Comparison of three methods for the pump energy analysis

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Abstract: *This paper presents a comparison of three methods for any pump energy analysis. Each method is used for the analysis of three different water pumps from the conventional steam thermal power plant – two feed water pumps (FWP1 and FWP2) and condensate pump (CP). For each pump three essential types of mechanical power which defines all energy analysis methods are: delivered power from power producer, real (polytropic) power and ideal (isentropic) power. Method 1 which compares delivered and real (polytropic) power show the best performances, while Method 3 which compare delivered and ideal (isentropic) power should be avoided because it results with too high energy power loss and too low energy efficiency of any pump. Method 2 which compares real (polytropic) and ideal (isentropic) pump power can be used as a good compromise for the pump energy analysis in the most of the cases – its results are similar to results of Method 1.*
KEYWORDS*: PUMP, VARIOUS ENERGY ANALYSIS METHODS, ENERGY LOSS, ENERGY EFFICIENCY*

1. Introduction

Pumps are unavoidable components of various steam power plants [1-3], combined-cycle power plants [4, 5], cogeneration plants [6] and many different power and energy systems [7, 8].

The pumps function is the same as a function of compressors or turbocompressors – increasing of operating medium pressure [9, 10]. The only difference between pumps and compressors is in operating medium – pumps operate with liquids, while compressors and turbocompressors operate with gases, vapors and its mixtures [11, 12].

In the literature can be found pumps of various types which operate in different operating regimes [13]. Regardless of pump type or operating regime, the crucial element in energy analysis of any pump is taking into account three types of mechanical power which defines various losses – delivered mechanical power to pump from the mechanical power producer, mechanical power required for real (polytropic) pressure increase of liquid and mechanical power required for ideal (isentropic) liquid pressure increase.

The comparison of the mentioned types of mechanical power defines all the methods for any pump energy analysis. In this paper is described and presented each energy analysis method for any pump and all the methods are compared at each of three pumps from the conventional steam thermal power plant. Those pumps are two feed water pumps (FWP1 and FWP2) as well as condensate pump (CP). For each observed pump are calculated energy power losses and energy efficiencies by using each of three energy analysis methods. The obtained results are discussed in detail.

2. Pump description and operating characteristics

As the analysis in this paper will be performed by using three water pumps, all of the descriptions and explanations are based on the water as the operating medium for the pump (again, it must be taken into account that operating medium can be any liquid).

Operation principle of any pump is presented in Fig. 1. The pump takes liquid (water) of a lower pressure, increases its pressure and delivers liquid with a higher pressure to a higher pressure component [14]. For the liquid pressure increase, any pump uses mechanical power delivered from the mechanical power producer which are in the most of the cases electrical motors or in some situations steam or gas low-power turbines [15-17]. Mechanical power delivered to pump from mechanical power producer is the highest mechanical power related to any pump – it takes into account all the losses in shafts, bearings, pump inner losses and all the other losses which occur in power distribution.

For proper pump energy analysis (regardless of used method) are required operating parameters of liquid at pump suction side (inlet) and at the pump compression side (outlet). Those liquid operating parameters at the pump inlet and outlet are liquid pressure, temperature and mass flow rate. Therefore, pump operation can be analyzed only by measuring described liquid operating parameters at both sides of any pump. In Fig. 1, the operating points in which the measurements should be obtained are marked with yellow dots and marked as water inlet (input) and water outlet (output).

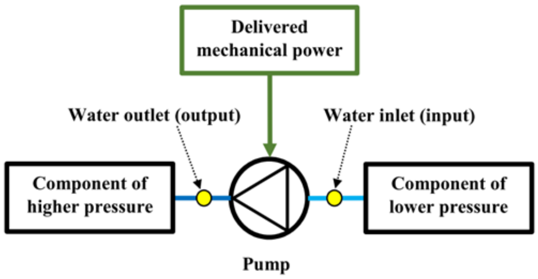

Fig. 1. *Scheme of the pump with two operating points (marked yellow) required for the analysis*

Measured operating parameters of liquid at the pump inlet (input) and at the pump outlet (output) defines real (polytropic) process of a pump, Fig. 2. This process takes into account all the losses which occur during liquid pressure increase. Losses during liquid pressure increase can be seen in liquid specific entropy increase at the pump outlet (in comparison to pump inlet). Second mechanical power related to any pump is real (polytropic) power, which is required for real (polytropic) pressure increase process. From the measured liquid operating parameters at the pump inlet and outlet can be calculated mechanical power required for the real (polytropic) process.

The third and final mechanical power of any pump is ideal (isentropic) mechanical power. This power can also be calculated from the measured liquid operating parameters at the pump inlet and from the calculated liquid operating parameters at the pump outlet. In ideal (isentropic) pump process, liquid operating parameters at the pump inlet are the same as in the real (polytropic) process. However, the difference in ideal and real pump pressure increase process can be seen in liquid operating parameters at the pump outlet. Ideal (isentropic) pressure increase process is a process between the same pressures but with assuming always the same liquid specific entropy, Fig. 2. Always the same liquid specific entropy during the pressure increase neglected any losses during such process. Any real process should be as close as possible to the ideal one, but due to losses, real process will never achieve the ideal one. By knowing the liquid specific entropy and pressure at the pump outlet can be calculated all the other liquid operating parameters in ideal process and therefore, from those parameters can be calculated ideal (isentropic) mechanical power.

Comparison of pump ideal and real pressure increase process, Fig. 2, leads to conclusion that in the real process pump will require more mechanical power (due to higher difference in liquid specific enthalpies at pump outlet and inlet). In both real and ideal pump processes, liquid mass flow rate is the same. As any pump is a mechanical power consumer, in ideal (isentropic) pressure increase process, between the same pressures as in the real process, pump will require the lowest mechanical power (in comparison to real and delivered mechanical power). From this point of view, for any pump is always valid following mechanical power relation:

$$P_{\text{delivered}} > P_{\text{real (polytropic)}} > P_{\text{ideal (isentropic)}}, \quad (1)$$

where P is mechanical power in (kW).

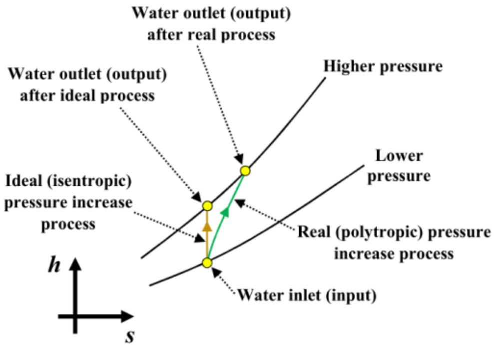

Fig. 2. *A comparison of real (polytropic) and ideal (isentropic) liquid pressure increase process in specific enthalpy-specific entropy diagram*

3. Equations for the energy analysis

3.1. General energy analysis equations and balances

The first law of thermodynamics defines energy analysis of any system or a control volume [18, 19]. The general energy balance equation, disregarding potential and kinetic energies, is [20]:

$$\dot{Q}_{\text{in}} + P_{\text{in}} + \sum \dot{En}_{\text{in}} = \dot{Q}_{\text{out}} + P_{\text{out}} + \sum \dot{En}_{\text{out}}, \quad (2)$$

where $\dot{Q}$ in (kW) is energy heat transfer, index in is related to the inlet (input) and index out is related to the outlet (output). $\dot{En}$ is a total energy of operating medium flow in (kW) [21], defined by the equation:

$$\dot{En} = \dot{m} \cdot h, \quad (3)$$

where $\dot{m}$ is operating medium mass flow rate in (kg/s) and h is operating medium specific enthalpy in (kJ/kg). Overall definition of the energy efficiency of any system or a control volume is [22, 23]:

$$\eta_{\text{en}} = \frac{\text{cumulative energy outlet (output)}}{\text{cumulative energy inlet (input)}}. \quad (4)$$

During the energy analysis of any system or a component usually did not occur any operating medium mass flow rate leakage, therefore it is also valid following mass flow rate balance [24]:

$$\sum \dot{m}_{\text{in}} = \sum \dot{m}_{\text{out}}. \quad (5)$$

All the general energy analysis equations and balances will be used in the following equations of three pump energy analysis methods.

3.2. Equations for three energy analysis methods of the pump

All pump energy analysis methods are based on the principles and operating parameters presented in Fig. 1 and Fig. 2. It should be noted that in any method must be fulfilled pump mechanical power relation presented in Eq. 1.

Method 1

The first method of pump energy analysis is based on comparison of mechanical power delivered to pump from power producer and real (polytropic) power required for real pump pressure increase process. The main problem of this method in practical calculations is that for many pumps, mechanical power delivered from mechanical power producer is not known or measured [25] because in each complex process pumps are auxiliary devices. Equations for this method will be derived from [26].

Pump energy power loss by using this method is:

$$\dot{En}_{\text{PL,M1}} = \dot{m}_{\text{in}} \cdot h_{\text{in}} + P_{\text{delivered}} - \dot{m}_{\text{out}} \cdot h_{\text{out}}, \quad (6)$$

where mechanical power delivered from mechanical power producer is measured variable. Mechanical power for the real (polytropic) pump pressure increase process is derived from measured fluid operating parameters at the pump inlet and outlet:

$$P_{\text{real (polytropic)}} = \dot{m}_{\text{out}} \cdot h_{\text{out}} - \dot{m}_{\text{in}} \cdot h_{\text{in}}, \quad (7)$$

therefore, pump energy power loss by using this method can be presented as:

$$\dot{En}_{\text{PL,M1}} = P_{\text{delivered}} - P_{\text{real (polytropic)}}. \quad (8)$$

Pump energy efficiency by using this method is:

$$\eta_{\text{en,M1}} = \frac{\dot{m}_{\text{out}} \cdot h_{\text{out}} - \dot{m}_{\text{in}} \cdot h_{\text{in}}}{P_{\text{delivered}}} = \frac{P_{\text{real (polytropic)}}}{P_{\text{delivered}}}. \quad (9)$$

Method 2

A second method of pump energy analysis is based on comparison of mechanical power which pump use in real (polytropic) pressure increase process and mechanical power which pump will use in ideal (isentropic) pressure increase process. This is the most common used method due to the highest data availability. This method, in fact, compared real pump process with the process which can be obtained in ideal situation.

Mechanical power for the ideal (isentropic) pump pressure increase process is:

$$P_{\text{ideal (isentropic)}} = \dot{m}_{\text{out}} \cdot h_{\text{out,IS}} - \dot{m}_{\text{in}} \cdot h_{\text{in}}, \quad (10)$$

where index IS denotes isentropic process. Mechanical power for the pump real (polytropic) pressure increase process is calculated by using Eq. 7. Pump energy power loss by using this method is:

$$\dot{En}_{\text{PL,M2}} = P_{\text{real (polytropic)}} - P_{\text{ideal (isentropic)}}. \quad (11)$$

Pump energy efficiency by using this method is:

$$\eta_{\text{en,M2}} = \frac{P_{\text{ideal (isentropic)}}}{P_{\text{real (polytropic)}}}. \quad (12)$$

Method 3

The third method of pump energy analysis is based on comparison of delivered mechanical power from mechanical power producer and mechanical power which pump will use in the ideal (isentropic) pressure increase process. Delivered mechanical power from mechanical power producer is measured inside the power plant, while the mechanical power for the ideal (isentropic) pump pressure increase process is calculated by using Eq. 10.

Pump energy power loss by using this method is:

$$\dot{En}_{\text{PL,M3}} = P_{\text{delivered}} - P_{\text{ideal (isentropic)}}. \quad (13)$$

Pump energy efficiency by using this method is:

$$\eta_{\text{en,M3}} = \frac{P_{\text{ideal (isentropic)}}}{P_{\text{delivered}}}. \quad (14)$$

4. Water operating parameters at pump input (inlet) and output (outlet) required for the analysis

Three described pump energy analysis methods are compared at each of three water pumps which required operating parameters are found in the literature [27].

Required water operating parameters (pressure, temperature and mass flow rate) at each pump inlet and outlet are presented in Table 1. Observed pumps are two feed water pumps (FWP1 and FWP2) as well as condensate pump (CP).

Table 1. *Water operating parameters at input (inlet) and output (outlet) of each pump [27]*

Pump	Operating point	Water mass flow rate (kg/s)	Water temperature (K)	Water pressure (kPa)
FWP1	Inlet	59.27	452.55	1032
	Outlet	59.27	456.34	18355
FWP2	Inlet	59.98	452.55	1030
	Outlet	59.98	456.34	18359
CP	Inlet	89.91	315.12	35.28
	Outlet	89.91	316.23	1618

By using water pressure and temperature at the inlet and outlet of each pump are calculated water specific enthalpies and specific entropies by using NIST REFPROP 9.0 software [28] and presented in Table 2. Required water specific enthalpies and specific entropies are calculated for both real (polytropic) as well as for the ideal (isentropic) water pressure increase process for each pump. From Table 2 can be seen that in the ideal (isentropic) pressure increase process water specific entropy at the inlet and outlet of each pump remains the same.

Table 2. *Water specific enthalpies and specific entropies at input (inlet) and output (outlet) of each observed pump in real (polytropic) and ideal (isentropic) pressure increase processes*

Pump	Operating point	Water specific enthalpy – real (polytropic) process (kJ/kg)	Water specific entropy – real (polytropic) process (kJ/kg·K)	Water specific entropy – ideal (isentropic) process (kJ/kg·K)	Water specific enthalpy – ideal (isentropic) process (kJ/kg)
FWP1	Inlet	760.43	2.1334	2.1334	760.43
	Outlet	785.97	2.1468	2.1334	779.86
FWP2	Inlet	760.43	2.1334	2.1334	760.43
	Outlet	785.97	2.1468	2.1334	779.86
CP	Inlet	175.79	0.5986	0.5986	175.79
	Outlet	181.82	0.6127	0.5986	177.39

5. Results and discussion

Energy power of water, calculated for each pump at inlet (input) and outlet (output) by using Eq. 3 is presented in Fig. 3. It can clearly be seen that both feed water pumps (FWP1 and FWP2) have much higher energy power inputs and outputs in comparison to condensate pump (CP).

It should be noted that FWP1 and FWP2 operates with much higher water pressures at inlet and outlet (Table 1) in comparison to CP, which is used for the pressure increase of condensate obtained in power plant main steam condenser.

The difference between energy power of water at each pump outlet and inlet denotes required mechanical power used in each pump (regardless of mechanical power type). Therefore, FWP2 will use the highest mechanical power, followed by FWP1, while the CP will use mechanical power much lower in comparison to both feed water pumps.

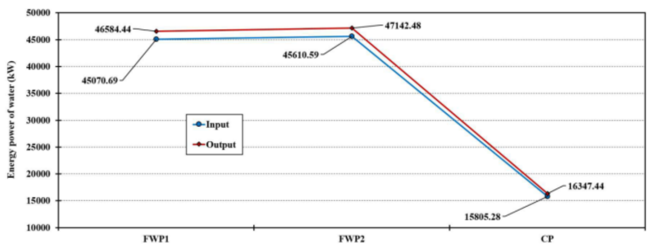

Fig. 3. *Comparison of water energy power input and output for three observed pumps*

The mechanical power relation for each pump, presented in Eq. 1 is clearly visible in Fig. 4. For each of three observed pumps delivered mechanical power is the highest one, followed by real (polytropic) power, while the lowest mechanical power is ideal (isentropic) one.

The conclusion obtained from Fig. 3 is also visible in Fig. 4 – FWP2 uses the highest mechanical power in comparison to other observed water pumps (regardless of the fact is that power delivered, real or ideal). Delivered mechanical power to FWP1, FWP2 and CP is equal to 1830 kW, 1860 kW and 850 kW, real (polytropic) power is equal to 1513.76 kW, 1531.89 kW and 542.16 kW, while ideal (isentropic) power is equal to 1151.62 kW, 1165.41 kW and 143.86 kW, respectively.

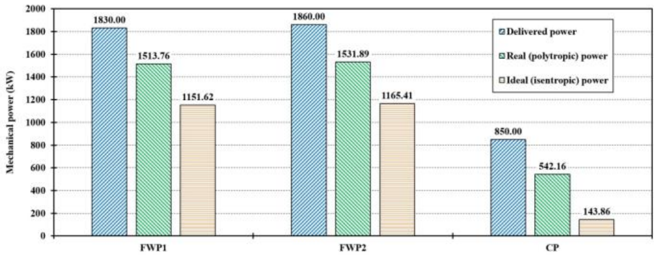

Fig. 4. *Comparison of delivered, real (polytropic) and ideal (isentropic) mechanical power for three observed water pumps*

Comparison of three methods for the pump energy analysis shows the same trends in energy power loss for each of three observed pumps, Fig. 5. The lowest energy power loss of each pump is obtained by using Method 1 (comparison of delivered and real power), followed by Method 2 (comparison of real and ideal power). It can be observed that Method 1 and Method 2 gives similar energy power loss for both feed water pumps, while the notable difference between those two methods in energy power loss can be seen only for condensate pump. Method 3 gives much higher energy power loss of each observed pump in comparison to the other two methods.

When comparing energy power loss trends between observed pumps, it can be seen that in Method 1 FWP2 has the highest, while CP has the lowest energy power loss, Fig. 5. Using Method 2 and Method 3 results with same trends in energy power loss – FWP1 has the lowest, while CP has the highest energy power loss.

A detail analysis and possible optimization of each of three observed pumps will be performed by using various artificial intelligence approaches [29-32].

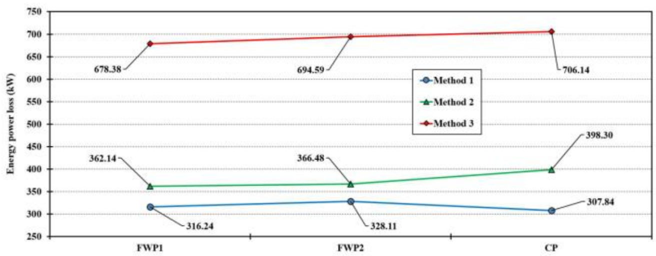

Fig. 5. *Energy power loss obtained in all energy analysis methods for three observed water pumps*

Comparison of Fig. 5 and Fig. 6 proves the fact that all of the observed pumps are components for which energy power loss and energy efficiency are reverse proportional.

Therefore, the highest energy efficiency of each observed pump will be obtained by using Method 1, while the lowest energy efficiency of each pump will be obtained by using Method 3.

Obtained energy efficiency for FWP1, FWP2 and CP is equal to 82.72%, 82.36% and 63.78% by using Method 1; 76.08%, 76.08% and 26.53% by using Method 2 and 62.93%, 62.66% and 16.92% by using Method 3, respectively, Fig. 6. Again, for both feed water pumps obtained energy efficiencies by using Method 1 and Method 2 are similar, while for the CP used energy analysis methods gives quite different results. For the CP, only Method 1 gives an acceptable energy efficiency result, while Method 2 and Method 3 gives unacceptably low energy efficiencies.

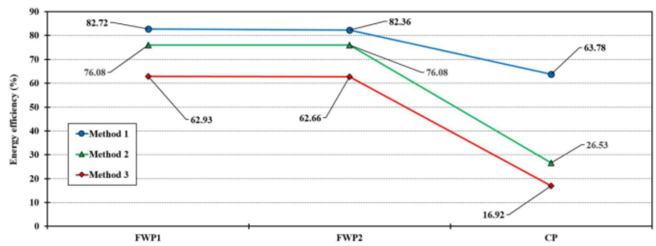

Fig. 6. *Energy efficiency obtained in all energy analysis methods for three observed water pumps*

Comparison in energy efficiency between all of the observed pumps gives as a result that the highest energy efficiency has FWP1, while the lowest energy efficiency has CP, regardless of used energy analysis method.

6. Conclusions

In this paper are presented three methods for any pump energy analysis. Each of observed methods is used for the analysis of three different water pumps from the conventional steam thermal power plant – two feed water pumps (FWP1 and FWP2) and condensate pump (CP). The most important conclusions are:

- The best energy analysis method for any pump is Method 1 which compare delivered and real (polytropic) mechanical power.
- Due to insufficient data (due to unknown delivered mechanical power from the mechanical power producer), Method 2 which compare real (polytropic) and ideal (isentropic) pump power can be used as a good compromise for the pump energy analysis – in the most of the cases obtained energy power loss and energy efficiency will be similar as in Method 1.
- The usage of Method 2 in the pump energy analysis can be questionable for the pumps which operate with low liquid pressure at the suction side.
- In any case, Method 3 should be avoided for the pump energy analysis, because it results with too high energy power loss and too low energy efficiency of any pump.

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