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Modeling of A Novel 3-Way Rotary Type Electro-hydraulic Valve

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***Abstract* - Hydraulic actuation is characterized by fast dynamics, high power density, high stiffness, large output force/torque, and in recent years it has become an increasing attractive form of robot actuation. Hydraulic control valves that not only provide the interface between hydraulic power element and actuators, but also receive feedback signal and adjust the system output accordingly, are key components of hydraulic actuation systems. This paper presents a novel 3-way rotary type electro-hydraulic valve which is driven by a DC motor. The valve is mainly composed of a rotary spool, a bush and a body. The operating principle is presented in details and a mathematical model is developed. In particular, the drag torque that majorly affects the valve performance is analyzed based on the theory of fluid mechanics.**

Index Terms - DC motor; Electro-hydraulic valve; Hydraulic actuation; Rotary spool.

I. INTRODUCTION

Hydraulic actuation is one of the classical actuation techniques. Compared to electro-mechanical drives and pneumatics, it has fast dynamics, high power density, high stiffness, large output force/torque, and in recent year is becoming an increasing attractive for robot actuation. For instances, it has been employed in robotic system, such as the exoskeleton system BLEEX [1], Raytheon SARCOS [2], the SARCOS hydraulically actuated humanoid robot CB [3], the legged robots Kenken [4], BigDog [5-7], Petman [8] and the leg prototype HyQ [9-11] and the wheel-legged robot Micro-hydraulic toolkit [12].

In all the robots mentioned above, valve-controlled hydraulic actuation systems are used. Within this kind of systems, the electro-hydraulic valves not only provide the interface between the hydraulic power element, i.e., the pump, and the hydraulic output device (a linear or rotary actuator) but also receive the feedback signal from the controller and adjust the system output accordingly. Therefore, the electro-hydraulic valves characteristics have great effects on the robot hydraulic control system performance.

Typically, two types of electro-hydraulic valves are used in robotic hydraulic actuation systems: proportional valve and servovalves.

Proportional valves, such as those used in [9-11], usually employ proportional solenoids to drive and control the spool motion and thereby change the orifice areas, which control system flow or pressure. The proportional valves are cheap and have good sealing performance in the neutral position. More importantly, they have low pressure losses ($<$ 1MPa) at the rated flow rate, thus saving energy, reducing temperature rise and improving the performance of the working medium and of the whole system. However, their poor accuracy (error factor $\geq$ 3%), low frequency response ($<$ 30Hz), large dead band (10%-25% of the rated input signal)[13-16] make them not a good choice for robot actuation where accurate control is required.

Servovalves, such as the control valves in BigDog actuation system [5-7] and the SARCOS hydraulically actuated humanoid robot CB [3], still meter flow proportionally but they have an internal closed-loop feedback mechanisms to realize precise control and are usually actuated by torque motors plus an internal hydraulic pilot stage. The servovalves have good dynamics performance (frequency response: 60-400Hz) and high control accuracy (error factor $<$ 1%). But they are expensive and sensitive to contamination. Furthermore, the pressure loss at rated flow rate is large (For example, the pressure loss of Moog servovalve E024 is 7MPa at rated flow rate [17]) and the leakage is much larger than that of a proportional valve due to critical center and the internal pilot stage, which make the servo valve less efficient [13-16]

From the discussion above, it can be concluded that the conventional electro-hydraulic valves are not the optimal choices for hydraulically actuated autonomous robot, where energy saving is an important issue provided dynamic response is acceptable. Therefore, there is scope to design an electro-hydraulic valve oriented to robotic applications.

The research [18] aims at developing a new electro-hydraulic valve with high efficiency and good dynamic performance. This paper presents the model of the valve, in which a spool driven by a DC motor is rotatably fitted in a valve bush. The flow area is controlled by the input signal to the DC motor, thus controlling the flow rate. A mathematical model is derived based on a detailed analysis of the drag torque.

The rest of this paper is organized as follows: Section II introduces the configuration and function of the electro-hydraulic valve. Section III presents the mathematical model. Section IV focuses on the drag torque analysis. Section V addresses the discussion and conclusions.

II. PHYSICAL MODEL

A. *Configuration*

The configuration of the rotary type 3-way electro-hydraulic spool valve [21] is illustrated in Figure 1. It consists of a:

a) DC motor
b) Valve bush
c) Rotary spool
d) Valve body
e) Rotary seal element and static seals
f) Thrust bearing
g) Stop pin
h) Connecting tube

The valve has 3 ports:

a) Port A connecting to actuator
b) Port P connecting to power source (pump)
c) Port T connecting to tank

The spool is rotatably fitted in a valve bush. One of the spool ends is coupled to a DC motor so as to be driven and the other end connects to port A with the hole H1. The grooves on the spool are symmetrical, which means that the resultant static pressure force acting on the spool equals zero, thus decreasing the friction torque. Turning the spool, port A connects to port P or port T by a controllable flow area. The stop pin, which is fixed on the spool by a nut, swings in the connecting tube and is limited by two lands. The hole H2 connects the bearing house to port T, and therefore, the rotary seal always works in low pressure medium which can reduces the friction torque and extends the life of the seal.

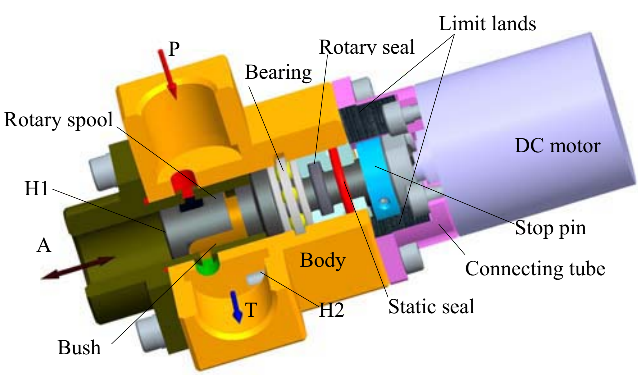


Fig.1 Schematic of DC driven rotary spool valve

B. *Function*

Based on Bernoulli's equation, the flow Q through an orifice can be written as:

$$Q = C_q A_{FA} \sqrt{\frac{2}{\rho} \Delta p} \tag{1}$$

where C_q is the flow coefficient, ρ the density of the fluid, Δp the pressure difference between the upstream and downstream of the orifice. A_{FA} the flow area.

The DC motor is used to drive and control the rotation of the valve spool, Fig.2. The spool rotary angle, speed and direction can be controlled, thus controlling the flow area A_{FA} and the flow rate. If we set the start point $\alpha = 0°$ corresponding to the fully closed status, Fig.3 (b), the sealing angle $\pm\alpha_s$, and the rotary range $\pm\alpha_r$, and the operating modes of this electro-hydraulic valve can be expressed as:

a) $|\alpha| \leq \alpha_s$, the valve is closed and there is no flow from port P to A and A to T, Fig.3(b);

b) $\alpha_s < \alpha \leq \alpha_r$, the valve is open and the flow is from port P to A, Fig.3(a);

c) $-\alpha_r \leq \alpha < -\alpha_s$, the valve is open and the flow is from port A to port T, Fig.3(c).

The flow area A_{FA} can be described as:

$$A_{FA} = 2Rb \sin \frac{(C_1\alpha - C_2\alpha_s)}{2} = Rbk_{\alpha s}(C_1\alpha - C_2\alpha_s) \tag{2}$$

$$\alpha_r - \alpha_s = 2\arcsin \frac{l}{2R} \tag{3}$$

where α is the rotary angle, R the radius of the rotary spool, l and b are the length and width of the orifice cross section, $C_1 = 0.5\big(\mathrm{sgn}(\alpha + \alpha_s) + \mathrm{sgn}(\alpha - \alpha_s)\big)$, $C_2 = 0.5\big(\mathrm{sgn}(|\alpha| + \alpha_s) + \mathrm{sgn}(|\alpha| - \alpha_s)\big)$, $k_{\alpha s} = \sin\frac{(|\alpha|-\alpha_s)}{2} / \frac{(|\alpha|-\alpha_s)}{2}$ and $k_{\alpha s} \cong 1$ when $(|\alpha| - \alpha_s)$ is small.

Therefore, the flow rate can be written as:

$$Q = k_p(C_1\alpha - C_2\alpha_s) \tag{4}$$

where $k_p = Rbk_{\alpha s}C_q\sqrt{2\Delta p/\rho}$

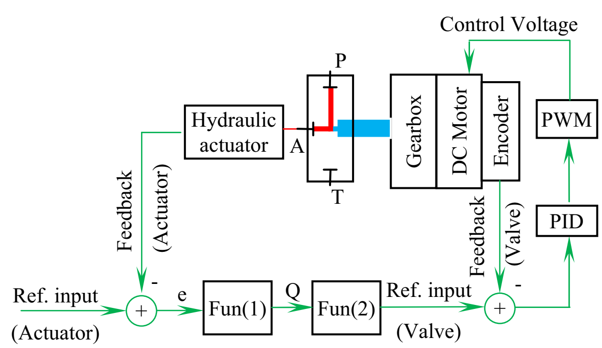
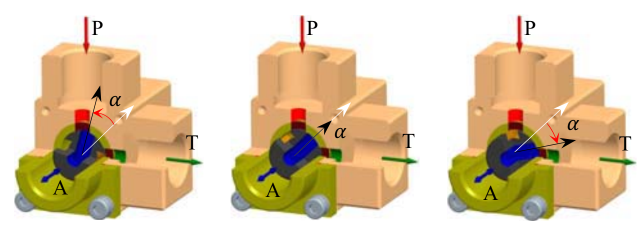


Fig.2 Operating principle



(a) P to A (b) Close (C) T to A

Fig.3 Operating modes

III. MATHEMATICAL MODEL

A functional block diagram of the rotary spool electro-hydraulic valve is illustrated in Fig.4. It consists of an electrical part (represented by a voltage source (u_a) across the coil of the armature, which is modelled by an inductance (L_a) in series with a resistance (R_a) and with an induced voltage (u_m) opposing the voltage source), a mechanical part (the valve spool) and the interconnection between them (gear box).

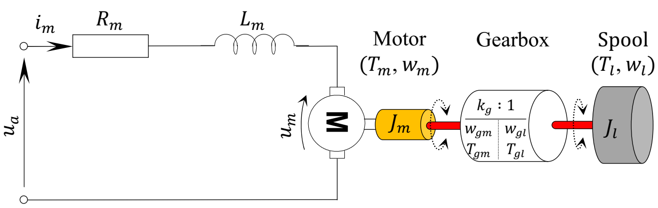


Fig.4 Model of the rotary valve

A. *Dc motor*

The DC motor [19] converts the electrical power into a mechanical power applied to the shaft. If the energy losses are neglected:

$$i_m u_m = T_m w_m \tag{5}$$

$$u_m = k_w \cdot w_m \tag{6}$$

which implies:

$$T_m = k_T \cdot i_m \tag{7}$$

where i_m is rotor circuit current (A), u_m the electromotive voltage (V), T_m the DC motor output torque (N·m), w_m the rotor speed of the DC motor (rad/s), k_w the speed constant $(\mathrm{V}/(\mathrm{rad/s}))$ and k_T the torque constant $(\mathrm{N \cdot m/A})$, $k_T = k_w$ (numerically-equal).

By applying Kirchhoff's law:

$$u_a = R_a \cdot i_m + L_m \frac{di_m}{dt} + u_m \tag{8}$$

where u_a is the input voltage (V), R_a is the circuit resistance (Ω) and L_m is the motor's inductance resistance

The applied torque produces an angular velocity w_m according to the inertia J_m and friction f_m of the motor and load T_{gm}:

$$T_m = J_m \dot{w}_m + f_m w_m + T_{gm} \tag{9}$$

B. *Gearbox*

Assuming that the gearbox is ideal, with no backlash and assuming that the shafts are stiff:

$$w_{gl} = w_{mg}/k_g = w_m/k_g \tag{10}$$

$$T_{gl} = k_g T_{gm} + J_g \dot{w}_{gl} \tag{11}$$

where J_g is the gearbox reflected inertia at the output of gear train (kg·m^2) and k_g the gear reduction radio.

C. *Spool*

The spool of the valve is the load for the DC motor:

$$T_{gl} = T_l + J_l \dot{w}_l \tag{12}$$

$$w_l = w_{gl} = \dot{\alpha} \tag{13}$$

where T_{gl} and w_{gl} are the gearbox output torque and rotary speed, J_l and w_l are the spool inertia and rotary speed respectively.

If only $\alpha \geq 0$ is considered, from (5)-(13), the system equations can be written as:

$$J_1 \ddot{\alpha} + f_1 \dot{\alpha} + T_l = k_g k_T i_m \tag{14}$$

$$u_a - k_w k_g \dot{\alpha} = R_a \cdot i_m + L_m \frac{di_m}{dt} \tag{15}$$

where $J_1 = k_g^2 J_m + J_l - J_g$, $f_1 = k_g^2 f_m$.

Taking the Laplace transform of (4), (14) and (15):

$$s^2 J_1 \alpha(s) + s f_1 \alpha(s) + T_l(s) = k_g k_T I_m(s) \tag{16}$$

$$u_a(s) - s k_w k_g \alpha(s) = (sL_m + R_a) I_m(s) \tag{17}$$

$$Q(s) = k_p [C_1 \alpha(s) - C_2\, \alpha_s / s] \tag{18}$$

A block diagram for the system can be developed from the differential equations given in (16), (17) and (18):

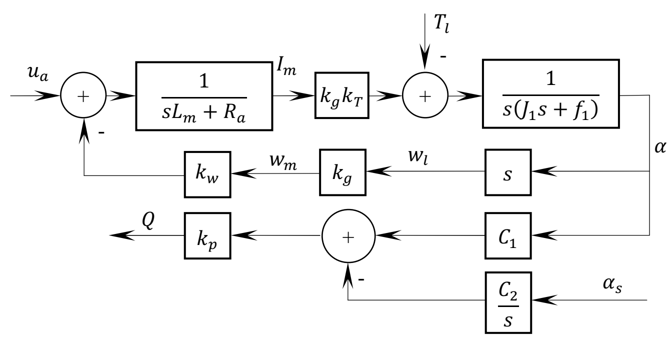


Fig.5. Block diagram representation

IV. DRAG TORQUE (T_l) ANALYSIS

From the block diagram, Fig.5, it can be seen that the DC motor should overcome the drag torque T_l. Therefore, it is essential to have a good understanding of the load.

When a spool rotates in a bore, there exists a friction torque caused by the viscosity of working medium (*viscous friction torque*, Fig.6). In order to protect the DC motor from the working medium, there is a rotary seal around the spool (shown in Fig.1) and the torque caused by this seal is called *seal friction torque*. When the fluid jets out from an orifice, there exists a tangential change of momentum, which reflects a torque preventing the spool rotation (*steady state flow torque*). If the spool is angularly accelerated, the fluid in the spool grooves should be accelerated synchronously and a torque is produced to meet with the acceleration (*transient flow torque*)

A. *Viscous friction torque*

Viscous friction torque is the resistance torque which is caused by the viscosity of the working medium. Due to the small clearance ($\delta \leq 5\mu$m) between the spool and bush, the working medium between the layers is considered to be a laminar incompressible Newtonian fluid and the shear stress, τ, is proportional to the velocity gradient, $\partial u/\partial r$, in the direction perpendicular to the layers [20]. The viscous friction torque is given by (7) under the assumption that the spool and bush are concentric.

$$\tau = \mu\,\partial u/\partial r \tag{19}$$

$$u = w_l R \tag{20}$$

$$\partial u/\partial r = w_l R/\delta \tag{21}$$

$$T_v = \tau R A_{sb} = 2\pi\mu \frac{R^2 L_{sb}}{\delta}\dot{\alpha} \tag{22}$$

where A_{sb} and L_{sb} are the contact area and length between the spool and bush respectively and, w_l is the rotary speed of the spool.

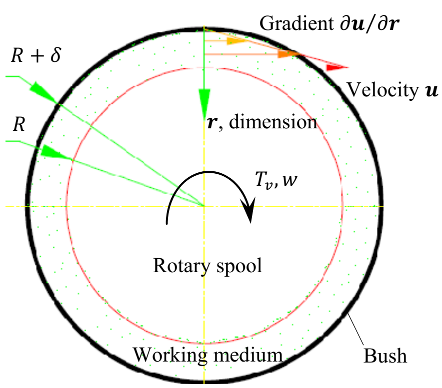


Fig.6 Laminar shear of the fluid between the spool and bush

B. *Seal friction torque*

For a rotary seal element, the friction torque is determined by the rotary speed, sealing pressure, pre-deformation and temperature [21]. In this electro-hydraulic valve, only the rotary speed changes frequently. According to Marin's studies [22], under low sealing pressure, the friction torque of a rotary seal approaches a constant. Therefore, in this research, the friction torque of the rotary seal is considered constant:

$$T_s = C_s \tag{23}$$

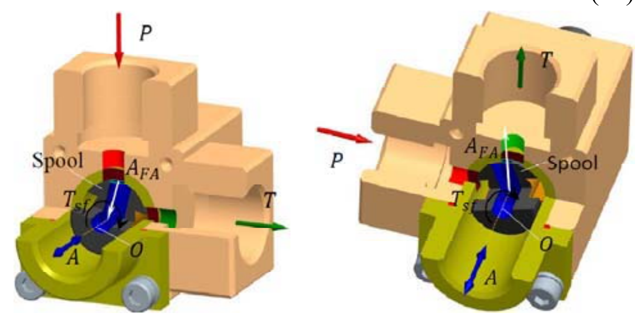
C. *Flow induced torque*

The flow induced torque/force is the torque/force acting on a valve spool as a result of fluid flowing in the valve chambers and through the valve orifices. It is considered to be either steady state or transient [23-25]. The steady state flow torque/force relates to a steady state of the spool valve due to relatively constant flow conditions, while the transient condition is due to the acceleration of the fluid in the valve.

1) Steady state flow torque

Theoretically, the direction of the steady-state flow force is perpendicular to the area A_{FA} and intersects the axis of the rotary spool "*O*", Fig.7. Therefore, the steady flow induced torque T_{sf} is considered to be zero in this research:

$$T_{sf} = 0 \tag{24}$$



(a) Meter-in (P to A) (b) Meter-out (A to T)

Fig.7 Direction of jet flow in the rotary spool valve

2) Transient flow torque

The transient flow torque is due to the angular acceleration of the fluid in the flow passage of the spool. The direction of the transient flow torque is the negative direction of $\ddot{\alpha}$

In the spool, the fluid in the flow passage has two types of motion, Fig.8:

(a) Linear motion

The linear motion is caused by the centrifugal force, and its direction is parallel to A_1. Obviously, the linear motion gives no contribution to the transient flow torque.

(b) Rotary motion

If the compressibility is neglected, the fluid in the flow passage should have the same rotary acceleration and speed at the spool. When the spool is angularly accelerated, the fluid element should be speeded up, while the pressure distribution on A_1 and A_2 is changed, thus a transient flow torque is produced and reacts on the spool:

$$T_{tf} = J_{fi}\ddot{\alpha} \tag{25}$$

where J_{fi} is the moment inertia of the control volume, Fig.8, to the axis "*O*".

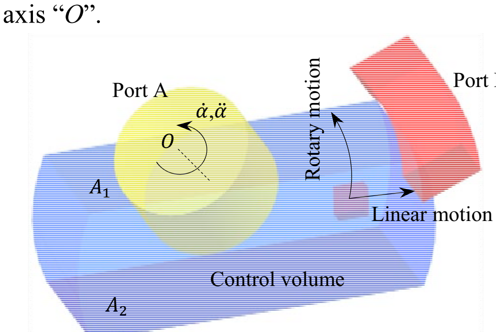


Fig.8 Fluid flow in the valve

In general, the load T_l can be written as:

$$\begin{aligned} T_l &= T_v + T_s + T_{sf} + T_{tf} \\ &= J_{fi}\ddot{\alpha} + 2\pi\mu \frac{R^2 L_{sb}}{\delta}\dot{\alpha} + C_s \end{aligned} \tag{26}$$

D. The electro-hydraulic valve model

According to (26), (14) can be written as:

$$(J_1 + J_{fi})\ddot{\alpha} + \left(f_1 + 2\pi\mu \frac{R^2 L_{sb}}{\delta}\right)\dot{\alpha} + C_s = k_g k_T i_m \tag{27}$$

If the initial conditions are zero, the Laplace transform of (27) is:

$$s^2 J\alpha(s) + sf\alpha(s) + C_s/s = k_g k_T I_m(s) \tag{28}$$

where $J = k_g^2 J_m + J_l + J_{fi} - J_g$, $f = k_g^2 f_m + 2\pi\mu \frac{R^2 L_{sb}}{\delta}$.

The transfer function, where the angle position is the output and the voltage is the input, can be expressed as:

$$\alpha(s) = \frac{k_g k_T}{R_a f + k_g^2 k_w k_T} \cdot \frac{u_a(s) - u_0(s)}{s\left(\frac{s^2}{w_h^2} + \frac{2\delta_h}{w_h}s + 1\right)} \tag{29}$$

where $u_0(s) = \frac{C_s}{k_g k_T s}(R_a + sL_m)$ is the starting voltage (V), $w_h = \sqrt{\frac{R_a f + k_g^2 k_w k_T}{J L_m}}$ is the valve natural frequency(rad/sec) and $\delta_h = \frac{L_m f + R_a J}{2\sqrt{J L_m (R_a f + k_g^2 k_w k_T)}}$ is the valve damping ratio.

Equation (29) models the dynamic of the electro-hydraulic rotary spool valve. The first term in the numerator can be identified as the required voltage without friction torque and the second term gives the starting voltage due to the friction torque.

Equations (15), (27) can also be represented in state-space form. If the angular position and speed, the armature current are chosen as the state variables, the state-space model can be expressed as:

$$d\begin{pmatrix}\alpha \\ \dot{\alpha} \\ i\end{pmatrix} = \begin{bmatrix} 0 & 1 & 0 \\ 0 & -\frac{f}{J} & \frac{k_T k_g}{J} \\ 0 & -\frac{k_w k_g}{L_m} & -\frac{R_a}{L_m} \end{bmatrix}\begin{pmatrix}\alpha \\ \dot{\alpha} \\ i\end{pmatrix} + \begin{pmatrix} 0 \\ 0 \\ \frac{1}{L_m}\end{pmatrix} u_a - \begin{pmatrix} 0 \\ \frac{1}{J} \\ 0 \end{pmatrix} C_s \tag{30}$$

The output:

$$y = \begin{pmatrix} Q \\ \alpha \end{pmatrix} = \begin{bmatrix} k_p C_1 & 0 & 0 \\ 1 & 0 & 0 \end{bmatrix}\begin{pmatrix}\alpha \\ \dot{\alpha} \\ i\end{pmatrix} - \begin{pmatrix} k_p C_2 \alpha_s \\ 0 \end{pmatrix} \tag{31}$$

V. Discussion and Conclusions

A. Discussion

In section IV, the drag torque is investigated theoretically. In the real situation, the spool and the bush are not concentric and the flow between them usually is turbulent, especially for high rotary speed. This to some extent affects the performance of the valve. However, it is less significant compared to the flow induced torque.

Due to the clearance between the spool and bush and the asymmetry in geometry, under the action of the valve inner walls, the flow direction deviates from the theoretical one with a jet angle θ (as shown in Fig.9). If the flow in the valve is considered irrotational, according to the conservation of moment of momentum [20], the steady state flow torque acting the rotary spool can be described as:

$$T_{sf} \cdot dt = I_F w_F \tag{32}$$

$$I_F = \rho(Q \cdot dt)R^2 \tag{33}$$

$$w_F = \frac{v_\tau}{R} = \frac{Q}{A_{FA}} \sin\theta \tag{34}$$

where I_F is the moment inertia of the controlled flow, w_F is the angular velocity.

According the equations (1), (2), (25), (26) and (27), the steady flow induced torque can be expressed as:

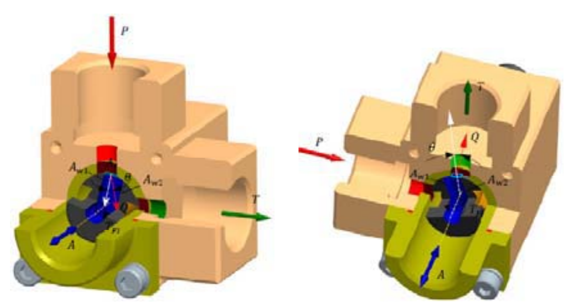
$$T_{sf} = \frac{\rho Q^2 R}{A_{FA}} \sin\theta = 4{C_q}^2 R^2 b\Delta p \sin\frac{(C_1\alpha - C_2\alpha_s)}{2} \sin\theta \tag{35}$$

From (35), it can be seen that the steady flow torque T_{sf} is strongly depends on the jet angle θ. There exist two special operation modes — fully open and close:

a) Fully open: when $|\alpha| = \alpha_r$, the flow direction points the center of the rotary spool and the jet angle $\theta = 0°$. There will be no steady state flow torque.

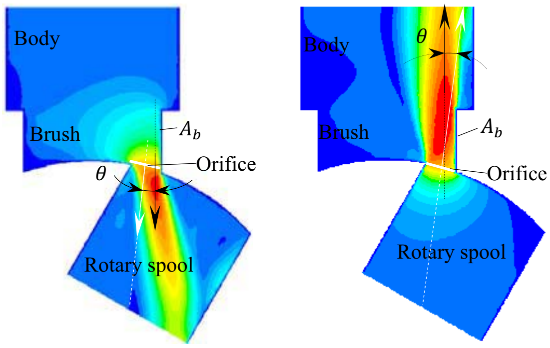
b) Fully close: when $|\alpha| \leq \alpha_s$, the flow through the valve is very small (only leakage). The jet angle approaches its biggest value θ_{max}. Therefore, the flow induced torque equals zero if the leakage flow is neglected.

Fig.10 shows the simulation results of the jet angle when the spool turns a angle α ($\alpha_s < |\alpha| < \alpha_r$), it can be concluded that the direction of the jet core deviates from the spool center. Further studies are necessary to determine the jet angle and steady state flow torque.



(a) Meter-in (P to A) (b) Meter-out (A to T)

Fig.9 Flow induced torque



(a) Meter-in (P to A) (b) Meter-out (A to T)

Fig.10 Jet angle

B. *Conclusions*

This research has presented the model of a novel rotary type electro-hydraulic valve, in which a spool driven by a DC motor is rotatably fitted in a valve bush. The flow area is controlled by the input signal of the DC motor, thus controlling the flow rate. A mathematical model is derived and a detailed analysis of the drag torque is described.

Future work will include the following:

1) Prototyping of the 3-way proportional valve, as shown in Fig.1.

2) Experimental and numerical studies to evaluate the performance in terms of the required torque, static/dynamic performance.

3) Optimization of the design based on experimental and numerical findings.

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REFERENCES

[1] A.B.Zoss, H.Kazerooni, and A.Chu, "Biomechanical design of the Berkeley lower extremity exoskeleton [BLEEX]," IEEE/ASME Transactions on Mechatronics, Vol. 11, No. 2, pp. 128-138, 2006.

[2] Raytheon, "The Exoskeleton: Advanced Robotics," Available at: http://www.raytheon.com/newsroom/technology/rtn08_exoskeleton

[3] S.H.Hyon, J.G.Hale, G.Cheng, "Full-Body Compliant Human–Humanoid Interaction: Balancing in the Presence of Unknown External Forces", IEEE Transactions on Robotics and Automation, vol. 23, Issue 5, pp. 884-898, 2007.

[4] S.Hyon, S.Abe, and T.Emura, "Development of a biologically inspired biped robot KenkenII," in Japan-France Congress on Mechatronics & 4th Asia Europe Congress on Mechatronics, pp. 404-409, Sept.2003.

[5] R.Playter, M.Buehler, and M.Raibert, "BigDog," Proceedings of SPIE 6230, pp. 896-901, April 2006.

[6] M.Buehler, R.Playter, and M.Raibert, "Robots Step Outside," Int. Symp. Adaptive Motion of Animals and Machines [AMAM], Ilmenau, Germany, Sept 2005.

[7] M.Raibert, K.Blankespoor, G.Nelson, R.Playter and the Bigdog Team, "BigDog, the Rough-Terrain Quadruped Robot," Proceeding of the 17th IFAC World Congress, pp. 10822-10825, 2008.

[8] Boston Dynamics, "Petman – BigDog gets a Big Brother," Available at: http://www.bostondynamics.com/robot_petman.html.

[9] C.Semini, N.G.Tsagarakis, B.Vanderborght, Y.S.Yang and D.G. Caldwell, "HyQ – hydraulically actuated quadruped robot: Hopping leg prototype," IEEE/RAS Int. Conf. on Biomedical Robotics and Biomechatronics (Biorob), pp.593-599, 2008.

[10] E.Guglielmino, C.Semini, Y.S.Yang, D.G.Caldwell, H.Kogler and R.Scheidl "Energy efficient fluid power in autonomous legged robots," Proceedings of the ASME 2009 Dynamic Systems and Control Conference, USA, Oct. 2009.

[11] Y.S.Yang, C.Semini, N.G.Tsagarakis, E.Guglielmino, and D.G. Caldwell, "Leg Mechanisms for Hydraulically Actuated Robots," The 2009 IEEE/RSJ International Conference on Intelligent Robots and Systems, pp.4699-4675, USA, Oct.2009.

[12] G.Lambert, Defence R&D Canada - Suffield, Ralston ALTA (CAN), Collineo Inc. Boucherville QC. (CAN), "Micro-hydraulic toolkit report," Report to Department of National Defence, DRDC No. DRDC-SUFFIELD-CR-2008-212, 2006.

[13] H.E.Merritt, "Hydraulic control systems," John Wiley and Sons, New York, 1967

[14] N.D.Manring, "Hydraulic control systems," John Wiley&Sons, Inc, New Jersey, 2005.

[15] D.McLoy and H. R. Martin "*Control of fluid power: analysis and design*," Wiley, 1980.

[16] S.C.John, "Fluid power circuits and controls," CRC Press, 2002.

[17] Moog, "Technical data E024 SUB miniature servo valve," Available at: http://www.moog.com/literature/ICD/E024_(LA_Series)_datasheet_EUROPE.pdf

[18] Y.S.Yang, E.Guglielmino, G.Pane, C.Semini, G.D.Caldwell, "Servovalvola rotativa, particolarmente di tipo idraulico (Rotary servo valve, particularly of hydraulic type)," Italian patent, No: TO2009A000271, 2009.

[19] W.Leonhard, "Control of electrical drives," Springer, Verlag Berlin Herdelberg New York, 2001.

[20] B.R.Munson, D.F.Young and T.F.Okiishi, "Fundamentals of fluid mechanics," John Wiley and Sons, Now York, 4th Edition, 2002.

[21] M.Gawlinski, "Friction and wear of elastomer seals," Archives of Civil and Mechanical Engineering, Vol. VII, No. 4, pp. 57-67, 2007.

[22] D.M.Goerres, "Torque analysis of rotary seals in HEES," Proc. Of 1st FPNI-PhD Symp, pp. 325-337, 2000.