Proceedings of the ASME/BATH 2019
Symposium on Fluid Power and Motion Control
FPMC2019
October 7-9, 2019, Sarasota, FL, USA

FPMC2019-1615

MEASURING THE LOSSES OF HYDROSTATIC PUMPS AND MOTORS – A CRITICAL REVIEW OF ISO4409:2007

Peter Achten
INNAS
Breda, the Netherlands

Robin Mommers
INNAS
Breda, the Netherlands

Takao Nishiumi
National Defense Academy
Yokosuka, Kanagawa, Japan

Hubertus Murrenhoff
ifas RWTH Aachen University
Aachen, Germany

Nariman Sepehri
University of Manitoba
Winnipeg, Manitoba, Canada

Kim Stelson
University of Minnesota
Minneapolis, MN, USA

Jan-Ove Palmberg
Linköping University
Linköping, Sweden

Katharina Schmitz
ifas RWTH Aachen University
Aachen, Germany

ABSTRACT

ISO 4409 is the most important international standard for measuring the efficiency of hydraulic pumps and motors, the latest edition being 4409:2007. The standard describes methods for determining the steady-state performance in terms of overall efficiency. It also defines equations for calculating the volumetric efficiency of pumps and motors. The hydro-mechanical efficiency is only defined for motors, not for pumps.

This paper analyses the efficiency and losses of pumps and motors in an alternative way. The preference is on loss analysis instead of efficiencies. Especially the effects of the bulk modulus are considered in a different and more inclusive manner. The new methodology results in a higher total loss for motor and a lower total loss for pumps than the current ISO 4409 standard. Furthermore, it results in significant changes of the hydro-mechanical and volumetric losses. The differences between the new methodology and ISO 4409 become larger for high load pressures.

The new methodology demands knowledge about the minimum volume of the displacement chamber. The ratio between this volume and the full displacement of a single displacement chamber strongly influences the hydro-mechanical and volumetric losses of the pump or motor. The new methodology is valid for all positive displacement hydrostatic pumps and motors. The volumetric efficiency, as defined in ISO 4409, can still be used as a flow rate factor, but should not be regarded as an energy conversion efficiency.

The importance of adopting the proposed methodology is further demonstrated by analyzing and comparing the measurement data about a fixed displacement pump and motor, showing the differences in the loss analysis by means of ISO 4409 and the new equations.

The methodology, observations and validation results presented in this paper are significant and can pave the road for improving the current ISO 4409:2007 standard, which would ultimately benefit the industry.

INTRODUCTION

Measurements always contain errors. Aside from human errors, measurement errors can be divided into two categories:

- Instrumental errors, due to instrumental accuracy;
- Methodological errors, due to imperfections in testing procedures and definitions.

This paper reviews standard ISO 4409:2007 [1] in terms of methodological errors. We argue that it is important that the standard prepared many years ago, to be revisited in terms of analysis of hydrostatic pumps and motors. In addition, ISO 4409 is not clear in the implementation of the test procedure, especially about the determination of the volumetric flow at different pressure levels, the way to handle external leakage flows, or what to do when a pump or motor does not have an external drainage. These additional topics, however, will not be discussed in this paper.

ISO 4409 gives equations for determining the volumetric efficiency for pumps and motors. The hydro-mechanical efficiency is only defined for motors. As will be shown in this paper, these equations may not be accurate enough for todays' applications. There are no definitions in ISO 4409 for the hydro-mechanical efficiency of pumps.

As will be discussed in this paper, for the purpose of energy conversion analysis, the concept of loss segregation by means of hydro-mechanical and volumetric efficiencies should be replaced by a segregation on the basis of power losses. The definitions and equations for the volumetric and hydro-mechanical efficiencies, which are defined in ISO 4409 (and also in ISO 4392 [2-4]) cannot be regarded as energy conversion efficiencies. Yet, for pump and motor sizing, the volumetric efficiency of ISO 4409 can still be used as a flow rate parameter.

In this paper, new definitions for the losses and (if relevant) efficiencies of hydrostatic pumps and motors will be derived on the basis of thermodynamic analysis. Not only the overall or total efficiency needs to be redefined, especially for operation at higher pressure levels, but also the hydro-mechanical efficiency. We also argue that the current definition of volumetric efficiency can be misleading, and that, at this moment, there is not a clear way to define this efficiency as an energy conversion efficiency.

The nomenclature in this paper is slightly different from ISO 4409. With reference to Figure 1:

- subscript 1 always refers to the low pressure side of the main hydraulic circuit
- subscript 2 always refers to the high pressure side of the main hydraulic circuit
- subscript 3 refers to the drain or external leakage port of the pump or motor

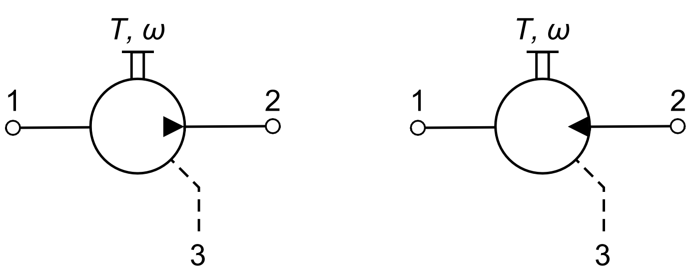


Fig. 1: Numbering of the connections to a pump (left) and motor (right)

Furthermore (as in ISO 4409), superscript P indicates that the corresponding parameter or variable is pump related; superscript M indicates that the parameter or variable is motor related.

ENERGY BALANCE

The following general assumptions are made for the energy balance of a pump or motor:

- There is no gravitational potential energy difference between inlet and exhaust;
- The kinetic energy of the flow at the inlet equals the kinetic energy of the flow at the outlet;
- The heat losses of the pump or motor can be neglected.

With these assumptions, the amount of hydraulic power being produced (by a pump) or consumed (by a motor) equals:

$$\begin{aligned} P_{hydr} &= \dot{m}_2 h_2 - \dot{m}_1 h_1 = \\ &= Q_2\rho_2\left(u_2 + \frac{p_2}{\rho_2}\right) - Q_1\rho_1\left(u_1 + \frac{p_1}{\rho_1}\right) = \\ &= \left(Q_2\rho_2 u_2 - Q_1\rho_1 u_1\right) + \left(p_2 Q_2 - p_1 Q_1\right) \end{aligned} \qquad (1)$$

In the above equations:

$\dot{m}$ = mass flow [kg/s]
h = specific enthalpy [J/kg]
u = specific internal energy [J/kg]
ρ = density of the oil [kg/m³]
p = pressure level [Pa]
Q = flow rate [m³/s]
P_{hydr} = hydraulic power [W]

The second part of this equation

$$\left(p_2 Q_2 - p_1 Q_1\right) \qquad (2)$$

is the basis for the efficiency definitions of ISO 4409. However, the first part of Eq. (1)

$$\left(Q_2 \rho_2 u_2 - Q_1 \rho_1 u_1\right) \qquad (3)$$

which describes the change of the internal energy of the fluid, is neglected in the methodology of ISO 4409. Yet, any pump needs to fulfil two functions:

- Oil must be compressed from a low pressure level to a higher pressure level;
- Oil needs to be displaced to the high pressure line.

As such, pumps could be regarded as two machines connected in series: one to perform the compression of the hydraulic fluid, and one for the actual fluid displacement to the high-pressure line. Eq. (1) can be regarded as a thermodynamic representation of these two functions. It is not sufficient to calculate the hydraulic power only on the basis of Eq. (2). Similarly, the process in a motor can be regarded as a high-pressure stroke, followed by an expansion of the oil volume in the displacement chamber.

The specific internal energy can be calculated assuming an isentropic compression and expansion.

$$
\begin{aligned}
\mathrm{d}u &= -p\,\mathrm{d}v = \\
&= -p\left[\left(\frac{\partial v}{\partial s}\right)_p \mathrm{d}s + \left(\frac{\partial v}{\partial p}\right)_s \mathrm{d}p\right] = \\
&= -p\left[0 + \left(-\frac{v}{\bar{K}_s}\mathrm{d}p\right)\right] = \\
&= \frac{v\,p}{\bar{K}_s}\mathrm{d}p = \\
&= \frac{p}{\rho\,\bar{K}_s}\mathrm{d}p
\end{aligned}
\tag{4}
$$

in which $\bar{K}_s$ is the average isentropic bulk modulus of the oil. Integrating this equation gives:

$$
u - u_0 = \frac{1}{\rho \bar{K}_s}\int_{p_0}^{p} p\ \mathrm{d}p = \frac{1}{2\rho \bar{K}_s}\left(p^2 - p_0^2\right) \tag{5}
$$

The hydraulic power defined by Eq. (1) can than be rewritten as:

$$
P_{hydr} = \left(1 + \frac{p_2}{2\bar{K}_s}\right) p_2 Q_2 - \left(1 + \frac{p_1}{2\bar{K}_s}\right) p_1 Q_1 \tag{6}
$$

The correction for the oil compressibility is only significant for relative high pressure levels, and can be neglected for the supply pressure p_1. The hydraulic power can therefore be approximated as:

$$
P_{hydr} \simeq \left(1 + \frac{p_2}{2\bar{K}_s}\right) p_2 Q_2 - p_1 Q_1 \tag{7}
$$

DEFINITION OF THE IDEAL CYCLE

Hydrostatic pumps and motors are positive displacement devices, having a low-pressure stroke, a high-pressure stroke and, in between these strokes, the commutation from one pressure level to the other.

Although ISO 4409 does not define an ideal cycle as such, it can be concluded from the equations used, that the ideal cycle is defined as a pure rectangle in the pV-diagram, i.e. the thermodynamic process in a single displacement chamber of a pump or motor (see Fig. 2).

Following the equations from ISO 4409, the indicated work E_i of the ideal cycle equals the product of Δp and ΔV. The cycle can be used to calculate the theoretical torque of the lossless ideal cycle:

$$
T_{th} = \frac{z\,E_i}{2\pi} \tag{8}
$$

where z is the number of displacement chambers. For ISO 4409, this can be redefined as:

$$
T_{th} = \frac{\Delta p\, z\, \Delta V}{2\pi} = \frac{\Delta p\, V_g}{2\pi} \tag{9}
$$

In ISO 4409 (equations (12) and (A.6) of the standard), the hydro-mechanical efficiency of motors is defined as:

$$
\eta_{hm}^{M} = \frac{T}{T_{th}} = \frac{2\pi\, n\, T}{\Delta p\, V_g} \tag{10}
$$

The parameter n in the numerator is clearly a mistake. The equation should be:

$$
\eta_{hm}^{M} = \frac{T}{T_{th}} = \frac{2\pi\, \cancel{n}\, T}{\Delta p\, V_g} = \frac{2\pi\, T}{\Delta p\, V_g} \tag{11}
$$

Equation (11) can be used to calculate the theoretical torque:

$$
T_{th} = \frac{\Delta p\, V_g}{2\pi} \tag{12}
$$

This equation is equal to the previous definition in equation (9).

ISO 4409 assumes an instantaneous commutation in the top and bottom dead centres. The oil is considered to be incompressible, i.e., to have an infinite bulk modulus. However, elsewhere in ISO 4409, the oil is considered to be compressible.

In order to be consistent, the compressibility of the oil should also be included in the definition of the ideal cycle. In that case, the ideal cycle would follow the grey coloured dashed line in the *pV*-diagram (see Fig. 2). Compared to the original rectangular cycle, the indicated work would then be reduced by the two hatched areas.

$$
\begin{aligned}
E_i &= \Delta p\, \Delta V - \frac{\Delta p^2\, \Delta V}{\bar{K}_s}\left(\frac{\tfrac{1}{2}\Delta V + V_{\min}}{\Delta V}\right) = \\
&= \Delta p\, \Delta V\left[1 - \frac{\Delta p}{\bar{K}_s}\left(\frac{1}{2} + \frac{V_{\min}}{\Delta V}\right)\right]
\end{aligned}
\tag{13}
$$

The theoretical torque of this more realistic cycle thus becomes:

$$
T_{th} = \frac{z\,E_i}{2\pi} = \frac{\Delta p\, V_g}{2\pi}\left[1 - \frac{\Delta p}{\bar{K}_s}\left(\frac{1}{2} + \frac{V_{\min}}{\Delta V}\right)\right] \tag{14}
$$

Defining correction factor a_1:

$$
a_1 = 1 - \frac{\Delta p}{\bar{K}_s}\left(\frac{1}{2} + \frac{V_{\min}}{\Delta V}\right) \tag{15}
$$

equation (14) can be rewritten as:

$$
T_{th} = \frac{\Delta p\, V_g}{2\pi} a_1 \tag{16}
$$

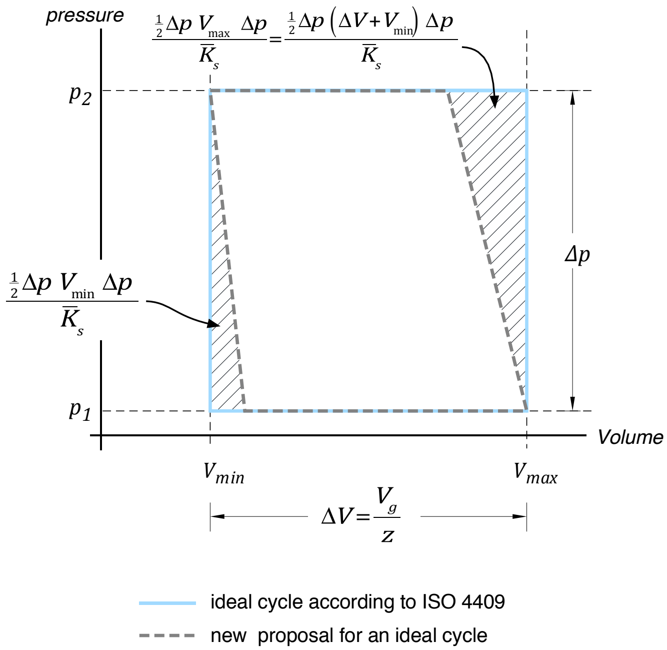


Fig. 2: Ideal cycle definitions. The blue (solid line) rectangle is the ideal cycle that is the basis for ISO 4409. The grey, skewed rectangle (dashed line) is the new 'ideal' cycle, including compressibility effects of the oil. The hatched areas mark the difference of the indicated work of both cycle definitions.

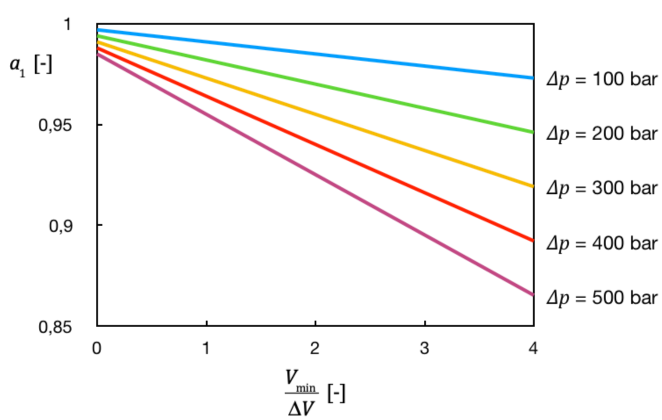


Fig. 3: Plot of a_1 for various pressures ($\overline{K}_S = 1.67 \cdot 10^9$ Pa)

Compared to the ISO definition, the new definition of the theoretical torque differs by a correction factor a_1. This factor is dependent on the pressure difference Δp, the average isentropic bulk modulus of the oil, and the ratio between the minimum volume of the displacement chamber and the displacement volume ΔV. All these parameters have a positive value, which means that $a_1 < 1$. Figure 3 shows a_1 for a range of values for Δp and $V_{min}/\Delta V$. For small pressure differentials and small dead volumes, the correction of the theoretical torque is not significant. However, there also exist situations in which ISO 4409 overestimates the theoretical torque by as much as 10% or even higher.

THE VOLUME RATIO

The correction factor a_1 strongly depends on the volume ratio $V_{min}/\Delta V$ (see Fig. 4). Knowing that:

$$V_g = z\,\Delta V \tag{17}$$

the volume ratio can be written as:

$$\frac{V_{min}}{\Delta V} = \frac{z\,V_{min}}{V_g} \tag{18}$$

The experimental determination of the total displacement V_g of a pump or motor has been described in ISO 8426 [5] or in [6]. However, at present, there is no experimental method for the determination of the minimum volume of the displacement chamber. This information will need either to be provided by the manufacturer, or, to be calculated from the dimensions of the pump or motor components.

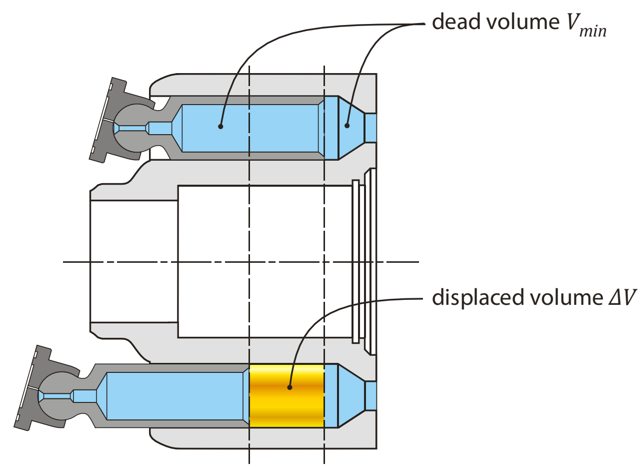


Fig. 4: V_{min} and ΔV of a slipper type axial piston machine

From an efficiency point of view, most pump and motor designers try to minimise the dead volume. For many pumps and motors the ratio $V_{min}/\Delta V$ is smaller than 1. There are, however, also designs in which the ratio $V_{min}/\Delta V$ is much larger, such as in Figure 4.

The situation becomes much more aggravated for variable displacement machines. If, for instance in the pump shown in Fig. 4, the swash angle is reduced, then the dead volume increases, whereas the displaced volume is reduced simultaneously. This can result in high values for the ratio $V_{min}/\Delta V$. The definition of ISO 4409 for the hydro-mechanical efficiency can then result in an error of even more than 15%. Also for digital displacement machines and machines using check valves for the commutation, the ratio will be dependent on the valve timing and valve dynamics of the inlet and outlet valves [7].

POWER LOSSES

The segregation of the overall losses into individual losses is important for many applications. The volumetric losses of a motor, for instance, define the amount of slip, whereas the torque loss determines the loss of load capacity, for instant when applied in a winch. Furthermore, the segregation of losses is also important for understanding the potential for design improvements. ISO 4409 does not specifically give any definitions for power losses, although they can be derived from the efficiency definitions.

In general, at any operating point (Δp and rotational speed n) the sum of all separate losses needs to be equal to the total loss:

$$P_{loss,t}\left(\Delta p,n\right)=\sum_{i}P_{losss,i}\left(\Delta p,n\right) \quad (19)$$

Whereas the standards for electric machines [8, 9] introduce up to six individual losses, ISO 4409 only divides the total loss into two losses: hydro-mechanical and volumetric losses. For hydrostatic machines, equation (19) thus becomes:

$$P_{loss,t}=P_{loss,hm}+P_{loss,v} \quad (20)$$

The total loss is defined as follows:

for pumps: $$P^{P}_{loss,t}=T\,\omega-P_{hydr} \quad (21)$$

for motors: $$P^{M}_{loss,t}=P_{hydr}-T\,\omega \quad (22)$$

where the hydraulic power P_{hydr} is defined in Eqs. (6) and (7). Similarly, the hydro-mechanical losses are defined as:

for pumps: $$P^{P}_{loss,hm}=T_{loss}\,\omega=\left(T-T_{th}\right)\omega \quad (23)$$

for motors: $$P^{M}_{loss,hm}=T_{loss}\,\omega=\left(T_{th}-T\right)\omega \quad (24)$$

Combining Eqs. (21), (22), (23), and (24) with Eq. (20), the volumetric losses can be defined as:

for pumps: $$P^{P}_{loss,v}=T_{th}\,\omega-P_{hydr} \quad (25)$$

for motors: $$P^{M}_{loss,v}=P_{hydr}-T_{th}\,\omega \quad (26)$$

The question is, if these last losses should be called volumetric losses or if they represent something more? A part of these losses are due to leakage, which can be measured at the drain port of the housing (if available). These losses are indeed real volumetric losses. But a large part of the losses defined in Eqs. (25) and (26) are related to commutation, i.e. throttling via the silencing grooves, which could also be regarded as hydro-mechanical losses.

EFFICIENCY DEFINITIONS

It is possible to define overall efficiency and hydro-mechanical efficiency equations for both pumps and motors. It is however much more difficult, if not impossible, to define the volumetric efficiency.

Overall Efficiency

The overall efficiency is calculated by comparing the mechanical power to the hydraulic power:

for pumps: $$\eta^{P}_{t}=\frac{P_{hydr}}{T\,\omega} \quad (27)$$

for motors: $$\eta^{M}_{t}=\frac{T\,\omega}{P_{hydr}} \quad (28)$$

Combined with Eq. (7) this results in:

for pumps: $$\eta^{P}_{t}=\frac{a_2\,p_2\,Q_2-p_1\,Q_1}{T\cdot\omega} \quad (29)$$

for motors: $$\eta^{M}_{t}=\frac{T\,\omega}{a_2\,p_2\,Q_2-p_1\,Q_1} \quad (30)$$

where a_2 is defined as:

$$a_2=1+\frac{p_2}{2\bar{K}_s} \quad (31)$$

Hydro-mechanical efficiency

As in ISO 4409, the hydro-mechanical efficiency of pumps and motors is defined by comparing the measured mechanical power to the theoretical torque, albeit now with the new definition (Eq. (16)), multiplied with the rotational speed:

for pumps: $$\eta^{P}_{hm}=\frac{T_{th}\,\omega}{T\,\omega}=\frac{\Delta p\,V_g\,a_1}{2\pi\,T} \quad (32)$$

for motors: $$\eta^{M}_{hm}=\frac{T\,\omega}{T_{th}\,\omega}=\frac{2\pi\,T}{\Delta p\,V_g\,a_1} \quad (33)$$

where a_1 is defined as:

$$a_1=1-\frac{\Delta p}{\bar{K}_s}\left(\frac{1}{2}+\frac{V_{\min}}{\Delta V}\right) \quad (34)$$

Volumetric efficiency

Following the definitions of ISO 4409, the volumetric efficiency is defined by:

for pumps: $$\eta_v^P = \frac{Q_2}{Q_{th}} = \frac{Q_2}{V_g\, n} \tag{35}$$

for motors: $$\eta_v^M = \frac{Q_{th}}{Q_2} = \frac{V_g\, n}{Q_2} \tag{36}$$

In these equations, the measured pressurised (compressed) oil flow is compared to the uncompressed flow rate (V_g is defined as the geometrical displacement at zero pressure). It should be noted, that the definition for the volumetric efficiency is not based on energy or power levels, but compares flow rates. In order to calculate a volumetric efficiency based on power or energy, both the measured and the theoretical flow could be multiplied with a pressure, but then the question is which pressure level should be used for the numerator and for the denominator. Any choice would be an arbitrary choice, and may question the validity of the definitions.

In general, for pumps and motors, the overall efficiency is assumed to be the product of the hydro-mechanical and the volumetric efficiencies [10-14]:

$$\eta_t = \eta_{hm}\,\eta_v \tag{37}$$

Following the equations (11) and (36) from ISO 4409:

$$\eta_t^M = \eta_{hm}^M\,\eta_v^M = \frac{2\pi\, T}{\Delta p\, V_g} \cdot \frac{V_g\, n}{Q_2} = \frac{T\,\omega}{\Delta p\, Q_2} \tag{38}$$

Yet, in ISO 4409, the overall efficiency for motors is defined as:

$$\eta_t^M = \frac{T\,\omega}{p_2 Q_2 - p_1 Q_1} \tag{39}$$

Equation (38) can only be equal to Eq. (39) if p_1 = 0 bar. ISO 4409 therefore does not support the general expression that the overall efficiency equals the product of the hydro-mechanical efficiency and the volumetric efficiency:

$$\eta_t \neq \eta_{hm}\,\eta_v \tag{40}$$

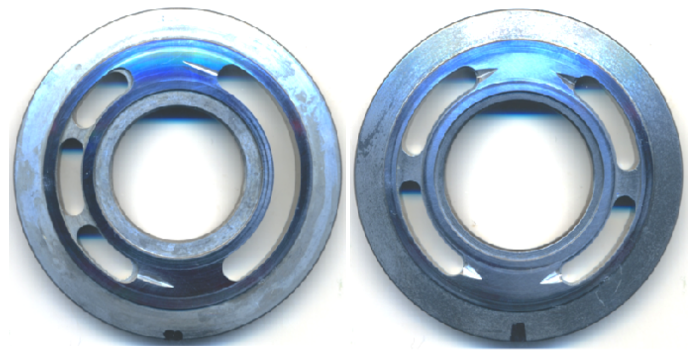
Furthermore, Eq. (37) assumes that the pump or motor consists of two machines in series, one causing only hydro-mechanical losses, and the second machine only creating volumetric losses. This does not correspond well with reality. The impracticability of Eq. (37) also becomes clear when the total loss would be separated into more than two losses. If, like in the standards for electric motors and generators, the total loss would be separated in six individual losses, it is obvious that in that case:

$$\eta_t \neq \eta_1\,\eta_2\,\eta_3\,\eta_4\,\eta_5\,\eta_6 \tag{41}$$

The equations show that the loss analysis by means of hydro-mechanical and volumetric efficiencies should be reconsidered and be replaced by analysis in terms of power losses.

CONSEQUENCES OF THE NEW LOSS ANALYSIS

As an example, efficiency tests of a slipper type pump and a slipper type motor are presented and evaluated. The tests have been performed at the INNAS test bench [15]. The pump and motor have a similar design, but have different port plates (Figure 5).



a) A4FO28 pump b) A4FM28 motor

Fig. 5: Port plates of the pump and motor being analysed

Both machines have been tested in a number of stationary operating points at an oil inlet temperature of 50°C (Shell Tellus VG46). The geometrical displacement has been measured with the method described in [6]:

- A4FO28 pump: V_g = 27.39 cc/rev
- A4FM28 motor: V_g = 27.75 cc/rev

The pump and the motor each have 9 pistons (z = 9). The dimensions of pump and motor have been measured. Based on these dimensions, the number of pistons and the measured geometrical displacement, it has been determined that the ratio $V_{min}/\Delta V = 0.78$.

The test results have been analysed according to ISO 4409, as well as using the new definitions for losses presented in this paper and summarized in Appendix B. Only the results for the measurements at 400 bar are shown.

Effects on the overall losses

Figure 6 shows the effect of the new definitions on the overall losses for both the pump and motor. Given the new definitions, the losses of the pump are 5 to 9% smaller, and for the motor 12 to 14% larger than those obtained using the ISO 4409 methods and definitions. This is due to the inclusion of the effect of the internal energy (see Eq. (1)), which results in an increased hydraulic power output for the pump, and a larger hydraulic input for the motor than with the ISO-definitions.

Effects on the hydro-mechanical losses

The new loss definitions given in this paper show a much larger effect on the calculated hydro-mechanical losses. The new definitions result in a 32 to 43% increase of the hydro-mechanical losses of pumps, whereas the same losses decrease with 41 to 68% for motors (see Figure 7).

Effects on the volumetric losses

The new definitions also have a strong influence on the volumetric losses (Figure 8). For the pump, the calculated losses are reduced by 48 to 75%. Employing the new definitions, the volumetric losses of the pump are almost identical to the measured drain flow losses. For the motor, however, the new calculation of the volumetric loss results in a much higher value than the one derived via the ISO 4409 equations. The losses are a factor 1,8 to 2,2 higher with the new definitions.

As was mentioned before, these results are analysed at a system pressure of 400 bar. The difference become smaller for lower pressure levels.

Differences between pump and motor operation

The new loss definitions reveal large differences between pump and motor operation of basically the same machine, aside from the port plate design. The differences in behaviour have also been seen during measurements at operating speeds below 1 rpm [16].

These differences can be explained by the losses due to commutation. The 'ideal' cycle is based on the assumption that during commutation, an instantaneous unthrottled opening and closing occurs of the connection between the displacement chamber and the high and low pressure lines. In order to achieve an isentropic compression and expansion of the hydraulic fluid in the displacement chamber, the chamber cannot have any connection to either the high or low pressure line, in order to allow the change of displacement to create a compression or expansion.

In reality, this ideal commutation is not possible. The displacement chamber (including the dead volume) is much longer and much more connected to the high pressure line than is assumed in the ideal cycle. This results in an internal leakage flow, especially from the high pressure line to the displacement chamber, which most of all increases the losses in the bottom dead centre, in the pump as well as in the motor.

The difference between the pump and the motor is, that a pump has to create its own flow, whereas the motor just absorbs any flow it needs or can receive from the high-pressure line. Therefore, the commutation losses are mainly measured as volumetric losses in a motor. But in a pump, the oil, which is used to compress the oil in the displacement chamber, can be seen as a loan, which has to be recuperated afterwards by the pumping action of the pump itself. This additional, non-effective pumping action needs extra mechanical input power. Therefore, contrary to a motor, the commutation losses of a pump are measured as hydro-mechanical losses.

CONCLUSIONS AND DISCUSSION

Given the analyses presented in this paper, it is concluded that the current ISO 4409 definitions are no longer sufficient in producing accurate information and may present inconsistencies especially for newly developed high-pressure, high-performance pumps and motors. More specifically:

- ISO 4409 is inconsistent in the calculation of the effects of oil compressibility: while it requires consideration for oil compressibility in the flow rates, it does not demand the same correction for the efficiency definition;
- The definitions of the overall and hydro-mechanical efficiencies for hydrostatic pumps and motors should be corrected by including compressibility effect;
- The current ISO 4409 definitions of volumetric efficiencies are based on flow rate ratios and not on power rates. For the purpose of power loss analysis and energy conversion analysis, new definitions are needed.
- It would be more beneficial to define equations for power losses instead of a segregation of the losses by means of efficiency definitions;

In this paper, new definitions of the overall loss, the hydro-mechanical loss and the volumetric loss were developed. It was shown that it is possible to define an overall energy efficiency as well as a hydro-mechanical efficiency. This paper did not result in a formula for calculating the volumetric efficiency.

Although ISO 4409 does not provide any equations for calculating the power losses, it is possible to derive these equations from the efficiency definitions given by ISO 4409. This paper thus proposes to correct these loss equations in order to include compressibility effects. The current ISO definitions as well as the proposed efficiency and loss equations are summarised in Appendices A and B for quick reference..

The determination of the torque losses and efficiencies , using the new definitions, demand knowledge about the $V_{min}/\Delta V$-ratio. There is yet no experimental methodology for the determination of this value. The value needs to be derived from the pump or motor dimensions. For fixed displacement machines, the ratio is a constant. However, for variable displacement machines, this value is variable. For digital displacement machines and check valve machines, the ratio depends on the timing and dynamics of the inlet and outlet valves.

The proposed loss and efficiency equations also demand information about the bulk modulus of the working fluid. The bulk modulus varies depending on the pressure level, the temperature of the fluid and the amount of air being dissolved in the fluid. For reasons of simplicity in the analyses, this paper used average secant values for the bulk modulus.

The proposed loss equations that are different from the ones currently adopted by the ISO 4409 methodology, are believed to lead to an improved analysis and result in a better understanding of the potential improvements and reduction of these losses.

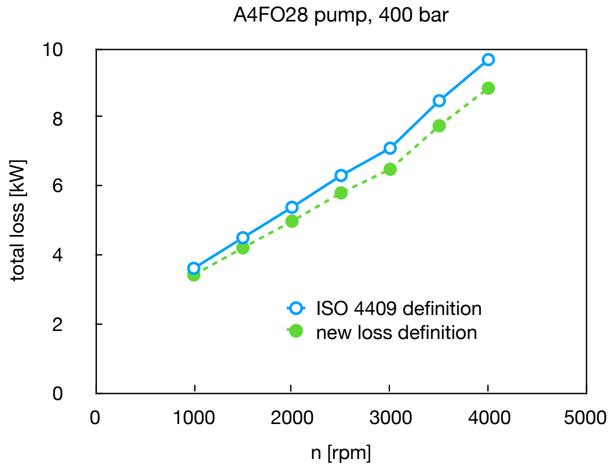


Fig. 6: Total loss of a slipper type pump (left) and motor (right)

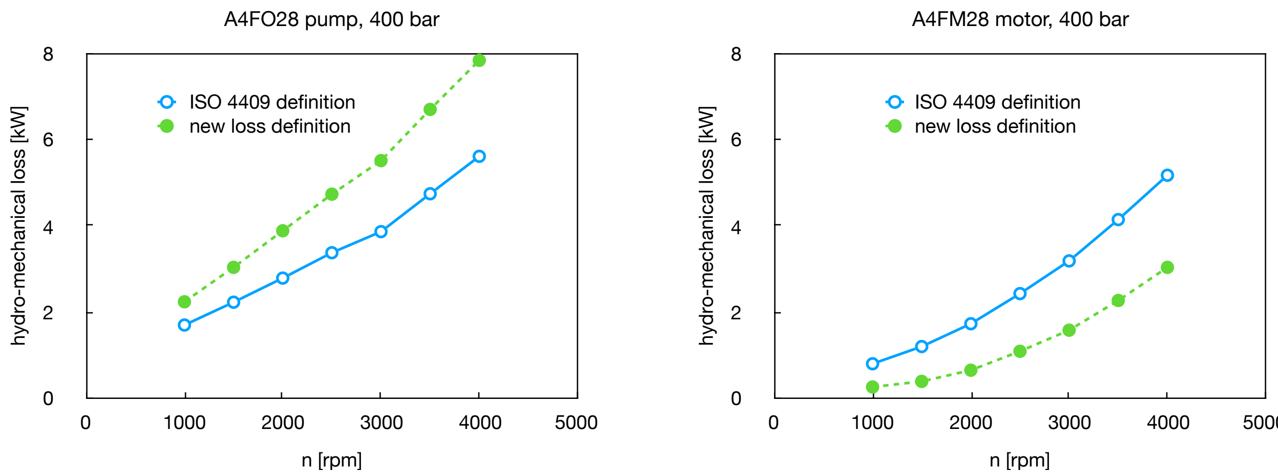


Fig. 7: Hydro-mechanical loss of a slipper type pump (left) and motor (right)

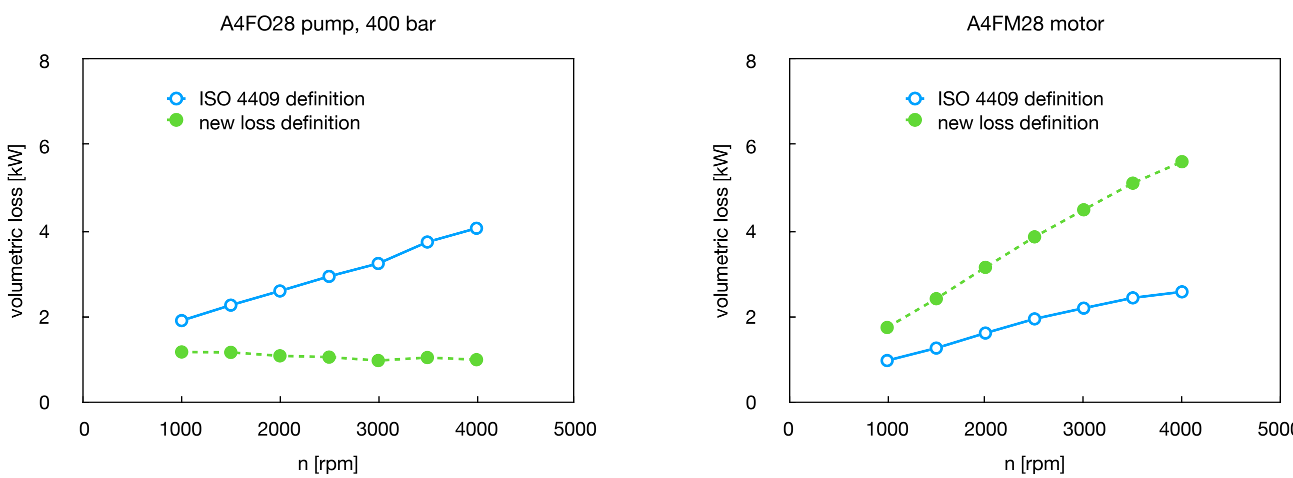


Fig. 8: Volumetric loss of a slipper type pump (left) and motor (right)

NOMENCLATURE

a_1	correction factor	[-]
a_2	correction factor	[-]
E_i	indicated work	[J]
h	specific enthalpy	[J/kg]
K_s	isentropic bulk modulus	[Pa]
K_T	isothermal bulk modulus	[Pa]
$\dot{m}$	mass flow rate	[kg/s]
n	rotational speed	[1/s]
p	pressure	[Pa]
P_{mech}	mechanical shaft power	[W]
P_{hyd}	hydraulic power	[W]
$P_{loss,t}$	total power loss	[W]
$P_{loss,hm}$	hydro-mechanical power loss	[W]
$P_{loss,v}$	volumetric power loss	[W]
Q	flow rate	[m³/s]
s	specific entropy	[J/kg K]
T	measured torque	[N m]
T_{th}	theoretical torque	[N m]
u	specific internal energy	[J/kg]
v	specific volume	[m³/kg]
V	volume	[m³]
ΔV	V_g/z	[m³]
V_g	geometrical displacement	[m³]
V_{min}	minimum volume displacement chamber	[m³]
V_{max}	maximum volume displacement chamber	[m³]
z	number of displacement chambers	[-]
α	thermal expansion coefficient	[1/K]
ρ	oil density	[kg/m³]
η_t	overall efficiency	[-]
η_{hm}	hydro-mechanical efficiency	[-]
η_v	volumetric efficiency	[-]
θ	temperature	[K]
ω	rotational speed	[1/s]

Subscripts

1	low pressure site of the circuit
2	high pressure site of the circuit
3	drainage port of the pump or motor
t	total or overall
hm	hydro-mechanical
v	volumetric
$mech$	mechanical
hyd	hydraulic
T	isothermal
s	isentropic

Superscripts

P	pump related
M	motor related

REFERENCES

1. ISO 4409:2007 Hydraulic fluid power -- Positive-displacement pumps, motors and integral transmissions -- Methods of testing and presenting basic steady state performance. 2007: 2007.
2. ISO 4392-3, First edition, 1993, Hydraulic fluid power - Determination of characteristics of motors - Part 3 At constant flow and at constant torque. 1993.
3. ISO 4392-1, Third edition 2002 Hydraulic Fluid Power – determination of characteristics of motors – Part 1: At constant low speed and constant pressure. 2002.
4. ISO 4392-2, Third edition 2002 Hydraulic Fluid Power – Determination of characteristics of motors– Part 2: Startability. 2002.
5. ISO 8426:2008 Hydraulic fluid power -- Positive displacement pumps and motors -- Determination of derived capacity. 2008.
6. Toet, G., et al., The Determination of the Theoretical Stroke Volume of Hydrostatic Positive Displacement Pumps and Motors from Volumetric Measurements. Energies, 2019. 12(3).
7. Manring, N. and C. Williamson, Calculating the Mechanical and Volumetric Efficiencies for Check-Valve Type, Digital Displacement Pumps. 2018
8. IEEE Standard Test Procedure for Polyphase Induction Motors and Generators IEEE Std 112™-2004. 2004.
9. IEC 60034-2-1:2014 Rotating electrical machines - Part 2-1: Standard methods for determining losses and efficiency from tests (excluding machines for traction vehicles). 2014.
10. Ivantysyn, J. and M. Ivantysynova, Hydrostatic pumps and motors, ed. N.D. Akademia Books International, India. 2001.
11. E., T.G. and V.J. De Negri, Handbook of hydraulic fluid technology, second edition, ed. C.P.T.a.F. Group. 2012.
12. Dieter, W. and G. Norbert, Hydraulik - Grundlagen, Komponenten, Systeme, ed. B. Springer Vieweg. 2014.
13. Dietmar, F. and H. Siegfried, Ölhydraulik - Handbuch der hydraulischen Antriebe und Steuerungen, ed. S. Vieweg. 2015.
14. Costa, G.K. and N. Sepehri, Understanding overall efficiency of hydrostatic pumps and motors. International Journal of Fluid Power, 2018. 19(2): p. 106-116.
15. Achten, P., J. Potma, and S. Eggenkamp. A New Hydraulic Pump and Motor Test Bench for Extremely Low Operating Speeds. in ASME/BATH 2017 Symposium on Fluid Power and Motion Control. 2017. Sarasota, Florida, USA.
16. Achten, P., J. Potma, and J. Achten, Low Speed Performance of Axial Piston Machines, in BATH/ASME 2018 Symposium on Fluid Power and Motion Control. 2018: Bath, UK

APPENDIX A

NEW PROPOSALS FOR EFFICIENCY CALCULATION IN ISO 4409

Table A1: Efficiency definitions for pumps

Overall efficiency:	
ISO 4409:	$\eta_t^P = \dfrac{p_2 Q_2 - p_1 Q_1}{T \cdot \omega}$
new:	$\eta_t^P = \dfrac{p_2 Q_2 a_2 - p_1 Q_1}{T\,\omega}$
Hydro-mechanical efficiency:	
ISO 4409:	No definition
new:	$\eta_{hm}^P = \dfrac{\left(p_2 - p_1\right) V_g}{2\pi\, T}\, a_1$
Volumetric efficiency	
ISO 4409:	$\eta_v^P = \dfrac{Q_2}{V_g\, n}$
new:	No definition

Table A2: Efficiency definitions for motors

Overall efficiency:	
ISO 4409:	$\eta_t^M = \dfrac{T\,\omega}{p_2 Q_2 - p_1 Q_1}$
new:	$\eta_t^M = \dfrac{T\,\omega}{p_2 Q_2 a_2 - p_1 Q_1}$
Hydro-mechanical efficiency:	
ISO 4409:	$\eta_{hm}^M = \dfrac{2\pi\, n\, T}{\left(p_2 - p_1\right) V_g}$
new:	$\eta_{hm}^M = \dfrac{2\pi\, T}{\left(p_2 - p_1\right) V_g} \dfrac{1}{a_1}$
Volumetric efficiency	
ISO 4409:	$\eta_v^M = \dfrac{V_g\, n}{Q_2}$
new:	No definition

$$a_1 = 1 - \frac{\Delta p}{\overline{K}_s}\left(\frac{1}{2} + \frac{V_{\min}}{\Delta V}\right)$$

$$a_2 = 1 + \frac{p_2}{2\overline{K}_s}$$

APPENDIX B

NEW PROPOSALS FOR THE CALCULATION OF POWER LOSSES

Table B1: Loss definitions for pumps. For ISO 4409, the losses are derived from the efficiency definitions.

Overall losses:	
ISO 4409:	$P^{P}_{loss,t} = T\,\omega - \left(p_2 Q_2 - p_1 Q_1\right)$
new:	$P^{P}_{loss,t} = T\,\omega - \left(p_2 Q_2 a_2 - p_1 Q_1\right)$
Hydro-mechanical losses:	
ISO 4409:	$P^{P}_{loss,hm} = T\,\omega - \left(p_2 - p_1\right)\dfrac{V^{P}_{g}\,\omega}{2\pi}$
new:	$P^{P}_{loss,hm} = T\,\omega - \left(p_2 - p_1\right)\dfrac{V_{g}\,\omega}{2\pi}\,a_1$
Volumetric losses	
ISO 4409:	$P^{P}_{loss,v} = p_2\left(V_g\,n - Q_2\right) - p_1\left(V_g\,n - Q_1\right)$
new:	$P^{P}_{loss,v} = p_2\left[V_g\,n\,a_1 - Q_2\,a_2\right] - p_1\left[V_g\,n\,a_1 - Q_1\right]$

Table B2: Loss definitions for motors. For ISO 4409, the losses are derived from the efficiency definitions.

Overall losses:	
ISO 4409:	$P^{M}_{loss,t} = \left(p_2 Q_2 - p_1 Q_1\right) - T\,\omega$
new:	$P^{M}_{loss,t} = \left[p_2 Q_2 a_2 - p_1 Q_1\right] - T\,\omega$
Hydro-mechanical losses:	
ISO 4409:	$P^{M}_{loss,hm} = \left(p_2 - p_1\right)\dfrac{V^{M}_{g}\,\omega}{2\pi} - T\,\omega$
new:	$P^{M}_{loss,hm} = \left(p_2 - p_1\right)\dfrac{V_{g}\,\omega}{2\pi}\,a_1 - T\,\omega$
Volumetric losses	
ISO 4409:	$P^{M}_{loss,v} = p_2\left(Q_2 - V_g\,n\right) - p_1\left(Q_1 - V_g\,n\right)$
new:	$P^{M}_{loss,v} = p_2\left[Q_2\,a_2 - V_g\,n\,a_1\right] - p_1\left[Q_1 - V_g\,n\,a_1\right]$

$$a_1 = 1 - \frac{\Delta p}{\bar{K}_s}\left(\frac{1}{2} + \frac{V_{\min}}{\Delta V}\right)$$

$$a_2 = 1 + \frac{p_2}{2\bar{K}_s}$$