Improving the Overall Efficiency of an Electro-hydraulic Drive System 
by using Efficiency Maps

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In recent years, numerous applications of the variable-speed variable-displacement (VS-VP) hydraulic power unit have been investigated. Especially in a valveless drive system, the necessary hydraulic power could be supplied effectively through a flexible motor speed and pump displacement ratio. This paper presents a control strategy to improve the overall efficiency of a valveless transmission system by regulating motor speed and the displacement ratio simultaneously based on the data of combined overall efficiency maps. In this way, the interaction of the servo motor and hydraulic pump – and whether it affects the overall efficiency of the system – are studied. A numerical simulation is implemented for a series of typical working points of three types of hydraulic power units. The results show that the proposed control strategy achieves better efficiency values in almost all simulation points compared with single-variable power units (variable speed unit (VS-FP) and variable displacement unit (FS-VP)). In particular, at a low pressure and low flow rate points, the efficiency of the proposed system is improved by up to 6% and 25%, respectively, compared with the VS-FP and FS-VP power units.

Keywords: Hydraulic Pump, Electric Motor, Efficiency, Closed Hydraulic Circuit, Power Transmission, Power Control

1. Introduction

In recent decades, researchers and manufacturers have been striving to determine how energy can be saved during the operating process – in other words, how the overall efficiency of electro-hydraulic drive systems (EHDSs) can be increased. One effective alternative drive system that has been proposed is the valveless electro-hydraulic drive system (Figure 1), which uses a servo motor and variable-displacement pump to supply hydraulic power to an actuator. The key advantage of this system is that it avoids both an excessive flow rate and power losses in the hydraulic control valves.

![Valveless electro-hydraulic drive system](image)

Fig. 1 Valveless electro-hydraulic drive system

In a valveless drive system, many ways to adjust the actuator flow exist: regulating the rotation speed of the motor, changing the displacement of the hydraulic pump, and simultaneously controlling the motor speed and pump displacement, as shown in Table 1. Moreover, most power losses in EHDSs occur in the stage during which electrical power (servo motor) is converted into hydraulic power (hydraulic pump). Therefore, the overall efficiency of EHDSs depends primarily on the efficiency of the hydraulic power unit. For these reasons, to improve the overall efficiency of the hydraulic power unit, many energy-saving methods have been proposed in recent decades [1-11].

| Type   | Description                           | Symbol |
|--------|---------------------------------------|--------|
| VS-FP  | Variable-speed motor and fixed-displacement pump | ![Symbol] |
| FS-VP  | Fixed-speed motor and variable-displacement pump | ![Symbol] |
| VS-VP  | Variable-speed motor and variable-displacement pump | ![Symbol] |

The first of these methods concerns the configuration of the hydraulic power unit. Helduser et al. [3] and Tašner et al. [2] presented three types of direct pump control for EHDSs: variable-speed fixed-displacement (VS-FP), fixed-speed variable-displacement (FS-VP), and variable-speed variable-displacement (VS-VP). Helduser found that the efficiency of VS-FP is higher than that of FS-VP and VS-VP, especially when the system operates with partial load and in idling mode. To the contrary, according to Tašner, VS-VP (two degrees of freedom) is more efficient than FS-VP and VS-VP (one degree of freedom each), although it exhibits a lower response. Haihong Huang et al. [3, 4] established the theoretical model of hydraulic power units and experimentally validated that the
overall efficiency of VS-VP could be improved by regulating the motor speed and the corresponding displacement under different load conditions; such efficiency improvement was not possible with VS-FP and FS-VP. Willkomm et al.\cite{9,10,11} determined the potential of the VS-VP system and focused on improving its efficiency; this has been the focus of much essential research\cite{9,12,13}. Based on the above discussion, it is possible that, in the near future, a VS-VP power unit with two degrees of freedom could be the most popular and efficient system.

Second, to improve the overall efficiency of the VS-VP power unit, experimental tests and control methods have also been proposed. Quan et al.\cite{5} investigated a power-matching method combined with pump speed and displacement control to improve energy efficiency under different working conditions. Haihong Huang et al.\cite{6,7,8} proposed separate controllers that estimate the optimal control parameters (speed, displacement) for separate working points of the hydraulic system (flow, pressure) to minimize total power losses. To determine a better motor speed for higher efficiency, artificial intelligence and big data were applied in the control method\cite{9}. Willkomm et al.\cite{9,10,11} transformed the dynamic loss model of the system into a mathematical problem and then proposed a novel predictive control concept to optimize the energy consumption by utilizing the degree of freedom in the VS-VP system. Montgomery et al.\cite{12} presented a dynamic engine power demand estimator that minimizes engine power subject to a load demand.

In summary, the methods of prior studies generally determine the efficient rotating speed first, based on the specific displacement value, and then calculate the working displacement that corresponds to the rotating speed. However, the interaction between the electric motor and hydraulic pump was not considered in these control methods. Moreover, the characteristics of the electric motor and pump were only estimated by mathematical equations, meaning that the exact characteristics of the equipment have not been determined. As such, the claim exists that existing control methods exhibit disadvantages that could be further improved.

This paper presents a novel method for improving the overall efficiency of EHDSs by utilizing efficiency maps of servo motor and hydraulic pump that can properly perform the specifications of these components through all working points. The interaction between these components will be considered based on the overall efficiency map, which is a combination of component maps based on simulation data. Then, the advantages of this method, particularly those pertaining to the typical working points, will be discussed and compared with those of the single-variable power units. This research builds on the authors’ previous study\cite{5}. The results in this paper will be utilized to design a novel EHDS control system.

2. Nomenclature

\[
\begin{align*}
C_s & : \text{leakage coefficient} \\
C_{\alpha} & : \text{viscous friction torque coefficient related to } \alpha \\
C_{\alpha_2} & : \text{viscous friction torque coefficient irrelevant to } \alpha \\
C_f & : \text{mechanical friction torque coefficient related to } \Delta p \\
D & : \text{displacement of pump} \\
D_{\max} & : \text{maximum displacement of pump} \\
I_M & : \text{motor current} \\
K_T & : \text{torque coefficient} \\
K_{\alpha,2} & : \text{iron losses coefficients} \\
K_c & : \text{current losses coefficient} \\
\eta_m & : \text{motor speed} \\
\eta_p & : \text{rotational speed of pump ( } \eta_p = \eta_m \text{) } \\
\eta_{p,\max} & : \text{motor speed correlated to maximum } \eta_p \text{ at the working point with } \Delta p_0, Q_0 \\
Q & : \text{pump flow rate} \\
R_M & : \text{motor winding resistance} \\
P_m & : \text{motor power} \\
T_m & : \text{motor shaft torque} \\
T_s & : \text{torque losses independent of } \eta_p \text{ and } \Delta p \\
\Delta p & : \text{pressure differential} \\
\Delta P_{Cu} & : \text{copper losses in electric motor} \\
\Delta P_{Fe} & : \text{iron losses in electric motor} \\
\Delta P_C & : \text{current losses in amplifier} \\
\alpha & : \text{displacement ratio of pump ( } \alpha = D/D_{\max} \text{) } \\
\alpha_{\max} & : \text{displacement ratio correlated to maximum } \eta_p \text{ at the working point with } \Delta p_0, Q_0 \\
\mu & : \text{coefficient of viscosity} \\
\eta_m & : \text{motor efficiency} \\
\eta_p & : \text{pump efficiency} \\
\eta_{v} & : \text{pump volumetric efficiency} \\
\eta_m & : \text{pump mechanical efficiency} \\
\eta & : \text{overall efficiency} \\
\eta_{\max} & : \text{maximum overall efficiency at the working point with } \Delta p_0, Q_0
\end{align*}
\]

3. Efficiency of EHDS components

The EHDS in this study includes a servo system and a variable swash plate piston pump. The overall efficiency of the system will depend on the individual efficiencies; in this section, these efficiencies will be evaluated based on power losses that occur during the working process.
3.1 Efficiency of the servo system

The servo system in the EHDS includes a servo amplifier and a servo motor. Three types of dominant losses are present in this system:\(^{9,11}\):

- Copper losses in the electric motor:
  \[ \Delta P_{cCu} = 3R_{Cu}I_{Cu}^2 = 3R_{Cu}\left(\frac{T_{\text{m}}}{K_f}\right)^2 \]  
  \(1\)

- Iron losses in the electric motor:
  \[ \Delta P_{cFe} = K_J\rho_a + K_f\rho_f^2 \]  
  \(2\)

- Current losses in the amplifier:
  \[ \Delta P_c = K_{c}\left|\frac{V_{\text{m}}}{I_{\text{m}}}\right| = K_{c}\frac{T_{\text{m}}}{K_f} \]  
  \(3\)

From Eqs. (1), (2), (3), it can be seen that the power losses in the servo system depend not only on working parameters (parameters measured during operation such as \(I_{\text{m}}, T_{\text{m}}\) and \(n_{\text{m}}\)) but also the experimental coefficients \(K_J, K_{f1}, K_{f2}, T_{\text{m}}\), and \(K_c\). These values will vary according to control strategy, operation mode, and the working environment of the motor. Hence, accurately determining the performance of the servo system for a specified working point is difficult (combination of shaft torque \(T_{\text{m}}\) and speed \(n_{\text{m}}\)).

3.2 Efficiency of the axial swash plate piston pump

The hydraulic pump losses include volumetric and hydraulic-mechanical losses and can be determined using Eq. (4):

\[ \eta_p = \eta_{pV}\eta_{pm} \]  
(4)

According to reference\(^9\), the experience calculation equation of the volumetric efficiency \(\eta_{pV}\) (when neglecting the flow loss due to inlet restriction) and the mechanical efficiency \(\eta_{pm}\) of hydraulic pump can be expressed as Eqs. (5) and (6), respectively:

\[ \eta_{pV} = 1 - \frac{C_s\Delta p}{\mu \frac{2\pi}{60} \eta_{\alpha}} \]  
(5)

\[ \eta_{pm} = \frac{1}{1 + \left(C_s, \alpha^2 + C_i\right)\left(\frac{2\pi}{60}\frac{\mu n_{\text{m}}}{\Delta p} + C_i + \frac{2\pi T_{\text{m}}}{\Delta p D_{\text{max}}}\right)\frac{1}{\alpha}} \]  
(6)

where \(C_s, C_{i1}, C_{i2}, C_j\) and \(\alpha\) are non-dimensional quantities. \(\mu, \Delta p, T_{\text{m}}, n_{\text{m}}\), and \(D_{\text{max}}\) are dimensional quantities with the unit of \(\text{Pa.s}, \text{Pa}, \text{Nm}, \text{rpm}\) and \(\text{m}^3/\text{rev}\), respectively. \(2\pi/60\) is the coefficient to convert the unit of \(n_{\text{m}}\) from rpm to rad/s. \(1/2\pi\) is a coefficient to convert the unit of \(D_{\text{max}}\) from \(\text{m}^3/\text{rev}\) to \(\text{m}^3/\text{rad}\).

It is apparent from Eqs. (5) and (6) that, similar to servo systems, the efficiency of hydraulic pumps is the function of not only the working parameters \(\Delta p, \mu, n_{\text{m}}\), and \(\alpha\) but also the experimental parameters \(C_s, C_{i1}, C_{i2}, C_j\) and \(T_{\text{m}}\). To determine the value of those experimental parameters, conducting many experiments and analyzing the experimental data using approximate methods is necessary. Moreover, when Eqs. (5) and (6) are used, considering the effect of flow loss due to the inlet restriction on the operating process of the hydraulic pump is not possible. As a result, using these equations to set up the control system to optimize the overall performance of the system for a given operating point is difficult. Furthermore, the required speed of the hydraulic pump exhibits a significant effect on the performance of the servo motor. Therefore, considering the interaction between all component efficiencies in the drive system is necessary, namely the interaction between the servo motor and hydraulic pump.

3.3 Efficiency maps of EHDS components

The proper controllers of any servo system and piston pump in an EHDS must be implemented to maximize overall efficiency. These controllers will often operate the servo motor and piston pump at the most efficient point for a given power requirement. To do so, the servo system and piston pump characteristics must be understood, and this is best done by creating an efficiency map. An efficiency map is a chart that shows how the efficiency changes based on output power. It involves operating the equipment at all possible torque and speed points. In existing control methods for improving transmission efficiency, an efficiency map has been effectively used to save energy\(^{15,18}\).

| Component                  | Specification          |
|----------------------------|------------------------|
| Servo motor                | Rated output: 12.5 kW  |
|                            | Rated torque: 60 Nm    |
|                            | Rated speed: 1500 rpm  |
|                            | Maximum speed: 3000 rpm|
| Hydraulic pump             | Swash plate axial piston pump |
|                            | Displacement: 16.3 cm\(^3\)/rev |
|                            | Rated operating pressure: 28 MPa |
|                            | Shaft speed: 600~3600 rpm |

To make an efficiency map of a servo motor and hydraulic pump, a series of experiments must be conducted, especially for typical working points of equipment. However, the aim of this paper is only to validate the effectiveness of the proposed control strategy; as such, simulation efficiency maps were created using the efficiency equations in Sections 3.1 and 3.2.
and the assumed power unit specifications in Table 2. The experimental parameters in the efficiency equations were chosen by referring to the performance characteristic curves of the servo motor and hydraulic piston pump\(^{7,19,20,21}\).

To create the efficiency data, the working ranges of the servo motor and piston pump were constrained. Based on the specifications in Table 2 and the controllability of motor speed, the limits of the working parameters in that drive system were as follows:

\[
\begin{align*}
\text{Servo motor: } & \ P_m \leq 25 \text{ kW} \\
& T_m \leq 64 \text{ Nm} \\
& 0 < n_m \leq 3000 \text{ rpm}
\end{align*}
\]

(7)

\[
\begin{align*}
\text{Hydraulic piston pump: } & \ \Delta p \leq 28 \text{ MPa} \\
& 0.2 \leq \alpha \leq 1 \\
& 0 < n_p \leq 3000 \text{ rpm}
\end{align*}
\]

(8)

To reduce the complexity of creating a pump efficiency map, the displacement ratio \(\alpha\) was set as a discrete parameter. In other words, the series of value \(\alpha\) is 0.2; 0.25; 0.3; 0.35; 0.4; 0.45; 0.5; 0.55; 0.6; 0.65; 0.7; 0.75; 0.8; 0.85; 0.9; 0.95 and 1. Hence, the number of pump efficiency maps is 17, whereas a single motor efficiency map exists. Due to the difference of displacement ratio \(\alpha\), the pressure and flow rate limits in the pump efficiency maps differ from one another.

Fig. 2 Servo motor efficiency map

Fig. 3 Typical efficiency maps of hydraulic piston pumps
The efficiency maps of the servo motor and hydraulic piston pump (at typical values of displacement ratio) are shown in Figures 2 and 3, respectively. As shown in Figure 2, the motor efficiency reaches the highest values when the speed is near the rated value 1500 rpm and the output torque is higher than 10 Nm. Moreover, the motor efficiency decreases quickly at a low-speed range and small torque range. However, it can be seen from the efficiency maps in Figure 3 that the efficiency of the hydraulic pump at the maximum displacement ratio \( \alpha = 1 \) is higher than that of smaller displacement ratios. In other words, for increased efficiency, the hydraulic pump should be operated at maximum displacement.

However, for a given flow rate \( Q \), there are series of combinations of motor speed \( n_m \) and pump displacement ratio \( \alpha \). If the hydraulic pump is operated at \( \alpha = 1 \) to achieve the highest efficiency value, the motor speed \( n_m \) must be reduced to achieve the required flow rate \( Q \). This results in decreased motor efficiency and overall efficiency. This issue also occurs when the opposite solution is implemented, in which the motor speed remains at the rated value, and the displacement ratio is controlled to produce a sufficient flow rate. This could result in an efficient motor but a less efficient hydraulic pump. From the above discussion, it can be seen that, when studying the overall efficiency of an EHDS, the interaction between the servo motor speed \( n_m \) and displacement ratio of the hydraulic pump \( \alpha \) should be considered entirely in working range of these parameters. This issue will be solved by utilizing the overall efficiency maps, which were created by combining the individual efficiency maps (servo motor and hydraulic pump) and will be presented in Section 4.

In this study, the efficiency maps were made by using theoretical equations, so the effect of transient characteristics on servo motor and flow loss due to the inlet restriction on hydraulic pump are not taken into account. Experimental efficiency maps, which are established by experimental data, could consider properly these effects.

4. Control strategy for improving the overall efficiency of the EHDS

The overall efficiency of the EHDS in this research is determined by servo system efficiency and hydraulic piston pump efficiency.

\[ \eta_t = \eta_m \eta_p \]  \hspace{1cm} (9)

Using Eq. (9), the overall efficiency map was established by combining the axes from efficiency maps of the servo system (amplifier and motor) and hydraulic piston pump, as shown in Figure 4. The coordinate axes of the servo system efficiency map (torque \( T_m \) and speed \( n_m \)) were converted to pressure differential \( \Delta p \) and flow rate \( Q \) using Eqs. (10) and (11), respectively, for a given displacement ratio value. In this stage, \( \eta_m \) and \( \eta_p \) were assumed as not affected by the values of \( Q \) and \( \Delta p \), respectively. In the next stage, these efficiencies were determined by experimental data corresponding to a given working point \( (\Delta p, Q) \). The overall efficiency maps for typical displacement values are shown in Figure 5.

\[ \Delta p = \frac{2\pi T_m \eta_m}{D_{mn} \alpha} \]  \hspace{1cm} (10)

\[ Q = n_m D_{mn} \alpha \eta_p \]  \hspace{1cm} (11)

![Fig 4 Combination of individual efficiency maps](image)

To analyze the overall efficiency maps in Figure 5, the working point with \( Q = 5 \) L/min, \( \Delta p = 10 \) MPa was taken as an example. The overall efficiency of EHDSs with different ratios \( \alpha \) is shown in Table 3.

| Displacement ratio \( \alpha \) | 0.25 | 0.5 | 0.75 | 1 |
|-------------------------------|------|-----|------|---|
| Overall efficiency \( \eta_t \) | 0.676 | 0.754 | 0.698 | 0.63 |
| Motor speed \( n_m \) rpm     | 1265 | 632.5 | 421.6 | 316.2 |
| Motor torque \( T_m \) Nm     | 6.745 | 13.23 | 19.72 | 26.2 |

Table 3 Overall efficiency of the EHDS for the given working point with \( Q = 5 \) L/min, \( \Delta p = 10 \) MPa

It can be seen that the overall efficiency achieves the highest value at \( \alpha = 0.5 \), although, in this case, the efficiencies of neither the servo motor nor the hydraulic pump are maximum values. The efficiency of pump with \( \alpha = 1 \) was higher than that with \( \alpha = 0.5 \); however, motor speed \( n_m \) and motor efficiency decreased. In this study, if the efficiency of the servo motor and hydraulic pump are taken into account separately, the system will not reach the highest efficiency.
To improve the overall efficiency of the hydraulic power unit compared with the conventional one, the control system in Figure 6 was proposed to control the motor speed $n_m$ and displacement ratio $\alpha$ simultaneously. The target of this system is to optimize overall efficiency by regulating $n_m$ and $\alpha$ for specified working conditions ($\Delta p$, $Q$) based on the data of the overall efficiency maps. In other words, for a given value of $\Delta p$, $Q$ the controller will determine the value of efficiency in every single overall efficiency map ($\alpha = \alpha_i$) and then compare these values while still considering the given constraints in Section 3.3. The working parameters ($n_m$, $\alpha$) were determined when the map was at its highest efficiency for the corresponding working point. Using overall efficiency maps, the EHDS could be controlled at this operating point. Moreover, the flexibility of $n_m$ and $\alpha$ could help the system operate within its permitted working range (motor speed and torque, hydraulic pump speed and pressure) and thus avoid overloading in the servo motor and hydraulic pump.

Fig. 6 Control strategy for improving overall efficiency

Fig. 5 Typical overall efficiency maps of the EHDS
5. Numerical simulation of the proposed control strategy

To evaluate the advantages of the proposed control strategy, numerical simulation was implemented for a series of working points using the simulation power unit detailed in Table 2. The efficiency of the working points for three cases were determined and compared:

Case 1: (VS-FP) ($\alpha = \alpha_{\text{max}}$)

Case 2: (FS-VP) ($n_n = n_{\text{rated}}$)

Case 3: (VS-VP)

Normally, in an EHDS, an induction motor is used as the constant rotation speed motor. The efficiency of the induction motor is inferior to that of the servo motor with a permanent magnet over the entire operating range. However, this study does not compare efficiencies due to differences in motor types, so a servo motor operating at its rated speed serves as the constant rotation speed motor (Case 2). The simulation results regarding flow control mode and pressure control mode are shown in Figures 7 and 8, respectively.

In the flow control model (Figure 7), the VS-VP unit achieved better overall efficiency than others at almost all given working points because of the flexibility of both $n_n$ and $\alpha$. The advantage of the VS-VP unit can be seen clearly at a low flow rate range ($Q = 5\sim12$ L/min). Considering the work point of $Q = 5$ L/min; $\Delta p = 24$ MPa, the efficiency of the VS-VP unit is 7% and 4% higher than that of the VS-FP unit and FS-VP unit, respectively. These differences were up to 6% and 25%, respectively, in the case of working point with $Q = 5$ L/min; $\Delta p = 4$ MPa. When comparing the efficiency of VS-VP unit with VS-FP unit in the high pressure and low flow operation with $Q \leq 9$ L/min; $\Delta p \geq 12.5$ MPa, the variance in overall efficiency of the VS-VP and FS-VP units is up to 7%. This could be explained by the decrease of motor efficiency at low speed, although the hydraulic pump of the VS-FP unit operates at maximum displacement (i.e., maximum pump efficiency).

In case of FS-VP unit, even though servo motor operates at a rated speed (i.e., high motor efficiency), its overall efficiency is significantly lower than that of the VS-VP unit due to the change of ratio $\alpha$, which exhibits a noticeable effect on the efficiency of the hydraulic pump. The overall efficiencies of all units are nearly identical at a large flow rate range since the hydraulic pump and servo motor must operate at maximum ratio $\alpha = 1$ and rated speed, respectively, to achieve the necessary flow rate $Q$.

Fig. 7 Overall efficiency $\eta$ comparison at pressure differential $\Delta p$ at 24, 18, 12.5, 6.3, and 4 MPa
In the pressure control model (Figure 8), the VS-VP unit is still the most efficient system at almost all required power points because of the flexibility of both \( n_m \) and \( \alpha \). At a low flow rate operation, VS-VP offers better \( \eta \) for both the low and high pressure mode because both \( n_m \) and \( \alpha \) are changed. The range of efficiency in the VS-VP unit remains 0.7–0.8 and, unlike in the FS-VP unit, does not decline when the pressure differential decreases. As shown in Figure 8, the FS-VP unit is the least efficient unit since the pump often works at low displacement. However, at a low flow rate range (\( Q = 5–9 \) L/min), the overall efficiency of all units decrease significantly when the pressure differential \( \Delta p \) increases. This could be explained by the increase of hydraulic oil leakage in the hydraulic pump at the high pressure.

6. Conclusion

In this paper, a novel control strategy was proposed to improve the overall efficiency of an EHDS by considering the efficiency of various EHDS components and the influence of these components on overall efficiency. Based on the knowledge of overall efficiency, the optimized values of the working parameters (motor speed and pump displacement ratio) can be determined to achieve the highest efficiency for a given working point. Next, the numerical simulation was implemented for a series of operating points. The results show that the efficiency of the proposed control strategy was improved significantly compared with conventional single-variable power units. This paper illustrates the concept of high-efficiency power transmission and the usefulness of two-degree-of-freedom control. To verify the results of this research, future work will evaluate the results via machine measurement.

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