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KAHRAMANMARAŞ SÜTÇÜ IMAM UNIVERSITY

INSTITUTE FOR

GRADUATE STUDIES IN SCIENCE AND TECHNOLOGY

COMPARISON OF ENERGY EFFICIENCIES OF CONSTANT AND VARIABLE FLOWRATE OPERATIONS WITH A SMALL SCALE IRRIGATION PUMP

ALAA A. SAHIB

MASTER THESIS

DEPARTMENT OF BIOENGINEERING AND SCIENCES

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ALAA A. SAHIB

This thesis was prepared at the

Department of Bioengineering and Sciences

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This is to certify that the MSc Thesis entitled “Comparison of Energy Efficiencies of Constant and Variable Flow Rate Operations with a Small Scale Irrigation Pump” and prepared by ALAA ABDULRADHA SAHIB has met the dissertation requirements of the Graduate School of Natural and Applied Sciences at Kahramanmaraş Sütçü Imam University.

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ALAA A. SAHIB

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KÜÇÜK GÜÇLÜ BİR SULAMA POMPASI İLE SABİT VE DEĞİŞKEN DEBİLİ ÇALIŞMADA ENERJİ ETKİNLİĞİNİN KARŞILAŞTIRILMASI
(YÜKSEK LİSANS TEZİ)

ALAA A. SAHIB

ÖZ

Bu çalışmanın amacı, küçük güçlü bir santrifüj pompaj tesisinin farklı çalışma şartları altında enerji etkinliğinin belirlenmesidir. Bu amaca ulaşmak için pompaj tesisinde debi, emme basıncı, basma hattı basıncı ve sistemin güç tüketimi ölçülmüştür. Denemeler; sabit debi ve değişken debi şartlarında dört, sabit basınç çalışma koşullarında üç tekrarlı yapılmıştır. Testler; emme vanası, basma vanası, by-pass vanası ve değişken hız kontrolü kullanılarak ve debi ve basıncın değiştirilmesi sureti ile gerçekleştirilmiştir. Testlerde debi 0-3.0 m^3 h^{-1} ve basınç 0-5 bar aralığında değiştirilmiştir.

Pompa ünitesinde değişken hız kontrolü yok iken gerekli debinin sağlanması için enerji etkinliğinin en iyi olduğu uygulama by-pass vanasının kullanılması (410 Watt) olarak bulunmuş, bunu emme vanası (420 Watt) ve basınç hattı vanası (920 Watt) izlemiştir. By-pass vanasının kullanımı, basma hattı vanası ile karşılaştırıldığında %45 enerji kazancı sağlamıştır. Emme hattı vanasının kullanımı, debinin değiştirilmesi için iyi bir alternatif olabilir, ancak artan vakum nedeniyle kavitasyon riskinin ortaya çıktığı gözlenmiştir. Debinin %20 azaltılması, vakumu -0.3 bardan -0.54 bara düşürmüştür. Sabit basınçta çalışma şartları, değişken hız ayarı veya by-pass vanası kullanılarak sağlanabilir. Ancak, eşit basınç gereksinimi için değişken hız kontrolü kullanıldığında güç tüketimi azalırken by-pass vanasının kullanımı güç tüketimini önemli ölçüde etkilememiştir. Değişken hız ayarlı çalışma koşulu, gerek duyulan debiye bağlı olarak enerji etkinliğini by-pass ayarına göre %0-28 oranında artırmıştır. Debi gereksinimi azaldıkça, değişken debili sistemin enerjisi daha etkin hale gelmektedir.

Anahtar Kelimeler: Santrifüj pompa, Sulama, Değişken hız, Güç tüketimi, Enerji etkinliği

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COMPARISON OF ENERGY EFFICIENCIES OF CONSTANT AND VARIABLE FLOWRATE OPERATIONS WITH A SMALL SCALE IRRIGATION PUMP (MASTER THESIS)

ALAA A. SAHIB

ABSTRACT

The objective of this study was to determine the most energy efficient operation of a small centrifugal pump under different operating conditions. A small centrifugal pump test bench was used to measure the flow rate, inlet and outlet pressures, and power consumption to meet the objective. The study included four replications of constant flow and variable flow experiments, and three replications of constant pressure experiments. The tests were done by varying the flow rate and pressure demand by using an outlet pressure valve, inlet pressure valve, by-pass valve, and variable speed drive. The flow rates and pressures were varied from 0 $m^3 h^{-1}$ to 3.0 $m^3 h^{-1}$ and 0 bar to 5 bar, respectively for the flow rate and the pressure.

When there was no variable speed drive (VSD) on the pumping unit, the most energy efficient way to deliver different flow rates was to use the by-pass line valve (410 Watt), followed by the suction line and the pressure line valves, respectively (420 Watt and 920 Watt). The energy saving was about 45% when the by-pass valve was used instead of the outlet valve on the pumping system. The use of inlet valve could be a good alternative in varying the flow rate for irrigation systems, but at the cost of cavitation due to increased vacuum pressure. The vacuum decreased from -0.3 bar to -0.54 bar for a 20% decrease in the flow rate when the inlet valve was used to vary the flow rate. Constant pressure at different flow rates could be achieved either by using the VSD or the by-pass valve. However, power consumption decreased in the case of VSD whereas it was almost the same when the by-pass valve was used to vary the flow rate at given head requirements. The VSD controlled system was more energy efficient by 0% to 28% depending on the flow rate demand. As the flow rate demand of the irrigation system decreased, the use of VSD became more efficient compared to the use of by-pass valve for varying the flow rate.

Keywords: Centrifugal pump, Irrigation, Variable Speed, Power consumption, Energy efficiency.

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KÜÇÜK GÜÇLÜ BİR SULAMA POMPASI İLE SABİT VE DEĞİŞKEN DEBİLİ ÇALIŞMADA ENERJİ ETKİNLİĞİNİN KARŞILAŞTIRILMASI

ÖZET

Bu çalışmanın amacı, küçük güçlü bir santrifüj pompaj tesisinin farklı çalışma şartları altında enerji etkinliğinin belirlenmesidir. Bu amaca ulaşmak için pompaj tesisinde debi, emme basıncı, basma hattı basıncı ve sistemin güç tüketimi ölçülmüştür. Denemeler; sabit debi ve değişken debi şartlarında dört, sabit basınç çalışma koşullarında üç tekrarlı yapılmıştır. Testler; emme vanası, basma vanası, by-pass vanası ve değişken hız kontrolü kullanılarak debi ve basıncın değiştirilmesi suretiyle gerçekleştirilmiştir. Testlerde debi 0-3.0 m^3 h^{-1} ve basınç 0-5 bar aralığında değiştirilmiştir.

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Sabit basınçta çalışma şartları, değişken hız ayarı veya by-pass vanası kullanılarak sağlanabilir. Ancak, eşit basınç gereksinimi için değişken hız kontrolü kullanıldığında güç tüketimi azalırken by-pass vanasının kullanımı güç tüketimini önemli ölçüde etkilememiştir. Güç tüketimi değerleri, pompaj tesisinin yüksek debilerinde (3.0 m^3 h^{-1} - 2.6 m^3 h^{-1}) değişken hız kontrolü(758 W) ve by-pass vanası(746 W) için yakın bulunmuştur, ancak debi daha fazla düşürülmek istenirse değişken hız kontrolü avantajlı duruma gelmiştir. Değişken hız ayarlı çalışma koşulu, gerek duyulan debiye bağlı olarak enerji etkinliğini by-pass vanası ayarına göre %0-28 oranında artırmıştır. Debi gereksinimi azaldıkça, değişken debili sistemde enerji daha etkin hale gelmiştir.

Debinin bir fonksiyonu olarak giriş basıncı, çıkış basıncı ve güç tüketimi arasındaki determinasyon katsayıları genellikle yüksek (R^2=0.97-0.99) bulunmuştur. Pompa karakteristiklerinin hesaplanması için santrifüj pompaları için bilinen eşitlikler kullanılamamıştır. Bunun iki nedeni vardır. Birincisi, bu araştırmada kullanılan pompanın karakteristik eğrisi pompalarda yaygın görülen normal bir eğri olmayıp doğrusal bir eğridir. İkincisi, araştırmada kullanılan sistem sadece bir pompadan ibaret olmayıp, bir sulama sisteminin çeşitli elemanlarını da üzerinde bulundurmakta, ayrıca küçük bir sulama sisteminde basınçlı ve basınçsız çalışma koşullarını simüle etmek için küçük emme ve basma yükseklikleri de bulunmaktadır. Bu nedenle, farklı çalışma koşullarında elde edilen enerji tüketimi değerleri, pompa ile değil, araştırmada kullanılan küçük sulama sistemi ile ilişkilendirilebilir.

Anahtar Kelimeler: Santrifüj pompa, Sulama, Değişken hız, Güç tüketimi, Enerji etkinliği

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SUMMARY

The objective of this study was to determine the most energy efficient operation of a small centrifugal pump under different operating conditions. A small centrifugal pump test bench was used to measure the flow rate, inlet and outlet pressures, and power consumption to meet the objective. The study included four replications of constant flow and variable flow experiments, and three replications of constant pressure experiments. The tests were done by varying the flow rate and pressure demand on the pump by using an outlet pressure valve, inlet pressure valve, by-pass valve, and variable speed drive. The flow rate and the pressure were varied from 0 to 3.0 m^3 h^{-1} and 0 bar to 5 bar, respectively for the flow rate and the pressure.

When there was no variable speed drive (VSD) on the pumping unit, the most energy efficient way to deliver different flow rates was to use the by-pass line valve (410 Watt), followed by the suction line and the pressure line valves, respectively (420 Watt and 920 Watt). The energy saving was about 45% when the by-pass valve was used instead of the outlet valve on the pumping system. When the use of outlet valve, inlet valve, and by-pass valve were considered, outlet valve caused rapid increase in the outlet pressure whereas the other two valves decreased the outlet pressure. The flow rate was reduced from 3.0 m^3 h^{-1} to only 1.8 m^3 h^{-1}using the outlet valve due to the increase in the discharge pressure whereas the flow rate could be reduced to 1.0 m^3 h^{-1}using the inlet valve and the by-pass valve. The use of outlet valve to decrease the flow rate increased the outlet pressure from 0.7 bar to 4.96 bar whereas the outlet pressure reduced from 0.7 bar to 0.02 bar in the case of using the inlet valve and from 0.69 bar to 0.05 bar in the case of using the by-pass valve. Therefore, it was the outlet pressure head that had the major effect on measured power consumptions. The use of inlet valve could be a good alternative in varying the flow rate for irrigation systems since it reduced the energy consumption efficiently, but at the cost of cavitation due to increased vacuum pressure. The vacuum decreased from -0.3 to -0.54 bar for a 20% decrease in the flow rate when the inlet valve was used to vary the flow rate. If the flow rate was decreased down to 1.0m^3 h^{-1}, then the measured vacuum pressure was -0.84 bar, implying that operating under this condition would be unlikely under practical conditions. The use of outlet valve and the by-pass valve caused significant reductions in the inlet pressure head, eliminating any potential hazards for cavitation due to reduced flow speed and head loss in the suction.

When the VSD was used for reducing the flow rate for non-pressurized irrigation simulation, the energy consumption reduced proportionally with the impeller speed, along with smaller outlet pressures. Compared to the inlet valve (420 W) and the by-pass valve (405 W), the use of VSD was more advantageous (287 W), resulting in 32% and 29% less energy consumption for a 20% reduction on the flow rate from 3.0 m^3 h^{-1} to 2.4 m^3 h^{-1}.

Constant pressure operation at different flow rates could be achieved either by using the VSD or the by-pass valve. However, the power consumption decreased in the case of VSD whereas it was almost the same when the by-pass valve was used to vary the flow rate at given head requirements. The power use at high flow rates (3.0 m^3 h^{-1} and 2.6 m^3 h^{-1}) was similar in using the VSD (758 W) and the by-pass valve (746 W) whereas the VSD became advantageous when the flow rate was further reduced. The VSD controlled system was more energy efficient by 0% to 28% depending on the flow rate demand of the system. As the flow rate demand of the irrigation system decreased, the use of VSD became more efficient compared to the use of by-pass valve for varying the flow rate.

High coefficient of determinations were found, usually about 0.97-0.99, for the inlet pressure, outlet pressure, and the power consumption as a function of flow rate throughout the experiments. The Affinity Laws could not be used to study the measured results to reach common conclusions on the pump characteristics because of two reasons. First, the characteristic curve of the pump was a straight line rather than an actual curve common to centrifugal pumps. Second, the test bench used in this study consisted not only the pump, but several hydraulic system components along with small static lifts in the suction and discharge lines to simulate non-pressurized and pressurized operation conditions in a small irrigation system. These had an effect on the inlet and outlet head requirements. Therefore, the energy consumptions under different settings could not be related to the pump but to the small irrigation pumping unit that was used in the study.

Keywords: Centrifugal pump, Irrigation, Variable Speed, Power consumption, Energy efficiency.

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ALAA A. SAHIB

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1. INTRODUCTION

1.1. Irrigation and Pumping Systems

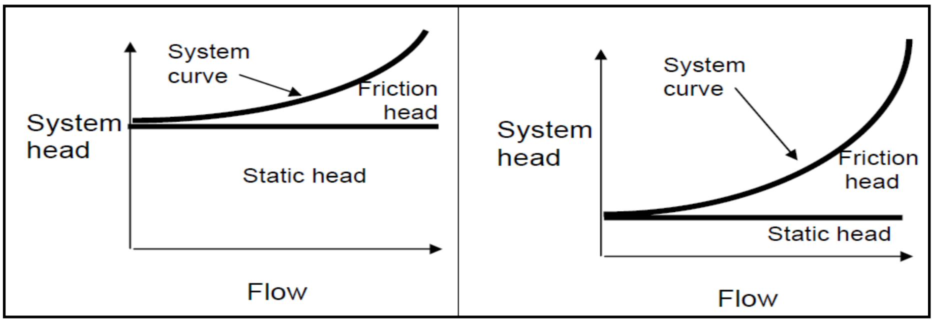
Irrigation is the process of adding water to the land, crops, trees, and orchards where sufficient water cannot be supplied to the plants and crops by the rainfall. It also is used to preserve the environment, landscape, and vegetation of the soil and reduce soil drought during the periods of rainfall deficit (Snyder and Melo-Abreu, 2005). Irrigation may also be helpful in suppressing weed growth in grain fields (Williams et al., 1990).

Irrigation has a critical role in the agricultural productivity because other inputs such as seeds and fertilizers may be useful provided that water is available in the soil (Islam et al., 2007). Furthermore, irrigation is an important factor in providing the food and meeting the fiber needs of increasing population in certain regions of the world (Howell, 2001). Irrigation systems reduce dust effects and are used for disposal of sewage as well (Kumar et al., 2013).

Traditional irrigation or surface irrigation allows the water to flood in the cultivated land, resulting in poor water efficiency from 20% to 50% (Shahidian and Serralheiro, 2012). The low water efficiencies in surface irrigation or in similar methods are due to high water runoff and poor infiltration. Pressurized or localized irrigation systems incorporate pipe networks for distribution of the water to the sections of the field, the plants, or near the root system of the plants. In drip irrigation, spray or micro-sprinkler irrigation and bubbler irrigation, water is delivered in a pre-determined pattern, and applied as a small discharge to each plant (Frenken, 2005). In pressured (closed conduit) networks, water efficiencies are much higher compared to surface irrigation methods. For instance, in drip irrigation water efficiency, evaporation, and runoff are minimized and water efficiency may be up to 80 to 90% (Provenzano, 2007). Systems such as center pivot or linear movement irrigation systems are other types of sprinkler irrigation. These types of irrigation systems consist of connected pipes (Mader and Kan, 2010).

In surface irrigation method, the system does not require an element demanding pressure since the water is delivered to the land freely. Therefore, this type of irrigation is known as non-pressurized irrigation system. On the other hand, sprinkler or drippers require certain pressures (head) to operate. Thus, these types of irrigation systems are known as pressurized irrigation systems. In both types of systems, the water needs to be supplied from a water source (river, well, canal) to the field via a pipe system. Therefore, all pressurized systems require a pump station for operation at specified flow rates and pressure settings while some surface irrigation systems may not require a pump if the water is delivered near the land through a canal system.

Pumping systems are responsible for 20% of the world's electrical energy demand. Energy use of a pumping system may be from 25% to 50% of the total in certain industrial plant operations (DOE, 2004). Pumping system should provide the total pressure head at the flow rate values needed in a system (Chaurette, 2005). In most cases the total pressure head of a system is a combination of static head and friction head and the friction head is proportional to the square of the flow rate (Fig.1.1).



(a) (b)

Figure 1.1 (a) System with high static head, (b) System with low static head (UNEP, 2006)

The irrigation systems rely usually on a centrifugal pump in meeting pressure and flow rate demands. The operational costs in an irrigation system may be high depending on the total head and duration of irrigation. Therefore the performance of the pump has a critical role in meeting the demands of an irrigation system.

Pump performance curve, head, and flow rate determine the performance of a pump as graphically shown in Fig.1.2. This figure is typical for a centrifugal pump where the head decreases with increasing flow rate (UNEP, 2006).

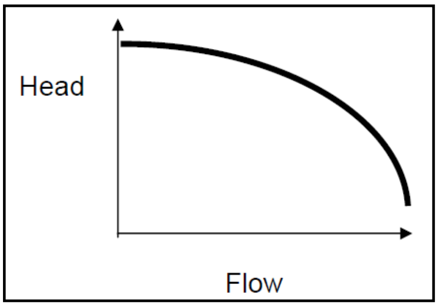


Figure 1.2 Performance curve of a pump (UNEP, 2006)

The operating point of a pump is found by using the system curve and the pump performance curve (UNEP, 2006). Fig.1.3 shows that the system curve and pump performance curve intersect at one point called the operating point. The pumps are selected to work at the specified flow rate and the head at the operation point provided that the efficiency of the particular pump is high at the intersection. When the flow rate or pressure requirement changes in a system, then the system is forced to operate a point other than the operating point. Therefore, whenever the flow rate or the pressure changes, the energy consumption of the system will also vary depending on the change in pump efficiency.

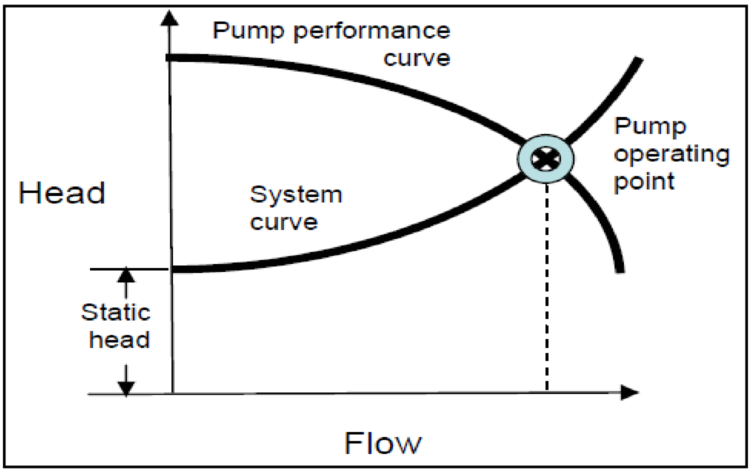


Figure 1.3 Pump operating point (UNEP, 2006)

1.2. Pump Efficiency, Irrigation Efficiency, Energy Efficiency

Pumps can be classified based on the basic operating principle as dynamic or positive displacement pumps (Fig.1.4).

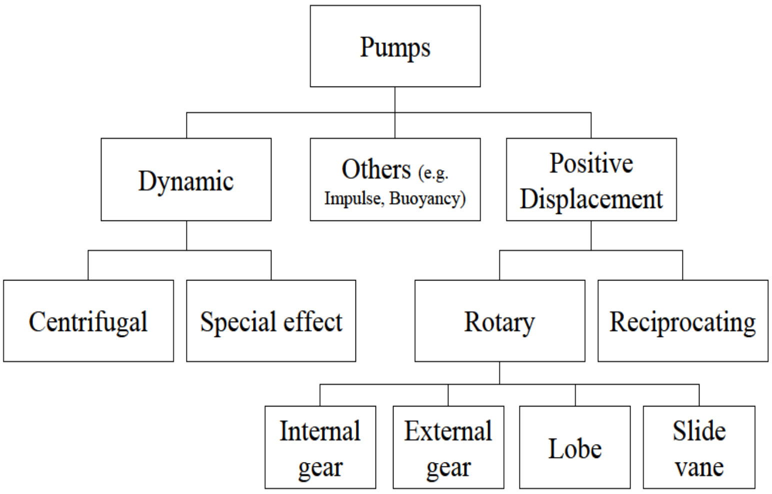


Figure 1.4 Different types of pumps (UNEP, 2006)

Dynamic pumps have a rotating impeller converting kinetic energy into pressure or velocity needed to pump the fluid. Centrifugal pumps are the most common pumps used for pumping water in industrial applications. More than 75% of the pumps in the industry are centrifugal pumps (Fernandez et al., 2002). Special effect pumps are used for specialized conditions in industrial sites (Karassik and McGuire, 1997). Almost all pumps in agricultural irrigation are centrifugal pumps.

Pump efficiency is a function of the hydraulic power at the outlet of the pump and the shaft power that is needed to operate the pump impeller. The actual horsepower delivered to the pump shaft (P_s) and the efficiency of the pump (ηp) can be calculated as follows:

$$Ps = \frac{\mathrm{H_p}}{\eta \mathrm{p}}$$

$$\eta \mathrm{p} = \frac{\text{Hydraulic power}}{\text{Pump shaft power}}$$

Pump output, water horsepower or hydraulic horsepower (H_p) is the liquid horsepower delivered by the pump, and can be calculated as follows:

$$H_p = \frac{\mathrm{Q}\ (\frac{m^3}{s})\ \mathrm{x}\ (\mathrm{Hd}(m) - \mathrm{Hs}(m))\ \mathrm{x}\ \rho\ (\frac{kg}{m^3})\ \mathrm{x}\ \mathrm{g}\ (\frac{m}{s^2})}{1000}$$

where;

Q: flow rate

H_d: discharge head

H_s: suction head

ρ: density of the fluid

g: acceleration due to gravity

Irrigation efficiency is a key measure of the effectiveness of irrigation management. High levels of irrigation efficiency means reduced operating costs, improved production per unit volume of water used and improved environmental management (IAA, 2005). Several parameters affect the irrigation efficiency, including the consumption of energy used to run the pumps. Regarding irrigation energy, the most important factors to consider include the following (Hill, 1999):

1. Total dynamic head (TDH)
2. Pump life
3. Sprinkler operating pressure
4. Hydraulic friction head losses

5. Minor losses in valves and joints
6. Elevation differences across farm
7. Pump horsepower requirements
8. Crop irrigation water needs
9. Application efficiency
10. Seasonal hours of operation

Energy efficiency may be affected by many factors. Pump performance and pumping systems may be improved by considering the above mentioned factors. Furthermore, the main areas for energy conservation include (UNEP, 2006)

1. Selecting the right pump
2. Controlling the flow rate by speed variation
3. Pumps in parallel to meet varying demand
4. Eliminating flow control valve
5. Eliminating by-pass control
6. Start/stop control of pump

The pump performance parameters (flow rate, head, efficiency, and power) will change with varying rotating speeds. The equations that explain these relationships are known as the "Affinity Laws":

1. Flow rate (Q) is proportional to the rotating speed (N).
2. Head (H) is proportional to the square of the rotating speed (N^2).
3. Power (P) is proportional to the cube of the rotating speed (N^3).

The power consumption increases by eight times by doubling the rotational speed of the pump. Also, a small reduction in rotational speed will result in a large reduction in power consumption. Therefore, the energy consumption of a pump depends mainly on the impeller speed, according to the Affinity Laws. As a result, the basis for energy conservation in centrifugal pumps comes from varying the speed of the pump.

1.3. Problem Statement

Regardless of irrigation system being used, there may be a need to change the flow rate in an irrigation system. This need may be driven by irrigating different sizes of fields or field sections using the same pump. When a pump is to be used in different fields, the static head may also differ due to variations in topography. Also, the hottest days during irrigation season may require more water than the early stages of growth of the plants. Thus, different reasons can motivate a farmer to adjust the flow rate in an irrigation system. In automated systems incorporating moisture sensors to manage irrigation, the flow rate needs to be variable as well. In such systems, the flow rate adjustment needs to be done by varying the pump speed automatically. In other cases, the farmers usually use a flow rate valve installed at the outlet (pressure) line or at the inlet (suction) line. Using valves is the usual and the most common methods of varying the flow rate in small scale irrigation systems. However, flow rate adjustment at the outlet may result in high energy consumption whereas flow rate adjustment at the inlet may be critical in terms of cavitation in an irrigation pump.

There was a need to quantify the effect of flow rate valves on energy consumption using an outlet valve, inlet valves, and a by-pass valve in very small irrigation systems. Also, variable flow irrigation pumps offer another alternative to change the flow rate when needed. The published references were not abundant on the energy consumptions of small irrigation systems comparing the effects of the inlet valve, outlet valve, by-pass valve, and VSD on energy efficiency. This was the motivation behind this thesis since there are many small irrigation systems used in agriculture in different parts of the world. Small pumps are also commonly used in urban areas mainly for landscape irrigation systems. In this sub-section, the use of "efficiency" was introduced to clarify the differences among pump efficiency, irrigation efficiency, and energy efficiency. The general and specific objectives of the thesis will be related to the energy efficiency of a centrifugal pump unit, as given in the next sub-section.

1.4. Objectives

The general objective of this study was to do different tests to determine the energy efficiency of a small centrifugal pumping station under different operation conditions. The specific objectives were to

1. Conduct constant flow rate tests to determine inlet pressure, outlet pressure, flow rate, and power consumption for different flow rate and pressure values,
2. Conduct variable flow rate tests in combination with valve adjustments to determine the inlet pressure, outlet pressure, flow rate, and power consumption for different flow rate and pressure values,
3. Do constant pressure tests for varying flow rate demands,
4. Compare constant flow rate, variable flow rate, and constant pressure test results to determine the best method of varying the flow rate for the most energy efficient operations.

2. LITERATURE REVIEW

Howard and Murphy (1998) exclaimed that during the last few decades, the VSD market changed significantly. Mechanical and electrical speed drives such as pulleys, gear changers, and eddy current clutches were replaced by electronic or solid state drives. The interest increased in reducing energy consumption and improving the motor efficiencies. As a result, the demand and use of ac drives consisted of more than 30% of the fixed speed motor market being converted with the application of variable speed ac motor drives. It is suggested that the most appropriate and most economical way of improving pump performance is the VSD.

Abu-Zeid (2002) stated that when pumping units are driven by VSD, adjustment and monitoring are flexible for pressure and flow over a very wide range of operation. Furthermore, the VSD improves the performance of pumping system and operates the system as near as possible to the maximum efficiency. This in turn results in energy saving over the range of field operation.

Boyadjis (2004) commented that in centrifugal pump applications with no static lift, power requirements vary with the cube of the pump speed, then small decreases in speed or flow can significantly reduce energy use. Thus, reducing the speed by 20% can reduce input power requirements by approximately 50%.

Barutcu et al. (2007) computed the characteristic curves of the pumps under field conditions. Demand curves of the irrigation network were determined by means of Indexed Characteristic Curve model. VSD was used to match pump characteristic curves and irrigation system demand curve. As a result, compared to the constant speed pump operation, total energy saving achieved was 33%.

ABB (2013) tested VSDs on the water pumps at one of its six pumping stations. Demand from the irrigation system dropped from 291 kW down to 175 kW, corresponding to 40% reduction in electric consumption for one pumping station. In addition to the financial savings on the energy, the farm also had further operational savings in terms of both labor and maintenance.

Shang (2004) introduced a new hydraulic circuit configuration demonstrating high performance and efficiency. Based on Shang’s statements, since PID or similar controllers can be properly integrated with automated systems, variable speed systems also are compatible with automated systems. They used a hydraulic motor and showed that the bypass flow control could reduce the overshoot of the motor speed about 50%. The relative efficiency of the circuit with the bypass flow control system was found to be 1% to 5% lower. They concluded that the postulated bypass control system could be used to generate an energy efficient circuit with outstanding dynamic transient responses.

DOE (2006) reported that the speed of the pumping system is nearly constant at most water pumping stations. The flow rate is controlled to match the load demand using the traditional control methods, including throttling valves or bypass techniques. In these techniques the operating point is changed by increasing the system's back pressure or resistance to flow, resulting in a shift in the pump's operating point, typically away from the most efficient operating point. These methods increase the energy losses and reduce the efficiency performance over the range of operation.

Kaya et al. (2008) measured the flow rate, pressure, temperature, and power consumption to study big industrial facility's pumps at different operating conditions and at maximum load. Energy saving potential was studied for each pump and electric motor studied. It was concluded that the main energy savings result from replacements of the current pumps with low efficiency, maintenance of the pumps whose efficiencies decline at certain range, replacements of high power electric motors with the ones having appropriate power, usage of high efficiency electric motors, and elimination of problems of cavitation.

Da Costa Bortoni et al. (2008) presented an optimization framework based on dynamic programming mathematic tool. This tool was suitable to be used in programmable logic controllers to improve the automated control of parallel pumping loops targeting energy conservation. They concluded that the use of VSDs combined with parallel pump arrangements show a great potential of energy-efficient utilization. The main advantage was maximized pump performance with increased pumping system efficiency, resulting in more profits.

Grandall (2010) carried out a study to assess the performance and efficiency of hydraulic pumps and motors both experimentally and theoretically. A test stand consisting of a pump and motor was developed to determine the efficiency of an axial piston swash plate pump/motor unit. A regenerative loop hydraulic system was used to minimize the power requirement of the system. Low displacement and low speed regimes were tested in the study. Efficiencies ranged from 0% to 82%.

Ahonen (2011) focused on using a frequency converter as a monitoring and analysis tool for a centrifugal pump. Firstly, they discussed the determination of energy efficiency and limits of reliability for the operating region of a VSD centrifugal pump for a laboratory pumping system. They studied three model-based estimation methods for operation points of the pump, and then the accuracy was determined through laboratory experiments. Additionally, a unique method was introduced to find the occurrence of cavitation or flow recirculation in a centrifugal pump by a frequency converter. Although the concentration was on the radial flow end-suction centrifugal pumps, it was stated that the methods studied could also be feasible with mixed and axial flow centrifugal pumps provided that their characteristics were suited.

Sobhy et al. (2011) conducted tests on a variable speed centrifugal pump unit at different operating speeds ranging from 25 Hz to 50 Hz. The objective was to study the effect of speed on vibration of the pump. Dangerous rotating speeds were determined to avoid operating at resonance conditions.

Lamaddalena and Khila (2012) identified the best pumping station operating mode to optimize energy consumption. It was explained that the pump characteristic curves can be adapted to the system curve by installing a VSD to the pumping station. It was shown that energy savings were about 27% and 35% for two districts compared to the current pumping station regulation.

Kang'au et al. (2012) evaluated the technical performance of the 10 pumping units in small scale irrigated agriculture. They studied the energy consumptions during pumping and identified the potential reasons of inefficient energy use, and evaluated the costs of pumping during irrigation. About 60% of the pumps operated below the suggested design efficiency. The lowest and highest fuel consumptions were 4 USD/ha and 96 USD/ha, respectively. The big differences were due to several factors including pump consumption rate.

Viholainen et al. (2013) introduced a new strategy using VSDs to control parallel pumps. The system was based on flow rate estimation and pump operation analysis using the VSDs. The potential for energy saving was studied for the new control strategy with simulations and laboratory tests. They found benefit of the new control strategy suggesting improved energy efficiency and lower risk of mechanical failure of the controlled pumps compared with traditional method of control.

Lamaddalena and Khila (2013) studied maximum energy savings in on-demand irrigation systems. This could be achieved by matching the discharge and the pressure head required by the system during the entire irrigation season. The system characteristic curve can be found by using a suitable stochastic generation and hydraulic model. The characteristic curves of the centrifugal pumps were adapted to the network curve using

AKLA model by equipping the pumping station with VSDs. Several types of regulation based on variable-speed techniques were identified and analyzed. Compared to the current pumping station regulation, maximum energy efficiency may be accomplished when VSDs are used in pump regulation.

García et al. (2013) evaluated water distribution systems for pressurized irrigation for better water use efficiency. Previous methods developed for branched irrigation networks with one single source caused high energy savings. However, these methods were not appropriate for networks having several water supply points. They developed an optimization methodology (WEBSOM) aiming at minimized energy use. The approach was based on operational sectoring for networks with several nodes of water supply. NSGA-II multi objective genetic algorithm and the optimal sectoring operation calendar were used to minimize energy consumption and to find the pressure deficit. This method was tested with three pumping stations and resulted in potential annual energy savings of from 20% to 29%.

Mora et al. (2013) concluded that sustainability and profitability of irrigation depends to a great extent on the energy efficiency of the pumping system, as water supply from wells accounts for most of the energy consumption in irrigation activities all over the world. So their methodology intended to calculate and generalize total maintenance costs in well pumping systems. The study was conducted over 22 well pumping stations in order to analyze the energy efficiency. The results showed the essential role of preventive maintenance works in the improvement of energy and economic efficiency.

NRCS (2010) conducted a test of center pivot sprinkler on a steeply-sloped field. Water was pumped from a well to supply a center-pivot sprinkler system. The pumping plant consisted of an electric motor and vertical turbine pump. Power costs were $0.07 per kWh. The sprinkler irrigation system was a MESA (Mid-Elevation Spray Application) pivot system with 20-psi pressure regulators and nozzles mounted at 6 feet. The estimated pivot flow rate was 877 gpm. The sprinkler irrigated 140 acres of corn with an estimated annual net water requirement of 28 inches, the pumping lift is 100 feet, and the pivot was irrigating a field that has a fairly uniform slope 4%. Pump selection was based on delivering the design flow with the pivot oriented uphill on a 4% slope. Due to the use of pressure regulators in the system, the flow rate was assumed to remain essentially constant, but it was necessary to determine the Total Dynamic Head (TDH) for the pivot at different positions. The pressure requirement at the pivot with no elevation change was 40 psi (operating pressure + friction loss + nozzle height). Estimated annual operating cost without VSD was 9596$/season, and with VSD was 7962$/season.

It was concluded from the above literature review that many studies have been conducted in different aspects of irrigated agriculture. Among the topics studied, researchers also focused on improving the efficiency of the irrigation systems by using the VSDs in the irrigation systems. However, scientific references were not easy to find on the energy efficiency of small powered irrigation pumps. There are many small farms and urban settlements in the world relying on small powered irrigation systems for agricultural production and landscape. Therefore, it is expected that the experimental data that will be obtained in this study will be useful for the end users.

3. MATERIALS AND METHODS

This experimental study was conducted in the Measurements Laboratory of the Biosystems Engineering Department, College of Agriculture, Kahramanmaraş Sütçü Imam University.

3.1. Materials

The experiments were conducted in this study by using a small pump centrifugal test bench (Fig 3.1). The experimental system consists of

a. A centrifugal pump(0.58 kW) driven by an electric motor with Variable Speed Drive (VSD),
b. Analog vacuum meter (0 bar to -1 bar) and outlet manometers (0 bar to 6 bar)
c. Outlet pressure sensor (Mikronetelektronik SBD AFR-455),
d. Flow meter gauge (LZS-32),
e. Flow rate valves (a globe valve for inlet (PiMTAŞ 32-1 DN 25), a globe valve for outlet (PiMTAŞ 32-1 DN 25) and a globe valve for by-pass (PiMTAŞ 20- ½ DN 15)),
f. Water tank,
g. Pipes and connectors (All pipes are 1 inch in diameter except the by-pass pipe with ½ inch),
h. Control panel, including digital wattmeter, digitaloutlet pressure indicator, variable speed potentiometer, emergency off switch, and on/off power switch.

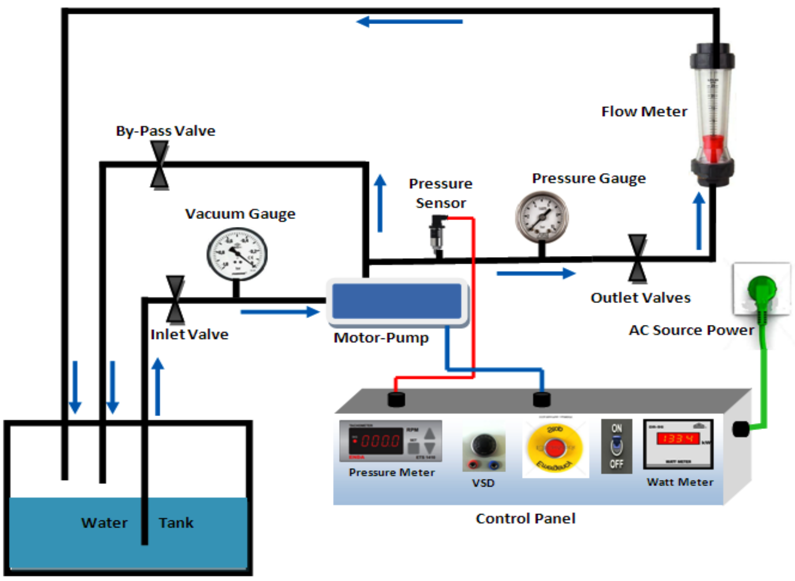


Figure 3.1 Schematic of the centrifugal pump system used in the study

The characteristics of the pump are given in Fig 3.2. The rotational speed of the pump is 2900 rpm with 70 m maximum head and 50 L min^{-1} (3.0 m3 h^{-1}) flow rate. Suction lift of the pump was reported to be 8 m.

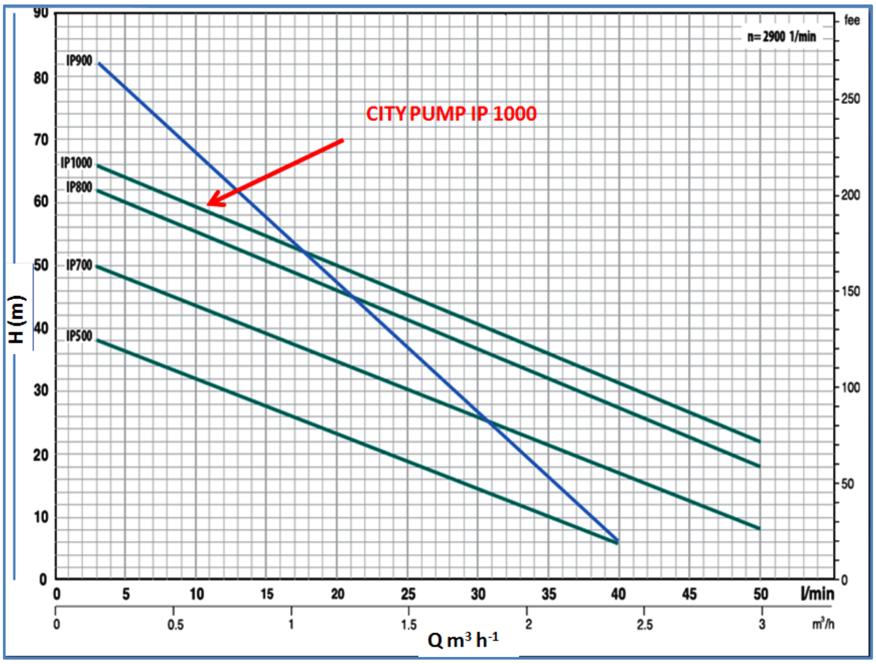


Figure 3.2 Performance curves of the pump used in the study (City Pump, 2014)

A thermo-hydro-anemometer (type delta OHM model DO 9847) was used to measure the ambient conditions, including air pressure and relative humidity. A J type thermocouple was used to measure the air and water temperatures during each set of tests in the preliminary experiments. Preliminary measurements were done to see whether the ambient conditions had an effect on measured parameters at different times under different ambient conditions. It was observed that the measured flow rate, pressure, and power consumption values in the early morning and afternoon tests were the same. Therefore, the effect of ambient conditions was not further tested in the study.

3.2. Methods

In order to meet the objectives of the study, eight experiments were conducted, including three constant flow rate experiments, three variable flow rate experiments, and two constant pressure experiments. Measured quantities were the flow rate, inlet pressure, outlet pressure, and power consumption in each test. All experiments were replicated four times in constant flow and variable flow tests and three times in constant pressure tests.

The average of measured values and associated standard deviations were calculated for comparison purposes.

3.2.1. Constant flow rate experiments

Pump was operated at its maximum speed of 2900 rpm in constant flow rate tests. The flow rate was changed by using the valve on the pressure line (outlet valve), the valve on the suction line (inlet valve), the valve on the by-pass line (by-pass valve), or a combination of these valves.

Experiment 1: The effect of outlet valve on flow rate, inlet pressure, outlet pressure, and power consumption at full pump speed

The pump operated at the maximum speed and the flow rate was varied by using the outlet valve from the highest flow rate of the pump that was 3.0 m^3 h^{-1} to 1 m^3 h^{-1} with 0.2 m^3 h^{-1} decrements.

Experiment 2: The effect of inlet valve on flow rate, inlet pressure, outlet pressure, and power consumption at full pump speed

The flow rate was changed the same way as in Experiment 1 by using the inlet valve in this experiment.

Experiment 3: The effect of by-pass valve on flow rate, inlet pressure, outlet pressure, and power consumption at full pump speed

The flow rates used in this test were 3.0, 2.8, 2.6, 2.4, 2.2, 2.0, 1.8, 1.6, 1.4, 1.2, and 1.0 m^3 h^{-1}.

Experiments 1, 2, and 3 were conducted with open circuit system, i.e., the water being delivered was circulated to the tank freely, simulating non-pressurized irrigation applications.

3.2.2. Variable flow rate experiments

Whereas different flow rate demands were met by using a flow rate valve in constant flow rate experiments, varying flow rate demand was met by changing the speed of the pump impeller in this sub-section. Different flow rates were accomplished by using the VSD available on the test system. VSD was used to adjust the frequency from 0 Hz to 50 Hz in order to change the motor speed from 0 to 2900 rpm, resulting in variable flow rates within a dynamic measurement range of 0 m^3 h^{-1} to 3.0 m^3 h^{-1}.

Experiment 4: The effect of VSD on flow rate, inlet pressure, outlet pressure, and power consumption

The pump speed was varied by using the VSD potentiometer knob to deliver the same flow rates that were tested in constant flow rate tests for comparison. Starting from the maximum speed, the flow rate was reduced gradually, as in the previous tests, to measure the necessary parameters at each step.

Experiment 5: The effect of VSD and outlet valve adjustment on flow rate, inlet pressure, outlet pressure, and power consumption

The flow rates of 2.8, 2.6, 2.4, 2.2, 2.0 m^3 h^{-1}were obtained step by step by using the VSD. Each of these flow rates was taken as the starting point of using the outlet valve for varying the flow rate further. Therefore, in these tests, a set of data were collected to observe the effect of outlet valve in combination of VSD for delivering a specific flow rate.

Experiment 6: The effect of VSD and inlet valve adjustment on flow rate, inlet pressure, outlet pressure, and power consumption

Similar to Experiment 5, the flow rate was first set to a specific value using the VSD and then the inlet valve was used to change the flow rate further. In this experiment, too, a unique set of data were obtained to see the effect of inlet valve on measured parameters in combination with the use of VSD for adjusting the flow rate.

3.2.3. Constant pressure experiments

The tests conducted up to this point represent non-pressurized irrigation systems since there was not any system element, such as a drippers or sprinklers, demanding certain pressures from the pump. The water delivered by the pump was circulated to the water tank on the test set-up. However, many irrigation systems require more head due to pressure demanding elements. Closed hydraulic circuits used in pressurized irrigation systems are operated at specified operating pressures. In irrigation systems, the pressure demands may generally be 1 bar to 3 bar and 2 bar 4.0 bar for drippers and sprinklers, respectively. With the friction losses added, the pump needs to deliver the desired flow rate at a certain total pressure head. Thus, tests were conducted to simulate the operation of a pressurized irrigation system by creating artificial resistance at the outlet of the pump by using a flow restricting valve (a second valve) on the pressure line. Therefore, the tests in this sub-section refer to different flow rates at certain discharge head requirements.

Experiment 7: The effect of VSD at constant outlet pressure settings on inlet pressure and power consumption

Test were done at four different constant pressure values (4.0, 3.5, 3.0, and 2.5 bar) and five different flow rate values (2.4, 2.2, 2.0, 1.8, and 1.6 m^3 h^{-1}) in the pressure line. The corresponding power consumptions and the inlet pressures were measured with three replications.

Experiment 8: The effect of by-pass valve at constant pressure settings on inlet pressure and power consumption

Power consumption and inlet pressure were measured at 4.0, 3.5, 3.0, and 2.5 bar with the flow rate values of 2.4, 2.2, 2.0, 1.8, and 1.6 m^3 h^{-1}. The corresponding power consumptions and inlet pressures were recorded at each pressure and flow rate setting.

The experiments 7 and 8 were replicated three times while all other tests were repeated four times. In all tests conducted, inlet pressure, outlet pressure, flow rate, and power consumption were recorded. The means and the standard deviations were calculated for each test and tabulated. Then the graphical representations were obtained depicting the relations between the flow rate and inlet pressure, flow rate and outlet pressure,and flow rate and power consumption.

Some other supporting graphs were also obtained as necessary. The setting that provided the lowest power to meet the demand (flow rate, or a combination of flow rate with certain head) indicated the most energy efficient method. Thus, the most energy efficient method was chosen by finding the least power consuming method amongst the experiments conducted in this study.

4. RESULTS AND DISCUSSION

4.1. Constant Flow Rate Experiments

Experiment 1: The effect of outlet valve on flow rate, inlet pressure, outlet pressure, and power consumption at full pump speed

Pressures at the inlet and outlet of the pump, and the power consumption of the pumping unit are summarized in Table 4.1. The outlet pressure increased from 0.7 bar to 4.96 bar as the flow rate decreased from 3.0 m^3 h^{-1} to 1.8 m^3 h^{-1}. Power consumption also increased significantly from 451.25 W to 1354.25 W during these tests. Inlet pressure reduced gradually as the flow rate decreased, due to fewer losses in the suction line.

Table 4.1 The effect of outlet valve on measured parameters in constant flow rate tests

flow rate (m^3 h^{-1})	inlet pressure (bar)	outlet pressure (bar)	power consumption (W)
3.0	-0.30 ± 0.000	0.70 ± 0.000	451.25 ± 03.500
2.8	-0.27 ± 0.006	2.86 ± 0.104	801.50 ± 16.523
2.6	-0.23 ± 0.000	3.46 ± 0.104	920.25 ± 18.661
2.4	-0.21 ± 0.000	3.92 ± 0.064	1026.75 ± 06.397
2.2	-0.19 ± 0.006	4.37 ± 0.029	1140.00 ± 11.518
2.0	-0.14 ± 0.000	4.65 ± 0.058	1241.25 ± 13.150
1.8	-0.12 ± 0.000	4.96 ± 0.050	1354.25 ± 14.198

The relationship between the flow rate variation and the inlet pressure of the pump was given in Fig. 4.1. Inlet pressure demand decreased with decreasing flow rate, posing fewer hazards in terms of cavitation. The minimum value of inlet pressure was -0.3 bar at the maximum flow rate.

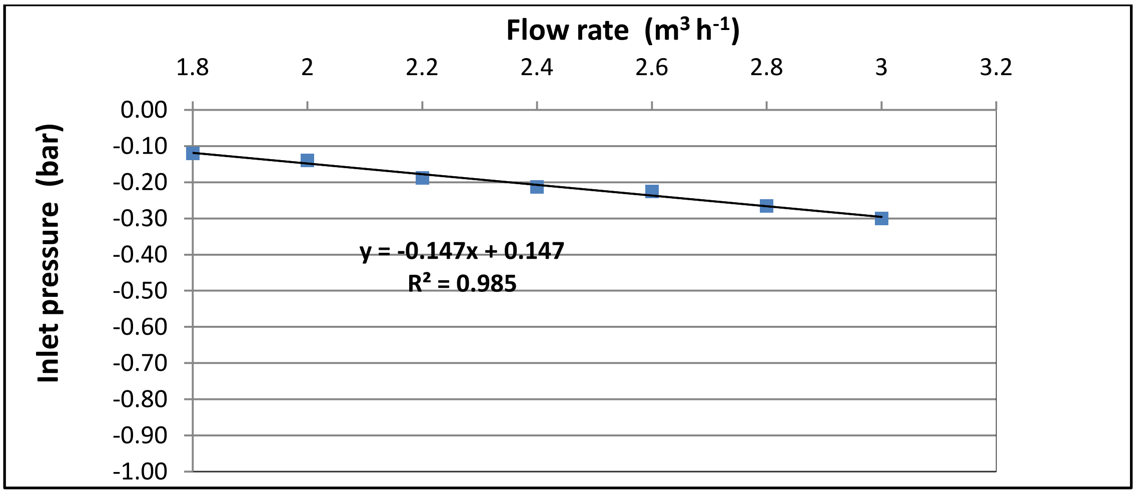


Figure 4.1 The effect of outlet valve on the relationship between the flow rate and inlet pressure in constant flow rate tests

The relationship between the flow rate variation and the outlet pressure of the pump at full engine speed was given in Fig. 4.2. Outlet pressure increased as the flow path was restricted more and more by the flow rate valve, resulting in high head demand from the pump. A high coefficient of determination (R^2=0.955) was obtained between the flow rate and measured outlet pressure. The coefficient of determination could be higher but this system does not relate only to the pump but the major and minor losses at the outlet due to the system components, which might have affected the regression equation and hence the coefficient of determination.

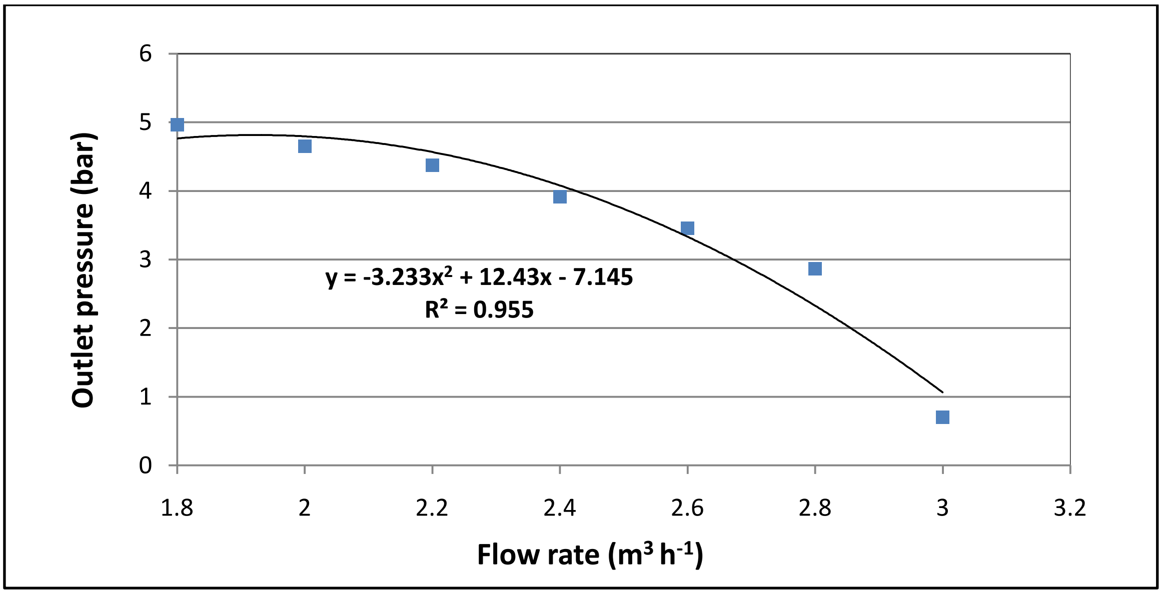


Figure 4.2 The effect of outlet valve on the relationship between the flow rate and outlet pressure in constant flow rate tests

The effect of using an outlet valve to vary the flow rate on power consumption was outstanding. The pump power is function of flow rate and pressure. Although the flow rate was decreased, the pressure demand was too high as a result of flow restriction in the outlet line, resulting in increased power consumption. Fig. 4.3 visually confirms the tabulated data in Table 4.1. Therefore, in a pumping system, less flow rate demand did not reduce the power consumption. In an attempt to reduce the flow rate by using the outlet valve, the head requirement dramatically increases, resulting in significantly higher power consumption. Power consumption increased from 451 W to about 1354 W for flow rates of 3 m^3 h^{-1} and 1.8 m^3 h^{-1}, respectively. For a 20% change in flow rate from 3.0 m^3 h^{-1} to 2.4 m^3 h^{-1}, the power consumption increased from 451 W to 1027 W. Therefore, a 20% reduction in flow rate resulted in about 100% more energy consumption. Further reduction in the flow rate resulted in even more power consumption. It was concluded that using an outlet valve was not an energy efficient method to vary the flow rate since it resulted in more power consumption at a lower flow rate delivery.

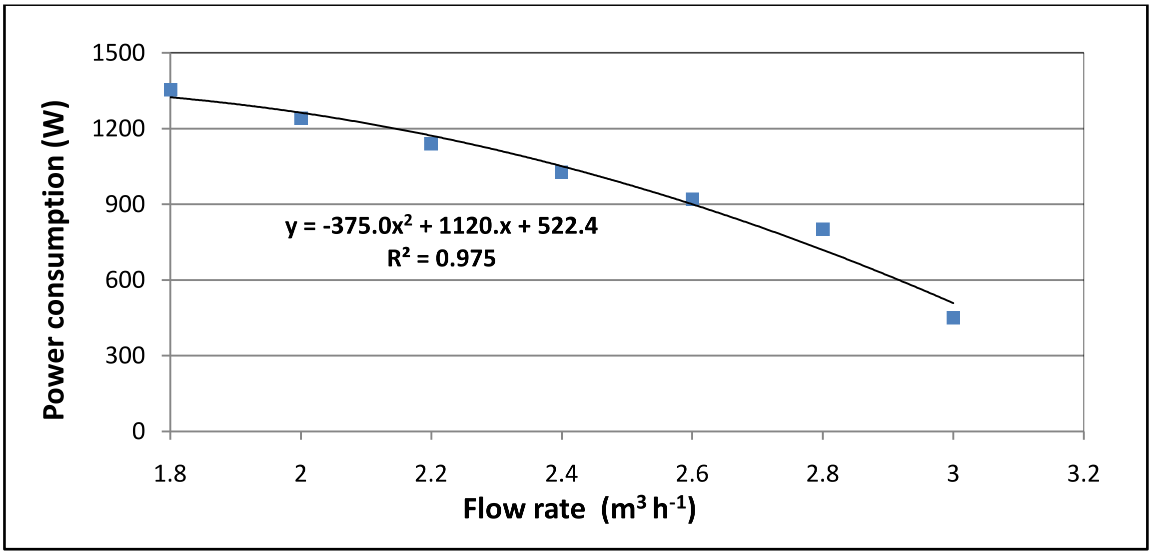


Figure 4.3 The effect of outlet valve on the relationship between the flow rate and power consumption in constant flow rate tests

Experiment 2: The effect of inlet valve on flow rate, inlet pressure, outlet pressure, and power consumption at full pump speed

As shown in Table 4.2, decreasing the flow rate by using an inlet valve increased the vacuum pressure in the inlet line significantly while reducing the overall head requirement due to reduced losses in the pressure line. Average power consumption reduced in accordance with the reduced flow rate.

Table 4.2 The effect of inlet valve on measured parameters in constant flow rate tests

flow rate ($m^3 h^{-1}$)	inlet pressure (bar)	outlet pressure (bar)	power consumption (W)
3.0	-0.30 ± 0.000	0.70 ± 0.000	451.50 ± 3.697
2.8	-0.39 ± 0.010	0.61 ± 0.010	443.50 ± 5.066
2.6	-0.44 ± 0.014	0.50 ± 0.010	433.00 ± 2.708
2.4	-0.54 ± 0.017	0.41 ± 0.010	420.00 ± 3.559
2.2	-0.61 ± 0.013	0.33 ± 0.015	409.00 ± 2.582
2.0	-0.66 ± 0.005	0.25 ± 0.000	397.25 ± 4.425
1.8	-0.70 ± 0.000	0.20 ± 0.000	386.50 ± 3.000
1.6	-0.75 ± 0.006	0.15 ± 0.000	377.00 ± 3.559
1.4	-0.77 ± 0.008	0.11 ± 0.012	369.00 ± 3.367
1.2	-0.81 ± 0.010	0.05 ± 0.005	358.00 ± 3.559
1.0	-0.84 ± 0.029	0.02 ± 0.005	348.00 ± 3.559

The relationship between flow rate variation and the inlet pressure of the pump is graphically shown in Fig. 4.4. The inlet pressure at the highest flow rate 3 $m^3 h^{-1}$ was -0.3 bar whereas it reduced to -0.54 bar at 2.6 $m^3 h^{-1}$. Under these conditions, in order to supply water from the source, the static lift could be only a few meters in the pumping system to avoid cavitation even under a 20% reduction in flow rate using the inlet valve. When the flow rate was decreased further up to about 50% (1.6 $m^3 h^{-1}$) the suction pressure dropped to -0.7 bar, implying critical operation that could cause cavitation even under very small static lifts.

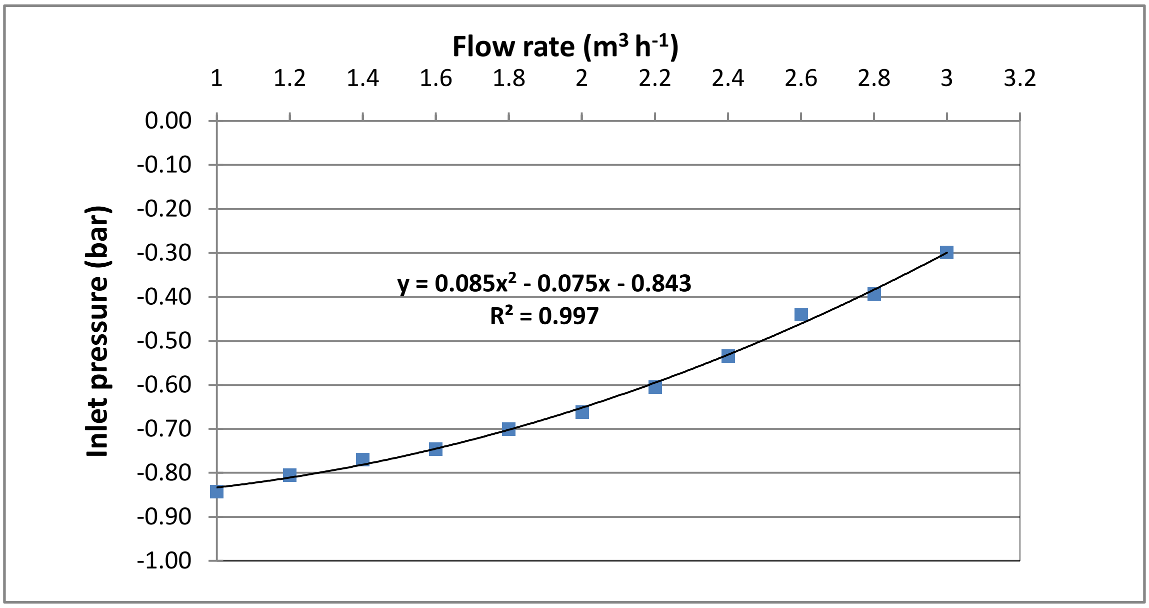


Figure 4.4 The effect of inlet valve on the relationship between the flow rate and inlet pressure in constant flow rate tests

The measured pressure at the pump outlet was significantly affected by the reduction in the flow rate (Fig. 4.5). Rapid decrease in the head demand can be explained by the mathematical equations known for hydraulic systems. Head loss is proportional to squared flow speed and the speed is proportional to the flow rate. Thus, decreasing flow rate rapidly decreases the head demand from the system. In this study, the purpose was to test the pumping unit as an irrigation system rather than testing the pump itself. Therefore, the measured parameters do not relate to the pump only, but to the pumping unit with all the elements installed as shown in Fig. 3.1.

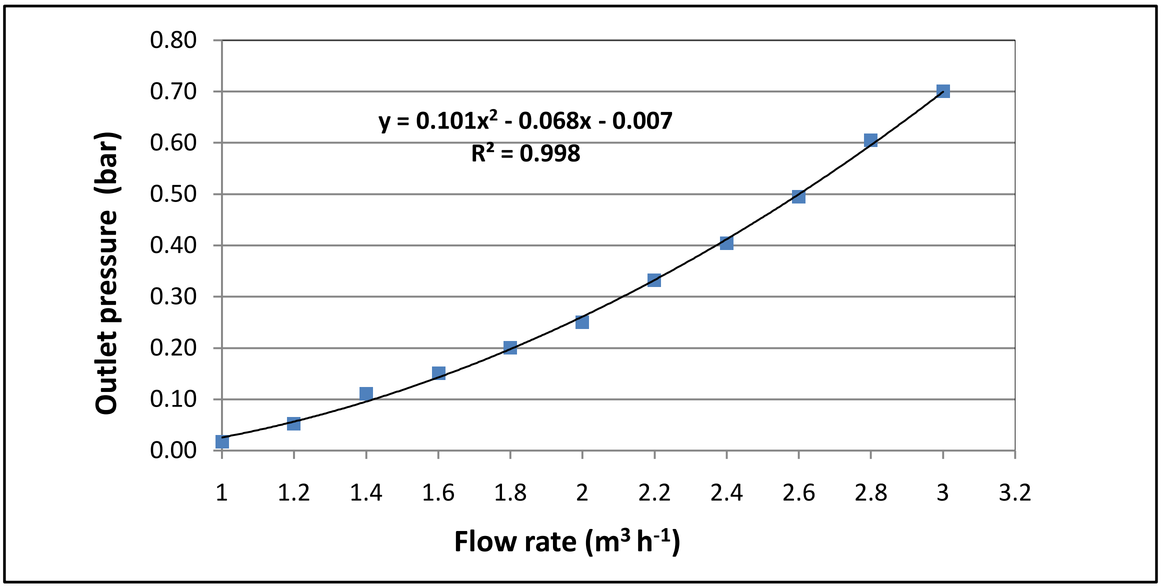


Figure 4.5 The effect of inlet valve on the relationship between the flow rate and outlet pressure in constant flow rate tests

As expected, power consumption was affected by decreased flow rate, resulting in reduced outlet head requirement (Fig. 4.6). The regression equation showed that the relationship between the pump power consumption and the flow rate was linear and did not obey the mathematical equations known for centrifugal pumps. This must have resulted from the fact that the characteristic curve of the pump (Fig. 3.2) is a straight line, showing an inverse but linear relationship between the flow rate and head.

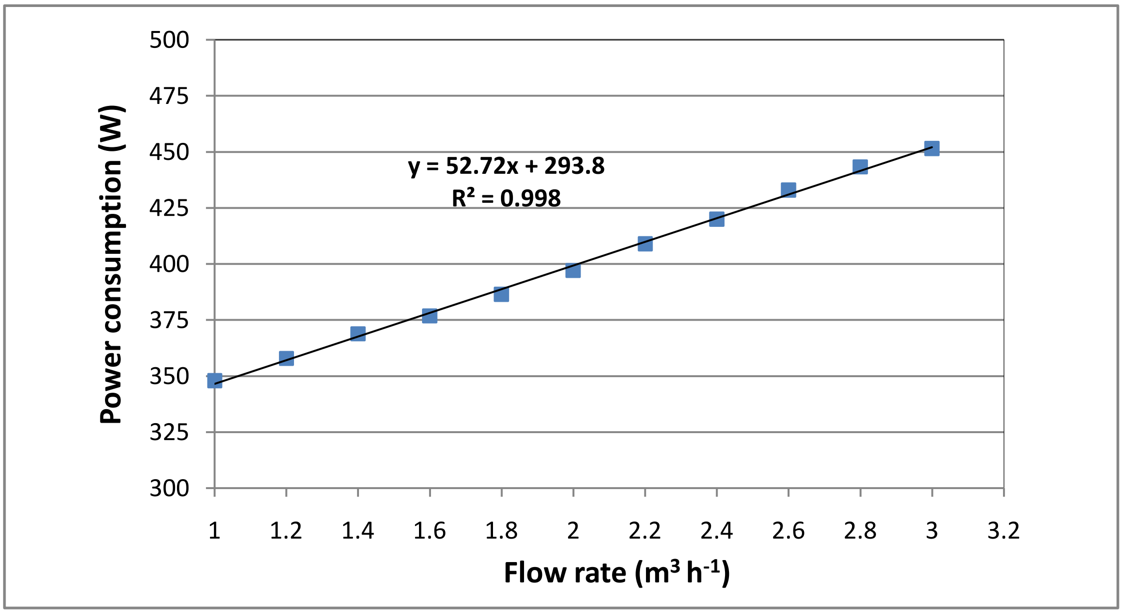


Figure 4.6 The effect of inlet valve on the relationship between the flow rate and power consumption in constant flow rate tests

In most centrifugal pumps, the characteristic curve is a curve where head is not linearly related to flow rate. In most cases, however, the pump characteristic curve looks like the one in Fig. 1.2. Power is known to be a function of cubed flow rate in centrifugal pumps. However, due to the different characteristic curve used in this study, the power was also affected in a different manner compared to widely used pumps in the market. As a result, the flow rate and power consumption was found to be linear.

A high coefficient of determination (R^2=0.998) was obtained between the flow rate and the power consumption (Fig. 4.6). Based on the high R^2 value, it can be stated that all the variation in the power consumption can be explained by the change in the flow rate. Thus, it was concluded that by using the inlet valve to reduce the flow rate, the head demand also decreases, resulting in a reduction in the power consumption as well. Also, the power consumption could be modeled by using a linear regression equation with the flow rate being the independent variable.

In this test, about 7% less energy was required as the flow rate was decreased by 20% from 3 m^3 h^{-1} to 2.4 m^3 h^{-1}. However, the inlet pressure demand increased significantly as a result of reduced flow rate by using a valve in the inlet line, posing a cavitation hazard in the pump.

Experiment 3: The effect of by-pass valve on flow rate, inlet pressure, outlet pressure, and power consumption at full pump speed

The measured average values as a result of using a by-pass valve to deliver different flow rates were given in Table 4.3. Based on the data in Table 4.3, the use of the by-pass valve for varying the flow rate did not have an effect on inlet pressure. However, outlet pressure reduced significantly. Therefore, water could be delivered with much less outlet head demand with decreasing flow rate. Power consumption decreased accordingly in these tests.

The resulting figures showing the relationships between the flow rate and inlet pressure, flow rate and outlet pressure, and flow rate and power consumption were shown in Figs. 4.7-9. It was interesting to note that the flow rate did not have an effect on measured inlet pressure (Fig. 4.7). The outlet pressure however was affected by the variation in the flow rate. The suction pipe was very short with a filter and a flow rate valve installed whereas the pressure line was longer with more valves and elbows with a small static lift. Thus, the effect of varying flow rate on outlet head demand was apparent and measurable.

Table 4.3 The effect of by-pass valve on measured parameters in constant flow rate tests

flow rate ($m^3 h^{-1}$)	inlet pressure (bar)	outlet pressure (bar)	power consumption (W)
3.0	-0.3	0.69 ± 0.010	425.50 ± 3.317
2.8	-0.3	0.64 ± 0.030	422.25 ± 8.221
2.6	-0.3	0.55 ± 0.034	410.75 ± 6.602
2.4	-0.3	0.47 ± 0.056	405.25 ±14.796
2.2	-0.3	0.40 ± 0.059	386.75 ±10.436
2.0	-0.3	0.32 ± 0.037	375.00 ± 9.092
1.8	-0.3	0.25 ± 0.038	363.25 ±6.946
1.6	-0.3	0.18 ± 0.029	354.25 ±6.238
1.4	-0.3	0.13 ± 0.021	345.75 ±4.193
1.2	-0.3	0.08 ± 0.017	338.25 ±4.646
1.0	-0.3	0.05 ± 0.014	333.50 ±1.291

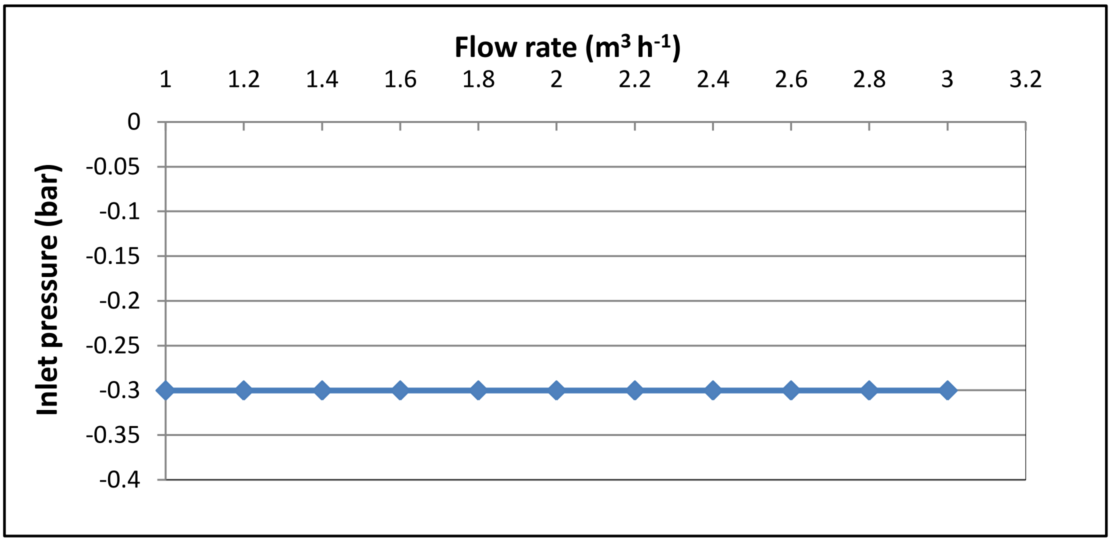


Figure 4.7 The effect of by-pass valve on the relationship between the flow rate and inlet pressure in constant flow rate tests

Since the system was operated as an open circuit in this experiment, it simulated a non-pressurized irrigation application. As a result, not much pressure was required during the tests, resulting in the outlet pressure demand of only about 0.7 bar for the maximum flow rate (Fig. 4.8). The power consumption was small as a result of reduced flow rate and reduced outlet head demand. Power was reduced from 425 W to 333 W, respectively for flow rates of 3 $m^3 h^{-1}$ and 1 $m^3 h^{-1}$. As a consequence, energy consumption was reduced by 22% by reducing the flow rate using a by-pass valve. Energy saving was about 5% for 20% change from in the flow rate 3 $m^3 h^{-1}$ to 2.4 $m^3 h^{-1}$. To recall, in the case of using the outlet valve, 20% reduction in the flow rate increased the energy consumption more than 100%.

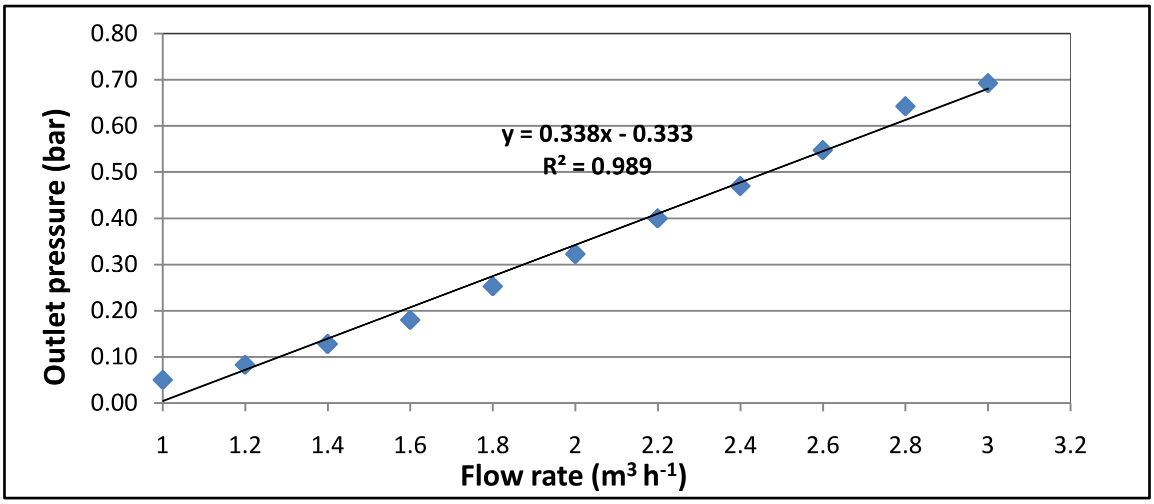


Figure4.8 The effect of by-pass valve on the relationship between the flow rate and outlet pressure in constant flow rate tests

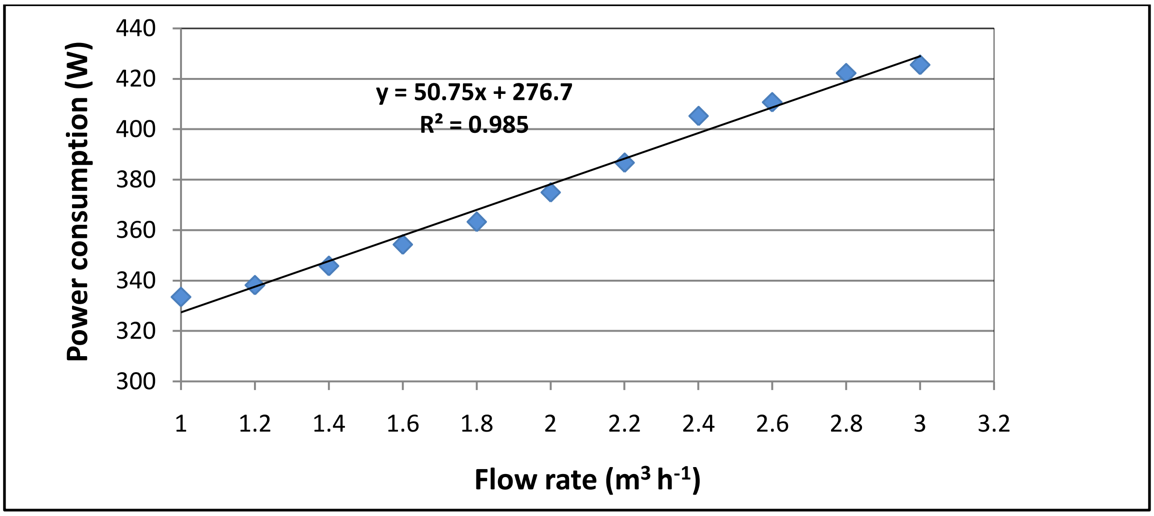


Figure 4.9 The effect of by-pass valve on the relationship between the flow rate and power consumption in constant flow rate tests

Therefore, using the inlet and by-pass valves, the power consumption decreased with decreasing flow rates whereas using the outlet valve, the power consumption increased with decreasing flow rate. In terms of inlet pressure demand, using an outlet valve created no disadvantages whereas using an inlet valve created a critical situation that might lead to cavitation. Thus, it was concluded that the use of by-pass valve for varying the flow rate was the best option among the three cases, namely, the use of inlet valve, outlet valve, and by-pass valve for delivering different flow rates at full pump speed operations.

4.2. Variable Flow Rate Experiments

Experiment 4: The effect of VSD on flow rate, inlet pressure, outlet pressure, and power consumption

Pump impeller speed was increased from 0 to 2900 rpm with equal increments of 290 rpm. The resulting average flow rate, inlet pressure, outlet pressure, and power consumptions were given in Table 4.4. Increasing the pump speed caused an increase in all measured parameters. Inlet pressure head demand gradually increased as a function of flow rate (Fig. 4.10). In Fig. 4.11, the regression equation shows that the relationship between the pump speed and the flow rate was not exactly linear and did not exactly obey the mathematical equations known for centrifugal pumps. Using the theory established by the Affinity Laws, the flow rate should have increased proportionally with the impeller speed. The coefficient of determination was 0.977 for a second degree polynomial whereas 0.962 was found for the linear fit. Although there was no big difference between the two coefficients, better estimation (R^2=0.977) was used in this study. This however, contradicts with the theory that suggests proportional increase in flow rate with increasing pump speed.

Table 4.4 The effect of VSD on measured parameters

pump speed (rpm)	flow rate ($m^3 h^{-1}$)	inlet pressure (bar)	outlet pressure (bar)	power consumption (W)
0	0.000 ± 0.000	0.000 ± 0.000	0.010 ± 0.011	53.625 ± 1.500
290	0.000 ± 0.000	-0.034 ± 0.009	0.027 ± 0.040	104.375 ± 0.816
580	0.460 ± 0.055	-0.073 ± 0.008	0.054 ± 0.021	111.375 ± 1.258
870	0.926 ± 0.048	-0.102 ± 0.002	0.114 ± 0.007	123.250 ± 1.155
1160	1.367 ± 0.058	-0.113 ± 0.005	0.196 ± 0.009	145.250 ± 2.217
1450	1.859 ± 0.070	-0.133 ± 0.007	0.237 ± 0.005	173.750 ± 1.708
1740	2.238 ± 0.094	-0.196 ± 0.005	0.371 ± 0.020	221.750 ± 6.683
2030	2.6245 ± 0.080	-0.223 ± 0.013	0.518 ± 0.018	287.250 ± 4.082
2320	2.9415 ± 0.075	-0.299 ± 0.002	0.674 ± 0.027	348.625 ± 3.464
2610	2.942 ± 0.069	-0.300 ± 0.000	0.703 ± 0.012	406.000 ± 3.697
2900	2.9435 ± 0.071	-0.300 ± 0.000	0.702 ± 0.014	449.250 ± 1.915

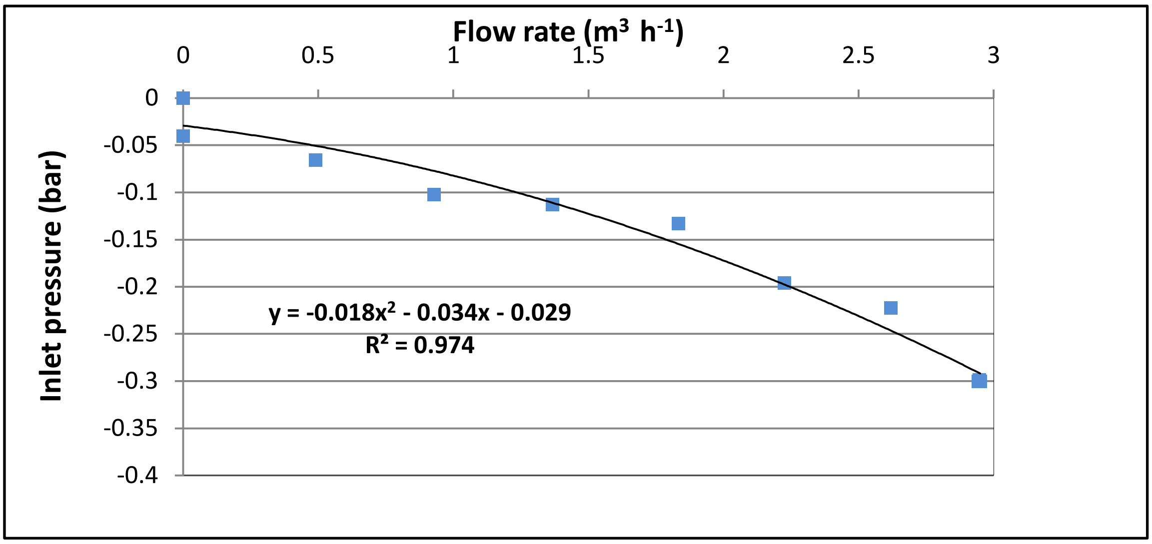


Figure 4.10. The effect of VSD on the relationship between flow rate and the inlet pressure

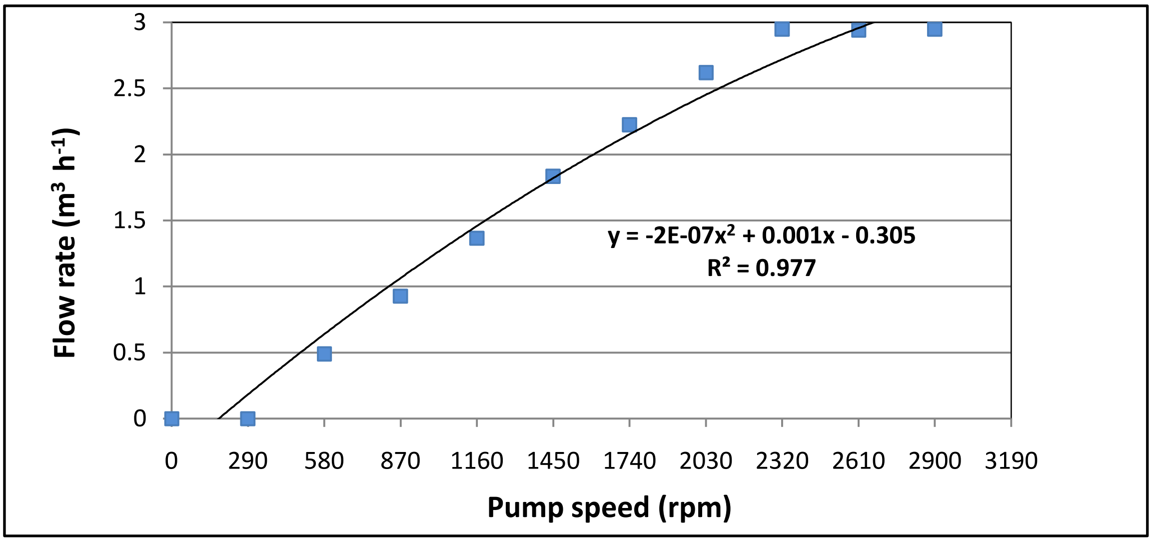


Figure 4.11 The effect of VSD on the relationship between the pump rotational speed and flow rate

The first and the last data points in Fig. 4.11 seem erroneous since they could not be explained either by the theory of pumps or by the system being used for the tests. Increasing the speed did not seem to change the flow rate at these values. This unexpected behavior might have resulted from erroneous potentiometer adjustment. These data points do not explain the change in the flow rate as a function of pump speed. Therefore, these points were excluded from the data set and a better representation was obtained as shown in Fig. 4.12. After the first and last measured data points were deleted, the coefficient of determination increased from R^2=0.977 to R^2=0.995, resulting in a better estimation between the pump speed and the measured flow rate. Deletion of data mentioned above was also applied to the data for the inlet pressure, outlet pressure, and power consumption.

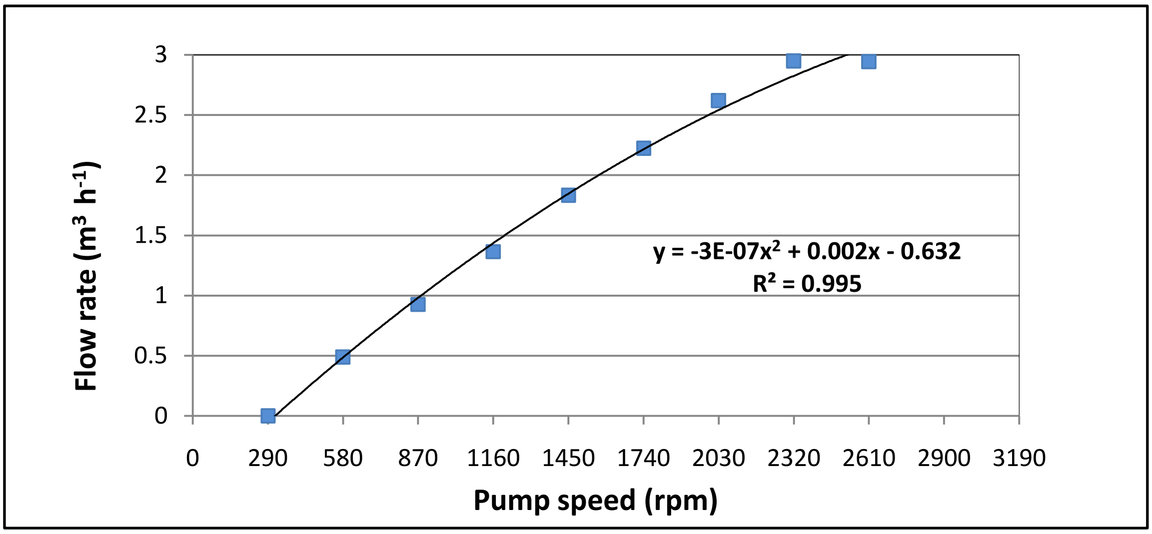


Figure 4.12 The effect of VSD on the relationship between the pump rotational speed and flow rate after excluding erroneous data

The change in the measured outlet pressure as a function of varying flow rate (or pump speed) is depicted in Fig. 4.13. The graph was typical for a system curve in a pumping unit with increasing outlet head as a result of increased flow rate. This test represents non-pressurized system and hence the maximum measured head value was about 0.7 bar. Delivering different volumes of flow rates in the open circuit did not require high head, as expected.

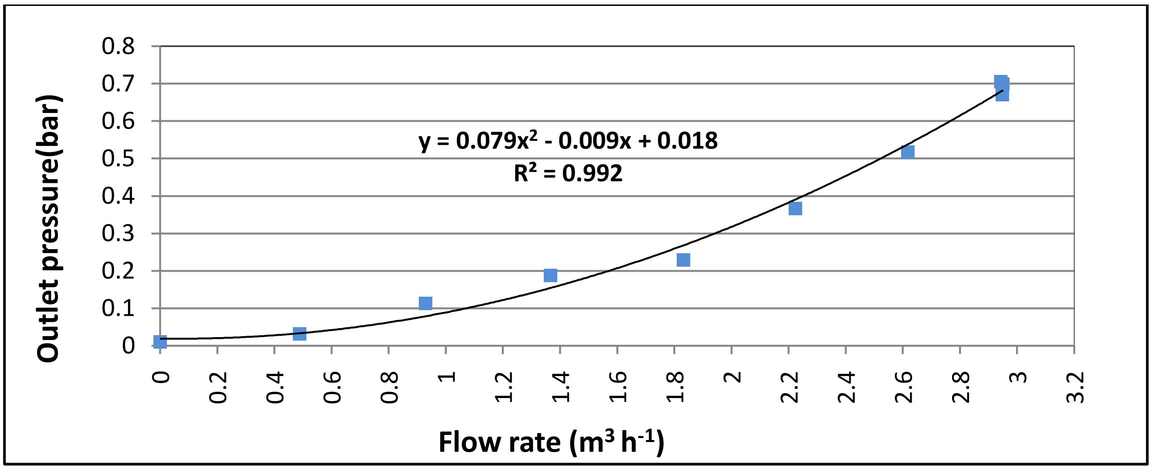


Figure 4.13 The effect of VSD on the relationship between the flow rate and outlet pressure

The trend in the power consumption was similar to the variation in the outlet head requirement (Fig. 4.14). At the lowest speed, although the pump operated at 290 rpm, the pump was not able to deliver water because of high resistance in the hydraulic system. Although no flow rate could be measured at this rotational speed, small amount of energy was consumed to operate the pump and the electrical devices on the control unit. Therefore, there was a power consumption value (104 W) recorded at 290 rpm in Table 4.4, which was also reflected in data shown graphically in Fig. 4.14.

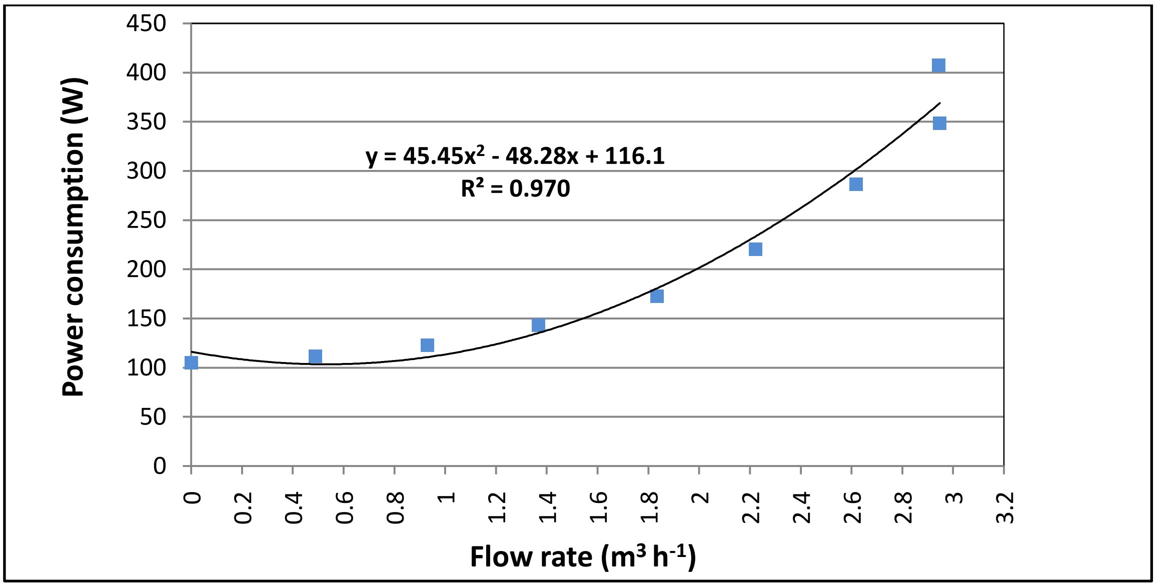


Figure 4.14 The effect of VSD on the relationship between the flow rate and power consumption

The power consumption increased gradually as a function of flow rate in VSD tests on non-pressurized hydraulic system. The power use at the highest speed was 449 W and reduced to 287 W for a 20% reduction in the flow rate, corresponding to 36% energy saving. Also, the energy saving was about 61% when the maximum flow rate was reduced by about 50%. It is apparent that when the flow rate demand changes, varying the pump speed is the most energy efficient method since the desired flow rate could be obtained with much less energy as the flow rate is decreased by the VSD. The energy consumption increased with increasing flow rate in the case of using an outlet valve, as discussed previously. In the case of inlet valve or by-pass valve, the energy could be saved but to a lesser degree compared to VSD. It was concluded that the flow rate should be varied by using the VSD to obtain the most energy efficient operation.

Experiment 5: The effect of VSD and outlet valve adjustment on flow rate, inlet pressure, outlet pressure, and power consumption

In this test, the flow rate was first varied step by step using the VSD to 3.0, 2.8, 2.6, 2.4, 2.2, and 2.0 $m^3\ h^{-1}$. At each flow rate, the outlet valve was used to reduce the flow rate further to observe the system behavior (Table 4.5).

Table 4.5 The effect of VSD and outlet valve on measured parameters

flow rate (m^3h^{-1}) by pump speed	flow rate (m^3h^{-1}) by outlet valve	inlet pressure (bar)	outlet pressure (bar)	power consumption (W)
3	2.8	-0.27 ± 0.01	2.90 ± 0.04	793.25 ± 19.82
	2.6	-0.24 ± 0.01	3.47 ± 0.04	903.75 ± 14.41
	2.4	-0.22 ± 0.00	3.95 ± 0.07	1012.00 ± 24.73
	2.2	-0.19 ± 0.01	4.37 ± 0.05	1116.50 ± 23.16
	2.0	-0.15 ± 0.01	4.69 ± 0.05	1209.00 ± 25.07
	1.8	-0.12 ± 0.00	4.95 ± 0.05	1306.00 ± 25.96
2.8	2.6	-0.23 ± 0.01	0.99 ± 0.03	350.25 ± 8.42
	2.4	-0.22 ± 0.01	1.42 ± 0.07	389.75 ± 12.26
	2.2	-0.19 ± 0.01	1.87 ± 0.05	433.25 ± 9.50
	2.0	-0.14 ± 0.01	2.33 ± 0.10	478.75 ± 12.04
	1.8	-0.12 ± 0.00	2.81 ± 0.10	529.00 ± 16.77
	1.6	-0.12 ± 0.00	3.28 ± 0.08	559.75 ± 40.29
2.6	2.4	-0.22 ± 0.00	0.84 ± 0.02	306.75 ± 3.30
	2.2	-0.19 ± 0.01	1.28 ± 0.02	344.25 ± 4.50
	2.0	-0.15 ± 0.01	1.66 ± 0.02	375.50 ± 5.80
	1.8	-0.12 ± 0.01	2.13 ± 0.02	421.50 ± 2.65
	1.6	-0.12 ± 0.01	2.58 ± 0.03	466.25 ± 6.85
	1.4	-0.11 ± 0.01	3.01 ± 0.02	514.25 ± 3.59
2.4	2.2	-0.19 ± 0.01	0.75 ± 0.08	269.25 ± 9.57
	2.0	-0.15 ± 0.01	1.14 ± 0.04	299.00 ± 7.16
	1.8	-0.13 ± 0.00	1.52 ± 0.03	331.25 ± 7.89
	1.6	-0.12 ± 0.00	1.93 ± 0.05	367.75 ± 9.91
	1.4	-0.11 ± 0.01	2.32 ± 0.05	406.25 ± 12.18
	1.2	-0.11 ± 0.00	2.71 ± 0.08	446.25 ± 15.28
2.2	2.0	-0.15 ± 0.01	0.62 ± 0.04	234.50 ± 2.08
	1.8	-0.12 ± 0.00	1.01 ± 0.02	265.00 ± 0.82
	1.6	-0.12 ± 0.00	1.36 ± 0.02	290.00 ± 2.16
	1.4	-0.12 ± 0.00	1.71 ± 0.03	318.75 ± 1.50
	1.2	-0.11 ± 0.00	2.09 ± 0.02	352.25 ± 4.35
	1.0	-0.11 ± 0.00	2.43 ± 0.03	382.25 ± 3.10
2.0	1.8	-0.12 ± 0.00	0.56 ± 0.05	208.50 ± 5.45
	1.6	-0.12 ± 0.00	0.89 ± 0.04	232.00 ± 4.69
	1.4	-0.12 ± 0.00	1.19 ± 0.03	254.50 ± 4.65
	1.2	-0.11 ± 0.00	1.53 ± 0.06	280.50 ± 6.45
	1.0	-0.10 ± 0.00	1.87 ± 0.06	308.00 ± 6.63
	0.8	-0.10 ± 0.00	2.22 ± 0.06	333.50 ± 3.70

The inlet pressure, outlet pressure, and the power consumption varied similar to Experiment 1 since the outlet valve was also used in both experiments, i.e. the power consumption increased with decreasing flow rate. The results of the test at 3 $m^3\ h^{-1}$ were the same as Experiment 1. As the flow rate was decreased by the VSD, the measured power consumption values decreased. The system was consistent in its behavior in that the energy consumption increased with decreased flow rate at each step. For instance, power consumptions were 479 W and 235 W, respectively for 2.8 $m^3 h^{-1}$ and 2.2 $m^3 h^{-1}$ adjusted by the VSD and then reduced to 2.0 $m^3 h^{-1}$ by using the outlet valve. It can be observed that as the flow rate was reduced further by the VSD, energy savings were more at the same flow rates.

The data from Table 4.5 was rearranged to show the effect of VSD in combination with the outlet valve at a flow rate of 1.8 $m^3 h^{-1}$ (Table 4.6). The first column represents the starting flow rate as adjusted by the VSD while the second column shows the final flow rate achieved by using the outlet valve. It is clear that the power consumption decreased significantly as a consequence of using the VSD to get the same flow rate.

Table 4.6 The effect of VSD and outlet valve at 1.8 $m^3 h^{-1}$ on measured parameters

flow rate ($m^3 h^{-1}$) by VSD	flow rate ($m^3 h^{-1}$) by outlet valve	inlet pressure (bar)	outlet pressure (bar)	power consumption (W)
3	1.8	-0.12	4.95	1306
2.8	1.8	-0.12	2.81	529
2.6	1.8	-0.12	2.13	421.5
2.4	1.8	-0.13	1.52	331.25
2.2	1.8	-0.12	1.01	265
2.0	1.8	-0.12	0.56	208.5

Inlet pressure was quite small and was not affected by using the combination of VSD and outlet valve whereas outlet pressure changed in a wide range from 4.95 bar to 0.56bar. Power consumption behaved the same way as the outlet pressure values (Figs. 4.15 - 4.16) As a result, to get the same flow rate, specifically at 1.8 $m^3 h^{-1}$, the power consumption reduced from 1306 W to 208 W, suggesting 84% energy saving. The saving in energy can all be attributed to the use of VSD, suggesting the irrelevancy of using a combination of VSD with other valves for setting a desired flow rate value.

The effect of VSD and outlet valve for delivering certain flow rates were shown graphically in Fig. 4.15 and Fig. 4.16, respectively for the outlet pressure and power consumption. The effect of VSD and the outlet valve on these parameters were also combined in a single graph as shown in Fig. 4.17. The slopes of the lines in Fig. 4.15 were similar. The gap that was observed between the tests at 2.8 $m^3 h^{-1}$and 2.6 $m^3 h^{-1}$ was relatively large compared to other cases. Similar observations can be made in the case of power consumptions graph in Fig. 4.16. These results suggest that this particular pumping unit should not be used at the maximum flow rate value that can technically be delivered by the pump. In this system, due to high head and energy demand at the highest pump speed, the system must have operated far from the best operating point. Using the VSD, the desired flow rates could be delivered with low energy use, suggesting that VSD expands the best operation region of the pump.

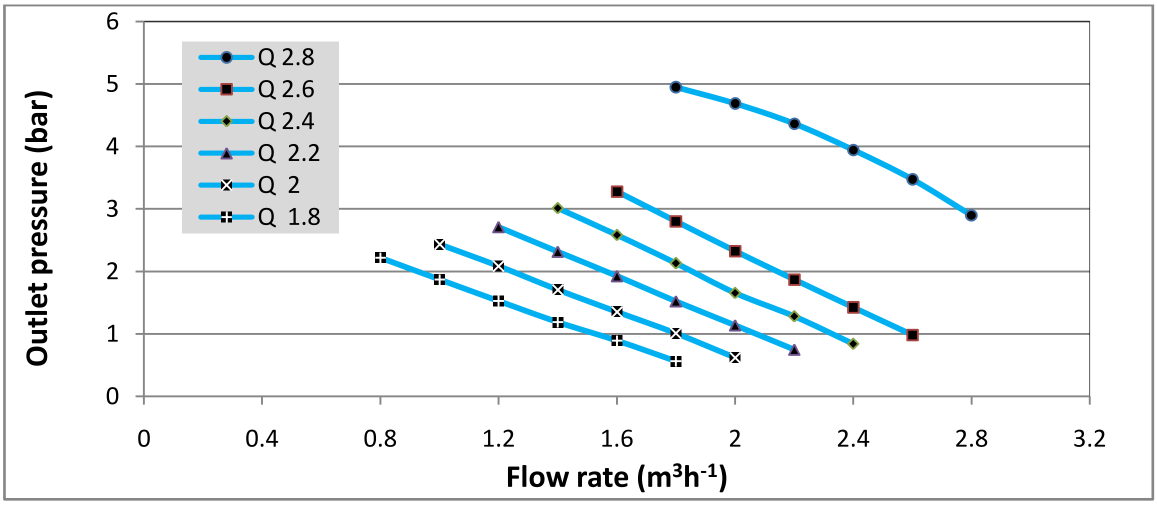


Figure 4.15 The effect of VSD and outlet valve on the relationship between the flow rate and outlet pressure

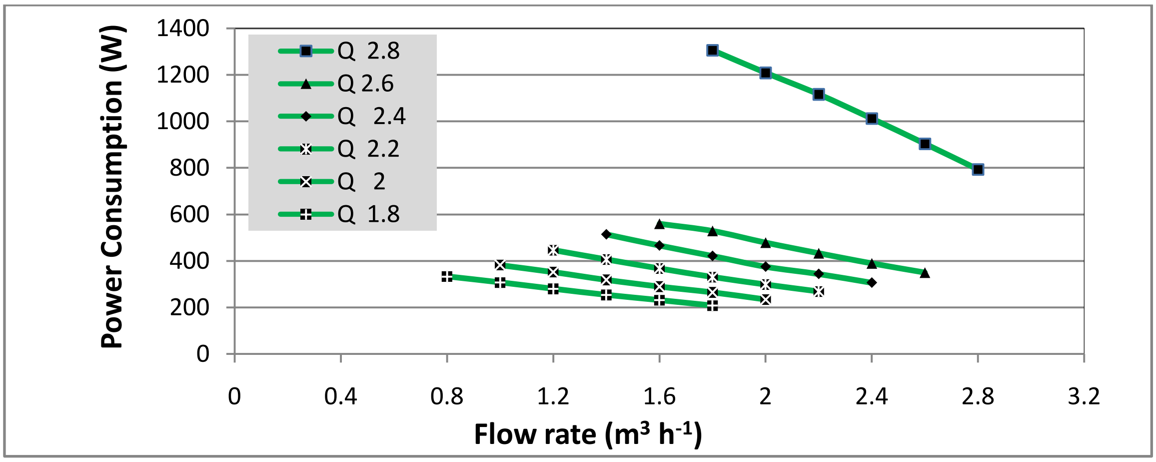


Figure 4.16 The effect of VSD and outlet valve on the relationship between the flow rate and power consumption

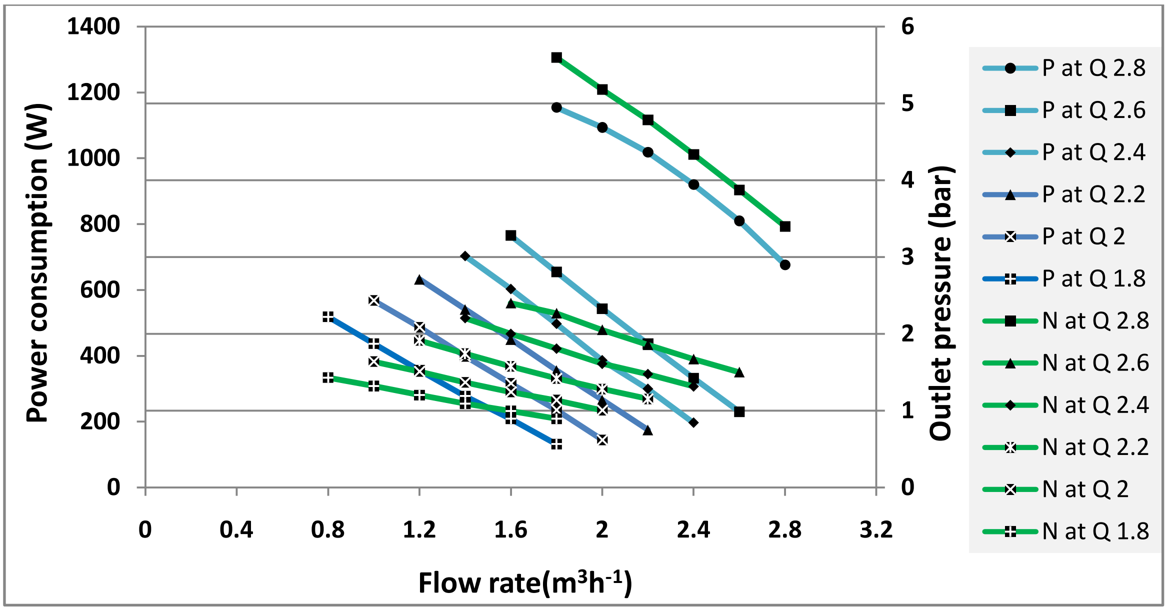


Figure 4.17 The effect of VSD and outlet valve on outlet pressure and power consumption as a function of flow rate (P: outlet pressure, N: power)

Experiment 6: The effect of VSD and inlet valve adjustment on flow rate, inlet pressure, outlet pressure, and power consumption

It was observed that for each starting flow rate value set by the VSD, the inlet pressure increased and the outlet pressure decreased as the flow rate was reduced further by the inlet valve (Table 4.7). In the meantime, as the flow rate was decreased, the power consumption always decreased accordingly. It can be noted that the tests done at a flow rate of 3 m^3 h^{-1} were the same as Test 3. Therefore the results were the same in these tests, except for 2.2 m^3 h^{-1}, which should have resulted from a measurement error or some repeatability problem in the measurement devices.

The pressure load in the suction line increased with decreased flow rate for each starting flow rate value (Table 4.7). For instance, for a starting flow rate of 2.8 m^3 h^{-1}, vacuum needed in the suction line varied from -0.448 bar to -0.753 bar. This obviously have serious implications about cavitation in the pump since vacuum pressure increased significantly as the flow rate was decreased by using the inlet valve. On the other hand, the outlet pressure and the power consumption decreased with varying flow rate. For example, for a starting flow rate of 2.8 m^3 h^{-1} in the first column, the power consumption reduced from 319 W to 296 W and the outlet pressure reduced from 0.49 bar to 0.15 bar. This could be explained by the fact that both the pressure and the flow rate decreased, resulting in less energy use.

Table 4.7 The effect of VSD and inlet valve on measured parameters

flow rate ($m^3 h^{-1}$) by pump speed	flow rate ($m^3 h^{-1}$) by inlet valve	inlet pressure (bar)	outlet pressure (bar)	power consumption (W)
3	2.8	-0.400 ± 0.000	0.60 ± 0.00	442.00 ± 3.46
	2.6	-0.438 ± 0.013	0.50 ± 0.00	429.50 ± 3.00
	2.4	-0.541 ± 0.002	0.43 ± 0.02	415.25 ± 1.89
	2.2	-0.593 ± 0.015	0.34 ± 0.01	405.75 ± 2.99
	2.0	-0.658 ± 0.010	0.26 ± 0.02	394.25 ± 3.30
	1.8	-0.696 ± 0.009	0.20 ± 0.00	383.50 ± 1.29
2.8	2.6	-0.448 ± 0.004	0.49 ± 0.01	318.50 ± 2.65
	2.4	-0.538 ± 0.015	0.43 ± 0.02	313.50 ± 3.00
	2.2	-0.605 ± 0.013	0.35 ± 0.02	309.75 ± 2.22
	2.0	-0.673 ± 0.005	0.26 ± 0.02	305.25 ± 2.36
	1.8	-0.725 ± 0.017	0.21 ± 0.01	300.75 ± 2.87
	1.6	-0.753 ± 0.005	0.15 ± 0.00	296.25 ± 1.71
2.6	2.4	-0.547 ± 0.009	0.42 ± 0.02	295.75 ± 2.36
	2.2	-0.614 ± 0.008	0.33 ± 0.02	290.25 ± 3.50
	2.0	-0.664 ± 0.003	0.26 ± 0.02	286.75 ± 3.59
	1.8	-0.730 ± 0.008	0.20 ± 0.00	283.75 ± 3.59
	1.6	-0.758 ± 0.010	0.15 ± 0.00	279.75 ± 2.36
	1.4	-0.798± 0.010	0.10 ± 0.01	275.75 ± 2.22
2.4	2.2	-0.536 ± 0.058	0.34 ± 0.00	264.25 ± 7.80
	2.0	-0.665 ± 0.010	0.26 ± 0.02	263.75 ± 3.40
	1.8	-0.735 ± 0.010	0.21 ± 0.01	260.50 ± 3.32
	1.6	-0.763 ± 0.012	0.16 ± 0.02	258.25 ± 2.99
	1.4	-0.805 ± 0.010	0.10 ± 0.00	254.75 ± 3.30
	1.2	-0.843 ± 0.015	0.05 ± 0.00	251.50 ± 3.70
2.2	2.0	-0.508 ± 0.068	0.27 ± 0.02	232.25 ± 6.70
	1.8	-0.734 ± 0.011	0.21 ± 0.01	240.00 ± 2.94
	1.6	-0.770 ± 0.014	0.16 ± 0.02	237.50 ± 2.65
	1.4	-0.805 ± 0.010	0.11 ± 0.01	234.25 ± 2.87
	1.2	-0.852 ± 0.019	0.07 ± 0.02	233.25 ± 2.22
	1.0	-0.869 ± 0.014	0.03 ± 0.01	231.00 ± 1.41
2.0	1.8	-0.511 ± 0.048	0.21 ± 0.01	209.50 ± 2.52
	1.6	-0.756 ± 0.007	0.15 ± 0.01	221.50 ± 2.65
	1.4	-0.813± 0.010	0.12 ± 0.01	218.25 ± 2.36
	1.2	-0.853 ± 0.005	0.06 ± 0.01	217.25 ± 1.50
	1.0	-0.871 ± 0.011	0.03 ± 0.01	215.50 ± 2.38
	0.8	-0.894 ± 0.005	0.00 ± 0.01	213.00 ± 2.94

The relationships between flow rate and outlet pressure, flow rate and inlet pressure, and flow rate and power consumption were given in Figs. 4.18-20. The change in flow rate did not significantly change from one step to another (Fig. 4.18). A similar observation can be made on the inlet pressure, as shown in Fig. 4.19. Power consumption was much higher for the highest flow rate, suggesting better energy efficiencies with a small reduction in the flow rate (Fig. 4.20).

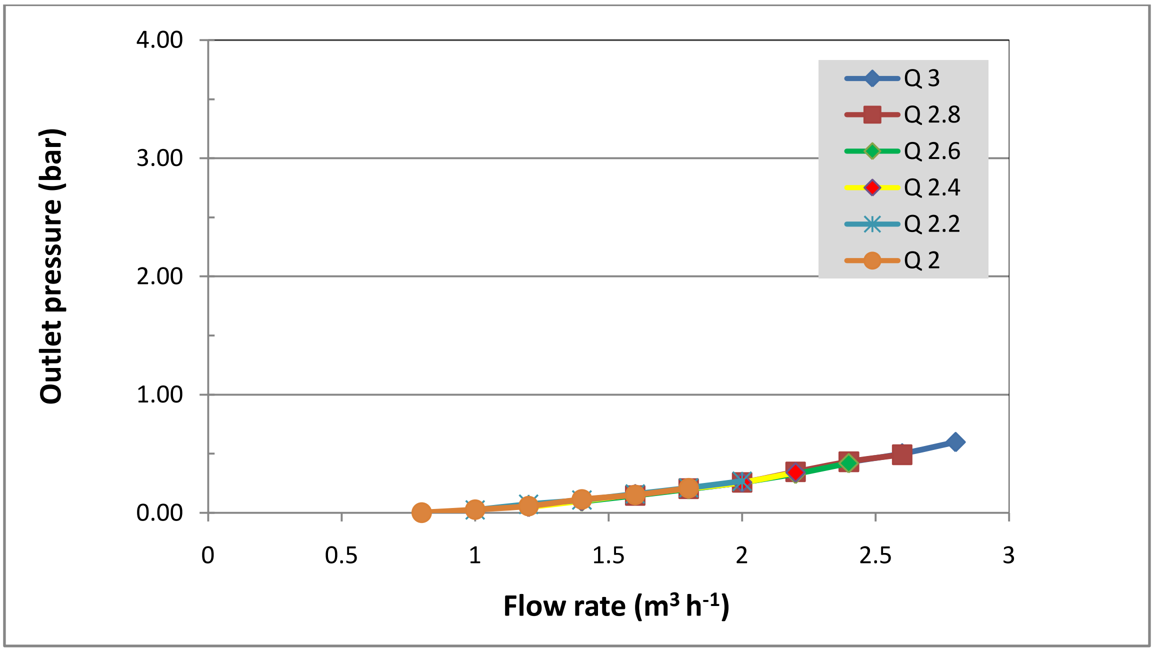


Figure 4.18 The effect of VSD and the inlet valve on the relationship between flow rate and outlet pressure

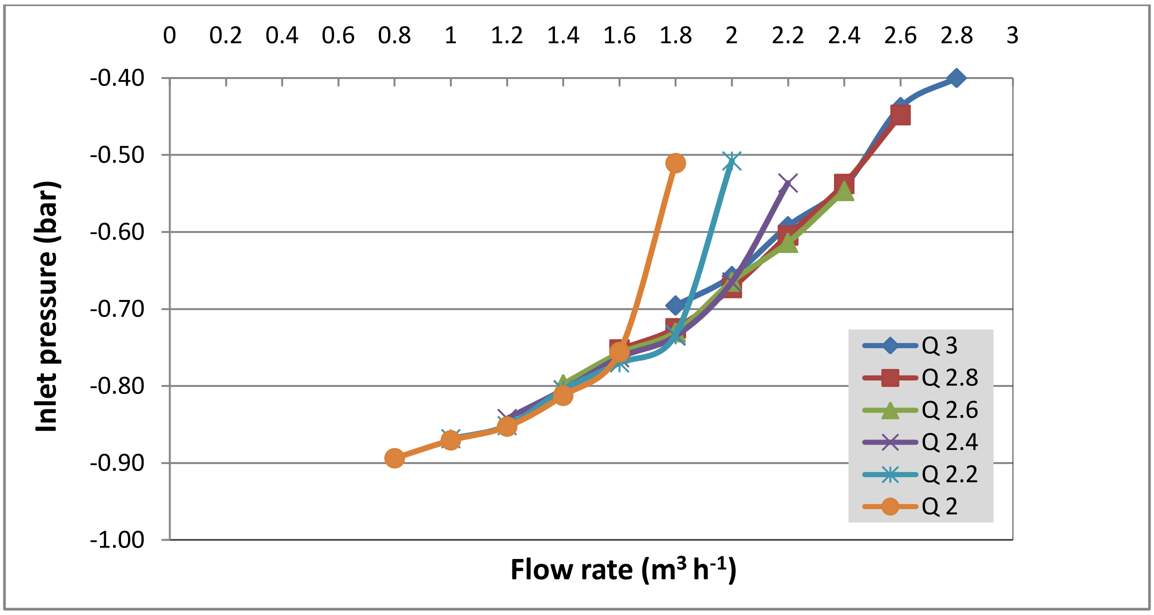


Figure 4.19 The effect of VSD and the inlet valve on the relationship between the flow rate and inlet pressure

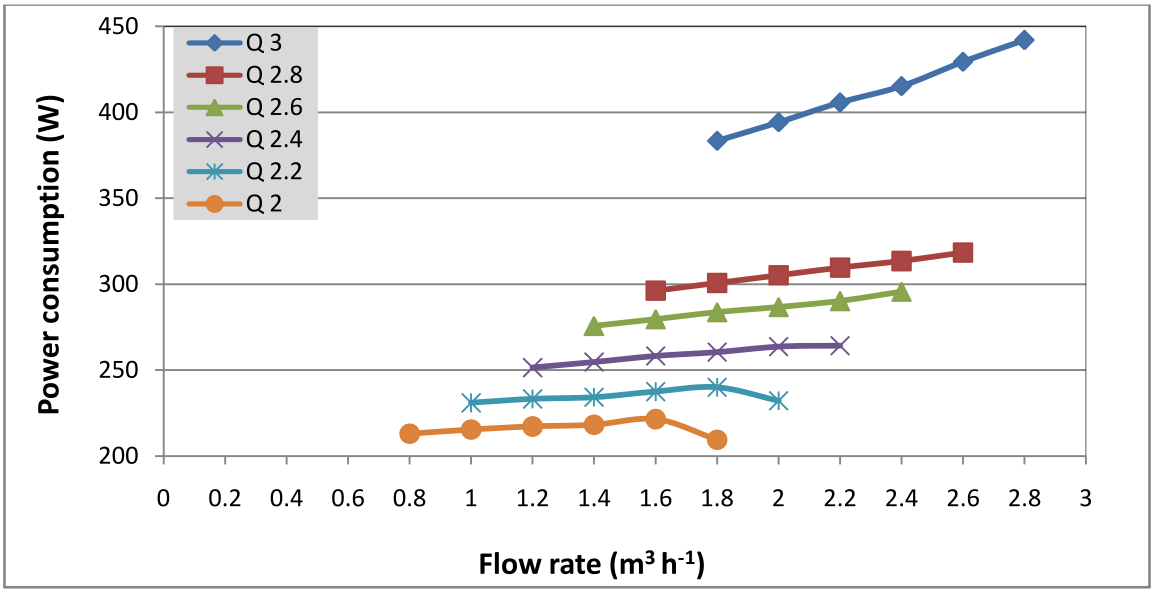


Figure 4.20 The effect of VSD and the inlet valve on the relationship between flow rate and power consumption

4.3. Constant Pressure Experiments

Experiment 7: The effect of VSD at constant pressure settings on inlet pressure and power consumption

In irrigation systems, a certain pressure level might be required at different flow rates depending on the flow rate demand of the irrigated land. Based on irrigated area, different volumes of water will be needed at the same pressure as long as the same pressurized irrigation system elements are in operation. Tests were done to simulate different flow rate demands at the same outlet pressure and the results were summarized in Table 4.8. The VSD was used to deliver the specified flow rate values that were given in the first column in order to supply the outlet head requirements specified in the first column in Table 4.8.

Power consumption decreased as the flow rate was reduced by using the VSD at given outlet pressure settings, which were shown with four curves in Fig. 4.21. The slopes of these curves were not big suggesting gradual reduction in the energy efficiency at a given pressure value as a result of reducing the flow rate. Similarly, the power consumption reduced gradually if the pressure reduced step by step from 4.0 bar to 2.5 bar at a given flow rate. For instance, at 4.0 bar the power consumption decreased from about 1000 W to 760 W in accordance with the flow rate variation. Similarly, at a flow rate of

2.2 m^3 h^{-1} the power consumption reduced from about 940 W to 550 W as the pressure demand reduced from 4.0 bar to 2.5 bar.

Table 4.8 The effect of VSD on measured parameters at constant pressure tests

outlet pressure (bar)	flow rate (m^3 h^{-1})	inlet pressure (bar)	power consumption (W)
4	2.4	-0.213 ± 0.006	1014.00 ± 3.61
	2.2	-0.203 ± 0.015	935.67 ± 5.51
	2.0	-0.143 ± 0.006	875.00 ± 7.00
	1.8	-0.120 ± 0.000	812.00 ± 5.29
	1.6	-0.120 ± 0.000	757.00 ± 8.54
3.5	2.6	-0.260 ± 0.006	950.00 ± 2.00
	2.4	-0.227 ± 0.006	886.67 ± 6.03
	2.2	-0.220 ± 0.000	817.67 ± 2.52
	2.0	-0.163 ± 0.006	738.00 ± 2.00
	1.8	-0.120 ± 0.000	689.33 ± 4.04
	1.6	-0.120 ± 0.000	650.67 ± 3.51
3	2.8	-0.270 ± 0.000	838.00 ± 2.00
	2.4	-0.243 ± 0.006	733.67 ± 1.53
	2.2	-0.213 ± 0.006	687.67 ± 5.51
	2.0	-0.143 ± 0.006	615.67 ± 1.53
	1.8	-0.127 ± 0.006	575.33 ± 5.03
	1.6	-0.120 ± 0.000	540.33 ± 2.52
2.5	3	-0.300 ± 0.000	758.00 ± 2.00
	2.4	-0.220 ± 0.000	596.00 ± 2.00
	2.2	-0.220 ± 0.010	545.67 ± 2.08
	2.0	-0.180 ± 0.010	510.00 ± 2.00
	1.8	-0.127 ± 0.006	479.33 ± 6.03
	1.6	-0.120 ± 0.000	457.33 ± 2.52

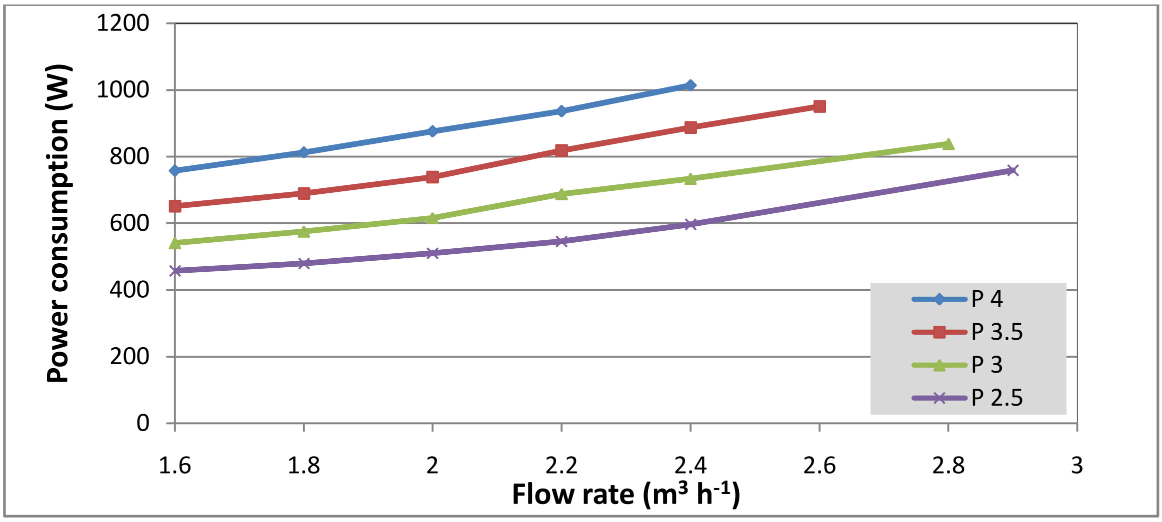


Figure 4.21 The effect of VSD on the relationship between flow rate and power consumption at constant pressure tests

Inlet pressure was affected by the flow rate variation but did not pose a hazard in terms of cavitation since the maximum suction pressure did not exceed -0.3 bar among all tests (Fig. 4.22). The use VSD could be considered safe for most cases in terms of changing the inlet pressure within the whole dynamic range of the flow rate at different head demands.

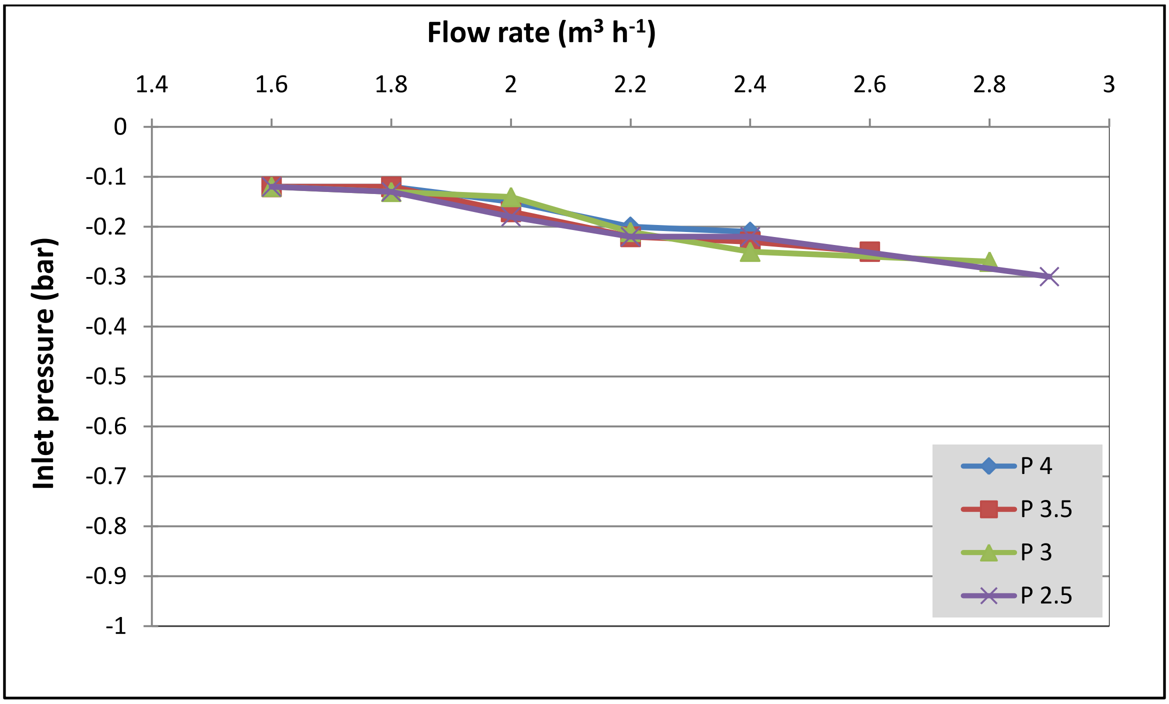


Figure 4.22 The effect of VSD on the relationship between flow rate and inlet pressure at constant pressure tests

Experiment 8: The effect of by-pass valve at constant pressure settings on inlet pressure and power consumption

Constant head demands were met at different flow rates by using the by-pass valve in this experiment. The outlet pressure values in the first column were demanded by the pipe system and the necessary flow rate values in the second column were adjusted by using the by-pass valve (Table 4.9).

The power consumption fluctuated but did not necessarily vary proportionally with the flow rate (Fig. 4.23). At each pressure setting, the power consumption was almost the same. For instance, the overall average of the power consumption values given at 4.0 bar was 1038.9 W with the measured results ranging from 1033 W to 1045 W.

Table 4.9 The effect of by-pass valve on measured parameters at constant pressure tests

outlet pressure (bar)	flow rate ($m^3 h^{-1}$)	inlet pressure (bar)	power consumption (W)
4	2.4	-0.217 ± 0.006	1038.00 ± 2.65
	2.2	-0.213 ± 0.006	1044.67 ± 3.06
	2.0	-0.210 ± 0.000	1045.33 ± 2.52
	1.8	-0.210 ± 0.000	1032.67 ± 1.53
	1.6	-0.217 ± 0.006	1040.33 ± 1.53
3.5	2.5	-0.24 ± 0.010	952.00 ± 2.00
	2.4	-0.217 ± 0.006	937.33 ± 2.52
	2.2	-0.220 ± 0.000	925.67 ± 2.08
	2.0	-0.220 ± 0.000	936.00 ± 2.65
	1.8	-0.220 ± 0.000	919.67 ± 1.53
	1.6	-0.220 ± 0.000	923.67 ± 1.53
3	2.7	-0.26 ± 0.010	861.00 ± 1.00
	2.4	-0.257 ± 0.006	813.67 ± 1.53
	2.2	-0.257 ± 0.006	838.33 ± 3.06
	2.0	-0.257 ± 0.006	836.00 ± 1.00
	1.8	-0.250 ± 0.000	817.00 ± 1.73
	1.6	-0.250 ± 0.000	827.67 ± 2.52
2.5	2.9	-0.300 ± 0.000	746.00 ± 1.00
	2.4	-0.287 ± 0.006	734.00 ± 1.00
	2.2	-0.277 ± 0.006	744.67 ± 2.52
	2.0	-0.297 ± 0.006	736.00 ± 2.00
	1.8	-0.297 ± 0.006	732.00 ± 2.00
	1.6	-0.297 ± 0.006	724.33 ± 1.53

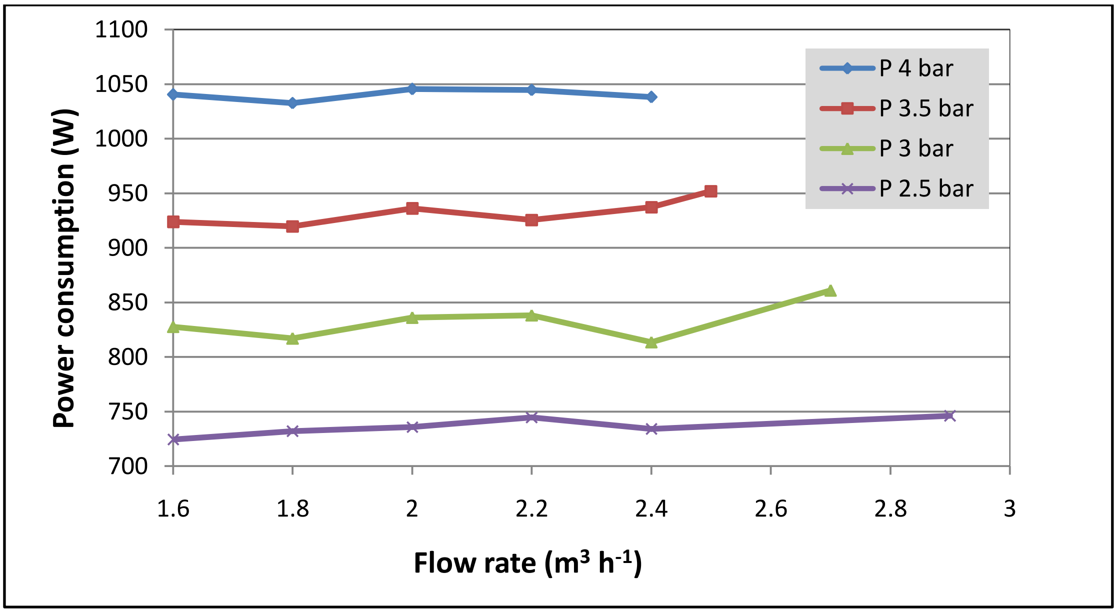


Figure 4.23 The effect of by-pass valve on the relationship between flow rate and power consumption in constant pressures tests

Similar variations could be observed for other tests done at different pressures. This can be explained by the fact that the pump operated at the maximum speed in all these tests. Also, the test system was not a complicated hydraulic network. The friction losses in the hydraulic system could not be a big factor to effect the power consumption. Therefore, the energy consumption of the pump operating at the highest speed must have determined the major part of the energy used during the tests. As pressure was reduced from one pressure setting to the next level, the power consumption reduced immediately.

Although the power consumption reduced with decreasing flow rate when using the VSD, the power consumption was not affected by the flow rate variation by using the by-pass valve. As explained, it was observed that the energy use was dominated by the pressure setting of the hydraulic system rather than the flow rate when a by-pass valve is used to vary the flow rates at that pressure setting. It was concluded that, for constant pressure operations, VSD was a better option compared to the by-pass valve.

Measured inlet pressures were not consistent and there were visible differences in measured values when different pressure settings were considered (Fig. 4.24). The inlet pressure tended to remain the same in these tests, on the contrary to the results found in Experiment 8. The reason for this can be explained by the fact that the flow rate of the pump was the same in all tests and was reduced at the outlet of the pump by using the by-pass valve. Therefore, using the by-pass valve reduced the flow rate in the pressure line only. In the case of using the VSD, however, the inlet pressure reduced with the flow rate, resulting in fewer potential hazards for the suction line.

4.4. General Discussion

Users might want to change the flow rate depending on the system requirements as a result of changing conditions in the soil, plants, season, or the field. An irrigation system with no VSD installed, the user can only adjust the flow rate by using a valve on the hydraulic system. This valve might be installed on the suction line (inlet valve), or on the pressure line (outlet valve), or on the by-pass line (by-pass valve). When a VSD is present in the irrigation pumping unit, the flow rate can be controlled be changing the speed of the pump rather than using a valve or a combination of valves in the system.

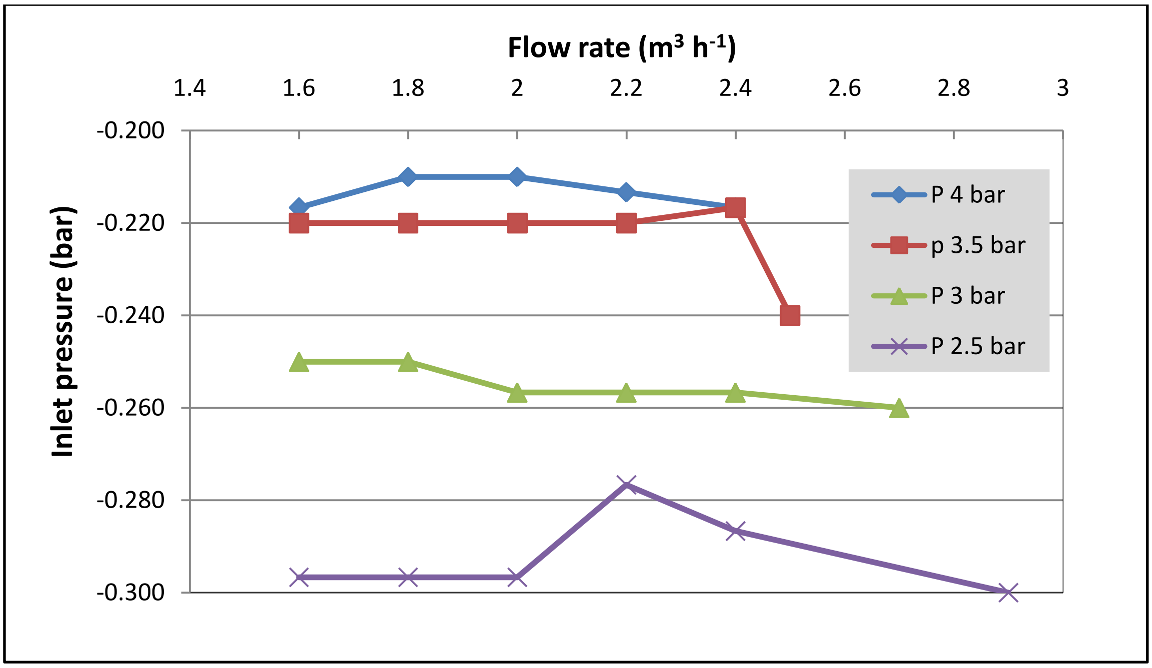


Figure 4.24 The effect of by-pass valve on the relationship between flow rate and inlet pressure in constant pressure tests

In this section, some data from the Experiments 1 through 8 will be used to discuss the findings to reach final conclusions. To do this, test results were classified as constant and variable flow tests and the data to be compared were given in Table 4.10. In the previous discussions, comparisons were made for a 20% reduction in the flow rate. The smallest flow rate values given in the table was limited to be 2.0 m^3 h^{-1} since the flow rate could not be reduced further in all tests. Additionally, experimental results regarding constant pressure tests were summarized and discussed.

When constant flow rate tests were compared among themselves, the least amount of energy was consumed by using the by-pass valve in obtaining different flow rate valves. When inlet valve was used for flow rate adjustment, the energy savings were 1%, 53%, and 68% compared to the use of outlet valve at 3.0m^3 h^{-1}, 2.6m^3 h^{-1}, and 2.0 m^3 h^{-1} whereas the savings were 7%, 55%, and 70% when by-pass valve was used at the same flow rates. The energy use was similar in using the by-pass valve and the inlet valve use, however, the suction pressure demand increased significantly in the case of inlet valve use. This makes the by-pass valve use the best option for changing the flow rate when needed. There was no apparent disadvantage in using the by-pass valve within a wide range, since the inlet pressure, outlet pressure, and the power consumption kept decreasing with the flow rate.

Table 4.10 Example results from the constant and variable flow rate experiments

experiment No.	explanation	flow rate (m^3 h^{-1})	inlet pressure(bar)	outlet pressure(bar)	power consumption(W)
		Constant flow rate experiments			
1	Constant flow rate test using the outlet valve - non-pressurized system	3.0	-0.300	0.700	451.25
		2.6	-0.230	3.460	920.25
		2.0	-0.140	4.650	1241.25
2	Constant flow rate test using the inlet valve -non-pressurized system	3.0	-0.300	0.700	451.50
		2.6	-0.440	0.500	433.00
		2.0	-0.660	0.250	397.25
3	Constant flow rate test using bypass valve - non-pressurized system	3.0	-0.300	0.690	425.50
		2.6	-0.300	0.550	410.75
		2.0	-0.300	0.320	375.00
		Variable flow rate experiments			
5	Variable flow rate test using the VSD	3.0	-0.300	0.700	427.50
		2.6	-0.250	0.520	287.25
		2.0	-0.165	0.275	197.75
6	Variable flow rate test with VSD and outlet valve	2.8	-0.270	2.900	793.25
		2.6	-0.240	3.470	903.75
		2.0	-0.150	4.690	1209.00
7	Variable flow rate test with VSD and inlet valve	2.8	-0.400	0.600	442.00
		2.6	-0.438	0.500	429.50
		2.0	-0.658	0.260	394.25

However, in the case of the inlet valve, while the power consumption decreased gradually with decreasing flow rate, the inlet pressure rose significantly. This in turn created unacceptable operating conditions for the suction line, especially when the reduction in the flow rate exceeded 30%.

Some results of the constant pressure tests were given Table 4.11. The power consumptions at high flow rates (3.0 m^3 $h^{-1}$and 2.6 m^3 h^{-1}) were similar in using VSD (758 W) and by-pass valve (746 W) whereas VSD became advantageous when the flow rate was further reduced. At 4.0 bar, the energy saving was about 21% when VSD was used instead of the by-pass valve.

Table 4.11 Example results from the constant pressure experiments

experiment No.	explanation	flow rate (m^3 h^{-1})	inlet pressure (bar)	outlet pressure (bar)	power consumption (W)
7	Constant pressure test using the VSD	3.0	-0.300	2.5	758.00
		2.6	-0.260	3.5	950.00
		1.8	-0.143	4.0	812.00
8	Constant pressure test using by-pass valve	3.0	-0.300	2.5	746.00
		2.5	-0.240	3.5	952.75
		1.8	-0.210	4.0	1032.0

The pump efficiency may change as the flow rates changes in a wide range during operations. Also the flow rate, pressure head, and the power of a pump follow the Affinity Laws, as follows:

$$\frac{Q1}{Q2} = \frac{N1}{N2} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(Eq. 1)}$$

$$\frac{H1}{H2} = \left(\frac{N1}{N2}\right)^2 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(Eq. 2)}$$

$$\frac{P1}{P2} = \left(\frac{N1}{N2}\right)^3 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(Eq. 3)}$$

where Q is the flow rate (m^3 h^{-1}), N is the speed (rpm), H is head (m), and P is power (W). Using these relationships pressure head and the power of the pump can be calculated based on the change in the pump speed. Therefore, these equations could be used for variable speed tests. As an example, for the following measured data, the following ratios were calculated:

Measured data: Q_1=0.46 m^3 h^{-1}, Q_2=0.926 m^3 h^{-1}, N_1= 580 rpm, N_2= 870 rpm

Measured ratios:

$$\frac{Q1}{Q2} = \frac{2.941}{2.944} = 0.99,$$

$$\frac{N1}{N2} = \frac{2320}{2900} = 0.8,$$

$0.99 \neq 0.80$

The ratios of the flow rate and power should have been the same by the Affinity Laws. However, the calculated ratios were different. The ratios for the head (H) and the speed (N) were also different:

$$\frac{H1}{H2} = \frac{0{,}701}{0{,}674} = 1.04,$$

$$\left(\frac{N1}{N2}\right)^2 = \left(\frac{2900}{2320}\right)^2 = 1.56,$$

$1.04 \neq 1.56$

The differences found in these calculations should have resulted from the fact that the test bench did not only consist of the motor and the pump, but additional valves, elbows, T connectors, gages, and pipes. Since the motor was of a quite small power, the system components of the small irrigation system must have affected the behavior of the pump. Therefore, in reality, the measured head and the flow rate values cannot only be related to the pump characteristics, but also to the hydraulic system.

Small motor pump power must have contributed to these differences as the effect of irrigation system components become more important in the system used in this research. If a large powered irrigation pump was used under similar conditions, the effect of major and minor losses on the power consumption might have been different.

Different tests resulted in different coefficient of determinations for the flow rate and pump speed, the flow rate and inlet pressure, the flow rate and outlet pressure, and the flow rate and power consumption. Experiment 4 was taken as an example, as shown in Table 4.12. The highest correlation was found for the flow rate and pump speed relationship whereas the smallest was found for estimating the power consumption. This was expected since power is a product of both the flow rate and the pressure head demand of the system.

Table 4.12 The regression equations obtained in Experiment 4

Relation	Regression equation	R^2
Flow rate – VSD	$y = -3E\text{-}07x^2 + 0.002x - 0.632$	0.995
Flow rate – Inlet pressure	$y = -0.024x^2 - 0.008x - 0.051$	0.982
Flow rate – Outlet pressure	$y = 0.079x^2 - 0.009x + 0.018$	0.992
Flow rate – Power consumption	$y = 45.45x^2 - 48.28x + 116.1$	0.971

5. CONCLUSIONS

The followings could be summarized and concluded as result of this study:

1. A small centrifugal pump was used on a test unit to measure the inlet pressure, outlet pressure, flow rate, and power consumption under different operating conditions.
2. High coefficient of determinations were found, usually about 0.97-0.99, for the inlet pressure, outlet pressure, and the power consumption as a function of flow rate throughout the experiments.
3. When there was no VSD on the pumping unit, the most energy efficient way to deliver different flow rates was to use the by-pass valve (410 W), followed by the inlet (suction) and the outlet (discharge) valves, respectively (420 W and 920 W). The energy saving was about 45% when by-pass valve was used instead of outlet valve on the irrigation system.
4. The use of inlet valve could be a good alternative in varying the flow rate for irrigation systems, but at the cost of cavitation due to increased vacuum pressure. The vacuum increased from -0.3 to -0.54 bar for a 20% decrease in the flow rate when inlet valve was used to vary the flow rate. The flow rate must not be varied over a wide range using the inlet pressure.
5. Different constant pressure demands could be effectively met either by using the VSD orby-pass valve. However, power consumption decreased in the case of VSD whereas it was almost the same when the by-pass valve was used to vary the flow rate at a given head requirement. VSD controlled system was more energy efficient by 0% to 28% depending on the flow rate demand of the system. As the flow rate demand of the irrigation system decreased, the VSD became more efficient compared to by-pass valve for varying the flow rate.
6. In non-pressurized operations, compared to the inlet valve (420 W) and the by-pass valve (405 W), the use of VSD was more advantageous (287 W), resulting in 32% and %29 less energy consumption for a 20% reduction on the flow rate.
7. In pressurized irrigation simulation tests, by-pass valve could be the only choice if no VSD existed. With the use of VSD to vary the flow rate at a given outlet pressure demand, power consumption decreased with decreasing flow rate. The power consumption did not reduce as a result of reducing flow rate when the by-pass valve was used.

8. The VSD expanded the dynamic range of the pressure head and flow rates that can be delivered. The VSD provided the most energy efficient delivery of the required flow rate and pressure demands. Thus, it was concluded that the use of VSD was more favorable in all types of operations, including pressurized and non-pressurized irrigation.
9. Ambient conditions (temperature of the air, water, relative humidity, and atmospheric pressure) may be important on the system performance, but the tests were done under similar conditions in this study, showing no effect of the ambient conditions on measured parameters.

In view of the results obtained in this study, following recommendations can be made for end users and for future studies:

1. VSD technology should be favored because of its significant impact on reducing energy consumption in the irrigation system.
2. By-pass technique should be used since it also plays a big role in raising the energy efficiency of the irrigation system if a VSD is not available on the system.
3. When an inlet valve is used to control the require flow rate, one should pay attention to the cavitation of the pump. Installation of a vacuum meter on the suction line might be helpful to observe the effect of varying flow rate on the suction line. The flow rate should not be varied over a wide range using an inlet valve.
4. Future work should focus on several small irrigation systems that are of different sizes to monitor and assess the potential benefits of installing VSDs in field conditions.

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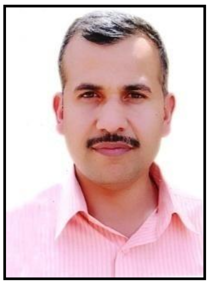

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