shows is the Preliminary Datasheet for Pump System at Plant A in Johor and Plant B in Sarawak Initial assessment has been done during walkthrough energy audit. This is important to get a clear picture of the pump condition and to understand the targeted pump well. The information is also important to be analysed in PSAT later. Each nameplate of the manufacturer, information on current, voltage, horsepower was gathered during the assessment for both selected pumps.

Table 3.1: Preliminary Datasheet for Pump System at Plant A in Johor and Plant B in Sarawak

Facility:	Date:	Time:	Pump Operation Hour:
System:			
Pump Name Plate	ID/Set		
Pump Style			
Service			
Name plate pump system	RPM		
Number of stages			
Motor Name Plate	NA		
Power	kW		
Power	HP		
Full load speed	RPM		
Full load efficiency	%		
Rated voltage	Volts		

3.5 Calibration of Ultrasonic Flow Meter

An unknown measurement, which is the standard was used to compare with the measurement when using the instrument. Comparison between the standard with measurement value is defined as calibration. Accuracy of the meter is depending on the flow rate of measurement, and it is critical to obtain the accuracy of the reading. Flow calibration is needed to all water meters that are built upon the transit-time ultrasonic principle. The meter can be calibrated across a specific known flow range with the deployment of a reference standard flow measurement device. The calibration operation was conducted at

Pump Bench University Malaysia Sarawak (UNIMAS). The result of calibration show that the flow rate of the pump bench is 2500 l/h and reading for ultrasonic flow meter approximately matches with reading of pump bench which is 2509 l/h.

A static flow offset can be determined from a method of calibrating an ultrasonic flow meter (Alvin, 1996). By subtracting the static flow offset and one half the dynamic flow offset from the flow speed, an adjusted flow speed can be determined.

A pair of transducers are the part of ultrasonic flow meters that are located through a conduit on either side of the flow path of a fluid flowing. There is a direction of the fluid flow in the line between the transducers. Transit time of an ultrasonic packet will increase in the upstream and decrease in the downstream. Flow rate is affected by the amount which the transit time as both is directly proportional. Ultrasonic flow meter has fixed offsets and normally, it can be measured by having the meter measure the flow rate and it is necessary that there is no fluid flow through it. Any reading is an offset when the meter reads zero, which is the value of offset is subtracted from all future reading. Adjustment of the potentiometer can be done manually to perform this process.

Figure 3.1 shows the calibration experiment conducted. In addition, calibration is also conducted to ensure silica gel used is compatible with the ultrasonic flow meter. Over time, the offset can be drifted, especially in the case of clamp on the transducer. Thus, the calibration method is necessary and a need to detect both static and dynamic offset automatically in an ultrasonic flow meter and compensate both offsets.



Figure 3.1: Calibration measurement

In Figure 3.2, flow rate of the pump is measured by using an ultrasonic flow meter. The instrument's TDS-100H ultrasonic flow meter comes with the capability of a full-size flow meter and is battery-powered. Clamp-on transit-time flow measurement principle has been used for this instrument. By using a pair of ultrasonic transducers, flow rate of liquid in a pipe is measured which is from outside of the pipe. In practice, the liquid should be full in the pipe while measuring the flow rate. It also must contain very little particles or bubbles. In addition, during measurement, it has been observed that there is no pressure drop, moving parts, leaking and contamination detected.

The TDS-100H is using advanced signal processing, low-voltage transmitting and small signal receiving with self-adapting. TDS-100H is capable of functioning as both an ultrasonic transmitter and receiver. On the outside of a closed pipe at specific distance from each other, the transducers are clamped. The upstream and downstream of the pump must be identified and put a mark on the pipe. The blue transducer is upstream, while the red transducer is downstream. V-method installation as per Figure 3.2 is highly used for pipe inner diameters ranging from 20mm to 300mm, and method is mounted in V-method where

the sound transverses the pipe twice. The suction pipe diameter is 230 mm and discharge diameter size are 125 mm.

In data reading on the flow meter shown in Figure 3.2, the measurement of flow rate of two identical pumps and reading of both pumps are approximately the same which is at 193 m³/h. Transducer type used is transducer B and it has its own range of pipe diameter which is 230 mm and discharge pipe diameter is 125 mm. The fluid flow direction can be checked by using transducers, if the direction of the flow will be from red to blue transducer, it indicates the positive value, meanwhile if the value is negative, the direction will be from blue to red transducer. From Figure 3.3, the negative sign is the flow of water which is from blue to red transducers and flow of water is from right to left toward the pump. Both identical pumps which are pump inlets 1 and 2 have velocity approximately the same at 1.47m/s.

In the instrument, the Q value is a signal quality. More accuracy can be achieved when a Q value is higher which means a higher signal to noise ratio (SNR). The Q value is in the range 40-90 under normal pipe condition, which the higher the value the better. In Figure 3.3, shows the signal is not good but does not give 0 value and manages to read the value of two pumps. Interference from other instruments and nearby devices causing the lower signal quality. For example, a power frequency transverter may cause strong interference while taking the measurement. In addition, the selected pipe section is difficult to conduct the measurement and transducers and the pipe are having bad sonic coupling. When encountering this issue while measurement, the pipe surface is polished again to ensure the surface is cleaner. The coupler is applied more to the pipe and transducers.

Reading shows that the R sign where the 'R' is displayed in the lower right corner of the LCD display, indicates the instrument is working properly. If the 'S' signal appears

which is 'S' indicates signal strength is not a good signal. Ultrasonic flowmeter works well when the signal strength shows from 400 to 900, where the higher the signal the better the result. From Figure 3.3(b) shows, the range obtained is 405-427 and this shows lower signal strength. Factors that contribute lower signal strength, S might be the same factors causing lower signal quality, Q. In general, the flow meter has three (3) totalize functions, but only POS totalizer set as most pipes will have flow in one direction only. Flow meter device will continuously integrate all the flow measure.



Figure 3.2: Installation of flow meter to calculate the flow rate of the pump



(a)

(b)

Figure 3.3: (a) Transducers Spacing (b) Flow rate Value

3.6 Data Measurement using PSAT

Figure 3.4 shows the PSAT main panel, which is the primary user interface. There are two independent conditions for the calculation to be performed and it will be calculated simultaneously. There is Condition A and B in the panel. There is a drop-down menu for user to select like type of pump, pump rotation, type of drive, unit used, line frequency, value of horsepower, motor rotation, type of efficient class, motor. During walkthrough audit, the parameters of the pump for Plant A and B were gathered and the values were analysed through PSAT.

There are three important buttons to be highlighted in this software. The buttons are “Estimate FLA” button (FLA= Full-Load Amps), Head tool button and “Fixed specific speed?” selector button. The FLA button is used to estimate motor output (shaft) power and if the motor nameplate is not provided, “Estimate FLA” will give the user a reasonable estimate for the specified motor speed, size, and its efficiency class. The pump style list used as per Figure 3.5.

For the Head tool button, the user will be able to open the head tool calculation. The type of measurement can be selected as per below button selection. Figure 3.6 shows Pump head calculation panel with suction and discharge pressure gauges. Meanwhile, Figure 3.7 shows pump head calculation for suction tank elevation, gas space pressure, and discharge line pressure. The user must determine which type of pump in terms of designation and choose correctly to get accurate head tool calculation.

Condition A

Condition B

Condition A

Condition B

Pump fluid
 End suction ANSI/API
 Pump rpm: 1780
 Drive: Direct drive
 Units: gpm, ft, hp
 Kinematic viscosity (cS): 1.00
 Specific gravity: 1.000
 # stages: 1
 Fixed specific speed? **YES**

Motor
 Line freq: 60 Hz
 HP: 200
 Motor rpm: 1780
 Eff. class: Energy efficient
 Voltage: 460
 Estimate FLA
 Full-load amps: 225.4
 Size margin, %: 0

Duty, unit cost
 Operating fraction: 1.000
 \$/kwhr: 0.0500
 Flow rate, gpm: 2000
 Head tool: Head, ft: 277
 Load estim. method: Power
 Motor kW: 150.0
 Voltage: 460

Field data
 Retrieve defaults Set defaults Copy A > to B >
 System curve tool: select below

Pump fluid
 Specified optimal eff (below)
 Achievable efficiency: 85.0
 Pump rpm: 1480
 Drive: Direct drive
 Units: L/s, m, kW
 Kinematic viscosity (cS): 1.00
 Specific gravity: 1.000
 # stages: 1
 Fixed specific speed? **NO**

Motor
 Line freq: 50 Hz
 kW: 145.0 <<Spec
 Motor rpm: 1480
 Eff. class: Specified (below)
 FL efficiency, %: 95.8
 Voltage: 380
 Estimate FLA
 Full-load amps: 266.0
 Size margin, %: 0

Duty, unit cost
 Operating fraction: 1.000
 \$/kwhr: 0.0500
 Flow rate, L/s: 126
 Head tool: Head, m: 84.4
 Load estim. method: Current
 Motor amps: 264.0
 Voltage: 380

Field data
 Copy B < to A < Background information

STOP

	Condition A			Condition B		
	Existing	Optimal	Units	Existing	Optimal	Units
Pump efficiency	72.6	84.8	%	72.5	85.0	%
Motor rated power	200	200	hp	145	132	kW
Motor shaft power	192.4	164.8	hp	143.9	122.7	kW
Pump shaft power	192.4	164.8	hp	143.9	122.7	kW
Motor efficiency	95.7	95.8	%	95.8	95.6	%
Motor power factor	86.7	85.7	%	86.4	86.4	%
Motor current	217.1	188.2	amps	264.0	225.7	amps
Motor power	150.0	128.4	kW	150.2	128.3	kW
Annual energy	1314.0	1125.1	MWh	1315.5	1124.1	MWh
Annual cost	65.7	56.3	\$1000	65.8	56.2	\$1000

Annual savings potential, \$1,000: 9.4 9.6
 Optimization rating, %: 85.8 85.4

Log file controls:
 Create new log Add to existing log
 Retrieve log entry Delete log entry

Summary file controls:
 Create new summary file
 Existing summary files: CREATE NEW

Condition A Notes **Documentation section**
 Facility: System: Date: Application: Evaluator: General comments:

Condition B Notes
 Facility: System: Date: Application: Evaluator: General comments:

Figure 3.4: PSAT Panel Overview (UNIDO, 2016)

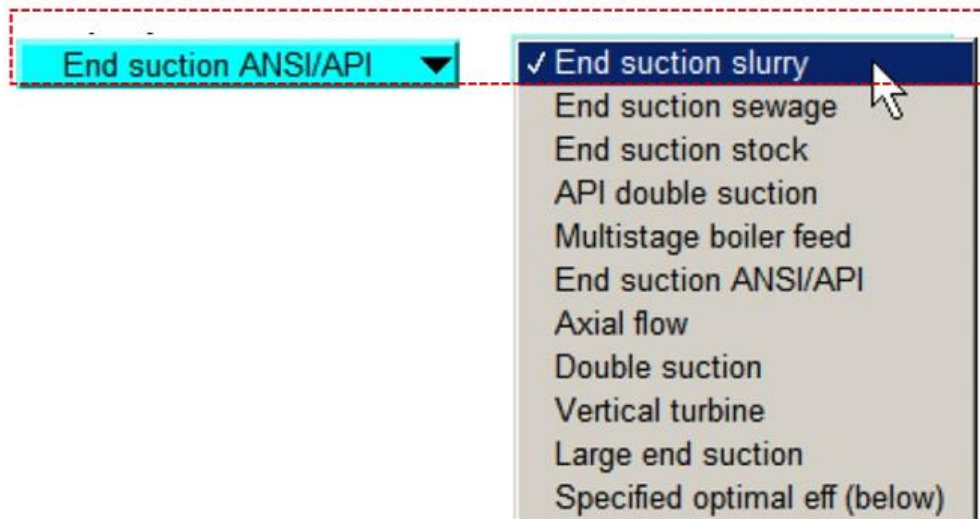


Figure 3.5: List of Pump Style Used in PSAT (UNIDO, 2016)

Click to access units converter tool

Type of measurement configuration
Suction and discharge line pressures

K_s represents all suction losses from gauge P_s to the pump
 K_d represents all discharge losses from the pump to gauge P_d

Suction pipe diameter (ID)	12.000	inches	Discharge pipe diameter (ID)	10.000	inches
Suction gauge pressure (P_s)	5.00	psig	Discharge gauge pressure (P_d)	124.00	psig
Suction gauge elevation (Z_s)	5.00	ft	Discharge gauge elevation (Z_d)	5.00	ft
Suction line loss coefficients, K_s	0.50		Discharge line loss coefficients, K_d	1.00	
Fluid specific gravity		1.000	Flow rate		2000.00 gpm
Don't update			Differential elevation head		0.00 ft
Accept and update			Differential pressure head		274.99 ft
Click to leave the main panel head unchanged			Differential velocity head		0.54 ft
Click to Accept and return the calculated head			Estimated suction friction head		0.25 ft
			Estimated discharge friction head		1.04 ft
			Pump head		276.81 ft

System of units: gpm, ft, hp

Figure 3.6: Suction tank elevation, gas space pressure, and discharge line pressure configuration (UNIDO, 2016)

Type of measurement configuration
 Suction tank elevation, gas space pressure, and discharge line pressure ▼

K_s represents all suction losses from the tank to the pump
 K_d represents all discharge losses from the pump to gauge P_d

Suction pipe diameter (ID) 12.000 inches Suction tank gas overpressure (P_g) 0.00 psig Suction tank fluid surface elevation (Z_s) 10.00 ft Suction line loss coefficients, K_s 0.50	Discharge pipe diameter (ID) 10.000 inches Discharge gauge pressure (P_d) 124.00 psig Discharge gauge elevation (Z_d) 5.00 ft Discharge line loss coefficients, K_d 1.00
Fluid specific gravity 1.000	Flow rate 2000.00 gpm

Don't update

Click to
leave the main panel
head unchanged

Accept and update

Click to
Accept and return
the calculated head

System of units: gpm, ft, hp	Differential elevation head -5.00 ft Differential pressure head 286.54 ft Differential velocity head 1.04 ft Estimated suction friction head 0.25 ft Estimated discharge friction head 1.04 ft Pump head 283.86 ft
------------------------------	---

Figure 3.7: Suction and discharge line pressure measurement (UNIDO, 2016)

The last important button that highlighted is “Fixed specified speed?” button selection. If the number of stages has been adjusted to improve achievable efficiency, “NO” can be selected. Other values in this software can be determined by the user as per targeted pump specification whether it must be measured or not, such as pump rotation, motor voltage and flow rate. According to Dey, 2013, the static suction head (h_s) can be obtained from lifting the liquid relative to the pump centreline and the value is positive if the liquid level is above and negative if vice versa. h_s value is also being called as suction lift. Meanwhile the value of static discharge (h_d) is measured by the vertical distance between pump centreline and the surface of liquid destination tank. In PSAT tool, this value can be observed in pump head calculation panel for value estimated suction friction head and estimated discharge friction head. Pump head value is important for the assessment. To make higher efficiency pump, higher pump head is used (Kim et al., 2008).

CHAPTER 4

RESULTS AND DISCUSSION

4.1 Targeted Pump at Plant A

Walkthrough audit has been done throughout this research. Figure 4.1 shows a graph of energy consumption percentage in Plant A. Pumps are contributing 22% of the plant energy consumption, which is the top three highest energy consumption of the plant. As the pump also consists of motors by themselves, thus there are two systems that can be targeted to do the savings.

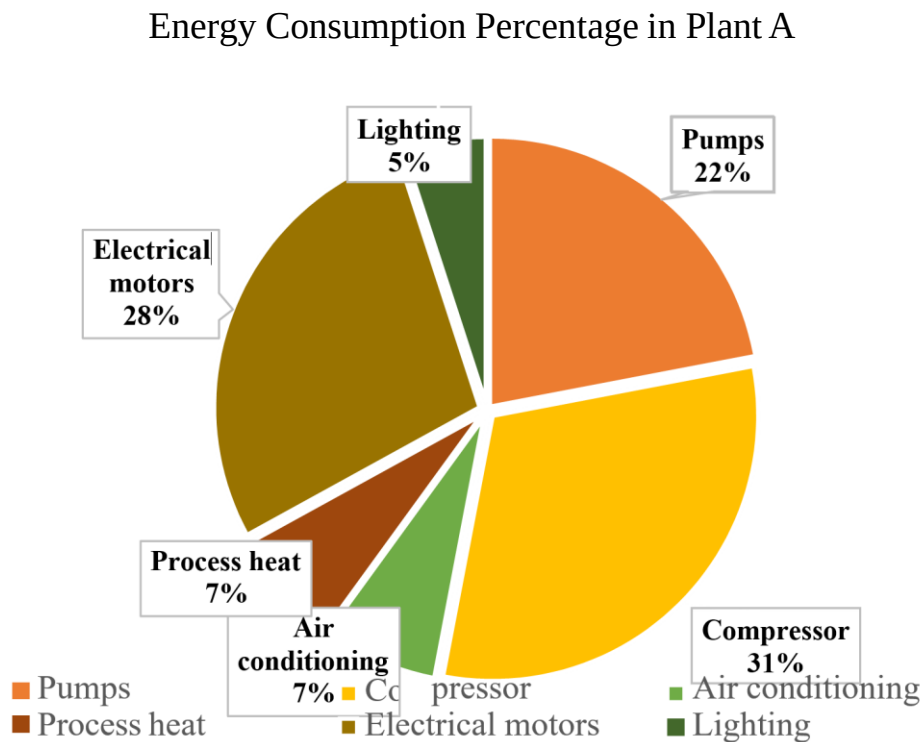


Figure 4.1: Energy Consumption in Plant A

Figure 4.2 shows annualized consumption in kWh from December 2013 until October 2016. Graph shows production output by glove pieces against total annual electricity

consumption. In order to gain manufacturing profit, production output must be higher than electricity consumption. From the graph, it can be seen from June 2016 to October 2016, electricity consumption is higher than output of the month. This has been caused by the stoppage of machineries. Plant downtime of the machineries could not support the production output. Thus, it is important to have preventive maintenance of the machine to ensure the machine is running in good condition.

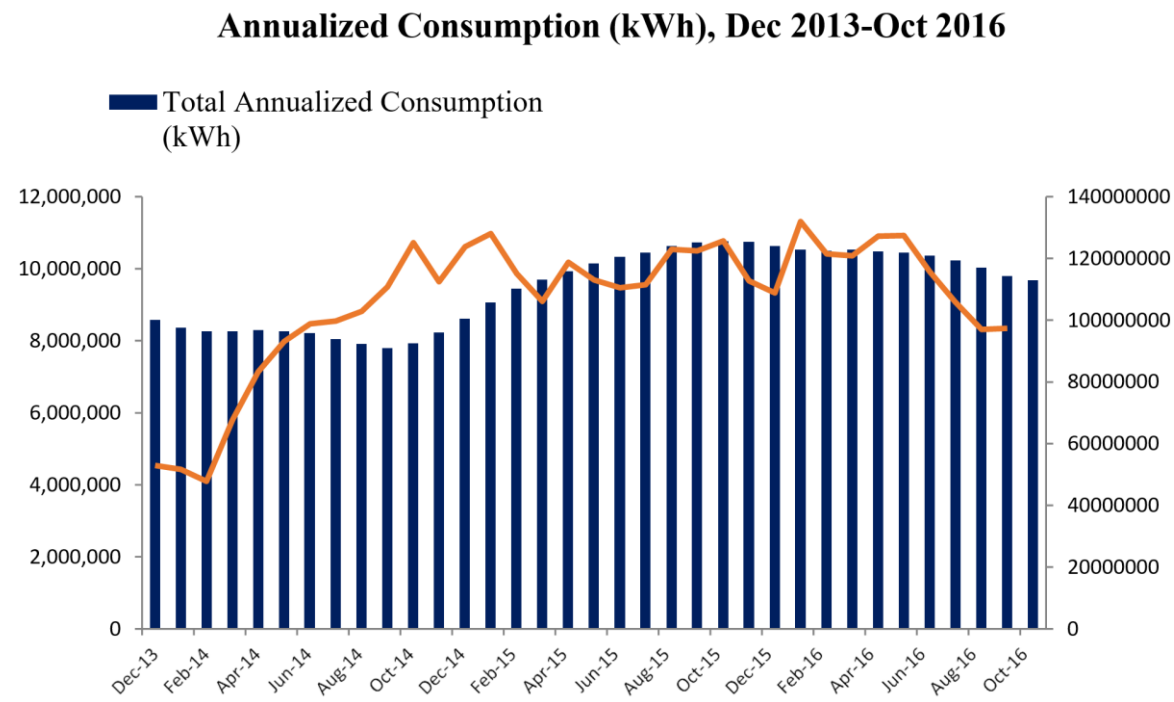


Figure 4.2: Annualized Consumption from Dec 2013 to Oct 2016

Figure 4.3 shows the relationship between energy consumption (kWh) against production output (pcs). The R^2 value is 0.8654, which shows that the higher the production output, the higher the energy consumption. This means that the energy consumption is increasing linearly with production output.

Energy Consumption (kWh) vs Production Output (Pcs)

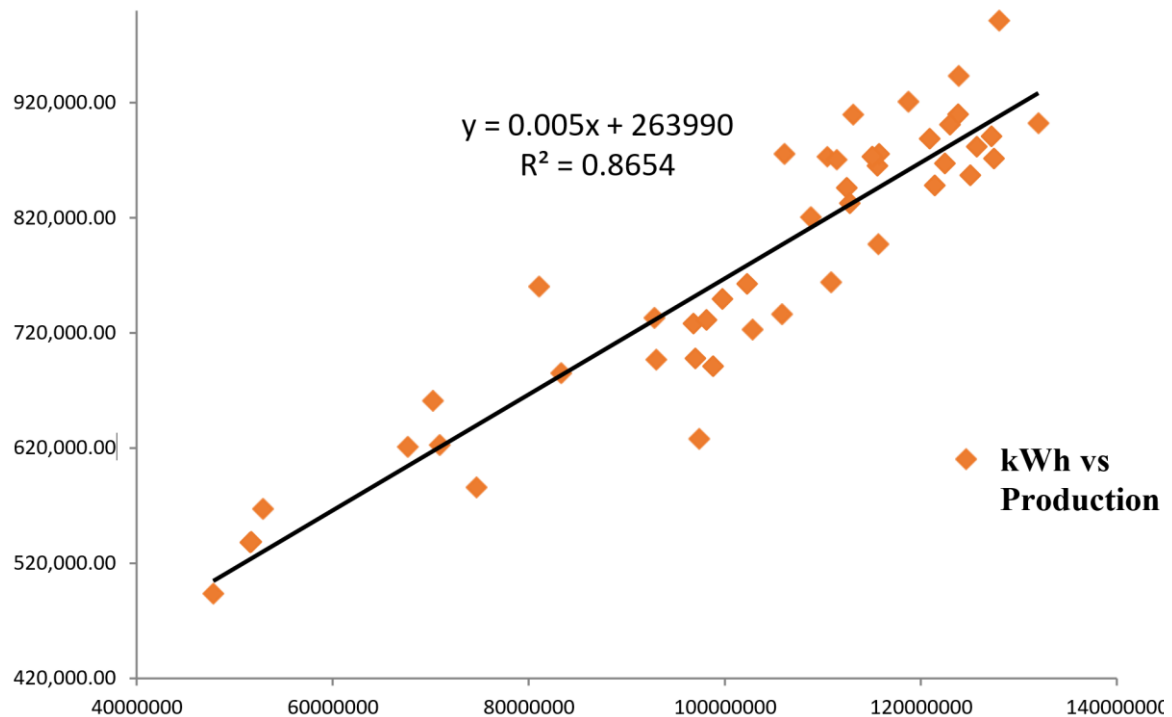


Figure 4.3: Energy Consumption (kWh) vs Production Output (Pcs)

It is important to have analysis on the baseline of the plant consumption. The power consumption and energy used by the organization will then be calculated based on its annual consumption. Any losses eliminated from the system will then contribute to the energy savings that can be obtained from the newly proposed operating conditions. This step is vital to determine whether the optimization will be much more drastic with a new capital investment involved or only with the newly proposed operating condition is sufficient to provide the organization with a reasonable savings with a short return of investment period.

For Plant A, pump chosen is a pump used to supply chilled water to chemical and leaching tanks, targeted pump as in Figure 4.4. As for the glove manufacturing process, chemical storage tanks which normally contain polymer if polymer-coated glove and

chlorine tank if the process is chlorine wash glove are very important to obtain good quality of final product. Thus, it is very important to maintain its respective temperature while production is running. Targeted pump in Plant A is running 24 hours, 365 days without any preventive maintenance taking place. Thus, whenever there is breakage of the pump, it will disturb the production. It is crucial for glove manufacturing to have chilled water in about ~ 6 to 7°C to be supplied in water and latex tank during production process to ensure the quality of the final product.



Figure 4.4: Targeted Pump at Plant A

The diagram illustrates a cooling system for Cooling Tower A. Water is circulated from the tower through a pump and a chiller to a production line. Key components and labels include:

- Cooling Tower A:** Located at the top left, with a height of 3.95 m.
- Pump:** A red dashed box highlights the pump area, containing a pump symbol and a pressure gauge showing 25 psi. The pump is labeled with specifications: **Pump : 11.3 A, 415 V, 0.85 PF**.
- Motor:** A red dashed box also highlights the motor area, containing a motor symbol and a pressure gauge showing 21 psi. The motor is labeled with specifications: **Motor : 11.0 kW, 15 HP, 50 Hz, 1455 rpm, 380-415 V, 21.5 A**.
- Chiller:** A rectangular unit labeled "Chiller 36°C" that cools the water.
- Production line:** A box on the right representing the industrial process being cooled.
- Flow and Pressure Labels:**
 - 6" pipes connect the tower to the pump and the pump to the tower.
 - 6" pipes connect the pump to the chiller and the chiller to the pump.
 - 4" pipes connect the chiller to the production line and the production line to the chiller.
 - Pressure gauges show 25 psi at the pump outlet and 21 psi at the motor outlet.
 - Temperature labels indicate $T_{Out} \sim 8-12^{\circ}C$ at the production line outlet and $T_{in} 23^{\circ}C$ at the production line inlet.
- Other Labels:** A vertical dimension of 0.155 m is shown near the pump, and a valve symbol is present on the line between the pump and the chiller.

Table 4.1: Preliminary Data for Pump System in Plant A

Facility: Chiller Water Pump	Date: 24 Feb 2017	Time: 1700pm	Pump Operation Hour: 24hrs/365days
System: Pump is used to pump back water from chiller to production			
Pump Name Plate	ID/Set	DA 125-20/C-5166	
Pump Style		Centrifugal pump	
Service		Pump water from chiller to production tank	
Name plate pump system	RPM	1450	
Number of stages		Single stage suction	
Motor Name Plate	NA		
Power	kW	11	
Power	HP	15	
Full load speed	RPM	1455	
Full load efficiency	%	20.1	
Rated voltage	Volts	415	

In Plant A, End suction ANSI/API is used. ANSI pump is the American National Standards Institute. For pump fluid part, pump rotation, which is 1450 RPM, type of drive, which is direct drive, units used for the measurement, which is m³/hr, m, kW. The kinematic viscosity value is 1.00 Cs, whereas the specific gravity value is 1.000.

Refer Figure 4.6 and Figure 4.7 for the pump head calculation. In this section, type of measurement configuration as per tank designed is suction tank elevation, gas space pressure and discharge line pressure. All the specifications for suction and discharge pipe diameter (ID), suction and discharge tank gas overpressure (Pg), suction and discharge tank fluid surface elevation (Zs) and suction and discharge line loss coefficients, Ks were inserted in head tool calculation below. In condition A, current flow rate is 165 m³/hr, giving its pump head 11.63m $\approx$ 11.6m as in Figure 4.8.

Pump head calculator 2007.vi

Type of measurement configuration: **Suction and discharge line pressures**

Click to access units converter tool

Schematic Diagram: A pump system with suction and discharge lines. Suction gauge P_s is at elevation Z_s . Discharge gauge P_d is at elevation Z_d . Loss coefficients K_s and K_d are indicated for the suction and discharge lines respectively.

K_s represents all suction losses from gauge P_s to the pump
 K_d represents all discharge losses from the pump to gauge P_d

Suction pipe diameter (ID)	304.6 mm	Discharge pipe diameter (ID)	254.0 mm
Suction gauge pressure (P_s)	59.3 kPa	Discharge gauge pressure (P_d)	172.4 kPa
Suction gauge elevation (Z_s)	1.52 m	Discharge gauge elevation (Z_d)	1.52 m
Suction line loss coefficients, K_s	0.50	Discharge line loss coefficients, K_d	1.00
Fluid specific gravity	1.000	Flow rate	165.0 m ³ /hr

Differential elevation head	0.00 m
Differential pressure head	11.55 m
Differential velocity head	0.02 m
Estimated suction friction head	0.01 m
Estimated discharge friction head	0.04 m
Pump head	11.63 m

System of units: m³/hr, m, kW

Important note about loss coefficients
 The loss coefficients used here apply to the velocity head in the line size represented by the suction and discharge pipe diameters at the points of pressure measurement.

If the loss elements are in different size lines than the points of pressure measurement, they need to be appropriately scaled. It is generally suggested that the losses be scaled in proportion to the 4th power of the diameter ratio. For example, if the discharge pressure is measured in a 12-inch header, and there is a 6-inch check valve with a nominal loss coefficient of 2 (applied to the 6-inch valve size), the K factor to use for the valve would be $2 \times (12/6)$ to the 4th power, or 32. The reason for this 4th power scaling is that the velocity varies with the square of the pipe diameter, and the velocity head (to which the loss coefficients apply) is proportional to the velocity squared.

Figure 4.6: Head Tool for Condition A for Targeted Plant A

Meanwhile, for Figure 4.7, which is pump head calculation for Condition B, the flow rate of the pump is increased from 165m³/hr to 208.8m³/hr as this is the flow rate for the targeted pump should be run as recommended by the pump manufacturer. As the flow rate of the pump is increased, the PSAT analysis in Figure 4.8 can be seen to compare the optimization percentage from both conditions.

Type of measurement configuration
Suction and discharge line pressures

Schematic Diagram:
A pump is shown between two gauges, P_s (suction) and P_d (discharge). The suction side has a gauge elevation Z_s and a loss coefficient K_s . The discharge side has a gauge elevation Z_d and a loss coefficient K_d . The flow rate is F .

Input Parameters:

Suction pipe diameter (ID)	304.8 mm	Discharge pipe diameter (ID)	254.0 mm
Suction gauge pressure (P_s)	59.3 kPa	Discharge gauge pressure (P_d)	172.4 kPa
Suction gauge elevation (Z_s)	1.52 m	Discharge gauge elevation (Z_d)	1.52 m
Suction line loss coefficients, K_s	0.50	Discharge line loss coefficients, K_d	1.00
Fluid specific gravity	1.000	Flow rate	208.8 m ³ /hr

Buttons:
 Don't update | Accept and update
 Click to leave the main panel head unchanged | Click to Accept and return the calculated head

System of units: m³/hr, m, kW

Important note about loss coefficients:
 The loss coefficients used here apply to the velocity head in the line size represented by the suction and discharge pipe diameters at the points of pressure measurement.
 If the loss elements are in different size lines than the points of pressure measurement, they need to be appropriately scaled. It is generally suggested that the losses be scaled in proportion to the 4th power of the diameter ratio. For example, if the discharge pressure is measured in a 12-inch header, and there is a 6-inch check valve with a nominal loss coefficient of 2 (applied to the 6-inch valve size), the K factor to use for the valve would be $2 \times (12/6)^4$ to the 4th power, or 32. The reason for this 4th power scaling is that the velocity varies with the square of the pipe diameter, and the velocity head (to which the loss coefficients apply) is proportional to the velocity squared.

Calculated Results:

Differential elevation head	0.00 m
Differential pressure head	11.55 m
Differential velocity head	0.03 m
Estimated suction friction head	0.02 m
Estimated discharge friction head	0.07 m
Pump head	11.67 m

Figure 4.7: Head Tool for Condition B for Targeted Plant A

From the PSAT overview in Figure 4.8, all the value for the current system can be inserted in Condition A, where the system will suggest better condition for the current system in Condition B. Through this PSAT application, user can determine existing and optimal condition for the pump efficiency (%), motor rated power (kW), motor shaft power (kW), pump shaft power (kW), motor efficiency (%), motor power factor (%), motor current (amps), motor power (kW), annual energy (MWh) and annual cost for the pump system.

Through the PSAT, annual savings potential and optimization rating for the pump can also be determined. The difference between Condition A and B, Condition B is the condition suggested by the system, where the improvement can be made to achieve better optimization rating as suggested in Condition B.

For the motor of the pump, value for line frequency, which is 50 Hz, with 11 kW power, rotation 1455 rpm and using energy efficiency for its efficiency class. This motor runs with 415V, and with 20.1 full load amps. For the duty, unit cost as in USD/kwh is 0.3550.

Condition A as in PSAT overview is a column where all the specification for current system running can be inserted here. First part in this section is type of pump is used. For targeted pumps. Meanwhile for Condition B, all the specifications are the same as Condition A except for flow rate.

The pump curve of the targeted pump in Plant A was plotted based on current condition as shown in Figure 4.9. From the graph plotted, it can be seen that the system should be operated at 80% efficiency, which is at flow rate of about 165m³/hr or 58 L/s but the curve shows that the pump is currently operated at 60% region which is flow rate of 208m³/hr or 45 L/s. The pump is operating off from its best efficiency point (BEP) as it should be. From this condition, it can be said that the pump will have reliability issues, which may lead to vibration problems, and shorter lifespan. When the pump operates at flow which is less than the flow designated BEP, this will also induce cavitation, vibration, impeller damages, suction and discharge recirculation to the pump system.

Pumping System Assessment Tool

File Tools Help

PSAT 2008

Condition A

End suction ANSI/API

Pump rpm: 1450

Drive: Direct drive

Units: m³/hr, m, kW

Kinematic viscosity (cS): 1.00

Specific gravity: 1.000

stages: 1

Fixed specific speed? YES

Line freq: 50 Hz

kW: 11

Motor rpm: 1455

Eff. class: Energy efficient

Voltage: 415

Estimate FLA

Full-load amps: 20.1

Size margin, %: 15

Operating fraction: 1.000

\$/kwhr: 0.3550

Flow rate, m³/h: 165

Head tool: Head, m: 11.6

Load estim. method: Power

Motor kW: 11.0

Voltage: 415

Condition B

End suction ANSI/API

Pump rpm: 1450

Drive: Direct drive

Units: m³/hr, m, kW

Kinematic viscosity (cS): 1.00

Specific gravity: 1.000

stages: 1

Fixed specific speed? YES

Line freq: 50 Hz

kW: 11.0 kW <<Spec

Motor rpm: 1455

Eff. class: Energy efficient

Voltage: 380

Estimate FLA

Full-load amps: 21.9

Size margin, %: 15

Operating fraction: 1.000

\$/kwhr: 0.3550

Flow rate, m³/h: 209

Head tool: Head, m: 11.7

Load estim. method: Power

Motor kW: 11.0

Voltage: 415

Condition A

	Existing	Optimal	Units
Pump efficiency	51.6	83.0	%
Motor rated power	11	7.5	kW
Motor shaft power	10.1	6.3	kW
Pump shaft power	10.1	6.3	kW
Motor efficiency	91.9	91.1	%
Motor power factor	82.5	80.9	%
Motor current	18.6	11.9	amps
Motor power	11.0	6.9	kW
Annual energy	96.4	60.4	MWh
Annual cost	34.2	21.5	\$1000

Annual savings potential, \$1,000: 12.7

Optimization rating, %: 62.7

Condition B

	Existing	Optimal	Units
Pump efficiency	65.6	84.2	%
Motor rated power	11	9.2	kW
Motor shaft power	10.1	7.9	kW
Pump shaft power	10.1	7.9	kW
Motor efficiency	91.9	91.6	%
Motor power factor	81.8	79.7	%
Motor current	18.7	15.0	amps
Motor power	11.0	8.6	kW
Annual energy	96.4	75.3	MWh
Annual cost	34.2	26.7	\$1000

Annual savings potential, \$1,000: 7.5

Optimization rating, %: 78.2

Log file controls:

Create new log

Add to existing log

Retrieve log entry

Delete log entry

Summary file controls:

Create new summary file

Existing summary files

CREATE NEW

Condition A Notes

Facility: Purnabina Sdn Bhd

System: Chiller water pump

Date: 24 Feb 2017

Application:

Evaluator: Farhah

General comments:

Documentation section

Condition B Notes

Facility:

System:

Date:

Application:

Evaluator:

General comments:

Retrieve defaults

Set defaults

Copy A > to B >

Copy B < to A <

Background information

System curve tool: select below

STOP

Figure 4.8: PSAT Result for Targeted Pump at Plant A

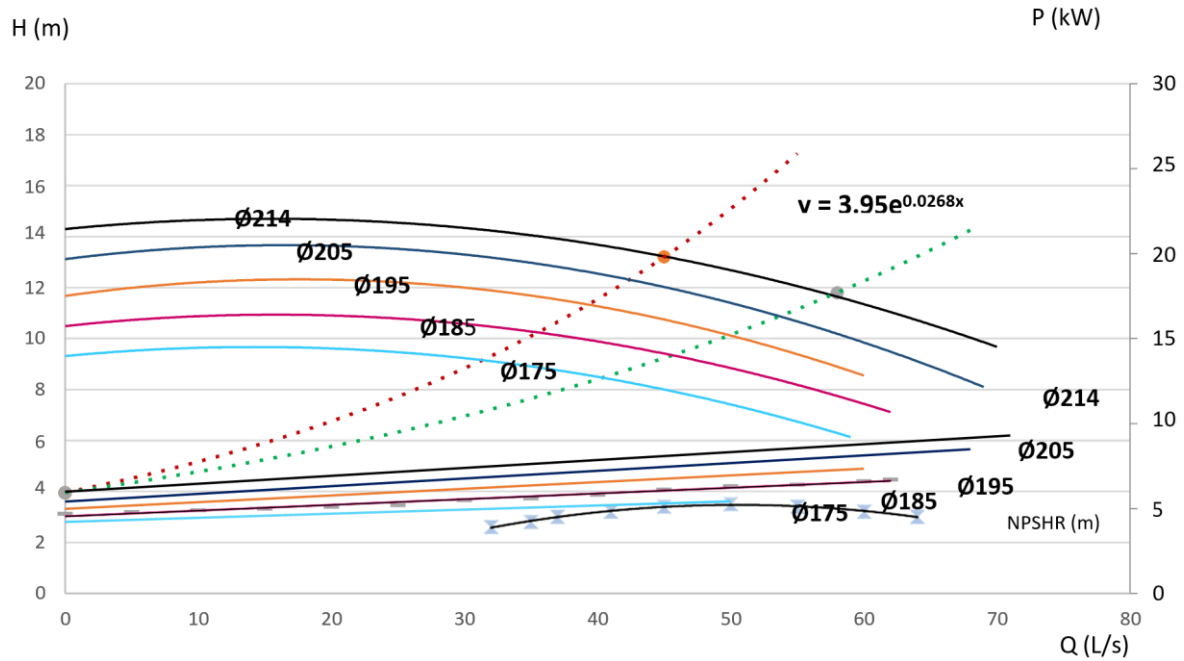


Figure 4.9: Pump Curve of Targeted Pump Plant A

Comparison of condition A and condition B as in Figure 4.10, right hand side of the PSAT result. In this section, pump efficiency, motor rated power, motor shaft power, pump shaft power, efficiency, power factor, current and power for the motor system. In this result section, annual energy and pump annual cost has been evaluated for both condition A and B too. Existing and optimal value for targeted pump for both conditions can be determined from the PSAT analysis. Details result for both conditions as per Figure 4.10 below. Existing annual energy for targeted pump in condition A is 96.4 MWh, but PSAT analysis shows that targeted pump can be run much lower annually, which is 60.4 MWh. Existing annual cost is about RM 34, 200, but targeted pump A can be run at RM 21, 500 if it runs in its optimal condition. From the result, it shows Condition A for targeted pumps with an annual savings potential is RM 12, 700, and its optimization rating 62.7%. If targeted pump can be run by increased its flow rate, from 165m³/h to 209m³/h pump result was analysed as in Condition B. In condition B, optimization rating of the pump is increased to 78.2% which in this

condition it can increase the lifespan of the pump if it can run with higher flow rate as per suggested by manufacturer.

	Condition A			Condition B		
	Existing	Optimal	Units	Existing	Optimal	Units
Pump efficiency	51.6	83.0	%	65.6	84.2	%
Motor rated power	11	7.5	kW	11	9.2	kW
Motor shaft power	10.1	6.3	kW	10.1	7.9	kW
Pump shaft power	10.1	6.3	kW	10.1	7.9	kW
Motor efficiency	91.9	91.1	%	91.9	91.6	%
Motor power factor	82.5	80.9	%	81.8	79.7	%
Motor current	18.6	11.9	amps	18.7	15.0	amps
Motor power	11.0	6.9	kW	11.0	8.6	kW
Annual energy	96.4	60.4	MWh	96.4	75.3	MWh
Annual cost	34.2	21.5	\$1000	34.2	26.7	\$1000
Annual savings potential, \$1,000	12.7			7.5		
Optimization rating, %	62.7			78.2		

Figure 4.10: Comparison on Condition A and B for Plant A PSAT Analysis

4.2 Targeted Pump at Plant B

Plant B is located at Cahya Mata Sarawak, which is a cement manufacturing industry. The plant is in Kuching, Sarawak. The cooling system is important to the plant as it supplies cooled water to support other machineries like compressor, bearing equipment, cooling tower conditioning and hydraulic oil station. Figure 4.11 shows a targeted pump at Plant B. This centrifugal pump is running at 200 m³/h, with 1450 rpm and 45 kW. The pump is running at maximum capacity to support the plant.



Figure 4.11: Targeted Pump at Plant B

From Figure 4.12, the targeted pump at Plant B is running with suction pressure - 68.9 kPa while discharge 382.7 kPa with pump inlet diameter 230 mm. The nearby river is the main source for CMS plants to get the water supplied for the pump to operate. Underground centrifugal pump is used to transport the river water to the underground tank. By using conversion of rotational kinetic energy to the hydrodynamics energy of the fluid flow, the underground pump is used to transport the fluids. Water is delivered by using a centrifugal pump in a continuous and uninterrupted flow. Inlet pump 1 and inlet pump 2 are the main pumps where water is supplied from the underground tank supply.

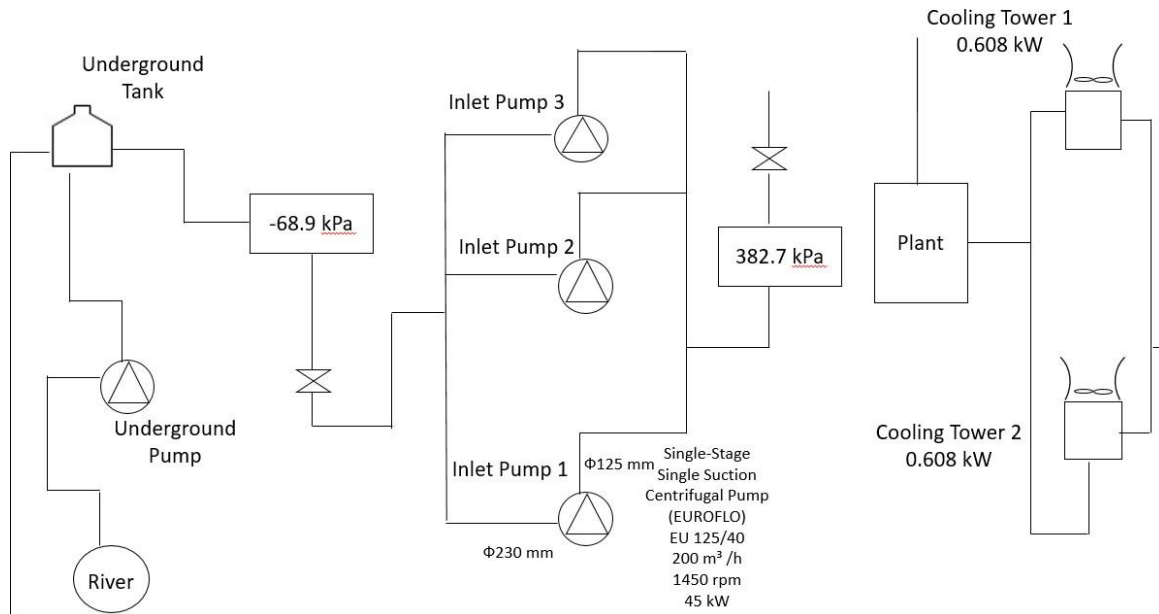


Figure 4.12: Flow Diagram of Targeted Pump at Plant B

Water elevation at underground tanks is used for PSAT results. Height of the water inside the tank is 9 feet/2.74 m. There is no water flow entering the pump as the inlet pump 3 is on standby mode, as it is not running thus there is no water flow entering the pump. Pressure gauge and butterfly valve are used for the system. From Figure 4.12, the diagram shows that the value of inlet pressure is -68.9 kPa, where the height of the pressure gauge from the ground is 1.0 and the butterfly valve is 100% opening position. Inlet pipe size is the same with the identical pump which is 230 mm and outlet size is 125 mm. Meanwhile for the outlet, the value for flow rate is measured by using ultrasonic flow meter is 193 m³/h. The outlet pressure measured is 55.5 PSIG/Kpa with 100% position butterfly valve. Then the water will be transferred to the plant to cool all machines. For example, compressor, bearing, cooling tower conditioning and hydraulic station/ Then this water will go back to the pump system by using a pipeline. Table 4.2 shows preliminary data for Pump at Plant A. Pump in plant A water pump house which is also centrifugal type as targeted pump in Plant A. Pump ID EU11325/40 MBC with serial number Y02520186. Pump is running with 1450

RPM with single stage suction. Pump motor ID Y225M-4/45-TH is running with 45 kW, 60 HP, 1455 RPM and 380V.

Table 4.2: Preliminary Data for Pump System in Plant B

Facility: Water Pump House	Date: 04 Oct 2017	Time: 1000 am	Pump Operation Hour: 24hrs/365days
System: Pump is used to cool out all machineries			
Pump Nameplate	ID/Set	EUI125/40 MBC S/N: Y02520186	
Pump Style		Centrifugal pump	
Service		Pumping circulating water	
Name plate pump system	RPM	1450	
Number of stages		Single stage suction	
Motor Name Plate	Y225M-4/45-TH		
Power	kW	45	
Power	HP	60	
Full load speed	RPM	1455	
Full load efficiency	%	-	
Rated voltage	Volts	380	

In Figure 4.13 and Figure 4.14, inlet pressure for both condition A and B are negative values. This occurs due to the booster having to suck water from tanks which are placed below the booster. The tank for the centrifugal pump is located underground in the plant near the pump house. If the booster draws from a break tank, the inlet pressure will derive from the water supply pressure. From the plant diagram for Plant B at Figure 4.12, it can be observed that the non-return valve is placed before the pump, thus during the standstill, the water will not drain from the pump. The optimum placement of the non-return valve is influenced by whether the inlet pressure is negative or positive.

Pump head calculator 2007.v1

File Tools Help

Type of measurement configuration
Suction and discharge line pressures

K_s represents all suction losses from gauge P_s to the pump
 K_d represents all discharge losses from the pump to gauge P_d

Click to access units converter tool

Suction pipe diameter (ID) 304.8 mm	Discharge pipe diameter (ID) 254.0 mm
Suction gauge pressure (P_s) 59.3 kPa	Discharge gauge pressure (P_d) 172.4 kPa
Suction gauge elevation (Z_s) 1.52 m	Discharge gauge elevation (Z_d) 1.52 m
Suction line loss coefficients, K_s 0.50	Discharge line loss coefficients, K_d 1.00

Fluid specific gravity Flow rate m³/hr

Don't update Accept and update

Click to leave the main panel head unchanged Click to Accept and return the calculated head

Differential elevation head	0.00 m
Differential pressure head	11.55 m
Differential velocity head	0.02 m
Estimated suction friction head	0.01 m
Estimated discharge friction head	0.04 m
Pump head	11.63 m

System of units: m³/hr, m, kW

Important note about loss coefficients
 The loss coefficients used here apply to the velocity head in the line size represented by the suction and discharge pipe diameters at the points of pressure measurement.

If the loss elements are in different size lines than the points of pressure measurement, they need to be appropriately scaled. It is generally suggested that the losses be scaled in proportion to the 4th power of the diameter ratio. For example, if the discharge pressure is measured in a 12-inch header, and there is a 6-inch check valve with a nominal loss coefficient of 2 (applied to the 6-inch valve size), the K factor to use for the valve would be $2 \times (12/6)$ to the 4th power, or 32. The reason for this 4th power scaling is that the velocity varies with the square of the pipe diameter, and the velocity head (to which the loss coefficients apply) is proportional to the velocity squared.

Figure 4.13: Head Tool for Condition A for Targeted Plant B

Pump head calculator 2007.v1

File Tools Help

Type of measurement configuration
Suction and discharge line pressures

K_s represents all suction losses from gauge P_s to the pump
 K_d represents all discharge losses from the pump to gauge P_d

Click to access units converter tool

Suction pipe diameter (ID) 230.0 mm	Discharge pipe diameter (ID) 125.0 mm
Suction gauge pressure (P_s) -30.5 kPa	Discharge gauge pressure (P_d) 382.7 kPa
Suction gauge elevation (Z_s) 2.74 m	Discharge gauge elevation (Z_d) 2.74 m
Suction line loss coefficients, K_s 0.50	Discharge line loss coefficients, K_d 1.00

Fluid specific gravity Flow rate m³/hr

Don't update Accept and update

Click to leave the main panel head unchanged Click to Accept and return the calculated head

Differential elevation head	0.00 m
Differential pressure head	42.21 m
Differential velocity head	1.31 m
Estimated suction friction head	0.06 m
Estimated discharge friction head	1.43 m
Pump head	45.01 m

System of units: m³/hr, m, kW

Important note about loss coefficients
 The loss coefficients used here apply to the velocity head in the line size represented by the suction and discharge pipe diameters at the points of pressure measurement.

If the loss elements are in different size lines than the points of pressure measurement, they need to be appropriately scaled. It is generally suggested that the losses be scaled in proportion to the 4th power of the diameter ratio. For example, if the discharge pressure is measured in a 12-inch header, and there is a 6-inch check valve with a nominal loss coefficient of 2 (applied to the 6-inch valve size), the K factor to use for the valve would be $2 \times (12/6)$ to the 4th power, or 32. The reason for this 4th power scaling is that the velocity varies with the square of the pipe diameter, and the velocity head (to which the loss coefficients apply) is proportional to the velocity squared.

Figure 4.14: Head Tool for Condition A for Targeted Plant B

Figure 4.15 shows PSAT analysis for the targeted pump at Plant B. The difference between Condition A and B is the flow rate of the pump is higher in Condition B. Both condition A and B run with pump rotation 1450 rpm, and pump is direct drive pump. For the motor parts of the pump, pump motor running at frequency 60 Hz, 45 kW, 380 V and rotation at 1455 rpm for both condition A and B.

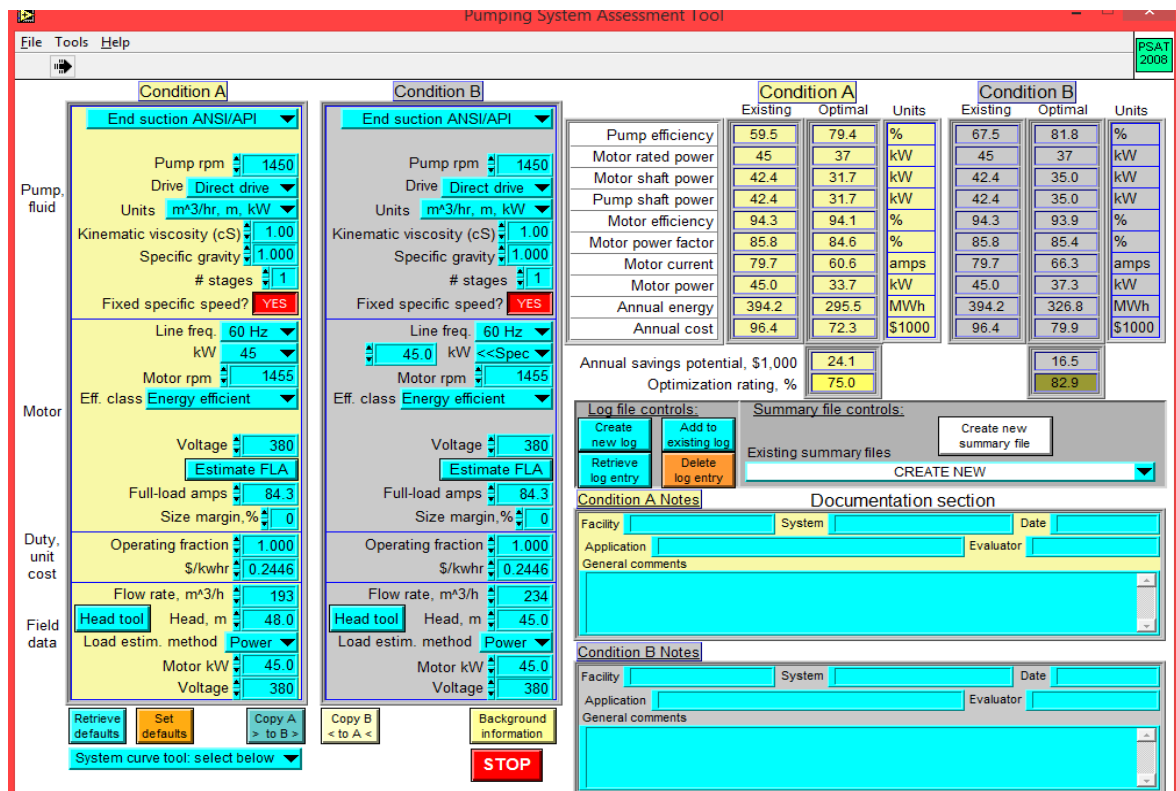


Figure 4.15: PSAT Result for Targeted Pump at Plant B

From Figure 4.15, it shows that from A condition, optimization rating of the system is 75.0% and it can achieve 82.9% if the flow rate of the pump is increased from 193 m³/h to 234 m³/h. Thus, the flow rate of the pump can also be changed to see the difference in the pump head. From the results, current pump efficiency is 59.5%, as shown in the upper right side (yellow column). The database suggests the system can be operated up to 79.4% in its optimum condition.

It can be concluded that the optimization rating of the targeted pump in Plant B can be increased more than 5% from its current operating condition as suggested by the system database in condition B. The significant amount of energy for this plant is MYR 96300 or USD 24100. The annual energy for the plant can be saved from 394.2 MWh to 295.5 MWh.

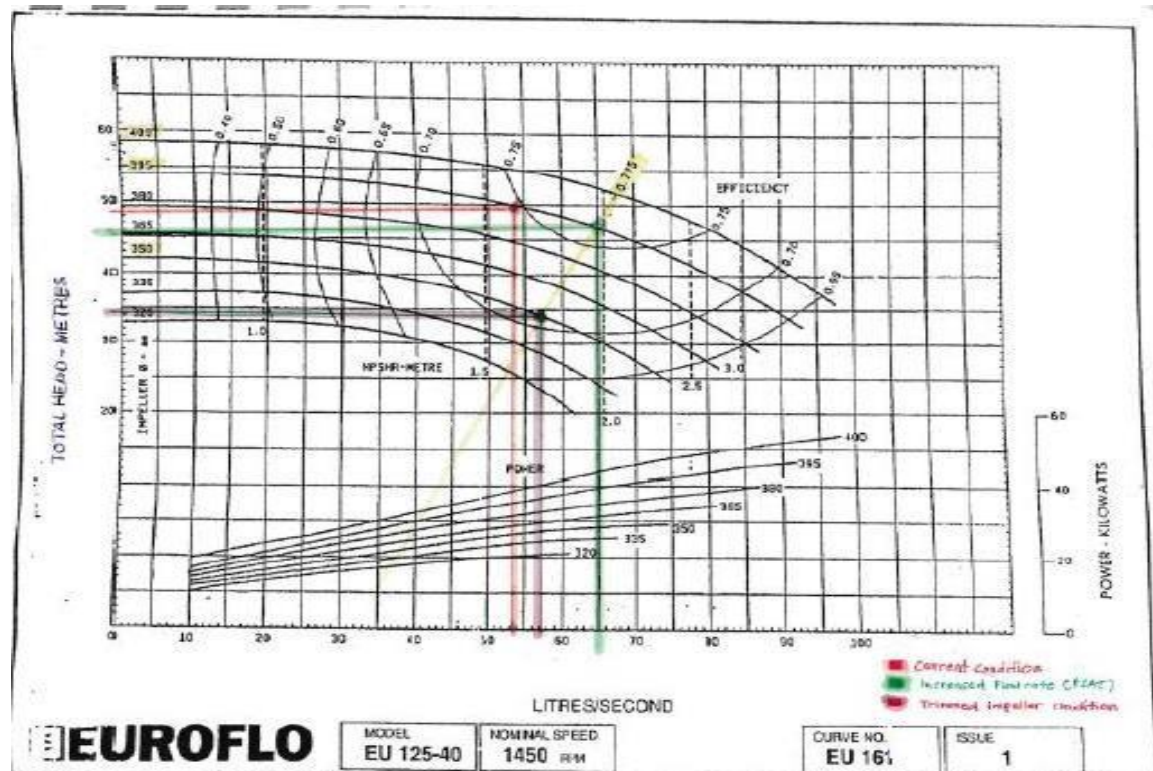


Figure 4.16: Pump Curve of Targeted Pump Plant

4.3 Operational Cost Savings

In both Plant A and B, both PSAT results show through pump system optimization annual energy savings can be achieved. Both plants also show reduction of energy consumption through optimization. Table 4.3 shows a summary of potential energysavings for both Plant A and B.

Table 4.3: Summary of Potential Energy Savings

		Pump Plant A	Pump Plant B
Current Annual Energy Usage	MWh	96.4	394.2
Current Operating Costs	RM	34,200.00	96,400.00
Optimum Annual Energy Usage	MWh	75.3	326.8
Optimum Operating Costs	RM	21,500.00	79,900.00
Potential Energy Saving	MWh	21.1	67.4
Potential Operating Costs Saving	RM	7,500.00	16,500.00

It can be said from this research study PSAT analysis can be used to do pre-assessment for the organization. By using PSAT, pump optimization can be achieved which can help to reduce energy consumption by machineries in manufacturing plants.

4.4 Carbon Dioxide Emission

About 82 percent of carbon emissions for all total greenhouse gas emissions in the U.S as of Year 2017. Carbon emissions is defined as a type of greenhouse gas emission when carbon dioxide enters the air after human activity or process (Osmanski, 2020). Emission of carbon dioxide will cause a greenhouse effect. The greenhouse gases will release into the atmosphere and trap the sun's heat and eventually cause global warming. Investigation study on the impact of energy consumption on the economic and financial development in 19 countries has been performed. High development for these countries causes three decades of increased CO₂ emission and it is important to reduce the level of pollution by utilizing energy protection policies.

Methods like rationing consumption of energy and controlling carbon dioxide emissions or clean energy of their total energy consumption to be increased can be performed (Al-Mulali & Binti Che Sab, 2012). High intensity of carbon emissions to energy consumption will increase use of fossil fuel due to rapid economic growth (Ahmad et al., 2017; Andreoni and Galmarini, 2016; Sohag et al., 2015). The demand for energy needs to be reduced to mitigate carbon emissions (Martinho, 2016).

Based on analysis of the study, the optimization of the pump system can help to reduce carbon dioxide emission by reducing total energy consumption used. As per a report by the Ministry of Energy, Water and Communication Malaysia, 1 kWh of energy emission is equivalent to 0.6989 metric tons. (Greentech Malaysia, 2016).

As for Plant A, from PSAT software, annual energy of the existing pump is 96.4 MWh and optimum annual energy usage is 75.3 MWh giving its potential energy saving is 21.1 MWh.

$$\frac{1 \text{ kWh}}{0.6989} = \frac{21,100 \text{ kWh}}{X} \quad (\text{Equation 4.1})$$

$$X = 146,76.9 \text{ metric ton}$$

CO₂ emission in a year and 10 years,

$$\begin{aligned} X &= 146,76.9 \times 10 \text{ years} \\ &= 146769 \text{ metric ton} \end{aligned}$$

Meanwhile for Plant B, annual energy of the existing pump is 394.2 MWh and its optimum annual energy usage is 326.8 MWh, which gives its potential energy saving is 67.4 MWh.

$$\frac{1 \text{ kWh}}{0.6989} = \frac{67,400 \text{ kWh}}{X} \quad (\text{Equation 4.2})$$

$$X = 47,105.8 \text{ metric ton}$$

CO₂ emission in a year and 10 years,

$$= 47,105.8 \times 10 \text{ years}$$

$$= 471058.6 \text{ metric ton}$$

Based on the results, 146769 metric ton CO₂ of emission can be reduced if there is improvement can be done in Plant A, meanwhile 471,058 .6 metric ton of CO₂ emission can be minimized from manufacturing of Plant B.

Improving the pump system as one of the major drives in a plant will automatically reduce the total consumption of the plant, therefore reducing the CO₂ emission.

CHAPTER 5

CONCLUSION AND RECOMMENDATIONS

5.1 Conclusion

The purpose of this research was to analyse and evaluate performance of selected two (2) pumps in Johor and Sarawak by using PSAT. Selected pumps in Plant A and Plant B were identified and the conditions of the pumps were analysed. The potential annual savings were assessed through a comparison between the existing pumping system and an optimized system utilizing the Pumping System Assessment Tool. This thesis also examined the favourable outcomes of increasing the flow rate of the pump, including improved pumping efficiency, increased flow rate, and greater cost savings. The flow rate of the pump was suggested to be increased higher to bring the pump curve to its best efficiency point (BEP). The study effectively investigated annual savings, thus contributes to the achievement of sustainable financial and energy optimization objectives.

Throughout this research, there is certain limitations exist to complete the study. Even though globally we would like to conserve energy and cost reduction, there is still negligence to the implementation of energy savings practices especially to the industrial end user. Our finding is pump is running without proper maintenance as the operation is only focus on output demand. This has impacted the pump lifespan and its performance. To overcome this gap, plant management team need to support on the energy savings opportunity through pump system and other machineries. Other limitation is to obtain details of the pump as the pump used is already ageing. Some of the information is also very limited as it is confidential. Thus, it is also cause difficulties to analyse the data further. Future

studies and details approach collaborate with other industrial partners and researchers could also facilitate access to get the information.

5.2 Recommendations

Based on the pump performance, there are few recommendations that can be done. First, maintaining pipeline cleanliness and flow rate of the pump can be improved by ensuring there is no restriction in the pump flow. Maintaining the service of the pipeline and preventing valve choking can be done in maintaining the pump flow to ensure the pump system is running in optimum condition. Redesign or modify the impeller to a lower diameter also can be done. To adjust the centrifugal pump head and flow to the actual needs, impeller trimming is a common practice performed by pump manufacturers and users to improve pump efficiency. Besides can improve pump efficiency, cross flow and secondary incidence flows also can be reduced.

Few proposals can be made for both Plant A and Plant B management for a better pump system operation, but some costs need to be considered. Return of investment percentage and payback period for the investment had to be included for the cost proposal. Discharge pressure, decrease operation fraction by replacing impeller with bigger size for Plant A and Plant B, were considered for estimated savings calculation.

- The estimated cost of the impeller is between MYR 45,000 to MYR 60,000.

Return of Investment for Plant A

$$ROI = \frac{\textit{Profit}}{\textit{Cost of Investment}} \quad (\text{Equation 5.1})$$

$$ROI = \frac{RM7,500}{RM60,000} \times 100\%$$

$$ROI = 12.5\%$$

Payback period

$$PBP = \frac{\textit{Cost of Investment}}{\textit{Cash Flow}} \quad (\text{Equation 5.2})$$

$$PBP = \frac{(\textit{Impeller cost})}{\textit{cash flows}} \quad (\text{Equation 5.3})$$

$$PBP = \frac{(RM\ 60,000)}{RM\ 7,500}$$

$$PBP = 4.58\ \textit{years}$$

~ Takes 4 years and 6 months to recover the investment

From the calculation above, the payback period calculated to recover the investment will take 4 years and 6 months. Percentage of ROI is also quite high which shows the investment for changing size of impeller to bigger size can be considered even though the payback period takes quite longer. Even though the profit return is not high, if the pump can be operating at optimal condition, this will increase the lifespan of the pump and reduce the maintenance cost.

Return of Investment for Plant B

$$ROI = \frac{\textit{Profit}}{\textit{Cost of Investment}} \quad (\text{Equation 5.4})$$

$$ROI = \frac{RM 16,500}{RM 60,000} \times 100\%$$

$$ROI = 27.5\%$$

Payback period

$$PBP = \frac{\textit{Cost of Investment}}{\textit{Cash Flow}} \quad (\text{Equation 5.5})$$

$$PBP = \frac{(\textit{Impeller cost})}{\textit{cash flows}} \quad (\text{Equation 5.6})$$

$$PBP = \frac{(RM 60,000)}{RM 16,500}$$

$$PBP = 3.63 \text{ years}$$

~ Takes 3 years and 6 months to recover the investment

From the calculation above, the payback period calculated to recover the investment will only take about 3 years and 6 months. Percentage of ROI is also high which shows the investment for changing size of impeller will be better for the pump performance and energy efficiency. Pump will have longer lifespan if the performance of the pump is higher and when the pump system is running in its optimum condition.

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APPENDICES

Pump Specification for Plant A

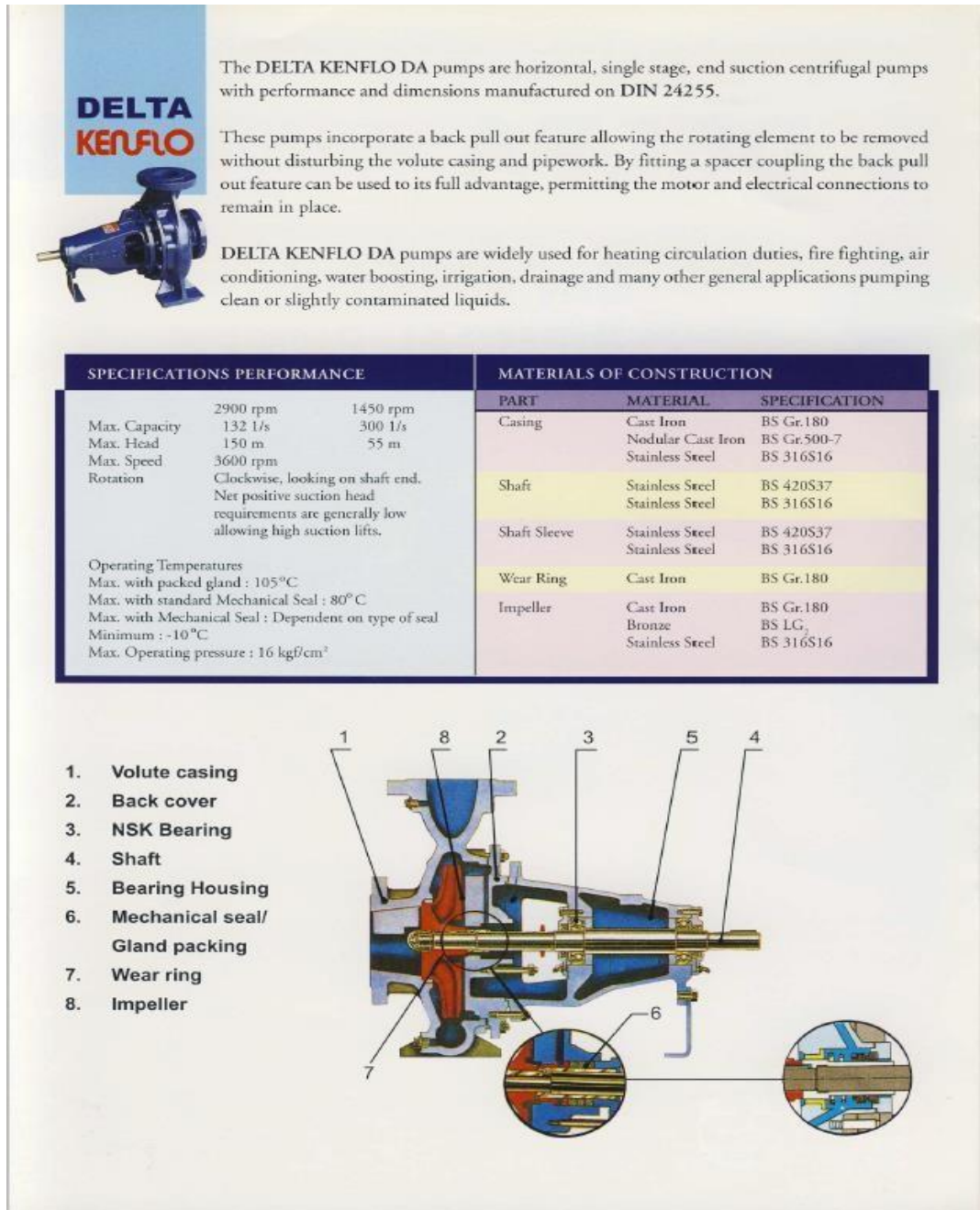


Figure A1.1: Delta Kenflo Pump Specification

SELECTION CHARTS

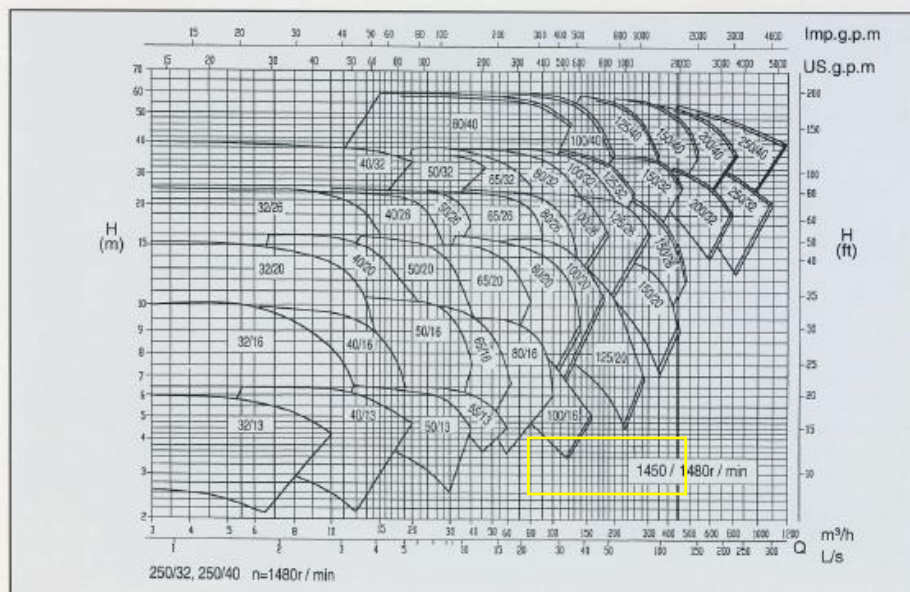
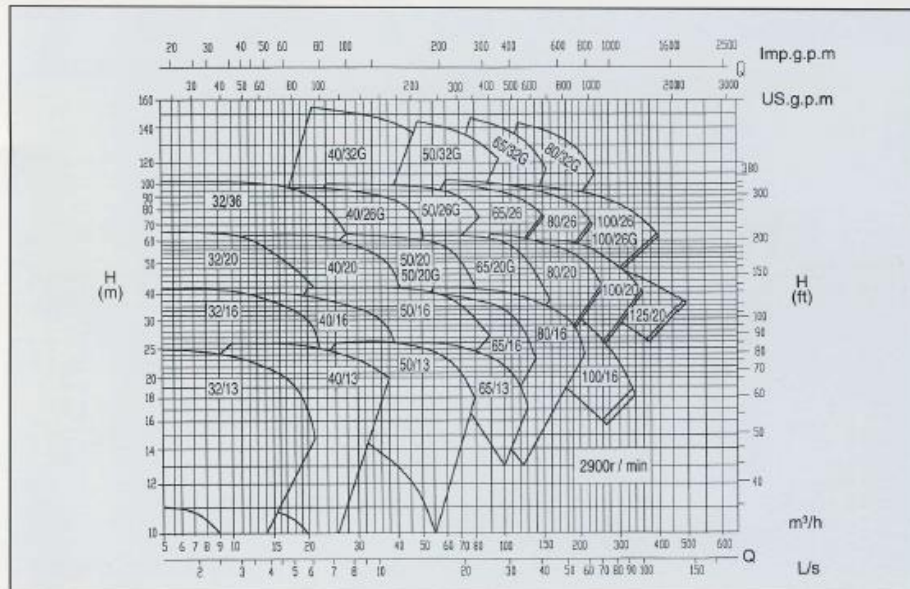
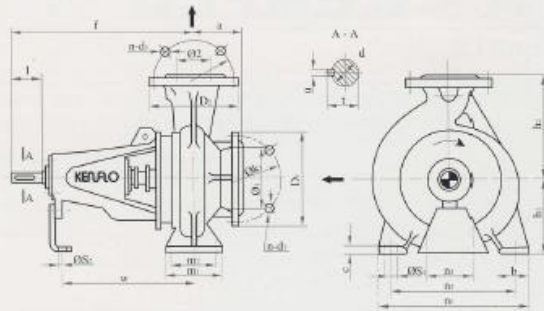


Figure A1.2: Delta Kenflo Pump Curve

DELTA-KENFLO OUTLINE & INSTALLATION DIMENSIONS (mm)



Dimensions :

Pump size	Shaft unit	Ø1	Ø2	a	f	h ₁	h ₂	b	c	m ₁	m ₂	n ₁	n ₂	ØS ₁	n ₃	ØS ₂	w	d	l	t	u	WGT. Kg
32/13	25	50	32	80	360	112	140	50	14	100	70	190	140	14	100	14	267	24	50	27	8	27.5
32/16	25	50	32	80	360	132	160	50	14	100	70	240	190	14	100	14	267	24	50	27	8	35
32/20	25	50	32	80	360	160	180	50	14	100	70	240	190	14	110	14	267	24	50	27	8	41
32/26	25	50	32	100	360	180	225	65	14	125	95	320	250	14	110	14	267	24	50	27	8	59
40/13	25	65	40	80	360	112	140	50	14	100	70	210	160	14	100	14	267	24	50	27	8	30
40/16	25	65	40	80	360	132	160	50	14	100	70	240	190	14	100	14	267	24	50	27	8	36
40/20	25	65	40	100	360	160	180	50	14	100	70	265	212	14	110	14	267	24	50	27	8	44
40/26	25	65	40	100	360	180	225	65	14	125	95	320	250	14	110	14	267	24	50	27	8	61
40/26G	35A	65	40	100	454	180	225	65	14	125	95	320	250	14	110	14	327	32	80	35	10	75
40/32	35	65	40	125	470	200	250	65	14	125	95	345	280	14	110	14	342	32	80	35	10	96
40/32G	45A	65	40	125	519	200	250	65	14	125	95	345	280	14	110	14	362	42	110	45	12	104
50/13	25	65	50	100	360	132	160	50	14	100	70	240	190	14	110	14	267	24	50	27	8	34
50/16	25	65	50	100	360	160	180	50	14	100	70	265	212	14	110	14	267	24	50	27	8	38
50/20	25	65	50	100	360	180	200	50	14	100	70	265	212	14	110	14	267	24	50	27	8	46
50/26G	35A	65	50	100	454	180	200	50	14	100	70	265	212	14	110	14	328	32	80	35	10	59
50/26	25	65	50	100	360	180	225	65	14	125	95	320	250	14	110	14	267	24	50	27	8	65
50/26G	35A	65	50	100	454	180	225	65	14	125	95	320	250	14	110	14	327	32	80	35	10	76
50/32	35	65	50	125	470	225	280	65	14	125	95	345	280	14	110	14	342	32	80	35	10	101
50/32G	45A	65	50	125	523.5	225	280	65	14	125	95	345	280	14	110	14	361	42	110	45	12	110
65/13	25	80	65	100	360	160	180	65	14	125	95	280	212	14	110	14	267	24	50	27	8	39
65/16	25	80	65	100	360	160	200	65	14	125	95	280	212	14	110	14	267	24	50	27	8	45
65/20	25	80	65	100	360	180	225	65	14	125	95	320	250	14	110	14	267	24	50	27	8	52
65/26G	35A	80	65	100	454	180	225	65	14	125	95	320	250	14	110	14	328	32	80	35	10	65
65/26	35	80	65	100	470	200	250	80	16	160	120	360	260	17.5	110	14	342	32	80	35	10	81
65/32	35	80	65	125	470	225	280	80	16	160	120	400	315	17.5	110	14	342	32	80	35	10	110
65/32G	45A	80	65	125	520	225	280	80	16	160	120	400	315	17.5	110	14	363	42	110	45	12	118
80/16	25	100	80	125	360	180	225	65	14	125	95	320	250	14	110	14	267	24	50	27	8	51
80/16G	35A	100	80	125	454	180	225	65	14	125	95	320	250	14	110	14	326	32	80	35	10	64
80/20	35	100	80	125	470	180	250	65	14	125	95	345	280	14	110	14	342	32	80	35	10	70
80/26	35	100	80	125	470	200	280	80	16	160	120	400	315	17.5	110	14	342	32	80	35	10	91
80/32	35	100	80	125	470	250	315	80	16	160	120	400	315	17.5	110	14	342	32	80	35	10	120
80/32G	45A	100	80	125	521	250	315	80	16	160	120	440	340	17.5	110	14	363	42	110	45	12	128
80/40	45	100	80	125	532	280	355	85	16	160	120	440	340	17.5	110	14	368	42	110	45	12	161
100/16	35	125	100	125	470	200	250	80	16	160	120	360	280	17.5	110	14	342	32	80	35	10	81
100/20	35	125	100	125	470	200	280	80	16	160	120	360	280	17.5	110	14	342	32	80	35	10	85
100/26	35	125	100	140	470	225	280	80	16	160	120	400	315	17.5	110	14	342	32	80	35	10	106
100/26G	45A	125	100	140	520	225	280	80	16	160	120	400	315	18	110	14	363	42	110	45	12	117
100/32	35	125	100	140	470	250	315	80	16	160	120	400	315	17.5	110	14	342	32	80	35	10	134
100/40	45	125	100	140	530	280	355	100	18	200	150	500	400	23	110	14	370	42	110	45	12	174
125/20	35	150	125	140	470	250	315	80	16	160	120	400	315	17.5	110	14	342	32	80	35	10	111
125/26	35	150	125	140	470	230	355	80	16	160	120	400	315	17.5	110	14	342	32	80	35	10	115
125/32	45	150	125	140	530	280	355	100	18	200	150	500	400	23	110	14	370	42	110	45	12	163
125/40	45	150	125	140	530	315	400	100	18	200	150	500	400	23	110	14	370	42	110	45	12	183
150/20	35G	200	150	160	495	280	400	100	20	200	150	550	450	23	110	14	367	32	80	35	10	140
150/26	45	200	150	160	530	250	355	100	18	200	150	450	350	23	110	14	370	42	110	45	12	164
150/32	45	200	150	160	530	280	400	100	18	200	150	550	450	23	110	14	370	42	110	45	12	170
150/40	45	200	150	160	530	315	430	100	18	200	150	550	450	23	110	14	370	42	110	45	12	209
200/32	55	250	200	180	670	315	480	120	20	220	170	660	480	28	110	14	504	48	110	51	14	251
200/40	55	250	200	180	670	335	480	120	20	220	170	660	480	28	110	14	504	48	110	51	14	295
250/32	55	300	250	220	691	355	520	150	22	250	200	660	510	28	110	14	525	48	110	51	14	311
250/40	55	300	250	220	681	400	560	150	22	250	200	660	510	28	110	14	516	48	110	51	14	390

Flange mating dimensions to ISO2084 PN16

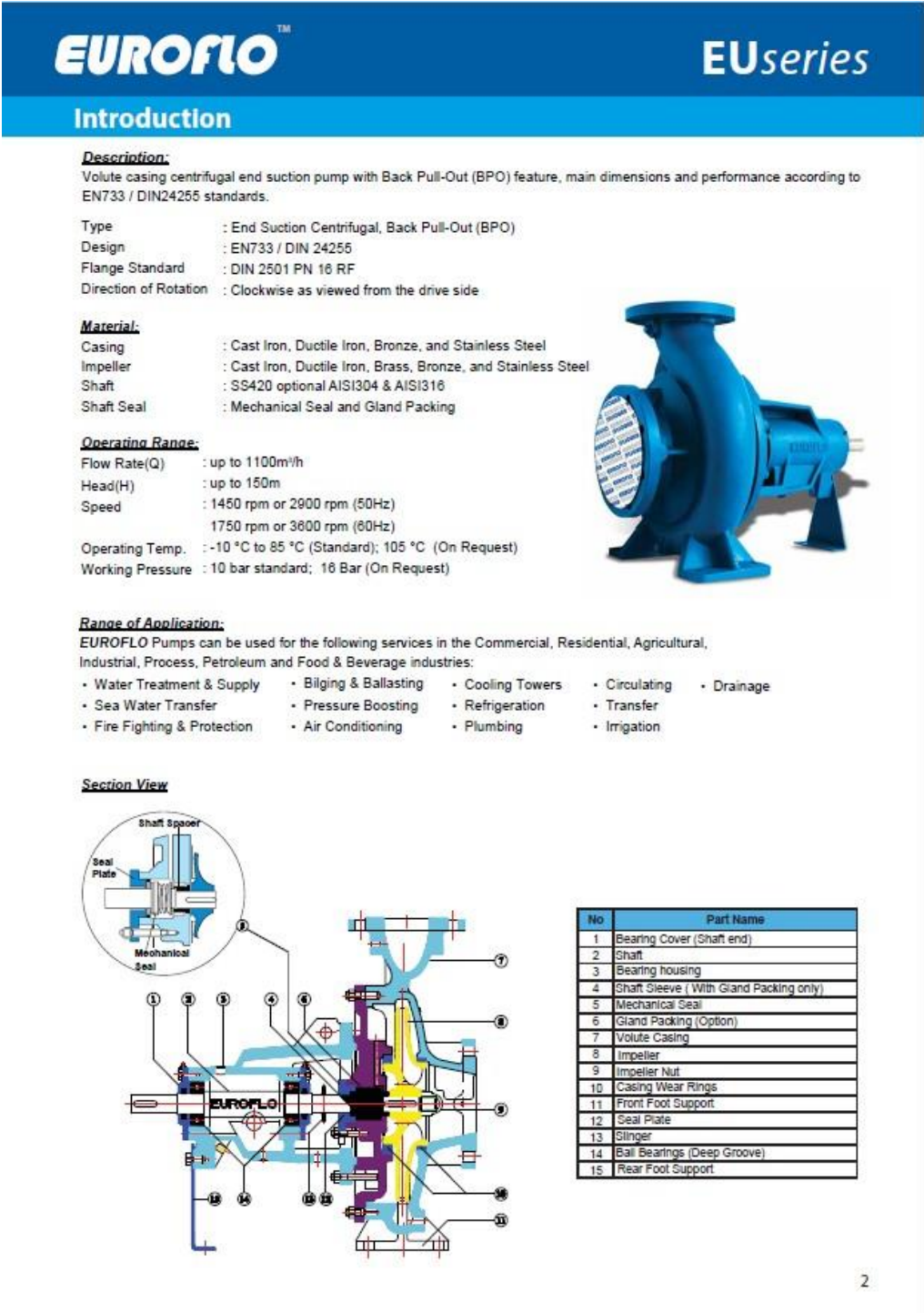
Ø1, Ø2	32	40	50	65	80	100	125	150	200	250	300
D ₁ , D ₂	140	150	165	185	200	220	250	285	340	405	469
ØK ₁ , ØK ₂	100	110	125	145	160	180	210	240	295	355	410
n-d	4-Ø18	4-Ø18	4-Ø18	4-Ø18	4-Ø18	4-Ø18	4-Ø18	4-Ø22	2-Ø24	2-Ø24	2-Ø24

The Manufacturer reserve the right to alter specification without prior notice.

DISTRIBUTOR :

Figure A1.3: Delta-Kenflo Outline and Installation Dimensions (mm)

Pump Specification for Plant B



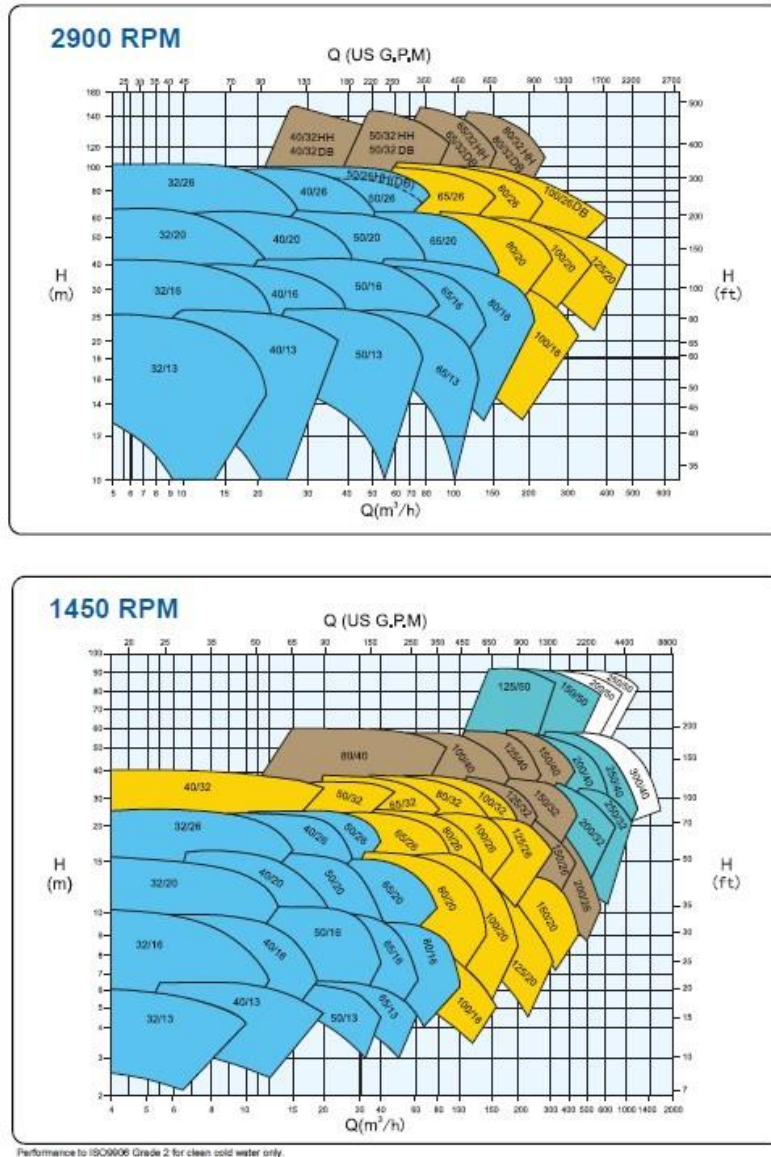


Figure A1.5: Euroflo Pump Curve

EU - Installing Dimensions

