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Numerical and experimental investigation on torque characteristics of seawater hydraulic axial piston motor for underwater tool system

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ABSTRACT

Seawater hydraulic axial piston motor is an important and elemental component in underwater tool system. The torque characteristics for a swash-plate-type seawater hydraulic axial piston motor is investigated, and an integrated torque model for the motor with symmetrical pre-compression angles has been developed, which consists of a torque sub-model and a dynamic pressure sub-model. Numerical simulations have been carried out to examine the effects of (a) pre-compression angle, (b) relief-groove obliquity, (c) motor speed, (d) piston chamber dead volume, (e) friction on the dynamic pressure and the output torque characteristics. The results indicate that the pre-compression angle, the friction coefficient, and the clearance between cylinder bore/piston have significant impact on the torque characteristics. The test verification has been undertaken with a five piston water hydraulic motor. This research contributes to the mechanism of output-torque fluctuation in a swash-plate-type seawater hydraulic axial piston motor, as well as the investigation of the torque transition phenomenon owing to the pre-compression angle. The research has laid the foundation for the development and improvement of the seawater hydraulic axial piston motor in underwater tool system.

1. Introduction

Underwater tool system is extensively applied in the sea salvage, marine resource investigation and marine structures or sea defense projects. The underwater tool system driven by seawater hydraulics has many advantages compared with hand-operated tool, electric power tool, pneumatic tool, oil hydraulic tool system, including non-flammability, low operating cost, and low pollution potential to marine environment. Moreover, the seawater hydraulic tool system has the function of pressure self-compensation, which can be designed into open system (Liu et al., 2011; Hu and Du, 2012). An underwater tool system driven by seawater hydraulics is illustrated in Fig. 1, which is employed to grind the hulls and underwater structures (Nie et al., 2004).

Seawater hydraulic axial piston motor (SAPM) is characterized by the higher volumetric efficiency and lower p_v values of the friction pairs in comparison with hydraulic gear and vane motors (Yin et al., 2013; Lim et al., 2003; Yang et al., 2013), which is the main type applied to rotational actuator in the underwater tool system. Output torque is one of the primary performance parameters in SAPM. Because of different physicochemical properties of raw water in comparison with mineral oil, the output torque of the SAPM would be different from the mineral oil one.

Previously, several researchers studied the effect of multiple factors on the dynamic pressure and torque of oil piston pumps/motors. Himmler (1969) presented experimental studies of the low speed-motion for different kinds of hydraulic axial piston motor (HAPM). It was concluded that the static friction in axial piston motor was high, and that the leakage between piston and cylinder had an important effect on the low speed behavior and torque characteristic of HAPM. However, it was not clear from this study whether the leakage increased the effects of static/kinetic friction and was stable at low speed. Hibi and Ichikawa (1975) developed a mathematical model of hydraulic motor to study the friction and torque characteristics in the entire operating process, *i.e.* from start to maximum speed. Dsagupta et al. (1996) studied the steady state performance of an orbital-rotor, low-speed, and high-torque hydraulic radial piston motor. Guo and Wang (1996) had made an analysis and

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Nomenclature

A	piston working area (mm^2)
A_{1-4}	cross-sectional area of triangle relief groove from the 1st to the 4th quadrant (mm^2)
B	viscous damping coefficient (N s/m^2)
C_d	discharge coefficient of flow through relief groove, $C_d=0.8$
C_e	impact coefficient for laminar entry flow
d	diameter of piston (mm)
E	bulk modulus of fluid (Pa), for water $E=2.43\times10^9$ Pa
F_1, F_2	acting forces of piston (N)
F_t	spring force of piston (N)
F_a	inertia force of piston (N)
F_l	centrifugal force of piston (N)
f	friction coefficient
l	theoretical length of piston (mm)
l_0	piston length remaining in cylinder bore (mm)
l_1, l_2	stress distribution lengths of piston (mm)
l_c	distance from the centroid to the spherical center of piston (mm)
m	number of pistons inside high-pressure zone
m_p	mass of piston-slipper assembly (kg)
n	motor speed (r/min)
N_i	reaction force between swash plate and slipper of i_{th} piston (N)
P_c	dynamic pressure inside piston chamber (MPa)
P_d	discharge pressure (MPa)
P_s	supply pressure (MPa)
Q_{in}	volumetric flow into the piston chamber by relief groove (m^3/s)
Q_{out}	leakage of piston chamber (m^3/s)
Q_{slip}	leakage between slipper and swash plate (m^3/s)
R	pitch radius of piston (mm)
r_1	inner radius of slipper (mm)
r_2	outer radius of slipper (mm)
T	output torque (N m)
V_i	initial control volume of piston chamber (m^3)
V_0	dead volume of piston chamber (m^3)
x_i	axial position of of ith piston (mm)
Z	number of pistons
α	interval angle of adjacent pistons (°)
γ	angle of swash plate (°)
δ	clearance between piston and cylinder bore (mm)
δ_1	clearance between slipper and swash plate (mm)
Δ	magnitude of torque transition (N)
ε_0	relative eccentric ratio (–)
θ	obliquity of triangle relief groove (°)
θ_2	angle of triangle relief groove (°)
λ	pressure ratio of the hydrostatic bearing
μ	dynamic viscosity of fluid (Pa s)
v	axial velocity of piston (m/s)
ρ	density of fluid (kg/m^3)
υ	fluid kinematic viscosity (m^2/s)
φ_i	angular position (°)
ϕ_0	pre-compression angle (°)
ω	angular velocity of rotor (rad/s)

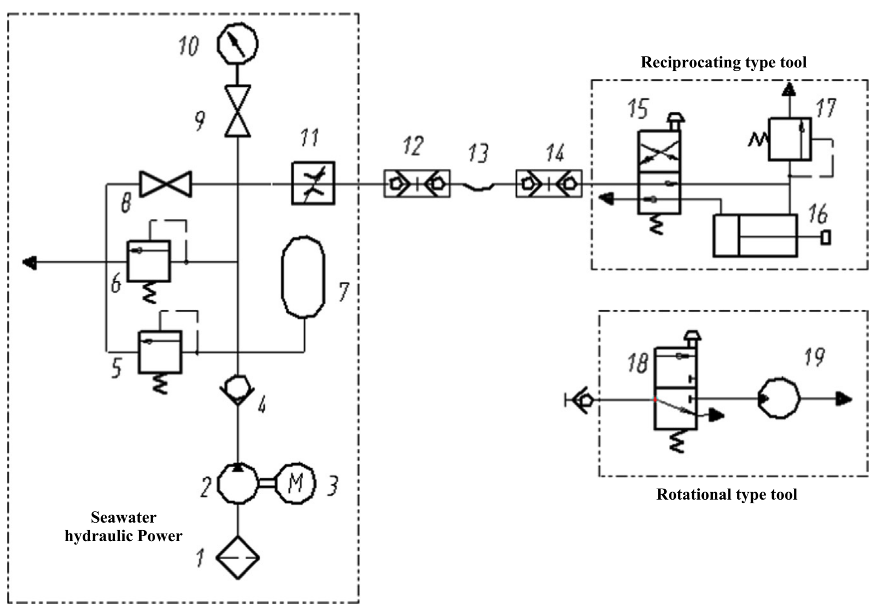

Fig. 1. Hydraulic diagram of underwater tool system driven by seawater hydraulics. 1-filter; 2-pump; 3-submersible motor; 4-check valve; 5-relief valve; 6-safe valve; 7-accumulator; 8,9-shut-off valve; 10-pressure gauge; 11-flow speed control valve; 12, 14- quick-change coupler; 13-soft pipe; 15,18-directional control valve; 16-cylinder;17-back pressure valve; 19-water hydraulic motor.

solving method of pressure inside piston chamber of oil pump when the piston was at transition zone of pre-expansion and pre-compression, respectively. It was proven through simulation that the piston pressure at transition zone was affected by the swashplate angle. Tan et al. (1999) analyzed the transient cylinder pressure in the radial piston hydraulic motors. They established the mathematical models for instantaneous pressure and got the simulation results that distribution overlap of motors must be strictly controlled. Meikandan et al. (1990, 1994) analyzed the effects of the tapered piston as well as the friction between piston and cylinder bore on the volumetric efficiency and torque characteristic of HAPM. Sadashivappa et al. (1996) investigated theoretically and experimentally the effects of piston-form deviations on the mechanical and volumetric efficiency of HAPM, through considering the viscous friction and the piston profiles. This study indicated that friction torque of the motor with three-lobe pistons was smaller than that with cylindrical piston. Ivantysynova et al. (2002) investigated the swash plate moment of