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A Novel Performance Model Given by the Physical Dimensions of Hydraulic Axial Piston Motors : Experimental Analysis

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Abstract

Huhtala (1997) discussed the strength and weakness of some existing models for hydraulic piston machines and unfortunately concluded that any model is not accurate enough for wide operating ranges. As an effort to develop an accurate model applicable for wide operating range, Jeong (2006) proposed a performance coefficient model for a hydraulic axial piston motor of swash plate design, which is given by the physical dimensions. In this paper, validity of the new performance model is addressed through experimental analysis. Estimation process for model parameters is established and then *performance decision parameters* are estimated from efficiency measurement data for a test motor. As the result, maximum and average efficiency estimation error over wide operating ranges of speed 300~2700 RPM and pressure 20~300 bar are just 2.33% and within 0.30%, respectively. Hence, it can be said that this paper opens a door toward developing true performance estimation model.

Keywords: Leakage flow loss; Hydro-mechanical loss; Efficiency; Performance characteristic coefficient; Performance decision parameter; Performance estimation model; Hydraulic piston motor

1. Introduction

Hydraulic units not only have a feature of high power density per unit weight and transmissibility of high powers, but also can realize rectilinear and circular motion easily. So it has been used in various industrial fields. In hydraulic units, pumps generating hydraulic power source and motors transforming hydraulic power into mechanical power are the essential components ruling the overall performance of a hydraulic system. Meanwhile, hydraulic systems are generally operated at certain designated speed and torque or pressure condition. Hence, the efficiency of hydraulic units especially at the operating point is one of the most important factor among many performance indices.

Since Wilson (1948) had firstly developed steady-state flow rate and moment loss model, several authors had tried to make precise performance model for hydraulic piston machines. Existing models may be grouped into two types. One of them is a kind of polynomial model represented with independent variables such as pressure difference, speed and displacement ratio. The other is a sort of performance coefficient model represented based on the physical nature of losses which is generally given by operating variables such as pressure difference and speed.

As for the polynomial model, all coefficients of the model are to be determined from measured experiment data by using curve fitting technique. And characteristic coefficients of the so-called performance coefficient model are also to be obtained through experiments. Hence, existing models are said

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to be models for indicating and/or calculating performance of an already manufactured and tested machine at certain operating points, rather than models for analyzing and evaluating the performance of a designed machine before test or designing machine in advance.

Some typical existing models are well summarized in Huhtala (1997), where weakness and strength of each model are discussed and compared with measurement data, concluding unfortunately that any model is not accurate enough for wide speed and pressure range of 500~3000 rpm and 20~210 bar. The reason originates primarily from the complex mechanism having rotating and reciprocating parts with friction working in viscous fluid of hydraulic oil, the physical properties of which is also dependent on the operating conditions.

As an effort to develop an accurate model applicable for wide operating range, Jeong (2006) proposed a performance coefficient model through theoretical analysis for a hydraulic axial piston motor of swash plate design with rotating cylinder block. In this paper, analysis and usage of the new model comparing calculated efficiency with measurement data are discussed. This paper is organized as follows: section 2 describes the new performance model and related oil properties and performance decision parameters and estimation process are established in section 3. In section 4, experimental analysis and estimation results are discussed and conclusions are followed in the last section.

2. New performance model for axial piston motors

A new performance model proposed by Jeong (2006) for a hydraulic axial piston motor of swash plate type with rotating cylinder block, shown in Fig. 1 is briefly described in this section. Flow rate to be delivered into a motor $\overline{Q_1}$ which equals the flow rate $\overline{Q_{vP}}$ induced by piston motion plus the sum of all leakages $\overline{Q_L}$ and output moment generated by a motor $\overline{M_2}$ which is the moment $\overline{M_{pP}}$ produced by the pressure force minus the sum of all loss moments $\overline{M_L}$ can be given as

$$\overline{Q_1} = \overline{Q_{vP}} + \overline{Q_L} = V_g n + \overline{Q_L} \tag{1}$$

$$\overline{M_2} = \overline{M_{pP}} - \overline{M_L} = V_g \Delta p / 2\pi - \overline{M_L} \tag{2}$$

And leakage $\overline{Q_L}$ and moment loss $\overline{M_L}$ derived through theoretical analysis on three facing gaps and other non-negligible sources in a motor are respectively repeated here.

$$\begin{aligned}\overline{Q_L} &= (\overline{Q_{LP}} + \overline{Q_{LS}} + \overline{Q_{LV}}) + Q_{LVN} + Q_{LPP} + Q_{Lc} + Q_{Lo} \\ &= (C_{\mu P} + C_{\mu S} + C_{\mu V})\frac{\Delta p}{\mu} + (C_{vP} + C_{vV})n \\ &\quad + C_{VN}\sqrt{\frac{\Delta p}{\rho}} + C_{PP}\frac{\rho n^3}{\Delta p} + C_c \frac{n \Delta p}{\beta} + Q_{Lo}\end{aligned} \tag{3}$$

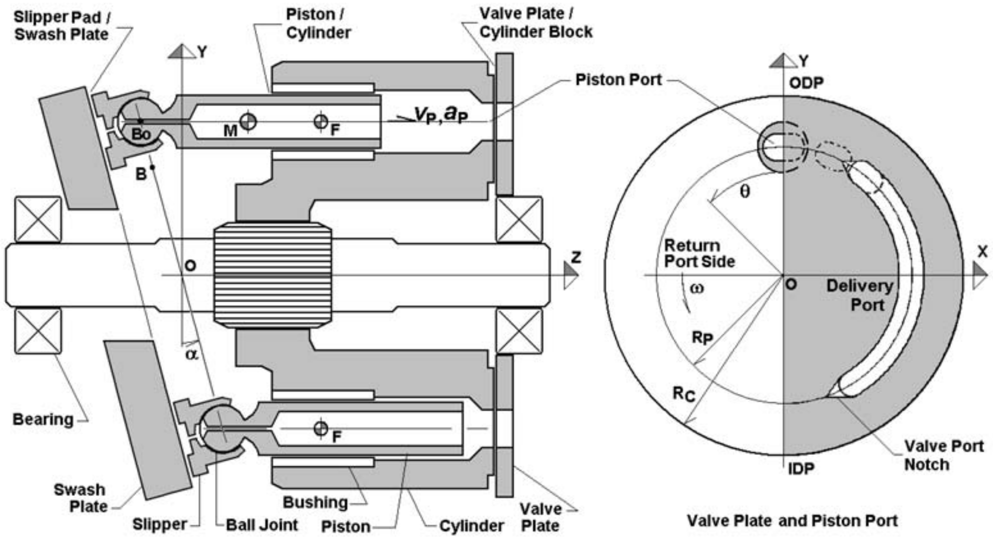

Fig. 1. Hydraulic axial piston machine of swash plate design.

$$
\begin{aligned}
&= C_{\mu PSV}\frac{\Delta p}{\mu} + C_{vPV}n + C_{VN}\sqrt{\frac{\Delta p}{\rho}} \\
&\quad + C_{PP}\frac{\rho n^3}{\Delta p} + C_c\frac{n\Delta p}{\beta} + Q_{Lo} \\
&\equiv C_\mu \Delta p + C_v n + C_{\rho VN}\sqrt{\Delta p} + C_{\rho PP}\frac{n^3}{\Delta p} + C_{\beta c} n\Delta p + Q_{Lo}
\end{aligned}
$$

$$
\begin{aligned}
\overline{M_L} &= M_{LP} + (\overline{M_{LS}} + M_{LV} + M_{LchCH} + M_{L\mu Br}) \\
&\quad + M_{LpBr} + (M_{LPP} + M_{LchSP}) \\
&\quad + M_{LVN} + (\overline{M_{L\mu P}} + \overline{M_{LfPp}}) + M_{Lp2} \\
&\quad + (M_{LoBr} + (M_{LoAux}) \\
&= (K_{\mu Pv} + K_{\mu S} + K_{\mu V} + K_{\mu CH} + K_{\mu Br})\mu n \\
&\quad + (K_{\mu Pp} + K_{fPp} + K_{pBr})\Delta p \qquad (4) \\
&\quad + (K_{PP} + K_{SP})\rho n^2 + K_{VN}\frac{\Delta p}{n}\sqrt{\frac{\Delta p}{\rho}} \\
&\quad + K_{p2}\Delta p^2 + (M_{LoBr} + M_{LoAux}) \\
&\equiv K_\mu n + K_p \Delta p + K_{\rho PSP} n^2 + K_{\rho VN}\Delta p\sqrt{\Delta p}/n \\
&\quad + K_{p2}\Delta p^2 + M_{Lo}
\end{aligned}
$$

At the last lines of Eqs. (3) and (4), loss terms of same character are grouped together. Each coefficient of loss terms in the second lines of Eqs. (3) and (4) is defined by physical dimensions of a motor in sections 4 and 5 of Jeong (2006) and it is repeated in appendix of this paper. The last term Q_{Lo} in Eq. (3) is not a physical leakage but a term introduced for compensating error of flow-meters used in measurements (Ivantysyn, 2001). M_{Lp2} in Eq. (4) stands for the pressure dependent moment loss due to mixed and/ or boundary friction which may happen especially at extremely high pressure. And M_{LoAux} is a term accounting for the neglected constant moment losses due to pre-loading of seals and springs, etc. in addition to the measurement error of the sensor used.

Normally accepted definition of overall efficiency for a motor (section 3.2.6 in Ivantysyn, 2001) is the product of volumetric efficiency $\eta_v = nV_g/\overline{Q_1}$ and hydro-mechanical efficiency $\eta_{hm} = \overline{M_2}/(\Delta p V_g/2\pi)$

$$
\begin{aligned}
\eta_t = \eta_v \cdot \eta_{hm} &= \frac{nV_g}{\overline{Q_1}} \cdot \frac{2\pi\overline{M_2}}{\Delta p V_g} = \frac{\overline{M_2}\cdot w}{\overline{Q_1}\cdot \Delta p} \\
&= \frac{(\overline{M_{pP}} - \overline{M_L})\cdot w}{(\overline{Q_{vP}} + \overline{Q_L})\cdot \Delta p}
\end{aligned} \tag{5}
$$

Following the definition, new formula for the overall efficiency of a motor yields

$$
\eta_t = \frac{2\pi n}{\Delta p} \frac{V_g \Delta p/2\pi - (K_\mu n + K_p\Delta p + K_{\rho PSP}n^2 + K_{\rho VN}\Delta P\sqrt{\Delta p}/n + K_{p2}\Delta p^2 + M_{Lo})}{V_g n + (C_\mu \Delta p + C_v n + C_{\rho VN}\sqrt{\Delta p} + C_{\rho PP} n3/\Delta p + C_{\beta c} n\Delta p + Q_{Lo})} \tag{6}
$$

Note that each term of the efficiency model as well as leakage and moment loss models are represented by operating variables such as pressure difference Δp and rotating speed n of a motor. One important thing that should be mentioned is that each coefficient of the proposed model is given by the physical dimensions of a motor.

On the other hand, hydraulic oil used in a motor affects its efficiency. State of hydraulic oil can be characterized by three parameters such as density ρ, viscosity μ and bulk modulus β (pp.18 in Ivantysyn, 2001)

$$
\rho = \rho_o\{1 + \gamma_{\rho p}P - r_{\rho T}(T - T_o)\} \tag{7}
$$

$$
\mu = \mu_o e^{\gamma_{\mu p}p} e^{-\gamma_{\mu T}(T - T_o)} \tag{8}
$$

$$
\begin{aligned}
&\beta = \frac{\beta_o/(1+\gamma_{AF})}{1+\gamma_\beta/p^{\kappa+1/\kappa}} \equiv \frac{\beta_{oAF}}{\xi_\beta} \simeq \beta_{oAF}(1 - \gamma_\beta p^{-(\kappa+1)/\kappa}), \\
&\gamma_{AF} = \frac{V_a}{V_F},\ \gamma_\beta = \frac{\beta_o \gamma_{AF} P_a^{1/\kappa}}{\kappa(1+\gamma_{AF})}
\end{aligned} \tag{9}
$$

where ρ_o, μ_o, β_o stand for corresponding datum values at reference temperature T_o or pressure p_o. That is, properties of hydraulic oil itself are dependent on the operating conditions of a motor. However, the effect of hydraulic oil's property on the performance can be easily included into the model just by replacing the parameters ρ, μ, β with Eqs. (7)~(9).

Meanwhile, at very low operating speed viscous friction acting on a certain facing gap is known to show nonlinear behaviour, that is so-called Stribeck effect (Ivantysyn, 2001). In order to account for the effect on the performance model, corresponding viscosity factor μ in moment loss terms of Eq. (4) can be modified as a speed dependent factor. But, such effect is not considered in the following analysis.

3. Estimation of performance decision parameters from efficiency measurement data

In order to validate the new performance model by experiment, test equipments verifying every equations given in sections 4 and 5 of Jeong (2006) or leakage and moment loss components of every parts in a motor are to be specially designed and manufac-tured for their own purpose. And component-wise or equation-wise measurements and analysis on the measured data are to be carried out at various operating conditions. Some example works are Jang (1997), Jeong (2004) and Ivantysynova (2004).

However, such a direct approach requires huge amounts of times, works and costs. On the other hand, efficiency measuring facilities for hydraulic machines are already spread in many manufacturers or research institutes. Hence in this paper, we choose an indirect approach instead. By using a readily provided test facility and measuring data for a specific hydraulic motor, some parameters for the motor will be estimated by comparison of the measured efficiency and calculated data. And analysis on the efficiency estimation error over wide operating range will be tried in the followings.

3.1 Remarks on the efficiency measurement

In order to obtain one efficiency data, physical variables of n, $\overline{M_2}, (\overline{Q_1}), \overline{Q_2}, \overline{Q_L}, \Delta p (= p_1 - p_2)$ are to be measured at a certain operating condition, which is set by changing the delivered flow $\overline{Q_1}$ and load $\overline{M_2}$ with some appropriate means. And by adjusting the operating condition and measuring above mentioned variables consecutively, volumetric, hydro-mechanical and overall efficiency over the operating range of a motor can be acquired.

During the measurement process, some noteworthy facts should be considered. First of all, indicated or derived displacement V_i for the test motor is to be determined. It is defined as the gradient of output flow rate with respect to the motor speed, $\partial \overline{Q_2} / \partial n$ at pressure difference Δp=0. Since maintaining such null pressure difference during the measurement needs special treatment, generally it is rather calculated from the data for efficiency measurement. But, care should be taken since incorrect V_i may yields volumetric efficiency over 100%. As for the leakage $\overline{Q_L}$, it should not be calculated by $\overline{Q_1} - \overline{Q_2}$ but measured by a separate high resolution and precision flow-meter. That is because inlet and outlet flow rates of a motor is too big compared with leakage and hence inlet and outlet flow-meters with large measuring capacity are not suitable for leakage measurement. As for the inlet flow rate $\overline{Q_1}$ in Eq. (5), one of measurement data $\overline{Q_1}$, $\overline{Q_2} + \overline{Q_L}$ or $n \cdot V_i + \overline{Q_L}$ can be used. Selection among the three data should be made based on the accuracy or reliability of the sensor involved for the measurement.

On the other hand, in order to display and analyze the performance of a motor it is convenient to measure efficiencies at equal-distance rectangular operating points over wide speed and torque or pressure ranges. However, setting such rectangular lattice type measuring point is not so easy, especially at low speed and high pressure ranges. Hence, instead of accurately adjusting the measuring points one can save effort by measuring data at nearby points and then calculating efficiencies at the desired lattice points by using a 3 dimensional curve fitting technique.

3.2 Performance decision parameters

According to Eq. (6), new efficiency model consists of 6 leakage-related *performance characteristic coefficients* $C_\mu, C_v, C_{\rho VN}, C_{\rho PP}, C_{\beta c}, Q_{Lo}$ and 6 moment loss-related performance characteristic coefficients $K_\mu, K_p, K_{\rho PSP}, K_{\rho VN}, K_{p2}, M_{Lo}$. All of them except K_{p2}, Q_{Lo} and M_{Lo} are defined by the physical dimensions of a motor and property of hydraulic oil. As for the 12 performance characteristic coefficients, after investigating definition of each coefficient one can find that they contain some hard-to-measure soft parameters, naming *performance decision parameters*, as follows

$$\begin{aligned} &h_p, h_S, h_V, f_P, f_{pBr}, f_{vBr}, C_{dPP}, \\ &C_{dVN}, C_{wSP}, \beta_{oAF}, K_{p2}, Q_{Lo}, M_{Lo} \end{aligned} \tag{10}$$

And the property of hydraulic oil is defined by 9 parameters such as

$$\beta_{oAF}, \kappa, \gamma_\beta, \mu_o, \gamma_{\mu p}, \gamma_{\mu T}, \rho_o, \gamma_{\rho p}, \gamma_{\rho T} \tag{11}$$

Hence, it can be said that performance of a motor is entirely characterized by 13 performance decision parameters of the motor and 9 hydraulic oil parameters.

3.3 Estimation process for performance characteristic coefficients and performance decision parameters

By using the measured experiment data of a motor, data M_2, Q_1, Q_2, Q_L at the desired rectangular test points are to be calculated first by using 3D curve fitting technique. And with those con verted data,

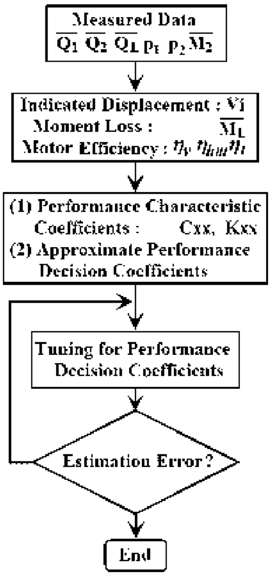

Fig. 2. Flow chart of estimation process.

indicated displacement Vi and moment loss $\overline{M}_L = V_i \Delta p/(2\pi) - \overline{M}_2$ are estimated and then volumetric, hydro-mechanical and overall efficiencies $\eta_v, \eta_{hm}, \eta_t$ are calculated according to their definitions. Calculation of 12 performance characteristic coefficients is the next step. When calculating the leakage-related and moment loss-related coefficients, data $\overline{Q}_L$ and $\overline{M}_L$ are to be used, respectively. Since many sets of data corresponding to the wide operating points are measured, one can get those 12 coefficients by applying least square method on Eqs. (3) and (4).

The third is a step for estimating 13 performance decision parameters of Eq. (10). However, we have 13 unknowns to be determined but there are only 12 relations corresponding to the definitions of 12 performance characteristic coefficients. Hence, the problem is indeterminate. But, with reasonable assumptions one can roughly get those parameters roughly in this step. Manual tuning to get accurate performance decision parameters is the final step to be done. New performance model is represented by the performance decision parameters in addition to the dimensional data of a motor. Since, in the 3rd step one already obtained approximate parameters and hence one can calculate efficiency corresponding to the approximate parameters. Hence, by comparing those calculated and measured efficiency data and evaluating efficiency estimation error, one can finally determine accurate value for each performance decision parameter.

Table 1. Principal dimensions of Rexroth A10VM.

Geometric displacement	V_g	45 cc/rev
tilting angle	α	18 degree
piston number	z	9 ea
piston diameter	d_P	17.0 mm
pitch circle radius	R_P	33.5 mm
piston length	l_P	17.0 mm
distance to cylinder	l_{Co}	17.4 mm
cylinder block radius	R_C	47.5 mm
cylinder block length	l_C	60.0 mm
slipper inner radius	r_{Si}	5.9 mm
slipper outer radius	r_{So}	10.7 mm
inner sealing ring	r_{V1}, r_{V2}	27.6/30.1 mm
outer sealing ring	r_{V3}, r_{V4}	36.9/39.5 mm

4. Experimental analysis and estimation results

As an experimental test target, a hydraulic swash plate type axial piston motor of Rexroth A10VM model with displacement 45 cc/rev without piston guide bushing is chosen, some principal dimensions of which are given in Table 1.

Among hydraulic parameters, magnitude of viscosity μ and bulk modulus β highly depends on the operating pressure as well as temperature, but density is known not to change so much over normal operating range. Hence, fluid density is assumed constant during the estimation process described in section 3.3. Also note that coefficient $C_{\beta c}$ reflecting the effect of fluid compressibility is the only one related with bulk modulus. So the fluid bulk modulus does not seem to affect much on the efficiency of a motor. Hence, the bulk modulus $\beta = \beta_{oAF} / \xi_\beta$ is simply assumed constant, representing the effective bulk modulus β_{oAF} of the fluid used in a motor.

As shown in Fig. 3, pressure, flow rate and moment data for the test motor are measured at 311 operating points of speed range 300~2700 RPM and pressure range of 20~300 bar. For the sake of post-processing, the data measured at operating points are converted into data at analysis points of rectangular lattice type, as shown in Fig. 3. With the converted flow data $\overline{Q}_2$, indicated displacement V_i of the motor is derived as shown in Fig. 4 and then with the converted output

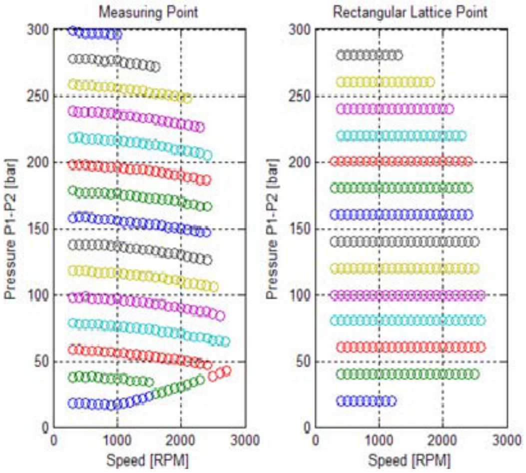

Fig. 3. Measured and rectangular operating points.

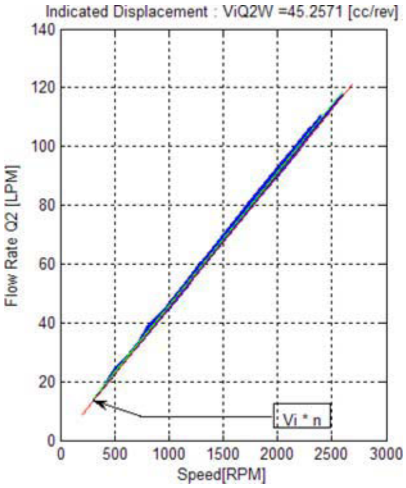

Fig. 4. Indicated displacement.

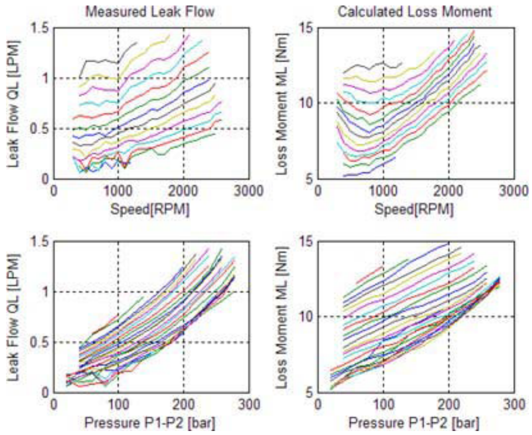

Fig. 5. Leak flow and loss moment data.

Table 2. Estimated performance coefficients and performance decision parameters (least square method).

C_{xx} coefficient	$C_{\mu PSV}$ 10^{-19}	$C_{\upsilon PV}$ 10^{-10}	C_{VN} 10^{-7}	C_{PP} 10^{-6}	C_c 10^{-13}	Q_{Lo} [LPM]
	0.1134	0.2669	0.8566	0.3265	0.1492	-0.3282
K_{xx} coefficient	$K_{\mu All}$	K_{Pp} 10^{-6}	K_{VN} 10^{-8}	K_{PSP} 10^{-5}	K_{p2} 10^{-15}	M_{Lo} [Nm]
	0.8540	0.1218	0.5965	0.3825	0.2740	4.5257
decision parameters	h_P [μm]	h_S [μm]	h_V [μm]	$f_{\upsilon Br}$	f_{pBr} 10^{-4}	f_P
	15.98	14.41	14.41	0.50	0.55	0.0180
C_{dPP}	C_{wSP}	C_{dVN}	β_{oAF} [bar]	K_{p2} 10^{-15}	Q_{Lo} [LPM]	M_{Lo} [Nm]
0.5554	0.8859	0.7495 /1.713	40793	0.2740	-0.328	4.5257

Table 3. Estimation result (least square method).

η_t : average estimation error	η_v : average estimation error	η_{hm} : average estimation error
0.2836 %	0.1754 %	0.2125 %
η_t : maximum estimation error	η_v : maximum estimation error	η_{hm} : maximum estimation error
1.7975 %	1.4432 %	1.7875 %

moment data $\overline{M_2}$, moment loss $V_i \Delta p/(2\pi) - \overline{M_2}$ is calculated as depicted in Fig. 5. With the measured leak flow data and estimated moment loss data, efficiencies data are calculated and depicted in Fig. 6, where efficiency data at measuring points are marked by ‘o’.

With those loss data, 6 leakage-related coefficients C_{xx} and 6 moment loss-related coefficients K_{xx} estimated by using least square technique with non-negativity constraints are given in Table 2. With the estimated performance characteristic coefficients and by using the relationships between performance characteristic coefficients and performance decision parameters, 13 performance decision parameters are derived and summarized in the lower two rows of Table 2.

Note that the relationships give two values for C_{dVN}. Also note that since the problem is indeterminate as mentioned in section 3.3, three gaps h_P, h_S, h_V in Table 2 are estimated under the assumption of f_{vBr}=0.50 and f_{pBr}=0.55×10^{-4}. Hence, manual tuning on performance decision parameters are carried out three times and selection minimizing average estimation error of overall and/or volumetric or mechanical efficiency is finally made as given in Table 4. corresponding performance characteristic coefficients C_{xx} and K_{xx} are given in the lower two rows of Table 4.

Note that the new performance model allows calculation of efficiency only with given the physical parameters. Hence, efficiencies of the motor can be estimated with those data of performance decision parameters in Table 4. And they are compared with measured efficiencies as shown in Figs. 7. In Fig. 8, error surface of estimated efficiency for the motor are displayed. And in Table 5 statistics for the estimation error are summarized.

Table 4. Estimated performance decision parameters and performance coefficients (after tuning).

decision parameters	h_P [μm]	h_S [μm]	h_V [μm]	f_{vBr}	f_{pBr} 10^{-4}	f_P
	11.36	14.23	13.92	0.292	0.344	0.0175
C_{dPP}	C_{wSP}	C_{dVN}	β_{oAF} [bar]	K_{p2} 10^{-15}	Q_{Lo} [LPM]	M_{Lo} [Nm]
0.784	0.672	0.786	47200	0.0401	-0.330	4.1238
C_{xx} coefficient	$C_{\mu PSV}$ 10^{-13}	C_{vPV} 10^{-7}	C_{VN} 10^{-7}	C_{PP} 10^{-6}	C_c 10^{-13}	Q_{Lo} [LPM]
	0.2042	0.9306	0.3930	0.1582	0.0673	-0.3301
K_{xx} coefficient	$K_{\mu All}$	K_{Pp} 10^{-6}	K_{VN} 10^{-8}	K_{PSP} 10^{-6}	K_{p2} 10^{-15}	M_{Lo} [Nm]
	2.3327	0.0985	0.6255	2.6452	0.0409	4.1238

Table 5. Estimation result (after tuning).

η_t : average estimation error	η_v : average estimation error	η_{hm} : average estimation error
0.2968 %	0.2034 %	0.2427 %
η_t : maximum estimation error	η_v : maximum estimation error	η_{hm} : maximum estimation error
2.3351 %	1.5929 %	2.1461 %

In Figs. 7 one may find that estimated efficiency curves match closely with measured curves over the whole operating range. From Table 5 one can also find that maximum estimation error and average estimation error for the efficiency is just 2.33% and within 0.30%, respectively. When recalling the fact that estimated efficiency is calculated by using the new performance model with 13 performance decision parameters in addition to the physical dimensional data of the motor and recalling the fact that efficiency estimation is done by using only one measurement data set over very wide operating speed and pressure range, it can be said that it is a quite amazing result.

Furthermore, performance decision parameters obtained in this analysis are within the ranges that are already known from many literatures. The heights of three facing gaps are normally in the range of 5~40 μm. The discharge coefficients are in the range of 0.5~1.0. The effective bulk modulus of oil is in the range of 1.9~3.5×10^{-4} bar (Ivantysyn, 2001). The friction coefficient of a piston is in the range of 0.02~0.3 (Ivantysyn, 2001). And the friction coefficients of bearings f_{vBr} and f_{pBr} are in the ranges of 0.2~0.6 and 0.2~2.0×10^{-4}, respectively (Jang, 1997). When considering that those ranges of above mentioned parameters are already proved in the literatures, it can be said that the experimental analysis result given in this section indirectly validates the newly proposed performance model.

One remaining problem to be clarified further is that the estimation error of volumetric efficiency in Fig. 8 shows a typical tendency with respect to the

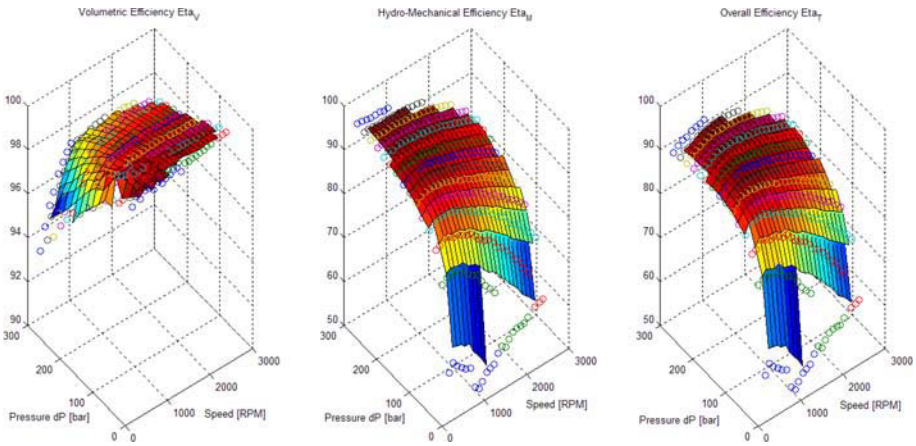

Fig. 6. Efficiencies data at measured and rectangular operating points.

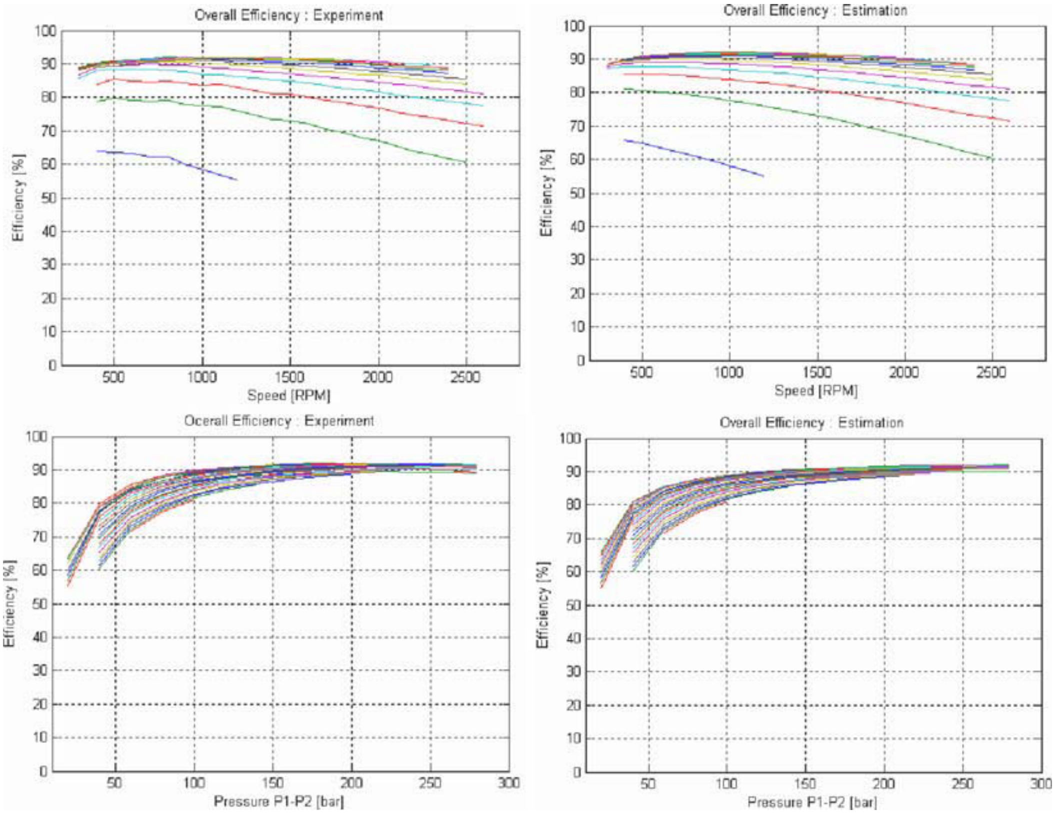

Fig. 7. Measured and estimated overall efficiency curves.

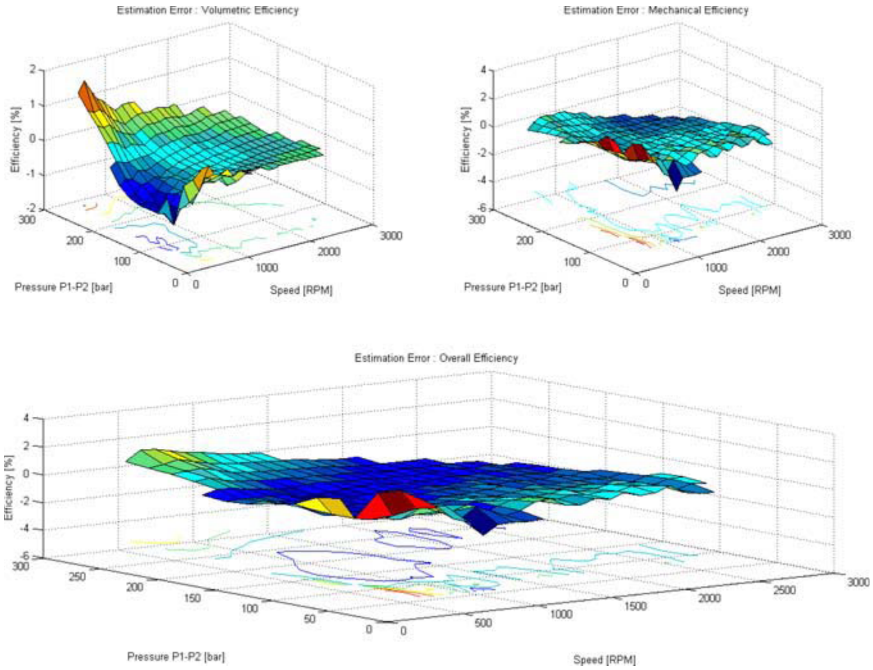

Fig. 8. Efficiency estimation error surfaces.

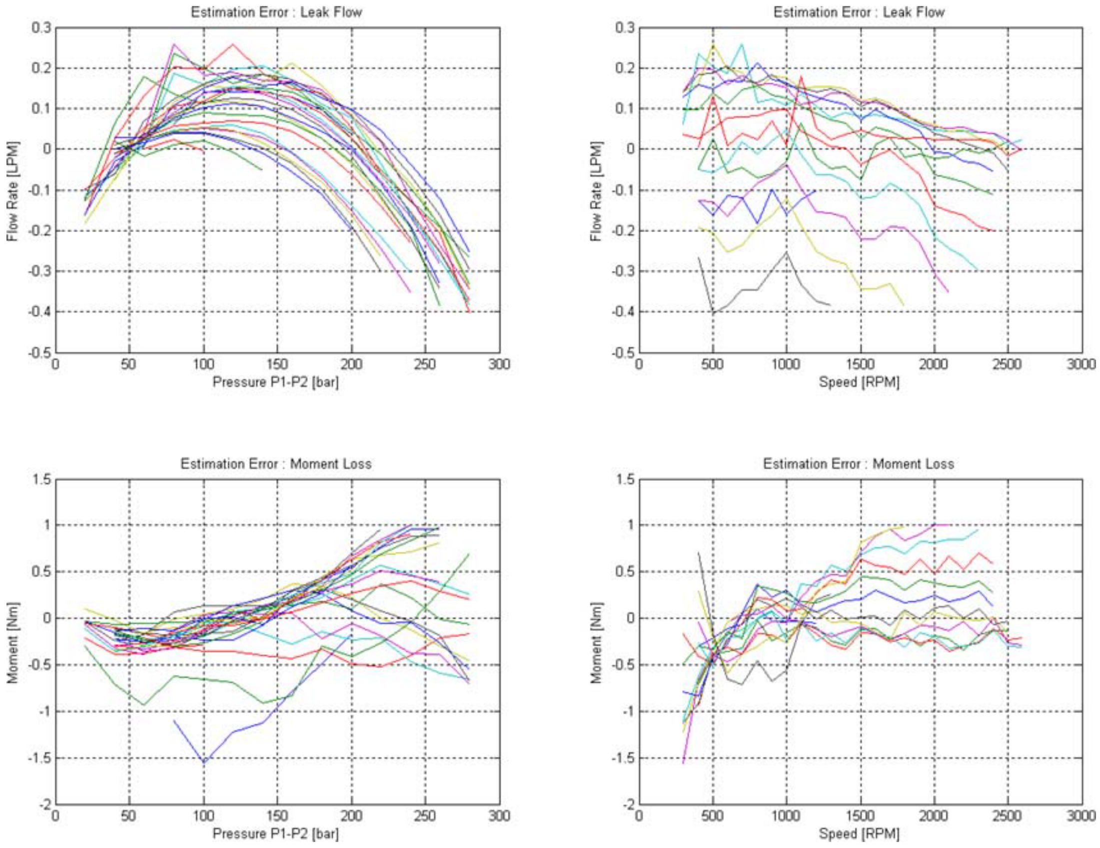

Fig. 9. Estimation error for leakage and moment loss.

variation of speed or pressure. Estimation errors for leakage and moment loss depicted in Fig. 9 show the phenomena more clearly. A remark to be made regarding this fact is that if we estimate $C_{\mu PSV}$ without non-negativity constraint it yields negative value. And it gives negligible random error with no tendency, like estimation error of moment loss in Fig. 9. But, physically negative coefficient does not make any sense. One possible explanation may originate from measure-ment error related with accuracy of the flow-meter for leakage. It needs further investigation.

5. Conclusions

As an effort to develop an accurate model applicable for wide operating range, Jeong (2006) proposed a performance coefficient model for a hydraulic axial piston motor of swash plate design, which is given by the physical dimensions. In this paper, experimental analysis is carried out for a test target motor of Rexroth A10VM model with displacement 45 cc/rev without piston guide bushing.

First of all, relationships between the physical dimensional data of a motor and performance characteristic coefficients are examined and performance decision parameters are clarified. And estimation process for such parameters from experimental data for measuring efficiency is established and then estimation is carried out. The experimental analysis gives that all of the performance decision parameters obtained through the estimation process are within the ranges already known from many literatures. Furthermore, maximum and average efficiency estimation error is just 2.33% and within 0.30%, respectively. Even though further investigation for the leakage estima-tion error is necessary, these facts indirectly validate the newly proposed performance model.

Recall that existing models are models for indicating and/or calculating the performance of an already manufactured and tested machine at certain operating points. For a model to be a certain model that can estimate the performance of a motor without test, so-called *performance estimation model*, two require-ments are to be satisfied. First of all, the model is to be described by the physical dimensional data of a machine and hydraulic oil involved in. Secondly, reliability and accuracy of the model over wide operating range should be verified. Possession of the performance estimation model means that it enables us to analyze and evaluate the performance of a designed machine before tests or a designing machine in advance. Moreover, it enables us to develop an efficient motor at a designated operating point, which is common requirement in industrial fields.

Note that all characteristic coefficients of the newly proposed performance model (Jeong 2006) is given by the physical dimensional data of a motor. And validity of the model is checked for a test case in this paper. Hence, it can be said that it has an eligibility to be called as a performance estimation model and that it opens a door toward developing true performance estimation model.

Nomenclature

C	:	Performance coefficient related with leakage
C_d	:	Discharge coefficient of an orifice
C_w	:	Drag coefficient of a cylindrical bar
f	:	Friction coefficient
h_P	:	Gap height between a piston and cylinder hole
h_S	:	Gap height between slipper and swash plate
h_V	:	Gap height between valve plate and cylinder
K	:	Performance coefficient related with moment
$\overline{M_{pP}}, \overline{M_L}$	:	Generated moment, moment loss
$\overline{M_1}, \overline{M_2}$	:	Input and output moment of a motor
n	:	Rotational speed of a motor in Hz,
p, p_e	:	Pressure in piston chamber, motor enclosure
p_1, p_2	:	Inlet and outlet pressure of a motor
Δp	:	Pressure difference, $p_1 - p_2$ or $p_1 - p_e$
$\overline{Q_1}, \overline{Q_2}$	:	Inlet and outlet flow of a motor
$\overline{Q_{vP}}, \overline{Q_L}$	:	Induced flow, leak flow at several part
V_g, V_i	:	Geometric and indicated displacement
α	:	Tilting angle of the swash plate
$\beta, \beta_o, \beta_{oAF}$	:	Bulk modulus of hydraulic oil
ρ, ρ_o	:	Density of hydraulic oil
η_{hm}	:	Hydro-mechanical efficiency of a motor
η_t	:	Overall efficiency of a motor
η_v	:	Volumetric efficiency of a motor
λ_P	:	Equivalent friction coefficient of a piston
μ, μ_o	:	Viscosity of hydraulic oil
w	:	Rotational speed of a motor in rad/sec,

Subscript

Br	:	Support bearing
C	:	Fluid compressibility
ch	:	Churning, swirl
CH	:	Cylinder and motor housing
fP	:	Friction force acting on a piston
o	:	Miscellaneous
p, *p*2	:	Pressure dependent term
P	:	Piston
pBr	:	Pressure acting on bearings
PP	:	Piston port
PSP	:	Piston port, slipper and piston
S	:	Slipper pad or swash plate
SP	:	Slipper pad, slipper and piston
V	:	Valve plate
VN	:	Valve port control notch
vBr	:	Velocity of bearings
v	:	Velocity dependent term
μP	:	Viscosity in a piston
μ	:	Viscosity dependent term
ρ	:	Density dependent term

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Appendix

As for the derivation of each term in Eqs (3) and (4) and meaning of the notations in this appendix, refer to sections 4 and 5 of Jeong (2006). Here, definition of each term used in performance characteristic coefficients are summarized. Note that equivalent friction coefficient $\overline{\lambda_P}$ in Eqs. (A.16) and (A.17) is a function of piston friction f_P.

$$C_{\mu P} \equiv \begin{Bmatrix} \dfrac{z}{2}\cdot\dfrac{\pi d_P}{12 l_{Fo}}\cdot h_P^3 & \text{with bushing} \\ \dfrac{z}{2\sqrt{a^2-b^2}}\cdot\dfrac{\pi d_P}{12}\cdot h_P^3 & \text{w/o bushing} \end{Bmatrix}, \tag{A.1}$$

$$\begin{cases} a = l_{Fo} + R_P\tan\alpha \\ b = R_P\tan\alpha \end{cases}$$

$$C_{\mu S} \equiv \begin{Bmatrix} \dfrac{z}{2}\cdot\dfrac{\pi d_B^4}{\left\{6 d_B^4 \ln\dfrac{r_{So}}{r_{Si}} + 128 h_S^3 l_B\right\}}\cdot h_S^3 & \text{capillary type} \\ \dfrac{z}{2}\cdot\dfrac{\pi}{6\ln r_{So}/r_{Si}}\cdot h_S^3 & \text{orifice type} \end{Bmatrix} \tag{A.2}$$

$$C_{\mu V} \equiv \frac{z}{2}\cdot\frac{\lambda_V}{12 l_V}\cdot h_S^3 \tag{A.3}$$

$$C_{vP} \equiv \frac{z}{\pi}\cdot\pi^2 d_P\cdot R_P\tan\alpha\cdot h_P \tag{A.4}$$

$$C_{vV} \equiv \frac{z}{2}\cdot\pi(b_{V1}+b_{V2})\cdot R_P\cdot h_V \tag{A.5}$$

$$C_{VN} \equiv C_{dVN}\cdot\frac{z\Delta\theta_{VN}A_{VN}}{2\pi} \tag{A.6}$$

$$C_{PP} \equiv \frac{V_g^3}{z^2 A_o^2}\cdot\frac{1}{C_{dPP}^2} \tag{A.7}$$

$$C_C \equiv z\cdot V_{P\max}\cdot\frac{1}{\beta_{oAF}} \tag{A.8}$$

$$K_{\mu All} \equiv (K_{\mu Pv} + K_{\mu S} + K_{\mu V} + K_{\mu CH} + K_{\mu Br}) \tag{A.9}$$

$$K_{\mu Pv} \equiv \begin{Bmatrix} \dfrac{z}{2}\cdot 2\pi^2 d_P (R_P\tan\alpha)^2 l_{Fo}\cdot\dfrac{1}{h_P} \\ \text{with bushing} \\ \dfrac{z}{2}\cdot 2\pi^2 d_P (R_P\tan\alpha)^2 (l_{Fo} + R_P\tan\alpha)\cdot\dfrac{1}{h_P} \\ \text{w/o bushing} \end{Bmatrix} \tag{A.10}$$

$$K_{\mu S} \equiv z\cdot 2\pi^2 R_P^2 (r_{So}^2 - r_{Si}^2)\cdot\frac{1}{h_S} \tag{A.11}$$

$$K_{\mu V} \equiv \pi^2 (r_{V4}^4 - r_{V3}^4 + r_{V2}^4 - r_{V1}^4)\cdot\frac{1}{h_V} \tag{A.12}$$

$$K_{\mu CH} \equiv 2\pi^2 R_C l_C (R_H + R_C) \tag{A.13}$$

$$K_{\mu Br} \equiv f_{vBr}\cdot(60/\rho_o)^{2/3} d_{Br}^3 \tag{A.14}$$

$$K_{\mu Pp} \equiv \frac{z}{\pi}\cdot\frac{\pi d_P}{2} R_P\tan\alpha\cdot h_P \tag{A.15}$$

$$K_{fPp} \equiv \frac{z}{\pi}\cdot A_P R_P\tan\alpha\cdot\left|\overline{\lambda_P}\right| \tag{A.16}$$

$$K_{pBr} \equiv f_{pBr} \cdot \frac{z}{2} \cdot d_{Br} A_P \tan\alpha \cdot (1 + \overline{\lambda_P}) \tag{A.17}$$

$$K_{PP} \equiv \frac{V_g^3}{\pi z A_o^2} \cdot \frac{1}{C_{dPP}^2} \tag{A.18}$$

$$K_{SP} \equiv C_{wSP} \cdot z \cdot 2\pi^2 A_{SP} R_P^3 \tag{A.19}$$

$$K_{VN} \equiv C_{dVN} \cdot \frac{z \Delta\theta_{VN} A_{VN}}{(2\pi)^2} \tag{A.20}$$