Efficient Utilization of Electrical Energy in Pumping Operations Based on Existing Conditions - A Case Study

P. Giridhar Kini
Dept of Electrical & Electronics Engineering,
Manipal Institute of Technology
Manipal – 576 104 (Karnataka State) INDIA
e-mail : giridhar.kini@manipal.edu

Dr. P. N. Sreedhar
Dept of Electrical & Electronics Engineering,
National Institute of Technology - Karnataka
Surathkal – 575 025 (Karnataka State) INDIA

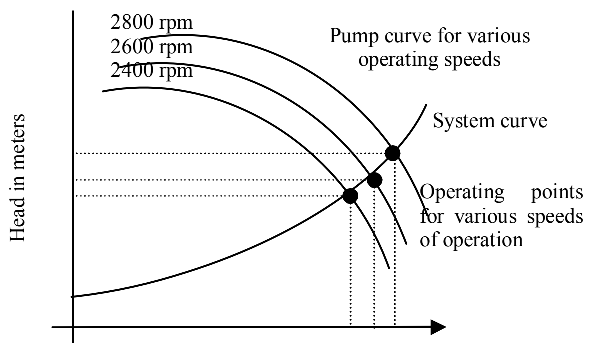
***Abstract*—The intersection of the pump characteristics with the system characteristics gives the design operating point for a given pumping system. Changes in conditions of fluid viscosity, total head, throttle positions, supply system parameters etc causes the operating point to shift on either side of the pump head-discharge curve, which affects not only the hydraulic output of the pump, but also results in the inefficient utilization of available electrical power. Use of variable speed drives involves additional investment, which if the pump is in operation for long hours every day throughout the year with the number of such units in operation being small in number, but is the best possible option to achieve energy efficiency. But if the industrial utility has large number of pump units for various processes, with individual operating times being quite insignificant, yet important in the process involved, variable speed drives may not be the best possible solution. This paper makes an attempt in presenting a simple method for making the pump to work at / or near its operating point based on the exiting system conditions, with the least possible changes to the working setup in addition to achieving the desired objective of efficient utilization of the available electrical power. The analysis is supported by a case study.**

I. INTRODUCTION

Due to improper operational and maintenance practices, equipments like motors and motor driven units like pumps operate at relatively lesser efficiencies leading to tremendous wastage of energy [1]. Most of the time it is found that small changes in the motor-pump system or piping network itself is quite sufficient to reduce the wastage. Only when it is not possible to redesign the system layout due to system constraints or if the equipment in question plays a major role in the process involved, it would be very sensible to go in for energy efficient equipments, as they will prove to be quite profitable in the long run.

A pump operates over a wide range of head - flow discharge rate, for a given value of speed and impeller diameter. Peak efficiency is possible only at a particular flow rate & discharge pressure, other parameters like speed, impeller diameter, and suction & discharge pipe diameters being kept constant. As induction motors are used as drives for pumping systems, efficiency of motor in addition to pump efficiency becomes an important factor; hence the system efficiency now plays a vital role in motor driven-pump system operation [4]. Moreover, imbalance in the supply voltage has a tremendous impact on the working performance [3]. Persistent fluctuations in supply system parameters like supply voltage, frequency etc are suitably reflected in varying operating speed, head to be supplied & discharge delivered, such that the operating point of the pump always deviates from the designed value, and the pumping operation tends to become unbalanced or unstable in nature [2].

The discharge of a pump operating at a particular speed will be different for different values of head [6]. For a given speed, there exists a head-discharge characteristic that is specified by the manufacturer. Therefore for a process utilizing a pump system, there exists two sets of head-discharge curves ie; pump curve & system curve. The intersection of these two curves gives the best operating point. Variation in the operating speed of the pump produces a corresponding change in the characteristic curves as shown in Fig 1 below.



For a given impeller diameter, the various mathematical relations that help in predicting the performance characteristics of a pump are :

$$\frac{\text{New Flow Rate}}{\text{Old Flow Rate}} = \frac{\text{New Speed}}{\text{Old Speed}} \quad \text{and} \quad \frac{\text{New Head}}{\text{Old Head}} = \frac{[\text{New Speed}]^2}{[\text{Old Speed}]^2}$$

As shown in Fig 1 above, the system curve is represented by $H = x + y\,Q^2$, where H = head to be supplied, meters, x = static head, meters, y = friction head & velocity head, meters, Q = discharge, Liters per second. It is clear from the above example that for a designed performance of 2800 rpm, if speed of operation deviates from its rated value, the operating point shifts along the curve and in some cases the practically obtained value of discharge Q and the calculated value of head H, may not match the theoretically estimated values of Q and H. This situation can be easily solved by throttling on either the suction side or on the delivery side. Throttling is always made on the delivery side and is never recommended on the suction side as it not only increases the suction head, but also gives rise to cavitations [6]. Also certain constraints may not allow the further operation of the throttle valve. In such situations, some measures need to taken so as to make sure the estimated operating point conditions are obtained practically. In the following study, the system parameters such as supply voltage and frequency, though slightly varying continuously over a 24-hour period was assumed to remain at the values existed during the start of the test performance.

II. Case Study Details

A study on efficient utilization of electrical energy in pumping operations in the first stage of the plant process was conducted in reputed local milk dairy. Reception is the first section in the dairy plant process wherein the milk received in cans from the nearby villages, is emptied into a weighing bowl manually for quantity measurement. The emptied cans are cleaned in a single cleaning chamber by placing the cans upside down, along with lids, on two separate chain conveyors. The different stages in the cleaning process being

(a) Initial rinsing of empty cans inside & outside with normal water at room temperature
(b) Hot water-detergent mixture at 45° C, is sprayed to remove substances attached to the can & lid, both inside & outside
(c) Once again clean hot water is sprayed to clean the can and lid, both inside and outside

In the above-mentioned stages, 3 separate 3ϕ induction motor driven pumps are used for the purpose. The cleaning unit was a semi closed box type structure wherein the water is sprayed onto the can from the top, sides as well as into can from bottom. There are a total of 62 nozzles distributed in a uniform manner with a suitable pipe network of a smaller diameter forming a cage. Considering the importance of the stage and capacity of the pump, the stage 2 system was taken for case study, the ratings of which are as follows : 3ϕ, 50 Hz, 5.5 kW, 415 V, 11 A, 2900 rpm, Δ connected, motor input : 6.11 kW, η_{ov} = 39 %, duty : S1, insulation : class E, temp rise : 65° C, size : 50 mm x 35 mm, total head : 45 m (53 m – 18.8 m), capacity : 5.4 Liters per second (2.7 Liters per second – 8.9 Liters per second).

III. Working Results

As it was not possible to disconnect the pump from its working setup and thereby disturb the process & equipment alignment, only the no-load test and load test is carried out on the motor. The cleaning setup has a throttle valve for controlling the discharge rate & pressure but the pressure gauge reading was found to be highly unreliable, hence not used in the analysis. Instead the approximation here is with regard to discharge pressure, the reading of which was taken by placing a pressure gauge in the opening (used for priming) provided on the pump casing located quite close to the discharge outlet. The test results are given in Table 1 below.

Table 1
Test Details of Induction Motor

	V	A	kW	kVA	Pf
No Load	412.52	3.4	0.75	2.40	0.3125
On Load	412.57	8.5	5.19	6.05	0.8580

In addition to the above readings, the other operating parameters are as follows: stator resistance R_s = 4.5 Ω / ph at 30°C, supply frequency = 48.62 Hz, working pressure = 4.45 Kg / cm^2, speed = 2860 rpm, discharge = 6.037 Liters per second, average time for discharge = 10.27 sec, working head = 31.83 m, friction factor = 0.01. From the above readings, the calculated motor parameters are as follows: slip = 0.048, actual output = 4.02 kW, % loading = 73.09 %, actual power input = 5.188 kW, operating pf = 0.8580, demand = 6.046 kVA, motor η = 77.45 %, η_{ov} = 46.57 %, η_{system} = 36.09 %.

The throttle valve placed on the delivery side of the pump is used in adjusting the delivery pressure for process requirements overlooked in system design. The throttle valve was adjusted from fully open to nearly fully closed condition and the following readings were obtained. As there were 62 nozzles placed at various locations in the pipe network the time taken for discharge was obtained by observing the time taken for 1 liter discharge from various nozzle points and taking the average time for the same. The other important parameters related to the motor side, calculated from the readings obtained are tabulated in Table 2A and 2B below.

Table 2A
Motor-Pump System Data for various Throttle Opening Positions

Pressure (Kg / cm^2)	Motor loading (%)	Motor η (%)	Power Input (kW)	Power factor
4.30	78.55	77.56	5.570	0.8504
4.35	78.55	77.56	5.570	0.8582
4.40	76.18	77.48	5.406	0.8513
4.45	73.09	77.45	5.188	0.8580
4.50	71.82	77.14	5.118	0.8376

Table 2B
Motor-Pump System Data for various Throttle Opening Positions

Pressure (Kg / cm^2)	kVA	Speed (Rpm)	Q (Liters per second)	Discharge time for 1 liter (Sec)
4.30	6.55	2844	7.469	8.30
4.35	6.49	2850	7.029	8.82
4.40	6.35	2858	6.378	9.72
4.45	6.05	2860	6.037	10.27
4.50	6.01	2871	5.688	10.90

IV. OBSERVATIONS

When the cans were being cleaned, the lids of the cans are also cleaned at the same time. The chain conveyer on which the can is placed upside down is different from the one used for the lids, in a manner such that there are separate sets of nozzles for can cleaning and lid cleaning. If the throttle valve were kept fully open, it was found by the company personnel that cans were not clean enough and if the throttle valve were kept almost partially open, it was found that force exerted by the water coming out of the nozzle onto the lid surface, used to displace it from its position on the conveyor belt, resulting in the lids getting displaced and sometimes getting stuck in the chain conveyor, subsequently interrupting the cleaning process.

As a result, by trial and error, the working pressure was more or less maintained at 4.45 Kg/cm^2. The characteristic curve as per nameplate details of the pump was plotted and is as shown as curve AB in Fig 2 below

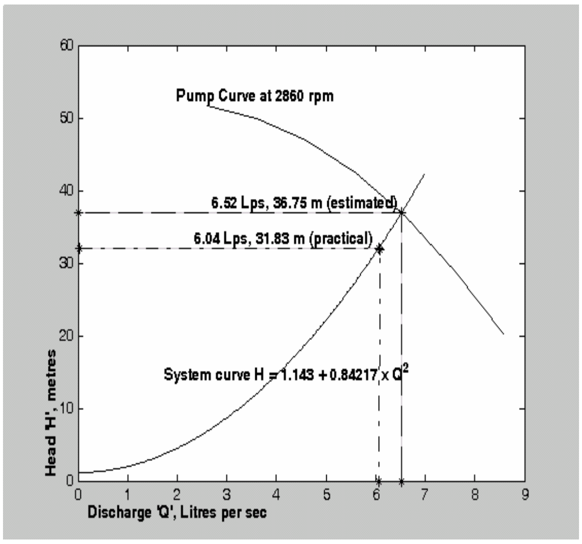


Fig 2 : Pump-System Characteristics at 2860 rpm

Applying Bernoulli's principle on the suction and discharge side of the pump, the head and the discharge are related by $H = [1.143 + 0.8421 * Q^2]$. As shown in Fig 2 above, system curve is plotted as per the above-mentioned relation. Practically, H = 31.83 m for Q = 6.037 Liters per second while theoretically the operating point should have been H = 36.75 m for Q = 6.5219 Liters per second. A strategy must therefore be thought of to ensure that the pump parameters are such that it operates at or closer to the operating point.

V. TEST RESULTS

The movement of operating point to achieve efficient utilization of electrical energy can be obtained in the following ways :

(a) Change in the suction side pipe diameter only, keeping all other dimensions unchanged
(b) Change in the discharge side pipe diameter only, keeping all other dimensions unchanged
(c) Reducing the suction side pipe length only, keeping all other dimensions unchanged
(d) Reducing the delivery side pipe length only, keeping all other dimensions unchanged
(e) Making use of energy efficient pump only, keeping all other things unchanged
(f) Making use of PVC pipes on the whole so as to reduce the pipe friction

Considering the main constraint that the working pressure cannot be changed, working setup cannot be changed whatsoever in construction & layout, modifications to the system needs to be done so that the energy saving potential achievable can be easily justified on an economic or cost benefit basis [5]. Keeping this in mind, the increase in head can be achieved by making use of a conical pipe passage as shown in Fig 3 below.

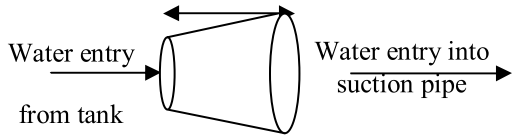


Fig 3 : Typical H-Q curves for a pump at different operating speeds

It was estimated by back calculations that with the exit diameter of the conical pipe piece kept constant at 50 mm, which is also the suction side pipe diameter, to obtain the best operating point, an entry diameter of 25 mm was sufficient for the conical pipe piece.

In order to justify the method proposed, two cast iron model pieces was prepared: one model had entry diameter of 25 mm while the other had a entry diameter of 33 mm, with the exit diameter remaining 50 mm in both models, as that of the suction side pipe diameter. As the suction side pipe opening was freely accessible, the detergent mixture entering the suction pipe is now made to enter the conical pipe passage with a bell mouthed entrance so as to reduce the loss at entry. The working details of the motor are as shown in Table 3

Table 3
MOTOR DATA WITH CONICAL PIPE ATTACHED TO SUCTION PIPE ENTRY

Fitting	A	KW	KVA	Pf	RPM
33 mm	8.64	5.233	6.205	0.8434	2863
25 mm	8.33	5.176	6.035	0.8577	2867

Now by placing the 33 mm pipe, speed of operation improved from 2860 rpm to 2863 rpm. As a result the pump curve now is redrawn for 2863 rpm, the system curve is now as per the relation $H = [1.143 + 0.8709 * Q^2]$. It is observed that the practical Q = 6.352 Liters per second & H = 36.28 m is also slightly away from the new operating point of Q = 6.4597 Liters per second & H = 37.46 m as shown in Fig 4 below. The result is quite satisfactory but in case of voltage variation or unbalance, the operating point would move further away towards the left and in the downward direction defeating the very purpose of the use of conical piece

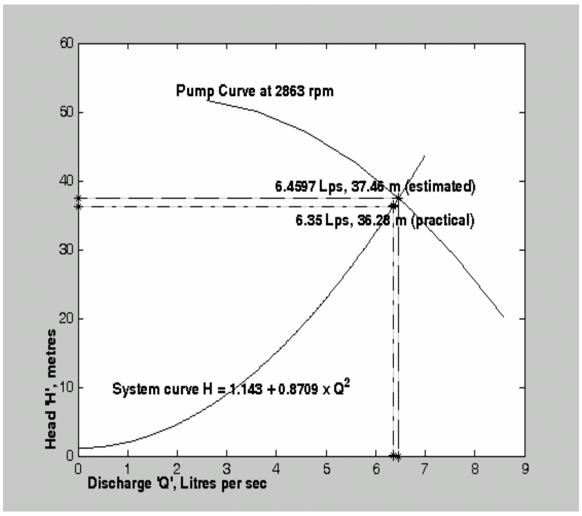


Fig 4 : Pump-System Characteristics at 2863 rpm

Now by placing the 25 mm pipe, speed improves to 2867 rpm, at the system curve is as per the relation H = [1.143 + 0.9428 * Q^2]. The pump curve is now redrawn for 2867 rpm and it is observed that practically obtained Q = 6.472 Liters per second & H = 40.63 m, while the operating point is Q = 6.2938 Liters per second & H = 38.33 m, as shown in Fig 5 below.

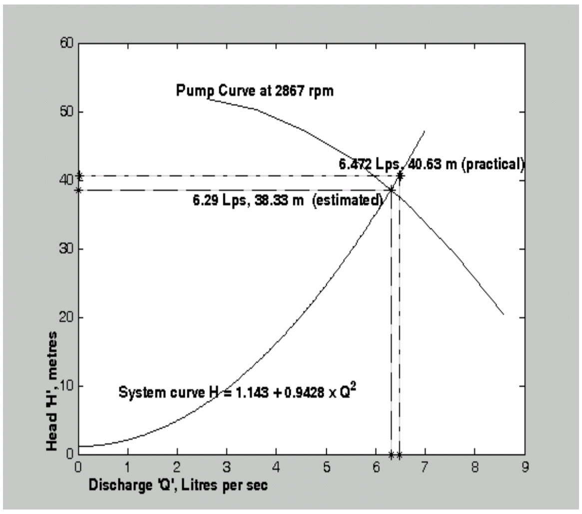


Fig 5 : Pump-System Characteristics at 2867 rpm

In this case, as the operating point is slightly beyond the pump characteristics, changes in supply system parameters would drag the operating point downwards, thereby achieving the intersection point, justifying the method proposed. The operating point is thus shifted as per the operating speed, which is now dependent on the system parameters. The operating parameters for the 25 mm pipefitting are as in Table 4A and Table 4B.

Table 4A
MOTOR DATA WITH 25 MM CONICAL PIPE ATTACHED TO SUCTION PIPE ENTRY

Pressure (Kg/cm^2)	Motor loading (%)	Motor η (%)	Pump η (%)
4.45	73.09	77.40	62.95

Table 4B
MOTOR DATA WITH 25 MM CONICAL PIPE ATTACHED TO SUCTION PIPE ENTRY

Pressure (Kg/cm^2)	Motor loading (%)	Speed (Rpm)	Q (Liters per second)	Discharge time for 1 liter (Sec)
4.45	73.09	2867	6.472	9.58

VI. CONCLUSIONS

From the pump curves plotted for various possible observed operating speeds, it is now possible to move the obtained H-Q point along the system curve so as to be closer to the pump curve. The cast iron piece when placed at the entrance of the suction side opening gave experimentally Q=6.472 Liters per second & H=40.63 m with the operating point for that speed being Q=6.2938 Liters per second for H=38.33 m. The summaries of changes in the actual working conditions are given in the Table 5A and Table 5B below.

Table 5A
COMPARATIVE SUMMARY OF RESULTS WITH AND WITHOUT 25 MM DIA CONICAL PIPE FITTING

	Pressure (Kg / cm^2)	H (m)	Q (Liters per second)	Time for 1 liter discharge (sec)
Before	4.45	31.83	6.037	10.27
After	4.45	40.63	6.472	9.58

Table 5B
COMPARATIVE SUMMARY OF RESULTS WITH AND WITHOUT 25 MM DIA CONICAL PIPE FITTING

	Motor η (%)	Pump η (%)	System η (%)	Input kW	Input kVA
Before	77.45	46.57	36.09	5.188	6.050
After	77.40	62.95	49.49	5.176	6.035

From the results tabulated so far, it can be concluded that based on the prevalent working conditions, it is still possible to achieve proper & effective utilization of available electrical energy. Though the results are not that attractive, still it is a method that leads to justifiable energy management with least possible additional investment.

REFERENCES

[1] Handbook on Energy Audits and Management, TERI Publication, Replika Press, New Delhi

[2] M. A. Choudhary, M. A. Rahman, "Determination of Operating Conditions of Submersible Induction Motors", IEEE Trans Industry Applications, Vol 28, No 3, *pp 680-684,* May / June 1992.

[3] Ching – Yin Lee, "Effects of Unbalanced Voltage on the Operation Performance of a Three Phase Induction Motor", IEEE Trans Energy Conversion, Vol 14, No 2, June 1999, *pp 202-208*

[4] P. Giridhar Kini, Dr. P. N. Sreedhar, U. Harish Padiyar, "Energy Audit & Conservation", Proceedings of UGC Sponsored National Conference On Recent Trends In Electrical Energy Conservation And Management, Annamalai University, Tamil Nadu, INDIA Mar 02 – 03, 2002.

[5] Wei-Jen Lee, Rasool Kenarangui, "Energy Management for Motors, Systems and Electrical Equipment", IEEE Trans Industry Applications, Vol 38, No 2, *pp 602-607*, March / April 2002.

[6] G. K. Sahu, "Pumps", New Age International, India, ISBN : 81-224-1224-6