A NOVAL MODEL FOR OPTIMIZING PUMPING ENERGY CONSUMPTIONIN BIOMASS HIGH PRESSURE HOT WATER GENERATOR

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***Abstract*—High-Pressure Hot Water (HPHW) for a variety of applications in the manufacturing industry, including leaching, curing, and drying. Nowadays HPHW is primarily generated using Biomass Hot Water Generators that require centrifugal pumps to maintain minimum hot water flow and pressure.**

Attributable to the criticalness in maintain pressure above saturation and evade steaming in the system, designers may allow for an excessive margin of error when deciding the capacity and developed head of centrifugal pumps used for HPHW circulation. An oversized pump will deliver more flow and discharge pressure than is required for the system and therefore impact the smooth synchronization of the HPHW system. To overcome such a scenario, throttling is used to influence the flow rate and discharge pressure in the pumping system by varying system resistance. It is the simplest way to shift the operating point of a centrifugal pump because it uses existing valves fitted inside, upstream, or downstream of the HPHW system. However, control by throttling is the least energy-efficient control method as it is based on the principle of control by generating losses.

Existing Lanka BIOMASS 2 HPHW GENERATOR circulation pumps operate in Star-Delta Circuit, while discharge pressure and flow rate are regulated by throttling manual valves at the pump discharge. This study is on the possibility of reducing pumping losses by changing the speed of the circulation pump used in the HPHW generator. The possibility of using Variable Frequency Drives (VFD) to reduce energy consumption was studied based on calculations and verified with pump OEM datasheets. Finally, the technical and economic viability of using VSD for improving pumping efficiency in existing BIOMASS 2 HPHW GENERATOR is presented and discussed.

Keywords-component; Energy Management, Centrifugal pumps energy conservation, VFD Flow Control, Valve Throttling.

I. INTRODUCTION *(VARIABLE FREQUNCY DRIVES APPLICATIONS)*

Process control in manufacturing processes was the typical use of variable frequency drive technology in its early stages. Motor generator sets and DC drives were replaced by VFDs because they enhanced process performance and decreased maintenance costs. When saving energy became a top priority, the use of VFDs quickly expanded to include big pump applications.

II. VARIABLE FREQUNCY DEVICES TO COMPARED THROTTLING DEVICES

A mechanical throttling device is used to restrict flow in numerous flow applications. Although this kind of control is effective, mechanical, and electrical energy are wasted. In Figure 1, a mechanical throttling valve-based pumping system is shown alongside a VFD-based version of the same system.

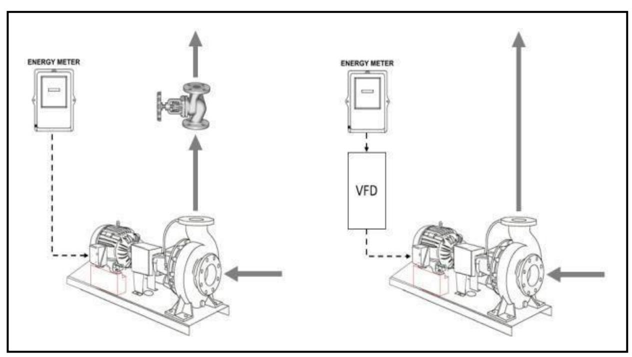


Figure 1: Discharge valve throttling vs VFD

If a throttling device is employed to control flow, energy usage is shown as the upper curve in Figure 2, while the lower curve demonstrates energy usage when using a VFD. Because a VFD alters the frequency of an AC motor, speed, flow, and energy consumption are reduced in the system. The energy saved is represented by the shaded area.

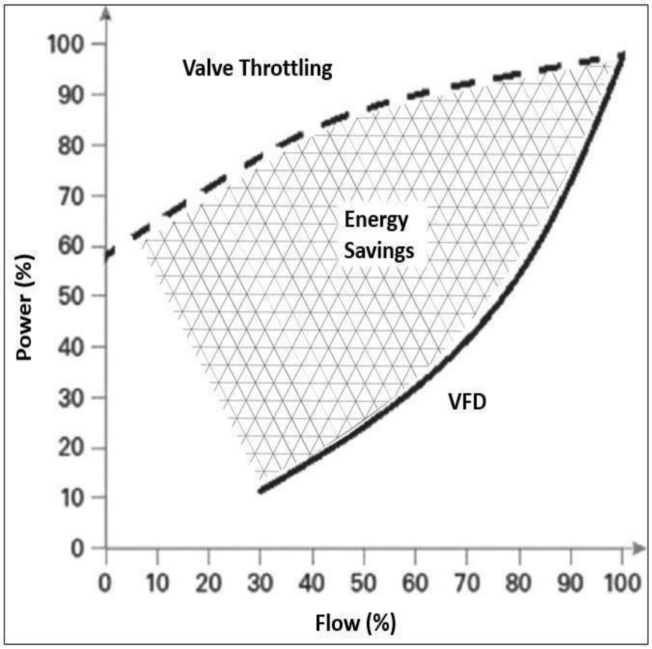


Figure 2: The amount of Energy Saved by Using a Variable Frequency Drive (versus a Valve) to Control Flow

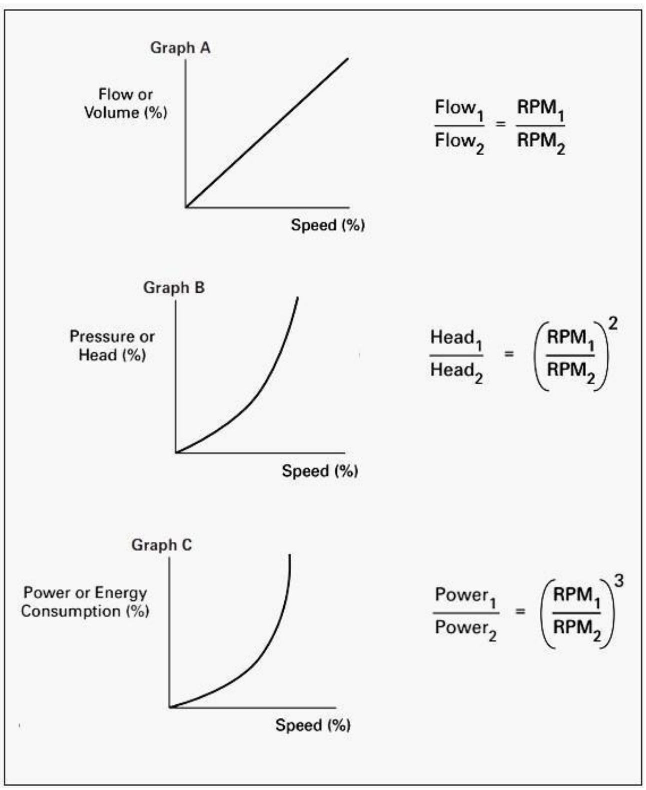


Figure 3: The Affinity Laws

III. VARIABLE FREQUNECY DRIVES THEORY

The system performance of centrifugal devices, including the potential for energy savings and the theoretical load needs, can be determined by the affinity laws. The three affinity laws are shown in Figure 3.

IV. PUMPING SYSTEM CHARACTERISITICS

Choosing the right pump for a system depends on knowing the system curve, which details what flow will happen given a certain pressure. Two components must be understood to determine an appropriate system curve:

- **Static Head**--The height that the fluid must be elevated to get from the source to the outlet is known as the static head or lift.
- **Friction Head--**The force necessary to compensate for losses brought on by fluid flow, valves, bends, and other piping-related components. These losses are nonlinear and totally flow-dependent.

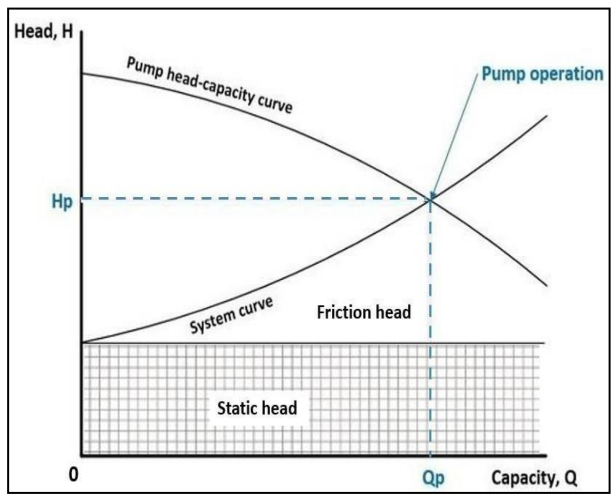


Figure 4: Combination of System and Pump Curves

Figure 4, the "system curve" is a graphical representation of the relationship between discharge and head loss in a system of pipes. The system curve is completely independent of the pump characteristics. The curve may start at zero flow and zero head if there is not static head. The head loss, or pressure loss, in a system containing a flowing fluid is to a large extent caused by friction between the pipe wall and the moving fluid. This curve has an upward (positive) slop because friction is related exponentially to the velocity at which the fluid is moving through the pipe, meaning that the faster the fluid is moving (i.e., the higher the flow rate), the larger the head loss will be. This is also true for other secondary losses, such as losses around bends or due to reducers.

In contrary, a pump curve is developed for each pump based on its design, pump speed, and impeller diameter. It displays a graph of the pump's head pressure as a function of flowrate. The head that a pump can provide decreases with increasing flow rate. As the velocity of the fluid flowing through the pump increases, the internal friction and secondary losses increase, up to the point where they equal the pressure imparted to the fluid by the spinning impeller. When this point is reached, the net head delivered by the pump will be zero. Conversely, when the flow rate through the pump is zero, there are no frictional and secondary losses, so the head delivered to the fluid is at a maximum. Due to the fact that produced head decreases as flow increases, the pump curve has a downward (negative) slope.

The intersection of the system head curve (typically convex down) and the pump curve (typically convex up) is the operating point. As the system filters clog, valves are throttled, etc., the system curve changes, and a new operating point is established. Similarly, when the pump speed changes, or impeller wears, the pump curve changes, and a new operating point is established.

A. Throttling device application in a pump system

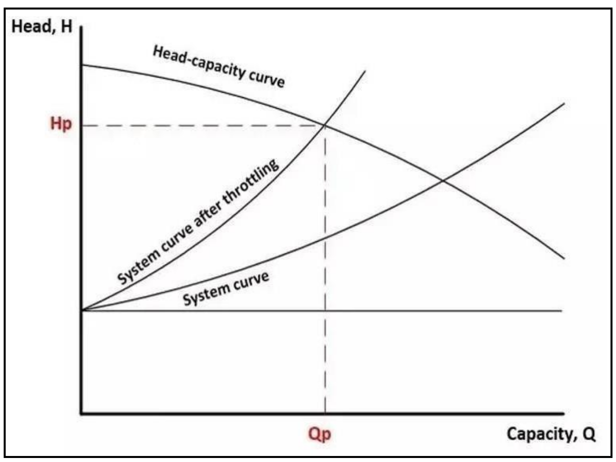


Figure 5: Change in System Characteristics using a Valve for Throttling

The flow rate of a pumping system is frequently reduced mechanically using a throttling mechanism. The pump curve is altered by adding a throttling mechanism to the system, as seen in Figure 5. This decreases the system's flow, yet the pump curve remains unchanged and keeps running at full speed. As a result, the pump system experiences mechanical stresses—excessive pressure and temperature—which may lead to premature seal or bearing failures. And perhaps more crucially, this uses a lot of energy. The energy consumed is represented by the blue shaded area in Figure 6.

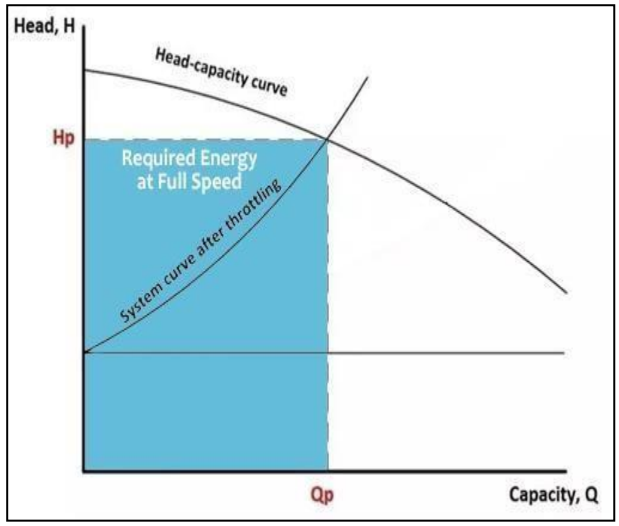


Figure 6: Pump Energy Requirement

B. Variable Frequency Drive Application in Pump System

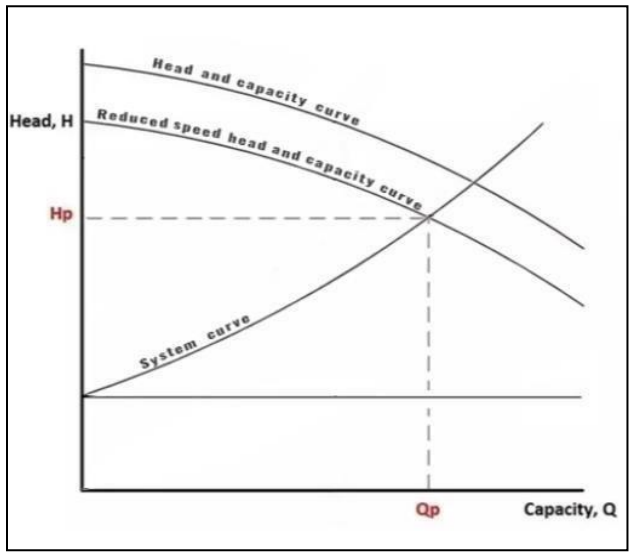


Figure 7: System Characteristics Using a VFD

Applying a VFD to the pump enables electrical speed control of the pump while utilizing only the energy required to create a specific flow. Applying a VFD to the pump motor enables to change the pump curve as shown in Figure 7. This reduces the flow and the developed head of the pump, but the system curve is not altered while pump continuous to operate at reduced speed.

Trimming existing pump impeller is comparable to this however it is subjective to minimum impeller diameter recommended by manufacture.

Figure 8 shows the updated pump curve and the amount of energy used by using VFD. Additionally, the pressure is decreased, which aids in lowering the mechanical strains brought on by throttling devices.

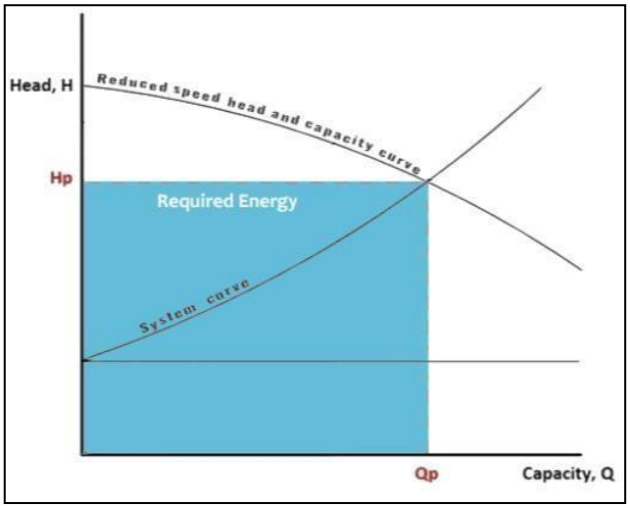


Figure 8: Energy Consumption Using VSD

Overlaying the two previous graphs, the difference is obvious in Figure 9. The blue shaded area is the energy saved by using a VFD instead of a throttling device.

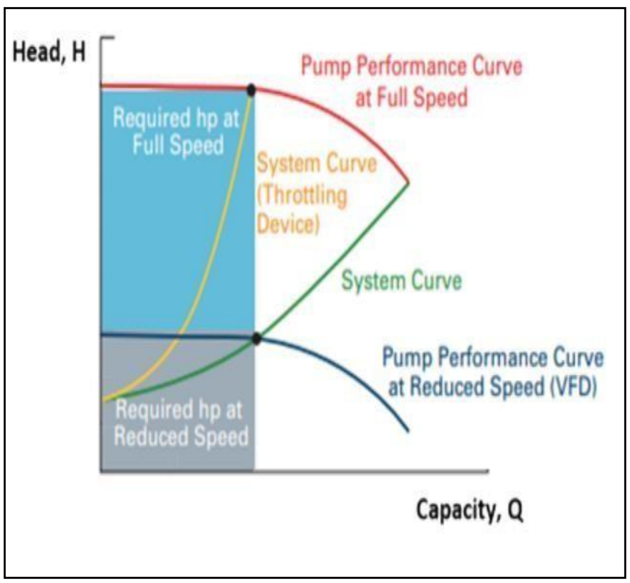


Figure 9: Difference in Energy Consumption Using a Throttling Valve versus a VFD

V. CALCULATING PUMP ENEGY CONSUMPTION AND COST SAVINGS

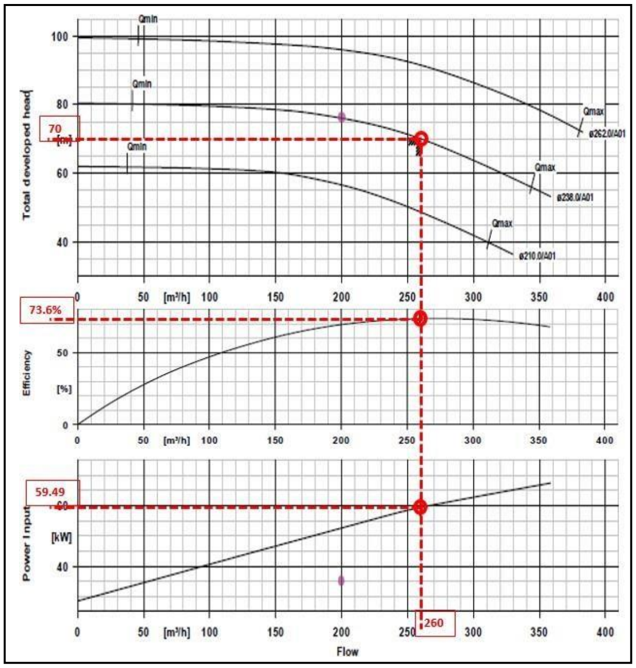
A. *Pump Rated Specifications*

HPHW circulation pump rated specifications and curve used for calculation are in Table 1 and Table 2. The pump is designed by the manufacture to operate at 260 m3/hr. while developing 70 m head. Furthermore, pump manufacture considers the fluid temperature to be 185 C (max temperature rating of system), however the fluid temperature in operation was measured at 160 ℃ with a density of 895.46 kg/m3. As result, rated pump power absorbed may somewhat differ from actual.

Table 1 Pump specifications and operational parameter

Pump Rated Flow - Q (m^3/hr)	260
Rated Head (m)	70
Pumped Medium	High Pressure Hot Water
Actual Fluid Temperature (℃)	160
Actual Fluid density - ρ (kg/m^3)	895.46
Rated Shutoff Head (m)	80.44
Pump Efficiency	73.60%
Pump Power Absorbed at Rated Spec - WHP (kW)	59.49
Pump Rotation Speed at rated spec (rpm)	2979
Minimum Pump Speed of Rotation (rpm)	800
Motor Efficiency Class	IE3
Motor Rated Power (kW)	75

Table 2: Pump designed operation point



B. Pump Operating Point With Valve Throttling

Pump break horsepower actual (BHP_a) or pump shaft power with discharge valve throttling can be calculated using equation *[1]*. The actual voltage and current were measured using a clip-on meter. Table 3 and Table 4 Pump BHP requirement and pump curve with valve throttled.

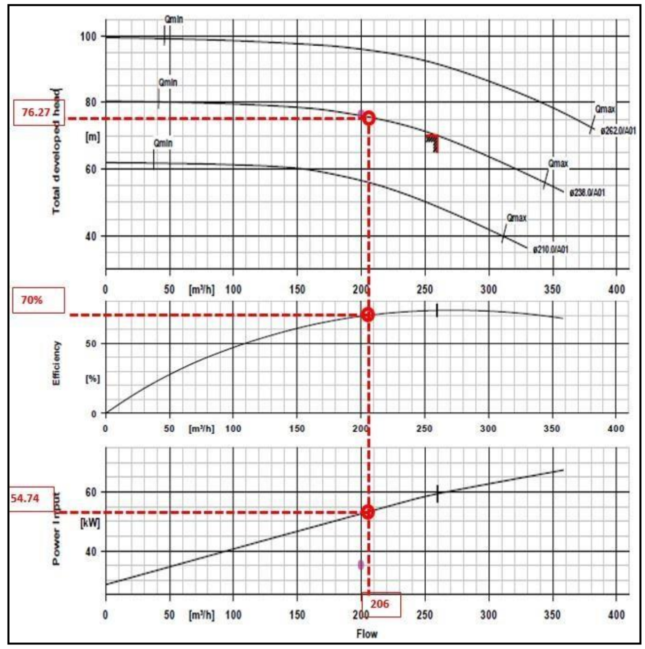
$$BHP_a = \frac{\sqrt{3}\, V I n_m PF_m}{1000} \qquad [1]$$

where,
BHP_a = pump brake horsepower actual (kW)
V = measured electricity voltage (v)
I = measure electrical current (A)
n_m = transmission efficiency of motor
PF_m = power factor

Table 3: Pump actual electricity consumption with valve throttling

Voltage (v)	405
Current (A)	98.00
n_m (%)	0.93
PF_m (%)	0.84
BHP_a (kW)	53.70
Motor Input Power (kW)	68.75

Table 4: Pump operating point with valve throttling



Based on BHP_a, actual developed head of pump can be calculated using equation *[2]*. For this study, fluid velocity difference is considered zero since suction and discharge are the same diameter. In addition, both suction and discharge pressure gauge taps are located at the same elevation.

$$H_{pa} = \frac{100000(P_{2D} - P_{1D})}{\rho g} + \frac{\mathcal{V}_{2D}^2 - \mathcal{V}_{1D}^2}{2g} + h_{gp} \qquad [2]$$

where,
H_{pa} = pump head actual (m)
P_{2D} = pump discharge pressure (bar. g)
P_{1D} = pump suction pressure (bar. g)
$\mathcal{V}_{2D}$ = average fluid velocity at pump discharge (m/s)
$\mathcal{V}_{1D}$ = average fluid velocity at pump suction (m/s)
ρ = fluid density (kg/m^3)
g = earths gravitational acceleration (9.81 m/s2)
h_{gp} = different in elevation of pressure taps at discharge and suction sites of the pump (m)

Table 5 Pump actual developed head

P_{1D} (Bar. g)	7.90
P_{2D} (Bar. g)	14.60
H_{pa} (m)	76.27

C. Drop in Head Across Throttling Valve

Head loss across throttling valve at the pump discharge can be calculated using equation *[3]*.

$$H_{Vloss} = \frac{100000(P_{Vin} - P_{Vout})}{\rho g} + \frac{\mathcal{V}_{Vin}^{2} - \mathcal{V}_{Vout}^{2}}{2g} + h_{gv} \qquad [3]$$

were,
H_{vloss} = head loss across valve (m)
P_{Vin} = valve inlet pressure (bar. g)
P_{Vout} = valve outlet pressure (bar. g)
$\mathcal{V}_{Vin}$ = average fluid velocity at valve inlet (m/s)
$\mathcal{V}_{Vout}$ = average fluid velocity at valve inlet (m/s)
h_{gv} = different in elevation of pressure taps at inlet and outlet sites of the throttling valve (m)

Table 6: Head loss across throttling valve

P_{Vin} (Bar. g)	10.25
P_{Vout} (Bar. g)	14.13
H_{Vloss} (m)	44.17

D. Pump new operating point with minimum throttling

The new pumping capacity shown in Table 7 is decided based on reducing the pressure drop across the throttling valve from 4 bar. g to 0.65 bar. g. While on paper it is possible to further reduce pump operating point and eliminate valve throttling, based on fluid temperature, it was decided to limit the pump minimum speed to 30 Hz to make sure adequate pump cooling is available. This would help to prolong the life of the pump bearings and the motor itself.

Table 7: Existing pump new operating point with VSD

Required at Pump Discharge	
Developed Head - H_{pr} (m)	45
Flow rate - Q (m^3/hr)	206

E. Pump energy consumption with VSD

The pump power output, which is referred to as the pump water horsepower (WHP) is calculated using the following equation.

$$WHP_r = \rho g Q H_{pr} \qquad [4]$$

Equation *[5]* calculates the motor output power required.

$$BHP_r = \frac{WHP_r}{n_p} \qquad [5]$$

$$Motor\ Input\ Load\ (kW) = \frac{BHP_r}{n_m PF_m} \qquad [6]$$

where,

WHP_r = water horsepower required (kWh)

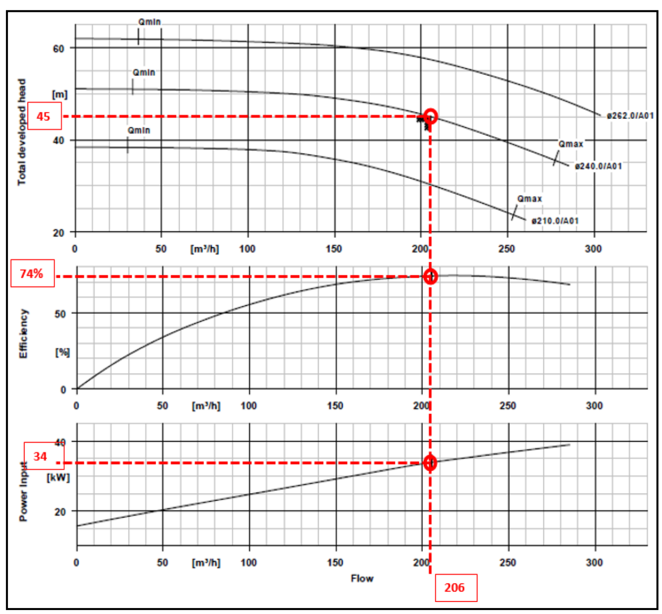
BHP_r = pump brake horsepower required (kWh)
n_p = pump efficiency

Table 8 and Table 9 show existing pump efficiency and energy consumption when operating with VSD at new operating point.

Table 8: Pump energy consumption with VSD

P_{1D} (Bar. g)	7.40
P_{2D} (Bar. g)	11.35
Q (m^3/hr)	206
H_{pr} (m)	45
WHP_r (kW)	22.57
n_p (%)	74%
BHP_r (kW)	30.50
Motor Input Power (kW)	41.27

Table 9: Pump operating point with VSD



The insignificant difference of values between BHP_r calculated in Eq. [5] vs Table 9 is due to the difference in fluid density considered by the manufacture datasheet. However, the accurate value for this study shall be the calculated BHP_r = 30.5 kW.

Table 10:Pump operational cost savings with VSD

	Valve Throttling	**VFD**
Operational hours per day	24	24
days per annum	350	350
Motor BHP (kW)	53.7	30.50
Pump motor input power (kW)	68.75	39.04
Daily energy consumption (kWhe)	1,650	937
Energy consumption per annum (kWhe)	577,459	327,962
Average electricity tariff (Rs. /kWhe)	32.04	32.04
Energy cost per annum (Rs.)	18,502,743	10,508,458

Based on results from Table 10, annual electricity savings from each pump equal to 498,993 kWhe and the cost saving of Rs. 7,994,285 shall be achieved from reducing pump RPM.

F. Financial Benefit

Table 11 shows the annual savings and simple payback of modifying existing pumping system to operate with VSD.

Table 11: Annual savings per pumping system

Number of pumps proposed to operate with VSD (nos)	3
Number of pumps working at a time (nos)	2
Number of pumps standby (nos)	1
Estimated cost of adding VSD to all 3 pumps (LKR)	15,888,471
Annual savings from pump system (LKR)	15,988,569
Simple payback (years)	0.99

VI. CONCLUSION

This study encompassed the comparison between two different methods of flow control methods which were studied in detail to find the feasibility of replacing existing manual valve throttling flow control with Variable Frequency Drive system (VFD).

Based on results obtained from onsite measurements and calculations, reducing pump RPM to required pump output without a throttling valve shall reduce electricity consumption of each pump by 43%. Therefore, the outcome of the study proves that switching to a VFD based flow control system provides significant increase in energy efficiency compared a throttling device.

In addition to energy savings, VFDs also offer improved process control, reduced maintenance costs, and increased equipment life. By providing precise control over the speed of the pump, VFDs help to reduce wear and tear on the pump and its components, resulting in longer equipment life and reduced maintenance costs. This results in a substantial energy, maintenance and downtime savings, further economically justifying the use of the VFDs.

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