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THE THEORY AND APPLICATION OF TRUE WEIGHTED EFFICIENCY -- A NEW METRIC TO EVALUATE PUMP ENERGY EFFICIENCY CONSIDERING MULTIPLE OPERATING CONDITIONS

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ABSTRACT

Energy efficiency is being emphasized more prominently in the pump industry. New legislation known as Ecodesign requirements for water pumps have been introduced in Europe and more recently, the United States Department of Energy (DOE) introduced legislation requiring specific types of pumps sold in the US to meet a minimum Pump Energy Index (PEI) benchmark. Neither of these metrics qualifies as pump efficiency values in the traditional sense, nor can they compare pumps using multiple customer specified load conditions. In pursuit of improved efficiency metrics some practitioners have proposed using a time weighted average of the efficiency values over a specified load case. Back-testing does not always guarantee that higher time averaged efficiency corresponds with lower energy consumption. These deficiencies inspired the development of a new efficiency metric called *True Weighted Efficiency* (TWE).

The TWE method is derived from first principles, using generalized load profiles for one or more system curves, multiple discrete condition points operating on those system curves, and varying time of operation at those condition points. Three numerical case studies are presented to illustrate the method. The first two case studies contrast two different pumps operating under different load cases. The TWE equation is illustrated using a simplified TWE equation where the system engineer is only required to choose the appropriate weighting factors, W, based on the applicable load case and tabulate pump efficiencies at each operating condition to determine the TWE value. The third case study compares the TWE, energy consumption, and energy cost for a high energy pipeline pump, considering either fixed or variable speed operation.

The TWE for any pump application scenario can be modeled; then confidently compared between 2 or more alternative pumps to determine the more energy efficient choice. The method is based on the well understood formulae for pump efficiency or wire-to-water efficiency. This method is generally applicable to rotodynamic or positive displacements pumps and other turbomachinery, warranting more widespread use in the industry.

INTRODUCTION

One of the most important and well known evaluation criteria in a pump selection is the pump efficiency. The pump efficiency is a numerical energy efficiency ratio between 0 and 100 percent used to compare two or more pumps, operating at a single operating condition. However, pumps rarely operate at a single operating condition. Pumps operate at different condition points and for different periods of time. How does an engineer measure and then describe an overall pump efficiency under these real world circumstances?

This article presents a new efficiency measurement called *True Weighted Efficiency (TWE)*. The TWE is a true measure of an overall pump efficiency for a specified pump operating at multiple operating points. TWE guarantees that the pump with the higher true weighted efficiency consumes less energy, based on the assumptions outlined in this method. Just as important, the TWE is easy to apply in practice, has meaning that is quickly understood in the context of the well-known definition of pump efficiency, and can be used on pump datasheets and performance evaluations. The TWE is suitable for broad use in the pump industry allowing engineers to confidently compare different pump alternatives in order to determine the more energy efficient choice.

BACKGROUND

The optimal selection of pumping equipment is a well-documented topic found in numerous papers and textbooks, including those from the author in Dahl (1997), Dahl and Patel (2008), and Dahl and Ochs (1997). The subject first and foremost requires that the equipment meets the essential functional requirements required by the application. Generally, there will be numerous alternative pumping system configurations, and different pump models that meet those requirements. The optimization process includes many considerations in the design process including pump type, pump configuration, and pump system design and operating control strategy (variable speed, on-off, throttle control, bypass control). These considerations are outlined in the references published by the Hydraulic Institute (2008), (2017). A general purpose evaluation method, Life Cycle Costing (LCC), considers the entire cost to purchase, install, operate, maintain and then dispose of that equipment during its lifetime. A methodology to perform LCC is published by the Hydraulic Institute and Europump (2001). For pumps operating under continuous service or for a significant portion of time, energy consumption is often the largest cost component of the LCC calculation. This paper will focus on energy consumption as a key decision criterion in the pump selection process.

Energy consumption is defined by the power consumed over a specified period of time. For a single operating condition with a constant input power, P, the energy consumption, E, over a specified period of time, t, is calculated as follows:

$$E = P \cdot t$$

Equation 1

The power consumed by the pump is typically denoted as P_2, and designated in literature as *pump power* or *pump shaft power*, which is the mechanical power input to the pump shaft. The total *pump input power* is typically denoted as P_1. When the input power is electrical power input from an electric motor or the combination of electric motor and control system, this is also known as the *wire-to-water power* as shown in Figure 1. The output from the pump is the *hydraulic power*, P_{hyd} , defined as follows where Q is pump flowrate, H is pump head, ρ is fluid density, and g is the gravitational constant as seen in Equation 2.

$$P_{hyd} = \rho \cdot g \cdot Q \cdot H$$

Equation 2

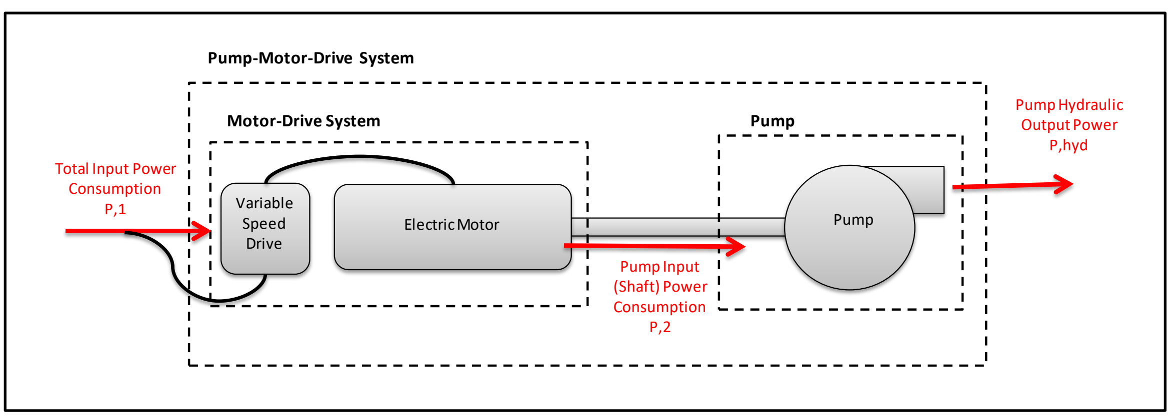

Figure 1: Power Consumption and Hydraulic Power Output in a Pump-Motor-Drive System

Efficiency, denoted by the Greek letter *Eta* or η, is defined as the ratio of useful output power to input power. Pump Efficiency, η_2, is the ratio of the hydraulic power of the pumped liquid to the pump shaft power, $\eta_2 = P_{hyd}/P_2$, whereas Total Pump Efficiency, η_1, is the ratio of the hydraulic power of the pumped liquid to the pump input power, $\eta_1 = P_{hyd}/P_1$.

Total Pump Energy Consumption, E_1, makes reference to pump input power, P_1, and Pump Energy Consumption, E_2 , to the pump shaft power, P_2. Total energy consumption directly affects the energy costs of a pump system, so managing the relationship between power, efficiency and operating time is an important goal in the design of energy efficient pumping systems. The duty point of a pumping system is determined by the intersection of the system curve with the pump curve as shown in Figure 2. The pump curve is provided by the pump manufacturer and is a function of the pump design and impeller trim, operating speed and properties of the process liquid. The system curve represents the differential head demanded by the pumping system between an input and output reference point, including losses that occur as a function of the flowrate through the system.

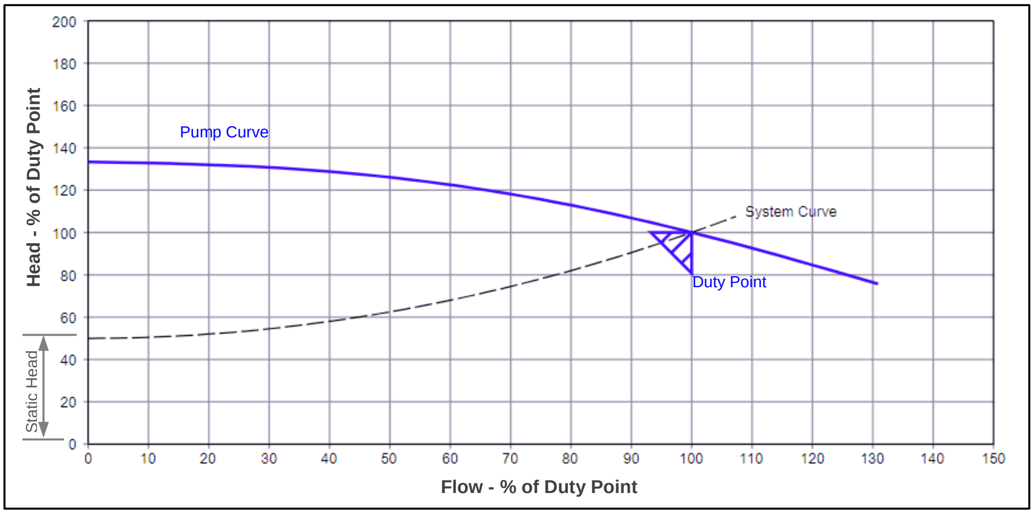

Figure 2: Typical Pump Curve and System Curve

The system curve may be mathematically depicted using the following equation:

$$H_{system} = H_{Static} + H_{loss}$$
Equation 3

$$H_{system} = H_{Static} + \left[\frac{f \cdot L}{D} + \sum K\right]\frac{V^2}{2g} = H_{Static} + C \cdot Q^2$$
Equation 4

where the static head, H_{static} includes the effect of pressure and elevation between the input and output reference points. The head loss due to friction, H_{loss} includes the frictional losses due to pipe friction, ($\frac{f \cdot L}{D}$), control valves and other components in the piping system ($\sum K$). The frictional losses are individually calculated for each loss component but may be reduced to a simple constant, C, as shown in Equation 4. This results in the typical parabolic mathematical curve shape depicted in many system curves.

Most pumping systems operate at multiple conditions, along a single system curve or multiple system curves. These operating points occur at different head and flow conditions, and each operates for a specified duration of time. This is known as the *Load Profile*, that considers the following factors:

- One or more System Curves
- Discrete operating points on the system curves, usually depicted as a percentage of a rated flow condition, *Q%*
- Time of operation at each operating point. The time of operation is usually determined over a fixed duration, often a 24 hour period or annually. The percentage of time operating at each point defined by *t%*.

The pump performance characteristic and system curve interactions with the load profile are governed by the pump control method. Figure 3 illustrates a flow control curve, denoted as System Curve A, with the pump operating at 100 percent, 75 percent, 50 percent, and 25 percent of the duty point flow. Both a fixed speed and a variable speed pump may be applied to achieve the desired flow control.

Since System Curve A is the desired flow control curve, the fixed speed pump may use throttle control to achieve the four desired operating flows. This results in the throttled system curves depicted in System Curve B, C, and D respectively. The differential head loss due to throttling is depicted by the vertical arrows in the area between the Pump Curve and the System Curve A. Alternatively, a pump operating with variable speed control can satisfy the 4 operating conditions by varying the speed from 100% to 52%, as shown.

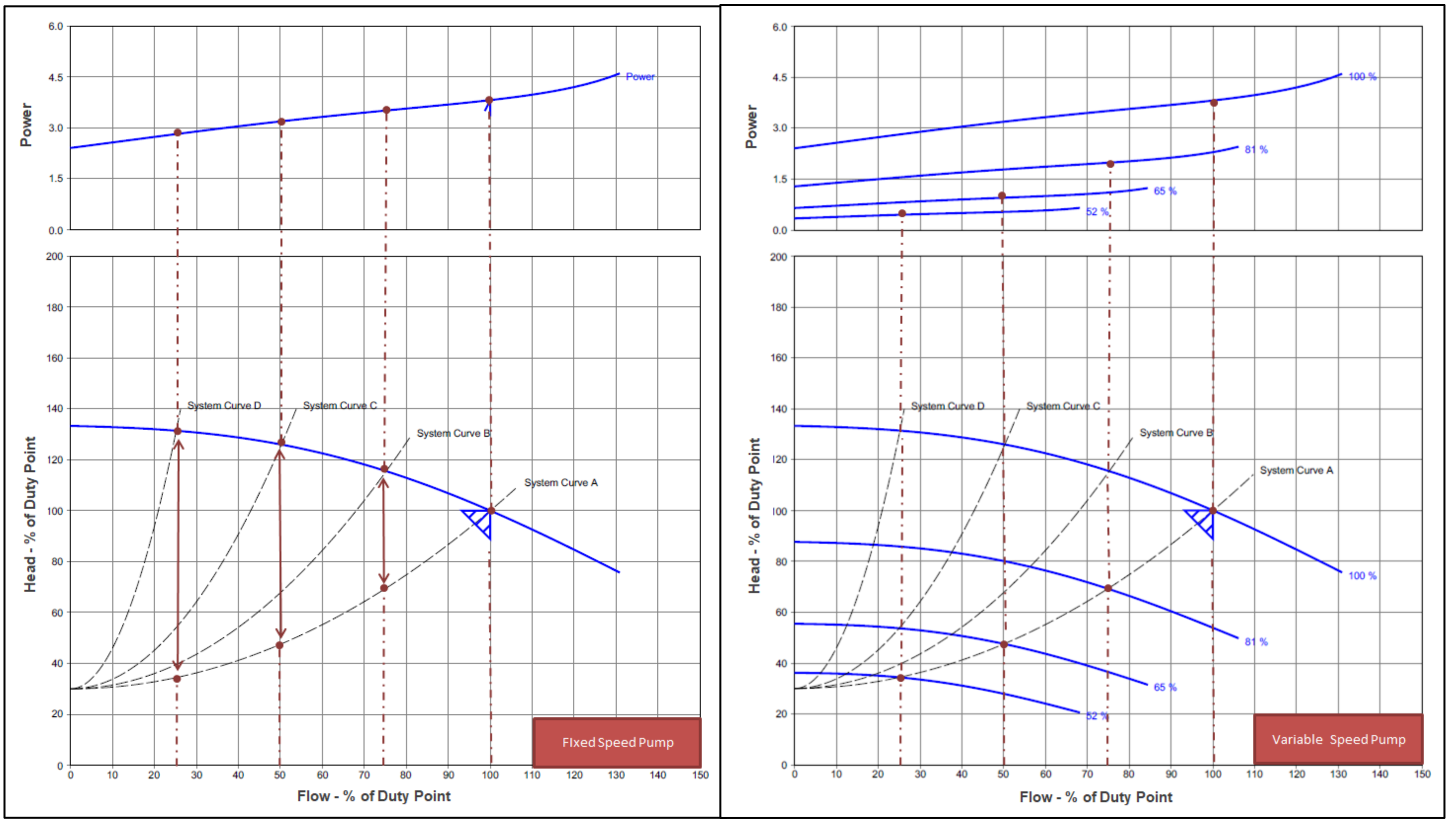

Figure 3: Fixed and Variable Speed Pump and System Curves

Total system energy consumption, E_{total}, at multiple operating points i = 1, 2, 3… N is defined in equation form as follows:

$$\mathrm{E}_{total} = \sum_{i=1}^{N}(E_i) = \sum_{i=1}^{N}(P_i \cdot t_i)$$

Equation 5

The lowest total energy consumption is often considered the optimal choice when comparing alterative pump systems that satisfy each of the other functional requirements. While total system energy consumption is a key optimization method when comparing alternatives, this calculation is not always performed in practice. Engineers prefer to use pump efficiency as a preferred metric, as it is well understood in the industry and readily comparable between different alternative pumping solutions. Unfortunately, there is no consistent or universally applied method for calculating efficiency involving multiple operating conditions.

ENERGY EFFICIENCY METRICS

The emphasis on energy savings has encouraged new government regulations in the European Union , the United States and other countries. These regulations have spawned new energy efficiency metrics to rate certain classes of pumps. Further, some practitioners have introduced weighted average schemes to assess pump efficiency for multiple operation conditions. The following is a brief summary of these metrics.

Ecodesign Regulation and MEI

The Ecodesign requirements for water pumps, EU Regulation No 547/2012 was introduced in 2012 to establish minimum efficiency thresholds for rotodynamic water pumps in clean water services. This regulation considered a comprehensive empirical study of commercially available pumps, commissioned by the European Union (EU), leading to a methodology for calculating the Minimum Efficiency Index, MEI, for different types of water pumps. The MEI calculation is based on the pump type, the pump operating speed, pump specific speed and a coefficient, currently 0.4, that is based on a set MEI value. The minimum efficiency for a regulated pump is determined at 3 different points; the best efficiency point (BEP), at 75% of the BEP known as the *Part Load (PL)* and at 110% of the

BEP flow known as the *Over Load (OL)* condition. The EU regulation therefore establishes a minimum efficiency that all regulated pumps must achieve, at a fixed speed, at each of three different operating conditions. See Figure 4.

Europump Extended Product Approach
Building off of the Ecodesign regulation, Europump recognized the need for an actual efficiency metric to rate each pump individually, considering multiple operating conditions associated with each pump. Since total pump efficiency, η_1, is a better indicator of total energy consumption than pump efficiency, η_2 alone, consideration of the pump plus its motor and drive system is required. Accordingly, Europump is developing the Extended Product Approach, EPA, as outlined in Europump (2013). The key metric introduced in this approach is the Energy Efficiency Index, EEI, defined as follows:

$$\mathrm{EEI} = \frac{P_{1,avg}}{P_{1,ref}}$$
Equation 6

In the numerator, $P_{1,avg}$,is the average power consumption of the actual pump model. The pump model considers the total power consumption, $P_{1,i}$ of the actual pump operating under different operating conditions, *i*, from 1 to N, and the fractional time of operation, $t\%_i$ across each of the N operating points in the prescribed load profile.

$$\mathrm{P}_{1,avg} = \sum_{i=1}^{N} \left(P_{1,i} \cdot t\%_i\right)$$
Equation 7

In the denominator, $P_{1,ref}$, is the reference total power consumption of the actual pump at its reference pump efficiency, $\eta_{pump,ref}$, based on the current MEI = 0.4, the reference motor efficiency, $\eta_{mot,ref}$, based on IEC 60034-30, and reference hydraulic power consumption, $P_{hyd,ref}$, at the BEP condition. This is defined in the following equation.

$$\mathrm{P}_{1,ref} = \frac{P_{2,ref}}{\eta_{mot,ref}} = \frac{P_{hyd,ref}}{\eta_{pump,ref}} \cdot \frac{1}{\eta_{mot,ref}}$$
Equation 8

Extended Products, EP, are used in a variety of applications with different load profiles and control methods. These load profiles and control methods are grouped into (1) closed loop systems or open loop systems or (2) constant flow systems or variable flow systems. In summary, the Europump EEI metric is a useful way of establishing a numerical efficiency index for any pump. Lower EEI values are more energy efficient. More details about the approach may be found in Europump (2013).

DOE Energy Conservation Standard
In the United States, the Department of Energy (DOE) also introduced new legislation in February 2016, establishing an Energy Conservation Standard and Test Procedure for certain clean water pumps, DOE (2016). This Standard requires specific types of pumps sold in the USA to meet a minimum Pump Energy Index, PEI.

The PEI metric was inspired by Ecodesign Regulations and the EEI, but has a different calculation procedure. The PEI is used to rate a basic pump model, where PEI is the ratio of the Pump Energy Rating, PER, for a given pump model compared to the standard Pump Energy Rating, $\mathrm{PER}_{\mathrm{STD}}$. The standard considers pumps operating with either a constant load (CL) or a variable load (VL). The PEI formulas for PEI Constant Load, $PEI_{CL,}$ and PEI Variable Load, $PEI_{VL,}$ are shown below:

$$\mathrm{PEI}_{CL} = \frac{PER_{CL}}{PER_{STD}}$$
Equation 9

$$\mathrm{PEI}_{VL} = \frac{PER_{VL}}{PER_{STD}}$$
Equation 10

With respect to both PEI_{CL} and PEI_{VL}, the denominator, PER_{STD}, is defined as the *DOE minimally compliant weighted average power* of a bare pump including the minimally compliant motor part load losses. The procedure for determining the $\mathrm{PER}_{\mathrm{STD}}$ for a basic model is defined in the DOE Standards for each DOE pump equipment category, using standard DOE minimally compliant pump and motor efficiency values for the BEP conditions of the basic model at three operating conditions of 75%, 100%, and 110% of the best efficiency point (BEP). See Equation 11 and Figure 4.

$$\mathrm{PER}_{STD} = \sum_{i=1}^{N} \left(P_{STD,i} \cdot t\%_i\right)$$

Equation 11

In the numerator, $\mathrm{PER}_{\mathrm{CL}}$, is defined as the *weighted average power of the basic model* including the corresponding motor part load losses. The procedure for determining the $\mathrm{PER}_{\mathrm{CL}}$ for a basic model is defined in the DOE Standards, for each DOE pump equipment category, using the pump and motor efficiency values for the basic pump and given motor system. See Equation 12 and Figure 4:

$$\mathrm{PER}_{CL} = \sum_{i=1}^{N} \left(P_{CL,i} \cdot t\%_i\right)$$

Equation 12

The proportion of operating time, $t\%_i$, for the three constant load operating conditions is 33.3 percent each.

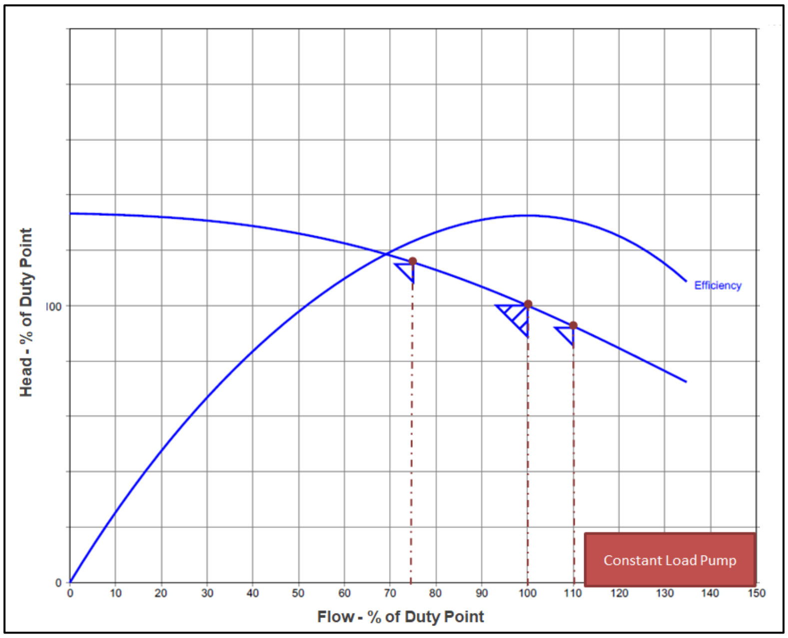

Figure 4: Flow versus Time Profiles for Constant Load Applications

Similarly, when calculating $\mathrm{PEI}_{\mathrm{VL}}$, the numerator is based on, PER_{VL}, defined as the *weighted average power of the basic model* including the corresponding motor part load losses. The load case is based on 4 different points at 25%, 50%, 75%, and 100% of the BEP flow. The proportion of operating time, $t\%_i$, for the four variable load operating conditions is 25 percent each. The procedure for determining the $\mathrm{PER}_{\mathrm{VL},}$ for a basic model is defined in the DOE Standards for each DOE pump equipment category, using the actual pump and motor efficiency values for the basic pump and given motor system. See Equation 13 and Figure 5.

$$\mathrm{PER}_{VL} = \sum_{i=1}^{N} \left(P_{VL,i} \cdot t\%_i\right)$$

Equation 13

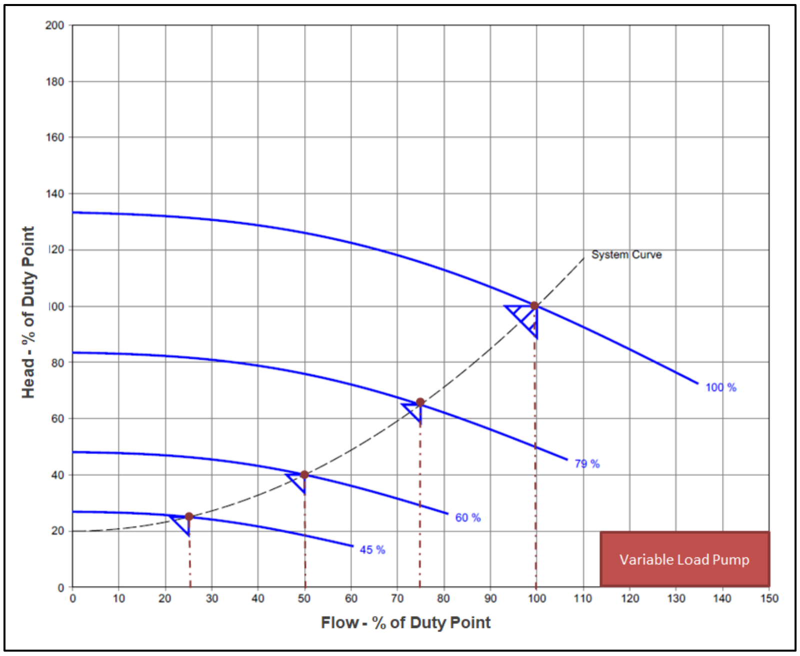

Figure 5: Flow versus Time Profiles for Variable Load Applications based on the DOE standard

Using the EEI and PEI Metrics
The energy efficiency of a regulated pump in the EU or the USA is intended to be measured using the EEI or the PEI metrics, respectively. These efficiency metrics are an improvement upon a conventional BEP efficiency comparison between two different pumps. These metrics consider multiple operating points, as well as constant load vs. variable load operation for PEI values, and constant flow vs. variable flow operation for EEI values. Any regulated pump is then easily compared, in a similar way that automobiles use an EPA fuel efficiency rating or an Energy Star rating is used on a household appliance.

While the MEI, EEI, PEI and PER are based on well documented calculation procedures and statistical analysis, these methods cannot be used to precisely determine the most energy efficient pump *for a specific pumping system application and load case.* These metrics are not *efficiency* values in the traditional sense as outlined later in Case Study #1.

Time Weighted Efficiency
In the quest to offer a pump efficiency metric that is calculated for specific customer applications involving multiple operating conditions, some practitioners have introduced calculations using a time weighted average of the efficiency values alone. One such approach is the *time weighted average efficiency*, designated, η_{TWA} , as shown in Equation 14. This method simply time weights each efficiency value for the i = 1 to N operating conditions to derived a time weighted efficiency.

$$\eta_{TWA} = \sum_{i=1}^{N} (\eta_i \cdot t\%_i)$$
Equation 14

The Part Load Efficiency Value, *PLEV*, is another time weighted average efficiency metric. This method was inspired by the Integrated Part Load Value, IPLV, applied to chillers and published in the AHRI Standard 550/590. The PLEV equation, shown in Equation 15, is applied to pumps as outlined in a white paper from Bell and Gossett (2016).

$$PLEV = \frac{1}{\frac{1\%}{Effy_A} + \frac{42\%}{Effy_B} + \frac{45\%}{Effy_C} + \frac{12\%}{Effy_D}}$$
Equation 15

Where
- $Effy_A$, is the efficiency at operating point A corresponding to 100 percent of the duty flow operating 1 percent of the time,
- $Effy_B$, is the efficiency at operating point B at 75 percent of the duty flow operating 42 percent of the time,
- $Effy_C$, is the efficiency at operating point C at 50 percent of the duty flow operating 45 percent of the time,
- $Effy_D$, is the efficiency at operating point D at 25 percent of the duty flow operating 12 percent of the time.

Both of these weighted efficiency measures are intriguing, for they imply a single efficiency metric that can be used to easily compare two or more pumps across multiple operating conditions. This inspired an investigation to derive a weighted average efficiency metric that is related identically to energy consumption. During the investigation, it was discovered that these two weighted average formulas, η_{TWA} and *PLEV*, only weigh pump efficiency based on the proportion of operating time. Since energy consumption is based on both time and power at each operating point, PLEV and η_{TWA} cannot guarantee that a higher weighted efficiency actually delivers lower overall energy consumption. This situation is illustrated in Case Study #2. An improved weighting method is needed in order to offer an accurate weighted efficiency metric.

TRUE WEIGHTED EFFICIENCY – TWE

TWE – Defined

True Weighed Efficiency overcomes the inconsistencies in the PLEV and η_{TWA} methods. TWE is a multi-condition efficiency value derived from basic engineering principles. TWE is based on the ratio of the total energy output, E_{output}, considering all operating points with the total energy input, E_{input}, at the same points. The general formula is defined as:

$$TWE = \frac{E_{output}}{E_{input}}$$

Equation 16

True Weighted Pump Efficiency, $TWE_2 = \frac{E_{hyd}}{E_2}$, or *True Weighted Total Efficiency*, $TWE_1 = \frac{E_{hyd}}{E_1}$ are similarly defined

Where

- E_{hyd} = The total hydraulic energy produced (output) by the pump,
- E_2 = The total shaft energy consumed (input) to the pump,
- E_1 = The total pump (input) energy consumed by the pump, motor, and drive system.

TWE_2 does not consider the losses consumed in the driver, including the motor or drive, while TWE_1 does consider these factors.

The general formula for True Weighted Total Pump Efficiency, TWE_1, for i = 1 to N operating conditions follows:

$$\boxed{TWE_1 = \frac{E_{hyd}}{E_1} = \frac{\sum_{i=1}^{N}(P_{hyd,i} \cdot t_i)}{\sum_{i=1}^{N}(P_{1,i} \cdot t_i)} = \frac{\sum_{i=1}^{N}\left(P_{hyd,i} \cdot t_i\right)}{\sum_{i=1}^{N}\left(\frac{P_{hyd,i}}{\eta_{1,i}} \cdot t_i\right)} = \frac{t_{total} \cdot \sum_{i=1}^{N}\left(P_{hyd,i} \cdot t\%_i\right)}{t_{total} \cdot \sum_{i=1}^{N}\left(\frac{P_{hyd,i}}{\eta_{1,i}} \cdot t\%_i\right)}}$$

Equation 17

While, True Weighted Pump Efficiency, TWE_2 is:

$$\boxed{TWE_2 = \frac{E_{hyd}}{E_2} = \frac{\sum_{i=1}^{N}(P_{hyd,i} \cdot t_i)}{\sum_{i=1}^{N}(P_{2,i} \cdot t_i)} = \frac{\sum_{i=1}^{N}\left(P_{hyd,i} \cdot t_i\right)}{\sum_{i=1}^{N}\left(\frac{P_{hyd,i}}{\eta_{2,i}} \cdot t_i\right)} = \frac{t_{total} \cdot \sum_{i=1}^{N}\left(P_{hyd,i} \cdot t\%_i\right)}{t_{total} \cdot \sum_{i=1}^{N}\left(\frac{P_{hyd,i}}{\eta_{2,i}} \cdot t\%_i\right)}}$$

Equation 18

TWE – Assumptions used for Variable Speed Applications

The following section develops the parameters that influence the TWE calculation for variable speed applications. This results in a simplified TWE formula based solely on weighting factors and efficiency values.

System Curve

Assume there is a system curve represented by a 100 percent duty condition with flowrate, $Q_{100\%}$, and head, $H_{100\%}$ passing through a the system static head, H_{static}. A parabolic shaped system curve is mathematically represented by the formulae:

$$H_{system,i} = H_{100\%} \cdot \left[Hsh0\% + ((1 - Hsh0\%) \cdot Q\%_i{}^2)\right]$$

Equation 19

$$H\%_i = \frac{H_{system,i}}{H_{100\%}} = \left[Hsh0\% + \left((1 - Hsh0\%) \cdot Q\%_i{}^2\right)\right]$$
Equation 20

Where
- $Q\%_i = Q_i / Q_{100\%}$, is the operating flowrate at duty condition, i, as a fraction of the 100 percent duty condition flowrate,
- $Hsh0\% = H_{static} / H_{100\%}$, is the static head at zero flow as a fraction of the 100 percent duty condition head,
- $H_{system,i}$ is the head operating at duty condition, i,
- $H\%_i$ is the head operating at duty condition, i, as a fraction of the 100 percent duty condition flowrate.

Operating Conditions

In general, there are i = 1 to N discrete operating conditions. To simplify, let's assume there are 4 discrete operating conditions, each designated as point A, B, C, and D in lieu of i = 1, 2, 3, and 4.

Operating Time

The total operating time, t_{total}, is composed of N operating times, t_i , from i= 1 to N.

$$t_{total} = \sum_{i=1}^{N} t_i$$
Equation 21

The percentage of operating time for each operating condition, i, is designated as $t\%_i$ such that:

$$t\%_i = {}^{t_i}/_{t_{total}} \quad ; \quad \text{and} \quad \textstyle\sum_{i=1}^{N}(t\%_i) = 1 = 100\%$$
Equation 22

Pump Control Method:

Variable speed pump operation is used such that the actual pump operating condition, designated by the intersection of the pump and system curve, occurs exactly on the system curve at each operating flow condition, $Q\%_i$. This is graphically depicted in Figure 5.

TWE – Derivation of Weighting Factors, W

The hydraulic power, $P_{hyd,i}$, along the system curve is calculated at any operating point, i, as follows:

$$P_{hyd,i} = \rho \cdot g \cdot Q_i \cdot H_i = (\rho \cdot g \cdot Q_{100\%} \cdot H_{100\%}) \cdot HPF_i$$
Equation 23

Where, HPF_i is the hydraulic power factor based on a parabolic shaped system curve.

$$HPF_i = (Q\%_i)^3 \cdot (1 - Hsh0\%) + (Q\%_i \cdot Hsh0\%)$$
Equation 24

One can see that the HPF_i is the fraction of the hydraulic power output at duty condition, i, compared to the hydraulic power at the 100% duty condition. Thus, the hydraulic energy output at any operating condition, i, is represented by the following equation:

$$E_{hyd,i} = P_{hyd,i} \cdot t_i = t_{total} \cdot (\rho \cdot g \cdot Q_{100\%} \cdot H_{100\%}) \cdot (HPF_i \cdot t\%_i)$$
Equation 25

and the total hydraulic energy output across all N operating conditions is,

$$E_{hyd,total} = \sum_{i=1}^{N}(E_{hyd,i}) = t_{total} \cdot (\rho \cdot g \cdot Q_{100\%} \cdot H_{100\%}) \cdot \sum_{i=1}^{N}(HPF_i \cdot t\%_i)$$
Equation 26

By defining $HPF_{TWA,}$ as the time weighted average of the hydraulic power factors as,

$$HPF_{TWA} = \sum_{i=1}^{N}(HPF_i \cdot t\%_i)$$

Equation 27

A Weighting Factor, W_i, is defined as shown in following equation:

$$W_i = \frac{E_{hyd,i}}{E_{hyd,total}} = \frac{HPF_i \cdot t\%_i}{HPF_{TWA}}$$

Equation 28

Allowing the TWE_1 to simplify to the following formula

$$TWE_1 = \frac{E_{hyd}}{E_1} = \frac{\sum_{i=1}^{N}(E_{hyd,i})}{\sum_{i=1}^{N}\left(\frac{E_{hyd,i}}{\eta_{1,i}}\right)} = \frac{E_{hyd,total}}{\sum_{i=1}^{N}\left(\frac{E_{hyd,i}}{\eta_{1,i}}\right)} = \frac{1}{\sum_{i=1}^{N}\left(\frac{E_{hyd,i}}{E_{hyd,total} \cdot \eta_{1,i}}\right)} = \frac{1}{\sum_{i=1}^{N}\left(\frac{W_i}{\eta_{1,i}}\right)}$$

In expanded form, the TWE_1 of a pump operating across four operation conditions A, B, C and D is

$$\boxed{TWE_1 = \frac{1}{\frac{W_A}{\eta_{1,A}} + \frac{W_B}{\eta_{1,B}} + \frac{W_C}{\eta_{1,C}} + \frac{W_D}{\eta_{1,D}}}}$$

Equation 29

where W is the individual weighting factor at each operating point, which is dependent upon the shape of the system curve, the *Q%* value, and the *t%* value at each operating point.

TWE_2 is written in the same way as follows:

$$\boxed{TWE_2 = \frac{1}{\frac{W_A}{\eta_{2,A}} + \frac{W_B}{\eta_{2,B}} + \frac{W_C}{\eta_{2,C}} + \frac{W_D}{\eta_{2,D}}}}$$

Equation 30

TWE – Representative Load Cases and Weighting Factors

Weighting factors are readily developed for various parabolic shaped system curves such as:

- @0% Hsh0 – a friction driven parabolic shaped system curve with a zero-static head
- @100% Hsh0 – a constant pressure system curve with the static head and all operating conditions equal to 100 percent of the duty head
- A combination of static head plus a friction component as depicted by the 30 percent and 60 percent static head curves shown in Figure 6.

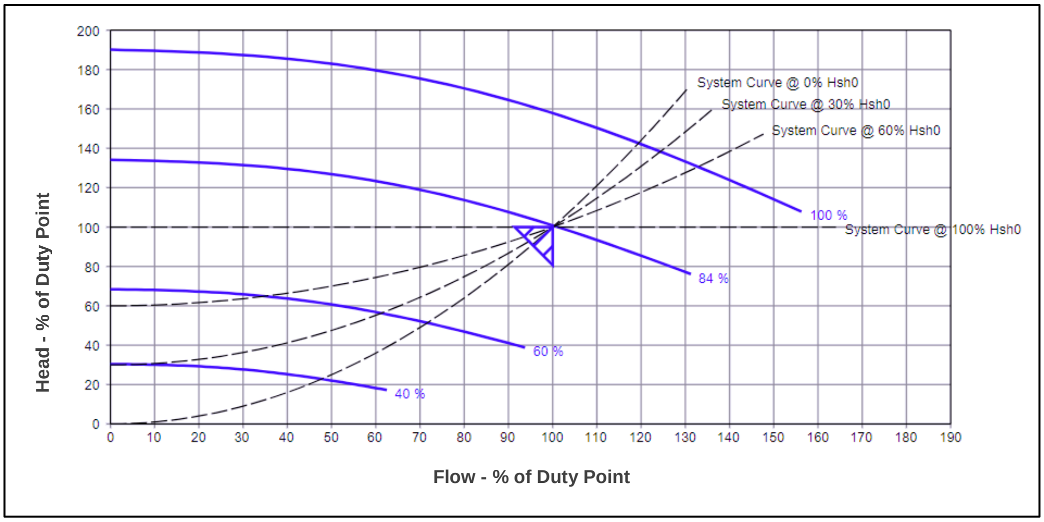

Figure 6: Variable speed pump performance with various system curves through the 100% duty condition

Example 1: Weighting Factors for a Parabolic System Curve with Equal Time Weighting

Weighting factors are developed based on the following:

- Four (4) operating conditions: A, B, C, and D with flowrates of 100 percent, 75 percent, 50 percent, and 25 percent of the $Q_{100\%}$ duty condition
- Uniform operating time of 25 percent at each of the four operating conditions
- Pump operating with a variable speed along a parabolic shaped system curve

In Table 1, the Weighting Factors, W, are depicted for each operating point A, B, C, and D and for different static head scenarios of Hsh0% between 0 percent and 100 percent. For comparison, the Hsh0% of 20 percent is highlighted as this is the same load case used in the DOE standard for the PEI_{VL} calculation.

Table 1: Weighting Factors for a Parabolic Shaped System Curve with Uniform Operating Time

Operating Point	% Duty Flow (Q%)	% Operating Time (t%)	Weighting Factor (W) based on Static Head, Hsh0%										
			0%	10%	20%	30%	40%	50%	60%	70%	80%	90%	100%
A	100%	25%	64.0%	60.4%	57.1%	54.2%	51.6%	49.2%	47.1%	45.1%	43.2%	41.6%	40.0%
B	75%	25%	27.0%	27.5%	27.9%	28.2%	28.5%	28.8%	29.1%	29.4%	29.6%	29.8%	30.0%
C	50%	25%	8.0%	9.8%	11.4%	12.9%	14.2%	15.4%	16.5%	17.5%	18.4%	19.2%	20.0%
D	25%	25%	1.0%	2.4%	3.6%	4.7%	5.6%	6.5%	7.4%	8.1%	8.8%	9.4%	10.0%
Totals		*100%*	*100%*	*100%*	*100%*	*100%*	*100%*	*100%*	*100%*	*100%*	*100%*	*100%*	*100%*

Example 2: Weighting Factors for a Parabolic System Curve using Unequal Time Weightings used in PLEV

Weighting factors are developed based on the following:

- Four (4) operating conditions: A, B, C, and D with flowrates of 100 percent, 75 percent, 50 percent, and 25 percent of the $Q_{100\%}$ duty condition
- Unequal operating time of 1 percent at point A, 42 percent at point B, 45 percent at point C, and 12 percent at point D, as used in the PLEV formula
- Pump operating with a variable speed along a parabolic shaped system curve

In Table 2, the Weighting Factors, W, are depicted for each operating point A, B, C, and D and for different static head scenarios of Hsh0% between 0 percent and 100 percent. The Hsh0% of 30 percent is a typical value used in PLEV studies, as highlighted in the table below. The weighting factors, W, in the TWE formula clearly differ from the time based weighted factors, t% used in the PLEV formula. The largest time-based weighting factor, t%, occurs at operating point C with a 45 percent weight whereas the largest TWE derived weighting factor, W, occurs at point B. This is due to the higher hydraulic power consumed at point B compared to point C, even though the time weighting is slightly higher at point C. Further, as the static head, Hsh0% increases from 0% to 100%, the W factors tend to become less dispersed and tend towards the time weighted factors, t%.

Table 2: Weighting Factors for a Parabolic Shaped System Curve with Operating Times adopted from PLEV

Operating Point	% Duty Flow (Q%)	% Operating Time (t%)	Weighting Factor (W) based on Static Head, Hsh0%										
			0%	10%	20%	30%	40%	50%	60%	70%	80%	90%	100%
A	100%	1%	4.1%	3.6%	3.2%	2.9%	2.6%	2.4%	2.2%	2.1%	1.9%	1.8%	1.7%
B	75%	42%	72.2%	68.5%	65.6%	63.2%	61.3%	59.6%	58.3%	57.1%	56.0%	55.1%	54.3%
C	50%	45%	22.9%	26.2%	28.8%	30.9%	32.6%	34.1%	35.3%	36.4%	37.3%	38.1%	38.8%
D	25%	12%	0.8%	1.7%	2.4%	3.0%	3.5%	3.9%	4.2%	4.5%	4.8%	5.0%	5.2%
Totals		*100%*	*100%*	*100%*	*100%*	*100%*	*100%*	*100%*	*100%*	*100%*	*100%*	*100%*	*100%*

Example 3: Weighting Factors for a Linear System Curve with using Unequal Time Weightings for an HVAC System

The Europump Extended Product Approach (Europump, 2013) provides a flow-time profile based on studies of HVAC systems under variable flow conditions. The load profile is as follows:

- Four (4) operating conditions: A, B, C, and D with flowrates of 100 percent, 75 percent, 50 percent, and 25 percent of the $Q_{100\%}$ duty condition
- Unequal operating time of 6 percent at point A, 15 percent at point B, 35 percent at point C, and 44 percent at point D
- Pump operating with variable speed along a *linear* shaped system (control) curve with Hso0% = 50 percent

The linear control curve modifies the formula for the System Head Curves as well as the Hydraulic Power Factor, HPF_i, as shown in equations Equation 31and Equation 32:

$$H\%_i = \frac{H_{system,i}}{H_{100\%}} = [Hsh0\% + ((1 - Hsh0\%) \cdot Q\%_i)]$$

Equation 31

$$HPF_i = (Q\%_i)^2 \cdot (1 - Hsh0\%) + (Q\%_i \cdot Hsh0\%)$$

Equation 32

In Table 3, the Weighting Factors, W, are depicted for each operating point A, B, C, and D and for different static head scenarios of Hsh0% between 0 percent and 100 percent. The Hsh0% of 50 percent is highlighted as this is the same load case used in the Europump guide for variable speed HVAC systems.

Table 3: Weighting Factors for a Linear Shaped System Curve with Operating Times for an HVAC system (Europump 2013)

Operating Point	% Duty Flow (Q%)	% Operating Time (t%)	Weighting Factor (W) based on Static Head, Hsh0%										
			0%	10%	20%	30%	40%	50%	60%	70%	80%	90%	100%
A	100%	6%	23.1%	21.5%	20.1%	18.8%	17.7%	16.7%	15.9%	15.1%	14.4%	13.7%	13.1%
B	75%	15%	32.5%	31.2%	30.1%	29.1%	28.2%	27.5%	26.8%	26.1%	25.6%	25.1%	24.6%
C	50%	35%	33.7%	34.5%	35.1%	35.7%	36.2%	36.6%	37.0%	37.4%	37.7%	38.0%	38.3%
D	25%	44%	10.6%	12.8%	14.7%	16.4%	17.9%	19.2%	20.4%	21.4%	22.4%	23.2%	24.0%
Totals		*100%*	*100%*	*100%*	*100%*	*100%*	*100%*	*100%*	*100%*	*100%*	*100%*	*100%*	*100%*

CASE STUDY #1

The following case study will illustrate the use of the TWE methodology. This study is modeled using the weighting factors from Example 1, based on the load case adopted for variable load, PEI_{VL}, calculations outlined in the DOE Standard.

Numerical Example – Case Study #1

The following numerical example is formulated as follows:

- 100% Duty Condition: $Q_{100\%}$ = 200 gpm; $H_{100\%}$ = 60 feet
- Static Head = 20% of $H_{100\%}$ = H_{sh0}% = 12 feet
- 4 Operating Conditions: A, B, C, and D with flowrates of 100 percent, 75 percent, 50 percent and 25 percent of the 100

percent duty condition with an equal operating time of 25 percent for each operating condition, respectively.

- Pump operating with a variable speed
- Weighting factors taken from the table in Example 1, for Hsh0% = 20% and shown in Table 4

Table 4: Case Study #1 Load Case and Weighting Factors

Operating Point	% Duty Flow (Q%)	% Operating Time (t%)	% Duty Head - (H%)	HPF	HPF * t%	Weighting Factor (W)
		Hsh0 =	20%			
A	100%	25%	100%	1.00	25%	57.1%
B	75%	25%	65%	0.49	12%	27.9%
C	50%	25%	40%	0.20	5%	11.4%
D	25%	25%	25%	0.06	2%	3.6%
Totals		*100%*			*44%*	*100%*

The TWE_1 formula for this load case and system curve with Hsh0= 20% is thus:

$$TWE_1 = \frac{1}{\frac{0.571}{\eta_{1,A}} + \frac{0.279}{\eta_{1,B}} + \frac{0.114}{\eta_{1,C}} + \frac{0.036}{\eta_{1,D}}}$$

Equation 33

One can see from the table above that the operating condition A will dominate the energy consumption with a weighting factor, W_A, of 57.1%. Operating condition B is at 75% flow is weighted at 27.9% and condition C at 11.4%. The relatively low hydraulic power output at the D condition at 25% flow only has a 3.6% weighting factor even though the pump is expected to operate 25 percent of the time at this condition. Based on these weighting factors, the most energy efficient pump will tend to have a peak near operating conditions A and B.

Discussion of Case Study #1

Pump selection software was used to analyze hundreds of hypothetical simulations across pump models in an end suction, frame mounted (ESFM) product line. For comparison purposes, two different pumps, denoted as Pump A and Pump B, are illustrated in Figure 7 and Figure 8 below.

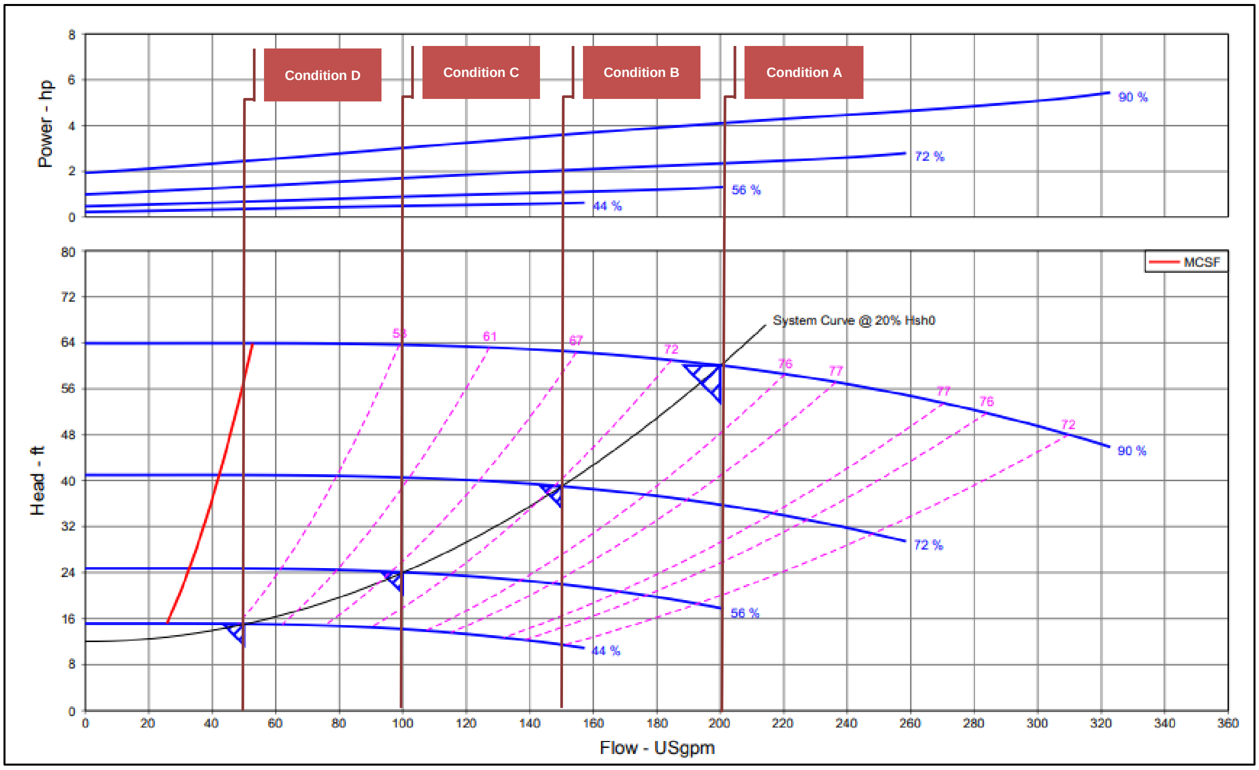

Figure 7: Case #1 – Performance Characteristic for larger Pump A

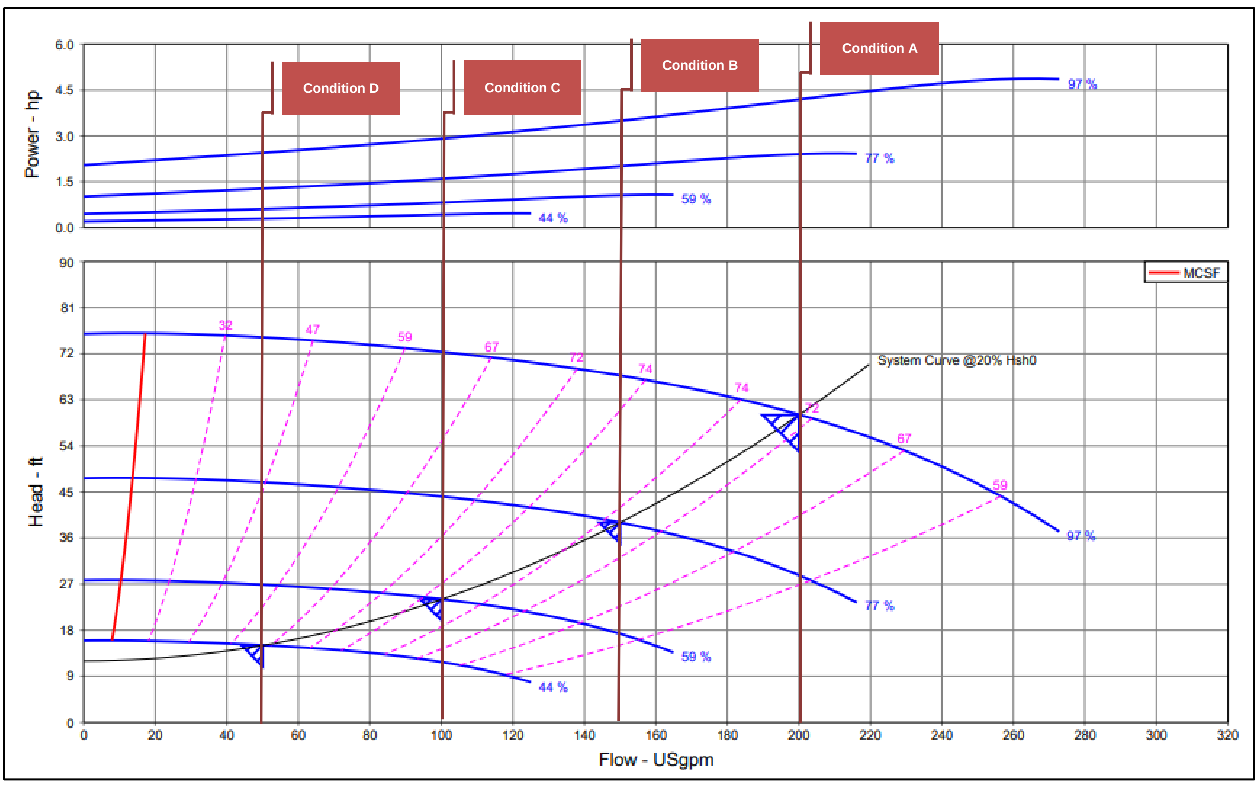

Figure 8: Case #1 – Performance Characteristic for smaller Pump B

The pump efficiencies for each of the 4 operating conditions are tabulated along with the respective TWE calculation in in Table 5. If the pump selection was based strictly on the 100% duty condition of 200 gpm and 60 ft head, Pump A with 74.0% pump efficiency would have been considered more attractive than Pump B at 72.5% pump efficiency. However, Pump B delivers higher efficiencies for operating points B, C, and D and therefore, in aggregate, has a higher true weighted efficiency, TWE, of 72.7% compared to 71.9% for Pump A. The ratio of the TWE of Pump B vs. Pump A is the same as the ratio of the energy consumption of Pump A vs. Pump B. The TWE is inversely proportional to the energy consumption, confirming that a higher TWE delivers lower energy consumption.

Table 5: Case Study #1 Pump Efficiencies and TWE Comparison

Operating Point	Pump A	Pump B
A	74.0%	72.5%
B	72.4%	73.6%
C	68.2%	74.3%
D	54.6%	65.6%

Metric	Pump A	Pump B
TWE	71.9%	72.7%
PEI,CL	0.94	0.88
PEI,VL	0.47	0.46
Energy (kW-hr)	32.98	32.62

PEI values are also shown, but must be interpreted differently than TWE. Recall that PEI is the ratio of the Pump Energy Rating, PER, for a given pump model compared to the standard Pump Energy Rating, PER_{STD}. Smaller PEI values are "better", as they indicate lower energy consumption. PEI values less than or equal to 1.00 indicate the PER values are lower than the PER_{STD} benchmark. In this case, the weighted average power consumption of Pump A under constant load conditions, PER_{CL}, is slightly better at 0.94 times the value of a *DOE minimally compliant weighted average power*, PER_{STD} , for a hypothetical pump at the same BEP as Pump A. Similarly, the weighted average power consumption of Pump A under variable load conditions, PER_{VL}, for Pump A is 0.47 times the value of a *DOE minimally compliant weighted average power*, PER_{STD}, for a hypothetical pump at the same BEP as Pump A at constant load. Since Pump B has PEI_{CL} and PEI_{VL} of 0.88 and 0.46 respectively, Pump B is nominally better than the DOE minimally compliant pump at its BEP condition, compared to Pump A at its BEP Condition. In summary, these PEI values are useful in comparing pumps with nearly identical BEP conditions but cannot be used to estimate energy consumption for pumps operating at non-BEP conditions, which typically occurs in practice.

CASE STUDY #2

The second case study will compare the TWE with the time-weighted average efficiency metrics as outlined in Example 2, using the time and flow weightings used in the PLEV formula.

Numerical Example – Case Study #2

The following numerical example is formulated as follows (Dahl (2018)):

- 100% Duty Condition: $Q_{100\%}$ = 230 gpm; $H_{100\%}$ = 54 feet
- Static Head = 30% of $H_{100\%}$ = H_{sh0}% = 16.2 feet
- 4 Operating Conditions: A, B, C, and D with flowrates of 100percent, 75percent, 50percent, and 25percent of the 100 percent duty condition and operating time of 1 percent, 42 percent, 45 percent, and 12 percent respectively
- Pump operating with a variable speed
- Weighting factors taken from the table in Example 2, for Hsh0% = 30% and shown in Table 6

Table 6: Case Study #2 Load Case and Weighting Factors

		Hsh0 = 30%				
Operating Point	% Duty Flow (Q%)	% Operating Time (t%)	% Duty Head - (H%)	HPF	HPF * t%	Weighting Factor (W)
A	100%	1%	100%	1.00	1%	2.9%
B	75%	42%	69%	0.52	22%	63.2%
C	50%	45%	48%	0.24	11%	30.9%
D	25%	12%	34%	0.09	1%	3.0%
Totals		*100%*			*35%*	*100%*

The TWE_1 formula for this load case and system curve with $H_{sh0}\% = 30\%$ is thus:

$$TWE_1 = \frac{1}{\frac{0.029}{\eta_{1,A}} + \frac{0.632}{\eta_{1,B}} + \frac{0.309}{\eta_{1,C}} + \frac{0.030}{\eta_{1,D}}}$$

Equation 34

In contrast to Case Study #1, the low operating time at Condition A results in a low weighting factor of 2.9%. The largest weighting factor, 63.2%, is shifted to operating condition B and the second largest weighting factor of 30.9% at condition C. The low hydraulic power output at the condition D delivers a small 3.0% weighting factor even though the pump is expected to operate 12% of the time at this condition. Based on these weighting factors, the most energy efficient pump will tend to have a higher efficiency near operating condition B and C.

Discussion of Case Study #2

The same two pumps, Pump A and Pump B, are illustrated in this case study as depicted in Figure 9 and Figure 10.

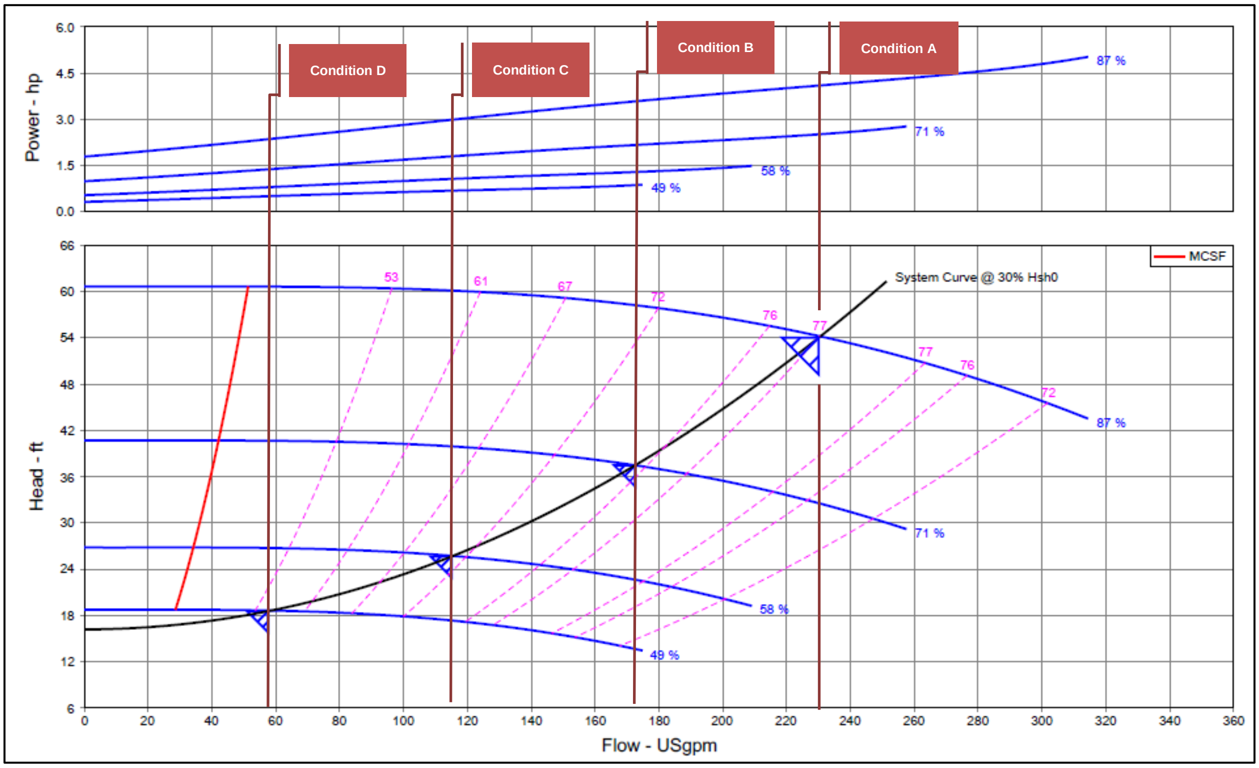

Figure 9: Case #2 – Performance Characteristic for larger Pump A

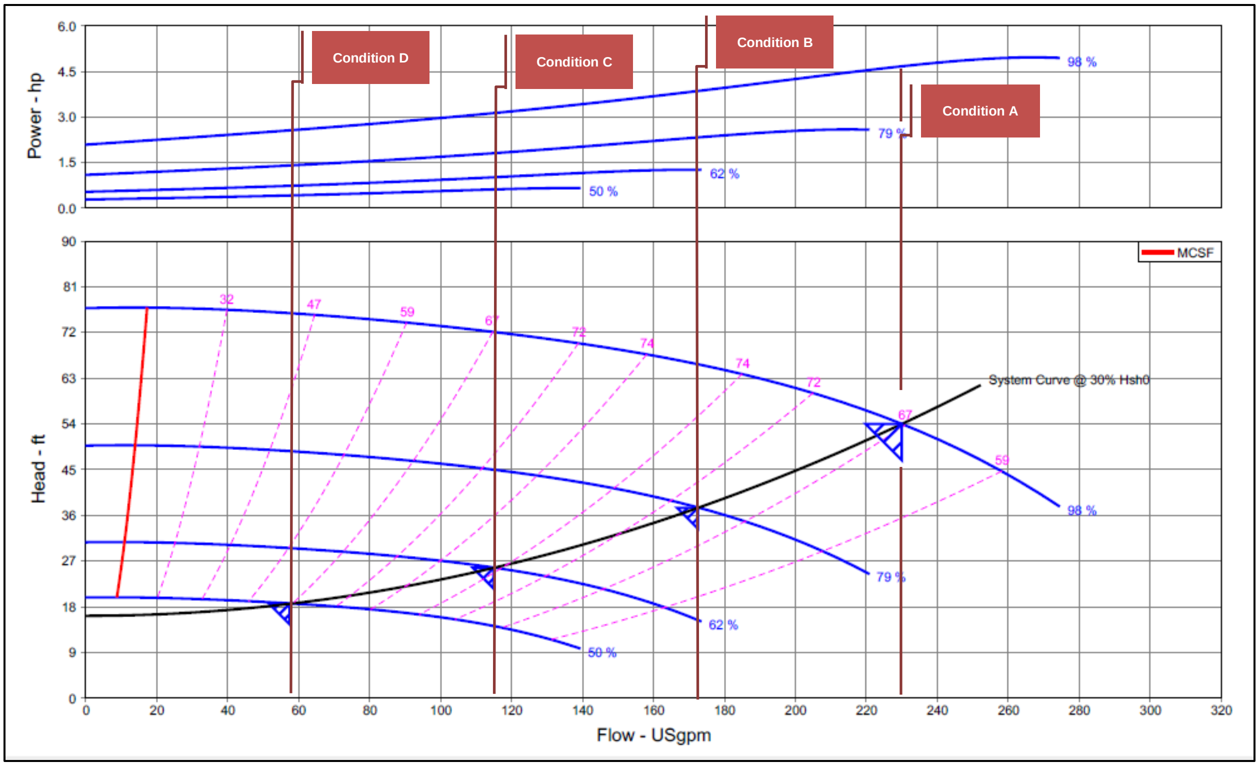

Figure 10: Case #2 – Performance Characteristic for smaller Pump B

The pump efficiencies for each of the 4 operating conditions are tabulated along with the TWE calculations in Table 7. If the pump selection was based strictly on the 100% duty condition point A of 230 gpm and 54 ft head, Pump A with 77.0% pump efficiency would have a significant advantage compared to Pump B at 67.2% pump efficiency. However, Pump B delivers higher efficiencies for operating points C, and D. Nonetheless, Pump A is the more energy efficient pump as indicated by its TWE of 73.4% compared to 71.4% for Pump B. The PLEV metric incorrectly suggests that Pump B is more efficient with a value of 71.5% compared to Pump A's lower PLEV value of 70.4%. Similarly, the time weighted average efficiency, η_{TWA}, of Pump B is 71.6% compared to the lower value of 71.1% for Pump A.

As found in Case Study #1, the ratio of the TWE of Pump B vs. Pump A is the same as the ratio of the energy consumption of Pump A vs. Pump B, confirming once again that the TWE is a faithful numeric indicator of energy consumption. Engineering practitioners should be cautioned that time weighted averages of pump efficiency alone, η_{TWA} or PLEV should be avoided when comparing two or more pumps as they may not accurately portray their relative energy efficiency, as revealed in this case study.

Table 7: Case Study #2 Pump Efficiencies and TWE Comparison

Operating Point	Pump A	Pump B
A	77.0%	67.2%
B	75.7%	70.5%
C	71.0%	74.2%
D	55.3%	66.6%

Metric	Pump A	Pump B
TWE	73.4%	71.4%
PLEV	70.4%	71.5%
η,TWA	71.1%	71.6%
Energy (kW-hr)	26.44	27.19

CASE STUDY #3

The third case study is applied to an existing pump system. One year of actual pump operating data is used to develop a load profile to determine the TWE for the pump operating along pump system curve with fixed speed operation. An alternative TWE is estimated for the same pump operating using variable speed.

Numerical Example – Case Study #3

The following numerical example is derived from data obtained from a pipeline pump application used to transport gasoline with a high pressure BB3 type multistage pump.

Initial Duty Conditions

- 100% Duty Condition: $Q_{100\%}$ = 2150 gpm; $H_{100\%}$ = 3150 feet with fluid density of 0.73 SG (propane)
- A between bearing split case multistage style pump (BB3) is installed. This 5 stage pump operating at 3560 rpm delivers 81.2 percent pump efficiency and 94.5 percent motor efficiency at the 100% duty condition.

Actual Operating Conditions

- The pump operates with a fixed speed motor. Flow control is achieved using a throttling control valve downstream of the discharge of the pump. The pump is simulated to operate for 6870 hours over a one year calendar period.
- The one year of operating conditions are used to approximate a parabolic shaped system curve with a load profile based on the following parameters:
  - 100% Duty Condition: $Q_{100\%}$ = 2150 gpm; $H_{100\%}$ = 3150 feet
  - Static Head = 20% of $H_{100\%}$ = H_{sh0}% = 630 feet
  - 4 Operating Conditions: A, B, C, and D with flowrates of 98 percent, 93 percent, 88 percent, and 77 percent of the 100 percent duty condition with operating times of 4 percent, 32 percent, 52 percent, and 12 percent respectively.

Case #3 – Fixed Speed Operation

Since this pump is operating with fixed speed, the pump curve and the system head curve only intersect at the 100% duty condition. The operating points A, B, C, and D along the system curve are achieved using a control valve downstream of the pump to throttle the pump discharge head. This head loss due to throttling is the difference between the pump head, H, and the system head, H_{system}, at each operating point. This is depicted in Figure 11 as labeled, “Throttled Head at Condition D”.

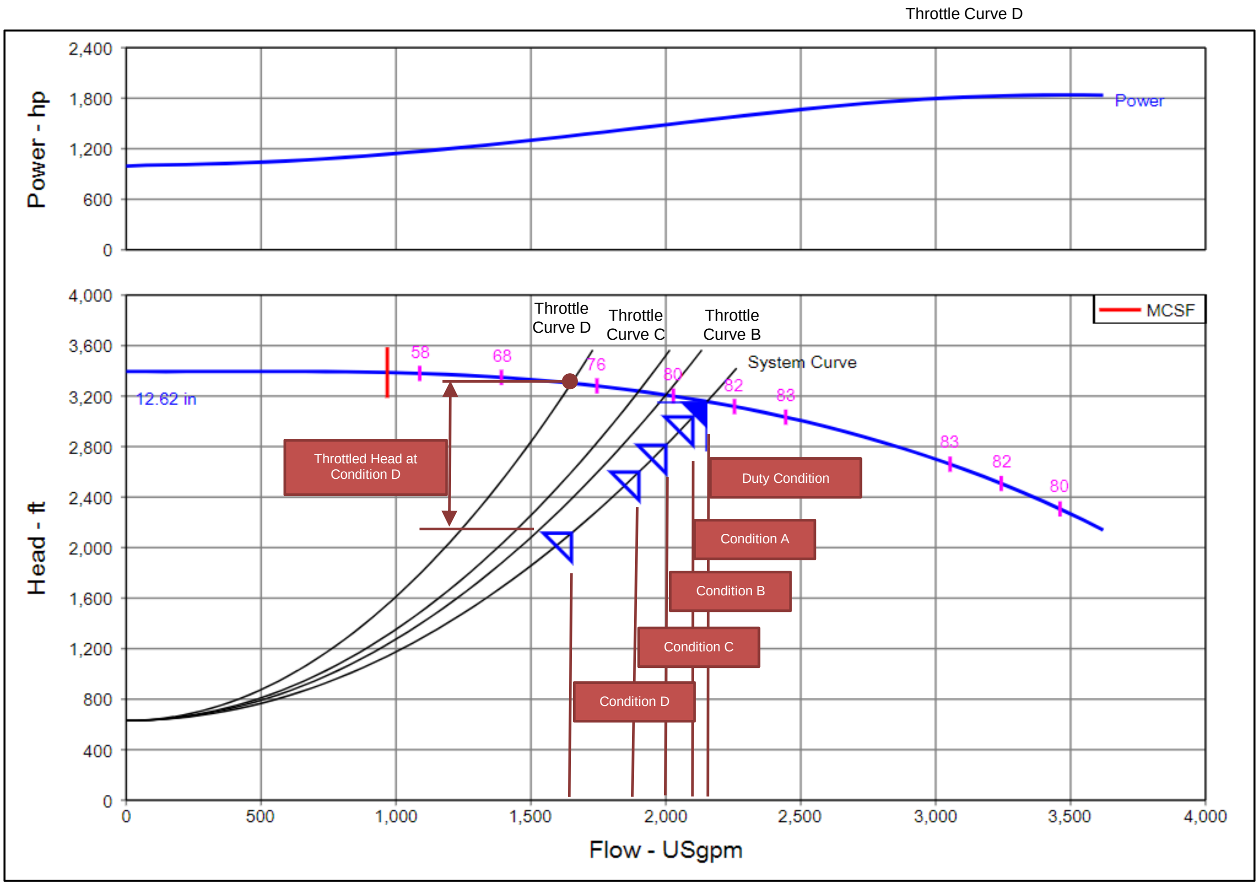

Figure 11: Case #3 – Fixed Speed Performance Characteristic for 5 stage BB3 Pump

The general formula for TWE in Equation 17 and Equation 18 is used to calculate the TWE_1 and TWE_2 for this fixed speed pump system. Details of the calculation procedure and the calculation of energy cost are found in the Table 8.

Table 8: Case Study #3 Fixed Speed TWE and Energy Cost Calculation

Value	Units	Load Case A	Load Case B	Load Case C	Load Case D	Total
Q%		98%	93%	88%	77%	
% Operating Time (t%)	%	4%	32%	52%	12%	100%
Operating Time (t)	hrs	274.8	2198.4	3572.4	824.4	6870.0
Rotating Speed	rpm	3,560	3,560	3,560	3,560	
Flow (Q)	gpm	2,100.0	2,000.0	1,900.0	1,650.0	
Head, pump (H)	ft	3,175.4	3,208.8	3,239.2	3,302.1	
Head, system (H,system)	ft	3,034.2	2,810.6	2,598.0	2,114.2	
Head,throttle loss	ft	141.2	398.2	641.2	1,187.9	
Specific Gravity (SG)	-	0.73	0.73	0.73	0.73	
Hydraulic Power (P,hyd,pump)	hp	1,229.26	1,183.04	1,134.54	1,004.39	
Hydraulic Energy Output (E,hyd,pump)	hp*hrs	337,801.9	2,600,800.5	4,053,023.5	828,018.1	7,819,644.0
Hydraulic Power (P,hyd,system)	hp	1,174.6	1,036.2	910.0	643.1	
Hydraulic Energy Output (E,hyd,system)	*hp*hrs*	*322,776.0*	*2,278,081.9*	*3,250,759.5*	*530,145.4*	*6,381,762.8*
Efficiency,pump (η2)	%	80.7%	79.7%	78.4%	74.2%	
Efficiency,motor	%	94.5%	94.5%	94.5%	94.5%	
Efficiency,drive	%	100.0%	100.0%	100.0%	100.0%	
Efficiency,pump (η1)	%	76.3%	75.3%	74.1%	70.1%	
P,1	hp	1611.11	1570.96	1530.95	1432.02	
Total Energy Consumption (E,1)	*hp*hrs*	*442,732.7*	*3,453,594.9*	*5,469,158.0*	*1,180,557.3*	*10,546,042.9*
TWE,1	%					**60.5%**
P,2	hp	1522.50	1484.56	1446.75	1353.26	
Pump Energy Consumption (E,2)	*hp*hrs*	*418,382.4*	*3,263,647.2*	*5,168,354.3*	*1,115,626.6*	*9,966,010.6*
TWE,2	%					**64.0%**
E,1	kWh	330,151.1	2,575,387.7	4,078,417.6	880,355.9	7,864,312.4
Energy Cost	$	$ 33,015	$ 257,539	$ 407,842	$ 88,036	$ 786,431

The TWE_2 of 64.0 percent is dramatically different from the original duty point pump efficiency of 81.2 percent. When including the effect of the motor efficiency, the TWE_1 of 60.5 percent gives a true indication of the ratio of the Hydraulic Energy Output from the system, 6,381,762.8 hp-hrs, versus the Total Energy Consumption, 10,546,042.9 hp-hrs, equivalent to 7,864,312 kWh, delivered during the 6870 hours of fixed speed pump operation. Using a representative energy cost of $0.10/kWh, the annual energy cost is $786,431. The energy lost due to throttling the pump hydraulic energy output to the system hydraulic energy output is 1,437,881 hp-hrs or 1,072,245 kWh. This equates to $107,225 in energy cost due to the pump throttling control.

Case #3 – Variable Speed Operation

Variable speed operation is often used as an alternative to discharge throttling to control the system flowrate. The following example depicts the predicted operating performance of the same pump using a variable speed control system with motor efficiency of 94.5 percent and a drive efficiency of 97 percent. Since this pump is operating with variable speed, the pump curve and the system head curve intersect at each of the operating points A, B, C, and D along the system curve, thus eliminating the throttling loss. This is depicted in Figure 12.

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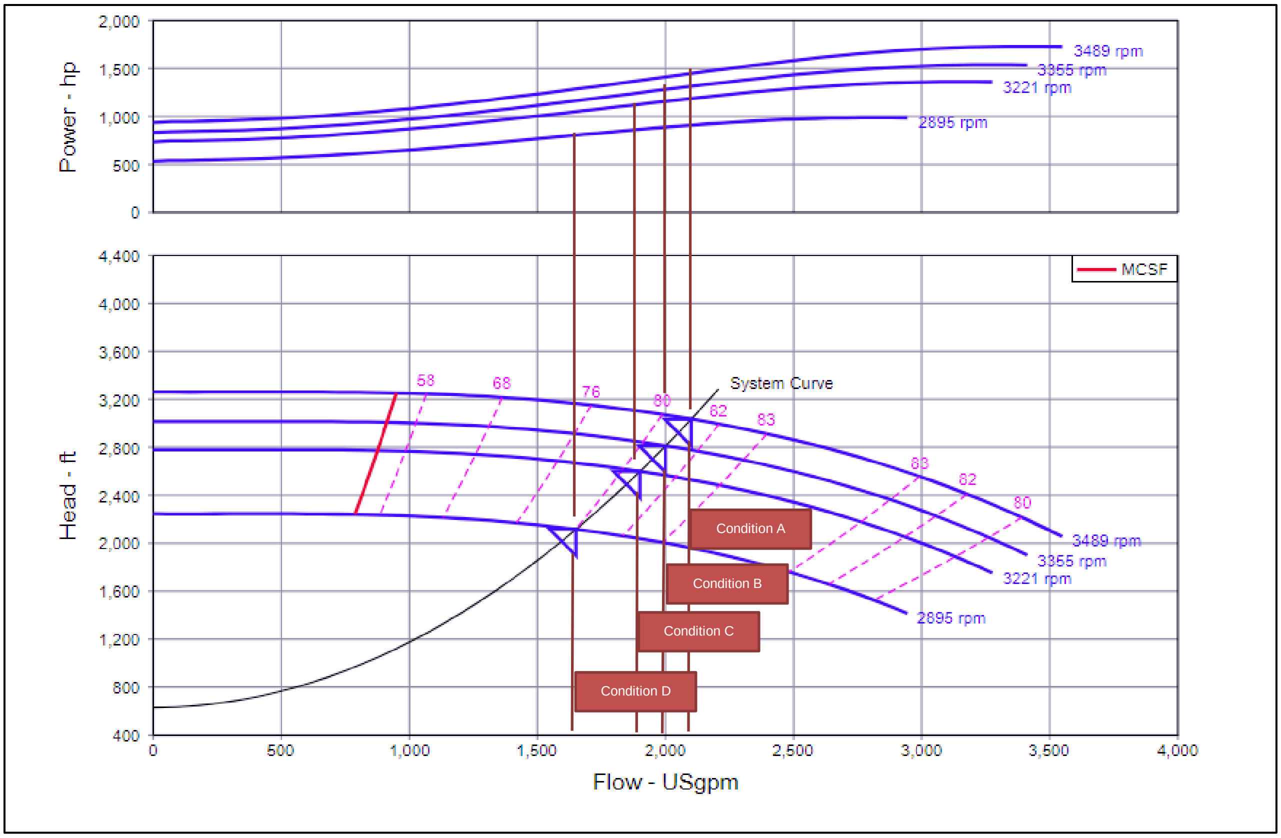

Figure 12: Case #3 – Fixed Speed Performance Characteristic for 5 stage BB3 Pump

The general formulae for TWE in Equation 17 and Equation 18 are once again used to calculate the TWE_1 and TWE_2 for this variable speed pump system. Details of the calculation procedure and the calculation of energy cost are found in Table 9.

Table 9 reveals that by variable speed operation, the TWE_2 of 80.8 percent is nearly the same as the original duty point pump efficiency of 81.2 percent. When including the effect of the motor and drive efficiency, the TWE_1 of 74.0 percent gives a true indication of ratio of the Hydraulic Energy Output from the system, 6,381,762.8 hp-hrs, versus the Total Energy Consumption, 8,619,164.8 hp-hrs, or 6,427,416.0 kWh, delivered during the 6870 hours of fixed speed pump operation. Using an energy cost of $0.10/kWh, the annual operating cost is $642,742.

Discussion of Case Study #3

The fixed speed and variable speed analysis outlined in this Case Study is fairly typical. This analysis is often performed to justify the incremental capital expenditures associated with fixed vs. variable speed operation. In this case, an annual energy savings of $143,690 could be realized if variable speed operation is used rather than fixed speed. These savings were achieved primarily by eliminating the $107,225 energy cost that was lost due to throttling. The remaining energy cost savings is attributed to the higher total pump efficiencies at each of the operating points with variable speed operation compared to fixed speed operation.

The insight provided by the variable speed weighting factors outlined in Equation 29 and Equation 30 could have also been used to optimize the pump selection, if variable speed operation was considered during the initial pump selection. The Variable Speed Weighting factors are readily calculated for Hsh0% = 20% using the load case and duty flow (Q%) and operating time (t%) assumptions outlined in this Case Study. For a parabolic shaped system curve, the Weighting Factors are shown in the Table 10.

Table 9: Case Study #3 Variable Speed TWE and Energy Cost Calculation

Value	Units	Load Case A	Load Case B	Load Case C	Load Case D	Total
Q%		98%	93%	88%	77%	
% Operating Time (t%)	%	4%	32%	52%	12%	100%
Operating Time (t)	hrs	274.8	2198.4	3572.4	824.4	6870.0
Rotating Speed	rpm	3,489	3,355	3,221	2,894	
Flow (Q)	gpm	2,100.0	2,000.0	1,900.0	1,650.0	
Head, pump (H)	ft	3,034.2	2,810.6	2,598.0	2,114.2	
Head, system (H,system)	ft	3,034.2	2,810.6	2,598.0	2,114.2	
Head,throttle loss	ft	-	-	-	-	
Specific Gravity (SG)	-	0.73	0.73	0.73	0.73	
Hydraulic Power (P,hyd,pump)	hp	1,174.59	1,036.25	909.97	643.07	
Hydraulic Energy Output (E,hyd,pump)	hp*hrs	322,776.0	2,278,081.9	3,250,759.5	530,145.4	6,381,762.8
Hydraulic Power (P,hyd,system)	hp	1,174.6	1,036.2	910.0	643.1	
Hydraulic Energy Output (E,hyd,system)	*hp*hrs*	*322,776.0*	*2,278,081.9*	*3,250,759.5*	*530,145.4*	*6,381,762.8*
Efficiency,pump (η2)	%	81.1%	81.0%	80.7%	80.0%	
Efficiency,motor	%	94.5%	94.5%	94.5%	94.5%	
Efficiency,drive	%	97.0%	97.0%	97.0%	97.0%	
Efficiency,pump (η1)	%	74.4%	74.2%	74.0%	73.4%	
P,1	hp	1579.43	1396.50	1229.51	876.71	
Total Energy Consumption (E,1)	*hp*hrs*	*434,026.5*	*3,070,074.6*	*4,392,305.7*	*722,758.0*	*8,619,164.8*
TWE,1	**%**					**74.0%**
P,2	hp	1447.78	1280.11	1127.03	803.63	
Pump Energy Consumption (E,2)	*hp*hrs*	*397,850.4*	*2,814,183.9*	*4,026,207.0*	*662,516.2*	*7,900,757.4*
TWE,2	**%**					**80.8%**
E,1	kWh	323,658.8	2,289,392.0	3,275,395.7	538,969.5	6,427,416.0
Energy Cost	$	$ 32,366	$ 228,939	$ 327,540	$ 53,897	$ 642,742

Table 10: Case Study #3 Variable Speed Load Case and Weighting Factors for a Parabolic System Curve

		Hsh0 =	20%			
Operating Point	**% Duty Flow (Q%)**	**% Operating Time (t%)**	**% Duty Head - (H%)**	**HPF**	**HPF * t%**	**Weighting Factor (W)**
	Q%	**t%**				
A	98%	4%	97%	0.95	4%	5.1%
B	93%	32%	89%	0.83	27%	35.8%
C	88%	52%	82%	0.72	38%	50.6%
D	77%	12%	67%	0.52	6%	8.4%
Totals		*100%*			*74%*	*100%*

The TWE_1 formula for this load case and system curve with $H_{sh0}\%$ = 20% is thus:

$$TWE_1 = \frac{1}{\frac{0.051}{\eta_{1,A}} + \frac{0.358}{\eta_{1,B}} + \frac{0.506}{\eta_{1,C}} + \frac{0.084}{\eta_{1,D}}}$$

Equation 35

This case study illustrates the use of the general TWE formulae shown in Equation 17 and Equation 18. The general TWA calculations produce the same result as the variable speed TWE_1 calculation for this load case shown in Equation 35. The Weighting

Factors indicate that a pump selection, biased to maximize the BEP near operating condition C will tend to deliver a higher TWE rather than attempting to match the BEP with the 100 percent duty condition.

TWE is a convenient way to quickly and accurately compare multiple different pump alternatives. In this example, the TWE_1 of 60.5 percent and $786,431 annual energy cost for the fixed speed pump operation compares in identical proportions to the TWE_1 of 74.0 percent and $642,742 annual energy cost for the variable speed pump operation. Accordingly, the TWE of alternative pump selections confidently indicate the proportional difference in energy consumption.

Engineering practitioners should be cautioned, however, that TWE analysis is not a substitute for thorough life cycle costing analysis that considers many other factors such as capital costs, installation or maintenance costs, or time value of money.

CONCLUSIONS

The spotlight on energy efficiency has brought new regulations from the EU and the US to promote the use of more energy efficient pump equipment and systems. New energy efficiency indices were introduced by this legislation, including MEI and EEI in the EU and PEI and PER in the US. Time weighted efficiency metrics were also introduced, all in the quest to offer more all-encompassing energy efficiency metrics to compare alternative pump models, under different load cases and multiple operating conditions.

True Weighted Efficiency, or TWE, is introduced as a general purpose method to provide a single pump efficiency value that may be used to measure the energy efficiency of any pump operating under multiple operating conditions. The TWE method is derived from first principles, using generalized load profiles applied to one or more system curves, multiple discrete condition points operating on those system curves, and the varying time of operation at each operating point. A simplified TWE method is also presented, based on pumps operating under variable speed operation and applicable to (1) parabolic system curves with varying static head and friction head components with equal time weightings, (2) parabolic system curves with varying static head and friction head components with unequal time weightings based on the PLEV formula, or (3) linear system curves operating along a specified flow or pressure control curve, modeled with unequal time weightings, typical for an HVAC application. The simplified TWE is calculated by choosing the appropriate weighting factors, W, based on the applicable load case, and tabulated pump efficiencies at each operating condition.

Three numerical case studies are outlined. The first two numerical case studies contrast two different pumps operating under different load cases. Case Study #1 revealed that a pump with the highest efficiency at the 100% duty condition is not always the most energy efficient pump as measured by the TWE. Further, one can gain insight on where to locate the highest pump efficiency for a variable speed pump by recognizing those operating points that have the highest weighting factors. Other energy indices such as PEI, while useful in determining the relative energy efficiency of a pump compared to a standard pump benchmark with the same BEP condition and load, are not particularly useful in determining the most energy efficient pump for a specific load case and application. Case Study #2 revealed the inconsistent conclusions one might encounter in using a time weighted average efficiency rather than the more accurate TWE approach. Case Study #3 compares the TWE and energy consumption calculations for as single multistage pipeline pump using either fixed speed or variable speed flow control. The general TWE calculation and the simplified TWE calculation for variable speed operation were both demonstrated.

The general TWE method is applicable to rotodynamic pumps operating at fixed or variable speeds, with on/off operation, throttle control, or by-pass control. This method is also applicable to positive displacement pumps and other types of turbomachinery. The ease in which the TWE method is applied, in addition to the simple derivation and explanation of the method, suggests an opportunity for broad adoption in the industry.

NOMENCLATURE

Symbol	Description
C	System Head Curve Coefficient
E	Energy Consumption
E_1	Total Energy Consumption by the Pump and Drive System
E_2	Pump Energy Consumption
E_{hyd}	Hydraulic energy output from a pump system
$E_{hyd,pump}$	Hydraulic energy output from the pump
$E_{hyd,system}$	Hydraulic energy output from the system
E_{input}	Energy input (consumed) within a system
E_{output}	Energy output from a system

E_{total}	Total Energy Consumption across multiple operating conditions
Effy	Pump or Total Efficiency, as used in the PLEV formula
$\frac{f \cdot L}{D}$	Dimensionless Pipe Friction Loss coefficient
H	Pump Head
H%	Pump Head as a fraction of the 100% rated head
$H_{100\%}$	The rated Pump Head corresponding to the 100% rated flow condition
H_{loss}	Head Loss due to friction in a pipe system
H_{system}	System Head
H_{static}	Static Head
Hsh0%	Static head at the zero flow on a system curve, as a fraction of the rated head at 100% flow
HPF	Hydraulic Power Factor
HPF_{TWA}	Time weighted average of all Hydraulic Power Factors
K	Dimensionless Piping System Loss Coefficient
P	Power Consumption
P_1	Power Consumed by the Pump and Drive System. Input Power. Wire-to-Water Power
P_2	Power Consumed by the Pump. Pump Power. Pump Shaft Power
P_{hyd}	Hydraulic Power
$P_{hyd,pump}$	Hydraulic Power delivered by the Pump
$P_{hyd,system}$	Hydraulic Power delivered by the System
PEI	Pump Energy Index
PER	Pump Energy Rating
PLEV	Part Load Efficiency Value
TWE	True Weighted Efficiency
TWE_1	True Weighted Total Efficiency
TWE_2	True Weighted Pump Efficiency
Q	Pump Flowrate
Q%	Pump Flowrate as a fraction of the 100% rated flow
$Q_{100\%}$	Pump Flowrate corresponding to the 100% rated flow condition
g	Gravitational Constant
SG	Specific Gravity
t	Operating time
t_{total}	Total operating time
t%	Fraction of operating time between 0 and 1
η	Efficiency. Greek letter (Eta)
η_1	Total Pump Efficiency including the pump and drive system
η_2	Pump Efficiency
ρ	Fluid density. Greek letter (Rho)
	Subscripts
$_{A,B,C}$ or $_{D}$	Operating conditions A, B, C, or D
$_{ave}$ or $_{average}$	Average
$_{CL}$	Constant Load. Used with PEI and PER
$_{i}$	The i'th value from a list from 1 to N
$_{mot}$	Motor
$_{ref}$	Reference
$_{STD}$	Standard. Used with PER
$_{VL}$	Variable Load. Used with PEI and PER

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