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Churning losses analysis on the thermal-hydraulic model of a high-speed electro-hydrostatic actuator pump

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ABSTRACT

The speed of Electro-Hydrostatic Actuator (EHA) pump can recently reach a maximum of 20,000 rpm due to its high power density requirements. The thermodynamics of EHA pumps become increasingly important because the oil in the casing is used to cooling the high speed electric motor and solving the "hot spots" problem in the EHA system. In order to realize the boundary condition for the motor cooling at high speeds, more investigations should be given to the churning losses in terms of the large centrifugal forces of piston/cylinder assemblies and high fluid kinetic energy in high-speed conditions. This paper analyzes the influence of churning losses on the thermal-hydraulic model. Using the control volume method, heat transfer analysis of the piston pump is presented and the heat flow inside the piston pump produced by churning losses is described in details. The steady-state leakage temperatures of the high-speed EHA pump are simulated and validated through experiments. The experimental results show that the churning power losses need to be considered on the thermal-hydraulic model to ensure good performance for the prediction of the leakage temperature in EHA pump.

1. Introduction

The Electro-Hydrostatic Actuator (EHA) has been successfully applied to More Electric Aircraft (MEA) to replace inefficient and centralized hydraulic systems [1–3]. As shown in Fig. 1, the EHA system consists of an electrically controlled pump, several check valves, an accumulator and a hydraulic actuator forming a closed circuit. The hydraulic fluid is used to move the hydraulic actuator of primary flight control surfaces in the EHA system, the velocity and direction of hydraulic actuator are controlled by the fluid flow from an electric motor driven hydraulic pump called EHA pump [4–6].

Fig. 2 shows a typical configuration of an EHA pump rotating group consisting of a valve plate, a cylinder block, nine pistons, nine slippers, a swash plate, a retainer mechanism, several bearings and a shaft. The pistons are arranged in a circular array within the block at equal angular intervals along the shaft centerline. The piston heads are mounted in the slippers. Under the limitation of the swash plate and retainer mechanism, pistons periodically reciprocate within the cylinder block bores suctioning and discharging oil through a valve plate which is fixed to the pump casing.

The EHA system has significant advantages, such as weight saving, high efficiency and improved maintainability. However, the closed circuit of an EHA system could cause the localized "hot spots" such as high-power motor because the EHA system removes the effective heat transfer network in the centralized hydraulic system [7,8]. The high temperature of the motor in the EHA system accelerates ageing of the winding insulation material, and the insulation performance of the material is greatly reduced. The service life of the motor is significantly reduced, which might cause the motor on fire and the electric shock hazard. Therefore, the EHA pump rotating group moves in a casing filled with oil and shaft is hollow as shown in Fig. 2. And the high-speed motor is cooled by the leakage oil in the casing of the EHA pump to reduce the high temperature of the motor. In order to solve thermal equilibrium of the high-speed motor in the EHA system, the leakage temperature of EHA pump is needed to be obtained as the boundary condition.

Theoretical and experimental studies have been carried out on the thermal-hydraulic models to investigate the leakage temperature in EHA pumps [9–14]. Andersson et al. firstly analyzed the thermodynamic model of hydraulic components and simulated the temperature of EHA systems in the HOPSAN simulation package [9,10]. Power losses including the volumetric and mechanical losses were considered in the mathematical model. Sidders et al. developed a set of lumped parameter mathematical models based

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Nomenclature

A_{fw}	area of fluid/wall heat transfer (m^2)
A_{wa}	area of wall/air heat transfer (m^2)
C_d	drag coefficient related to the Reynolds number
c_p	specific heat (J/kg·°C)
$\bar{c}_p$	mean value of specific heat (J/kg·°C)
D	displacement of the pump (m^3/r)
d_1	diameter of the piston (m)
d_2	outer diameter of sealing land at the bottom of slipper (m)
d_3	inner diameter of sealing land at the bottom of slipper (m)
E	sum of the internal energy, the kinetic energy and the potential energy (J)
$\dot{E}$	rate of E (J/s)
h	specific enthalpy (J/kg)
h_{in}	enthalpy of mass which enters the control volume (J/kg)
h_{out}	enthalpy of mass which leaves the control volume (J/kg)
KE	kinetic energy (J)
k_{fw}	coefficient of fluid/wall heat transfer (W/m^2·°C)
k_{wa}	coefficient of wall/air heat transfer (W/m^2·°C)
l_0	length of the piston out of the cylinder block (m)
l_1	length of the piston guide in cylinder block (m)
l_c	length of the cylinder block (m)
L_p	additional power losses of per revolution (J/r)
m	mass of the internal energy (kg)
$\dot{m}_{in}$	rate of mass which enters the control volume (kg/s)
$\dot{m}_{out}$	rate of mass which leaves the control volume (kg/s)
m_l	mass of the leakage node (kg)
m_p	mass of the outlet node (kg)
m_t	mass of the inlet node (kg)
m_w	mass of the mass node (kg)
n	rotational speed of shaft (rpm)
PE	potential energy (J)
P_c	churning losses in the casing filled with oil (W)
P_{cc}	churning losses power due to the cylinder block (W)
P_{cp}	churning losses power due to the piston/slipper units (W)
P_{cv}	viscous friction losses of the cylinder block/valve plate pair (W)
P_{pc}	viscous friction losses of the piston/cylinder block pair (W)
P_{ps}	viscous friction losses of the piston/slipper pair (W)
P_v	viscous friction losses of three friction pairs (W)
p	pressure (Pa)
p_l	pressure in the leakage node (Pa)
p_p	pressure in the outlet fluid node (Pa)
p_t	pressure in the inlet fluid node (Pa)
$\dot{Q}$	rate of heat transfer (J/s)
Q_{il}	internal leakage flow rate (m^3/s)
Q_{el}	external leakage flow rate (m^3/s)
R	pitch circle radius of piston bores (m)
R_1	inside radius of the inner region of the sealing land on the valve plate (m)
R_2	outside radius of the inner region of the sealing land on the valve plate (m)
R_3	inside radius of the outer region of the sealing land on the valve plate (m)
R_4	outside radius of the outer region of the sealing land on the valve plate (m)
R_5	inner radius of auxiliary support on the valve plate (m)
R_6	outer radius of auxiliary support on the valve plate (m)
R_c	radius of the cylinder block (m)
$\bar{T}$	mean value of temperature (°C)
T	temperature (°C)
T_a	temperature in the air (°C)
T_{in}	temperature in the oil tank (°C)
T_l	temperature in the leakage node (°C)
T_t	temperature in the inlet fluid node (°C)
T_w	temperature of the mass node (°C)
T_p	temperature in the outlet fluid node (°C)
t	time (s)
t_1	gap between the piston and cylinder block (m)
t_2	gap between the slipper and swash plate (m)
t_3	gap between the cylinder block and valve plate (m)
t_4	gap between the cylinder block and the housing internal surface (m)
U	internal energy (J)
u	internal energy of per mass (J/kg)
V	control volume (m^3)
V_l	volume in the leakage fluid node (m^3)
V_p	volume in the outlet fluid node (m^3)
V_t	volume in the inlet fluid node (m^3)
$\dot{W}$	rate of work done by the control volume (J/s)
$\dot{W}_b$	rate of boundary work (J/s)
$\dot{W}_s$	rate of shaft work (J/s)
z	number of pistons
α	reduction coefficient of the multi-pistons
α_p	volume expansion coefficient of the control volume (m^3/°C)
$\bar{\alpha}_p$	mean value of volume expansion coefficient of the control volume (m^3/°C)
ρ	fluid density (kg/m^3)
γ	swashplate angle (°)
μ	dynamic viscosity of oil (Pa·s)
υ	specific volume of the control volume (m^3/kg)
$\bar{\upsilon}$	mean value of specific volume (m^3/kg)
ω	angular velocity of shaft (rad/s)
η_m	mechanical efficiency of axial piston pump
η'_m	modified mechanical efficiency of axial piston pump
θ_1	wrap angle of auxiliary support on the valve plate (rad)
φ_0	sealing angle of the valve plate (rad)

on conservation of mass and energy for the hydraulic system [11]. After that, Jiao et al. presented a kind of modeling approach to study the thermal-hydraulic piston pump used in the EHA system [12–14]. Heat transfer analysis of the EHA pump was also performed and the heat flow inside the EHA pump was described in detail in the model. However, heat generation of high-speed EHA pump comes from not only the power losses of friction pairs but also the churning losses caused by the internal rotating components stirring the hydraulic fluid in the casing. Although a considerable amount of research has been devoted to the thermal-hydraulic model of EHA pumps in terms of power losses, little information is available on the churning losses because its effect is minor for pumps with the nominal speeds of 1500–3000 rpm [15–18]. With an increasing speed, the proportion of churning losses in axial piston pumps gradually increases, especially for high-speed EHA pumps. In general, the EHA pump is characterized as a very small displacement so that it can be easily mounted next to or even the inside actuator. However, an EHA system usually needs a large flow to drive hydraulic actuators with good speeds. It means that a very high rotating speed is required for the EHA