

# 负载敏感变量泵结构建模与性能分析

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**摘要:** 作为现代农业装备液压系统关键零部件, 负载敏感变量泵为农业绿色生产提供了保障。为深入研究负载敏感变量泵的工作性能, 该文重点分析了其内部机械结构和工作机理, 充分考虑了各运动部件的有效行程范围, 应用现代控制理论状态空间法建立了基于边界条件的负载敏感变量泵非线性数学模型, 基于 Matlab/Simulink 软件, 采用四阶龙格-库塔算法对其稳态和动态性能进行了仿真分析, 并搭建闭心式负载敏感液压系统试验平台, 完成其性能试验, 通过对比分析负载敏感变量泵动态特性试验与仿真结果, 得到负载补偿压力误差约为 0.1 MPa, 验证了负载敏感变量泵非线性数学模型的正确性。试验结果表明: 负载敏感变量泵输出流量和压力可实时与负载相适应, 补偿压力约为 1.5 MPa, 可有效提高液压系统效率, 减少系统发热, 满足现代农业装备作业机组的田间作业需求。

**关键词:** 泵; 计算机仿真; 农业装备; 负载敏感; 变量泵; 非线性建模; 边界条件

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## 0 引言

近年来, 随着中国农业生产机械化、自动化水平不断提高, 农机装备逐步向大型化、复合化、智能化方向发展<sup>[1-3]</sup>。液压传动系统以其质量功率比大、调速范围广、低速稳定性好、易于布局等优点, 在农机产业发展进程中得到了广泛应用<sup>[4-5]</sup>。以大型农机底盘静液压驱动系统<sup>[6-8]</sup>和重型拖拉机电液提升器<sup>[9]</sup>为例, 前者可有效提高驾驶员操作舒适性, 降低劳动强度; 后者可显著提高拖拉机悬挂作业机组<sup>[10-13]</sup>的田间作业质量和作业效率。

国外农机装备整机及作业机组配套液压系统为适应田间作业环境复杂、负载波动大、行驶速度低等特点, 广泛采用闭心式负载敏感液压系统回路<sup>[14-16]</sup>, 该回路主要由变量泵、液压阀、液压执行机构等组成, 可根据实际负载工况进行压力和流量补偿, 最大限度降低液压系统的功率损失, 减少系统发热, 达到节能环保的目的。而国内由于缺乏液压关键零部件自主研发能力, 农机装备液压系统多数仍采用开式定量泵液压系统回路, 降低了液压系统工作效率, 已难以适应现代化农业生产节

能增效的发展要求。

目前, 国内学者针对农机装备液压系统核心元件及关键技术的研究尚处于起步阶段, 且主要集中在液压阀仿真优化及特性分析方面<sup>[17-21]</sup>, 而在液压泵建模仿真及性能试验方面研究相对较少<sup>[22-24]</sup>。为此, 本文拟以负载敏感变量泵为研究对象, 在对比分析国内外现有液压泵建模方法<sup>[25-30]</sup>, 结合变量泵内部结构和工作原理, 并充分考虑变量泵非线性因素及流量调节机构有效行程范围的基础上, 建立基于边界条件的负载敏感变量泵非线性数学模型, 并进行仿真分析与试验研究。

## 1 负载敏感变量泵数学建模

### 1.1 负载敏感变量泵结构及工作原理

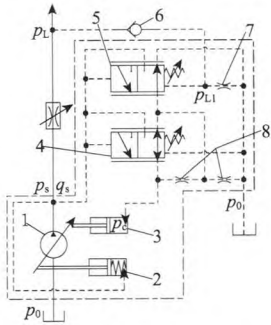
负载敏感变量泵液压系统原理图如图 1 所示, 主要由斜盘式柱塞泵 1、有弹簧变量机构柱塞缸 2、无弹簧变量机构柱塞缸 3、压力控制阀 4、流量控制阀 5、负载反馈单向阀 6、泄压阻尼孔 7 和减震阻尼孔 8 组成, 额定压力为 25 MPa, 公称流量为 68 L/min。由图 1 可知, 当系统压力低于压力控制阀调定压力时, 压力控制阀右位工作, 变量泵通过流量控制阀调整斜盘倾角, 为系统提供所需流量, 变量泵出口压力始终高出负载压力一定值; 当系统压力高于压力控制阀调定压力时, 压力控制阀左位工作, 切断流量控制阀与无弹簧变量机构柱塞缸之间的油路, 变量泵输出的高压油进入无弹簧变量机构柱塞缸的无杆腔, 使变量泵斜盘倾角变小, 直至接近零排量, 满足变量泵在超载工况下输出高压小流量的功能需求, 减少了液压系统的功率损失。

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1. 斜盘式柱塞泵 2. 有弹簧变量机构柱塞缸 3. 无弹簧变量机构柱塞缸 4. 压力控制阀 5. 流量控制阀 6. 负载反馈单向阀 7. 泄压阻尼孔 8. 减震阻尼孔

1. Swash plate piston pump 2. Plunger cylinder with spring variable mechanism 3. Plunger cylinder without spring variable mechanism 4. Pressure control valve 5. Flow control valve 6. Load-sensing check valve 7. Pressure relief damping orifice 8. Shock absorbing damping orifice

注： $p_L$  为负载压力，Pa； $p_{L1}$  为流量控制阀低压控制油腔压力，Pa； $p_0$  为回油压力，Pa； $p_s$  为变量泵出口压力，Pa； $q_s$  为变量泵出口流量， $m^3 \cdot s^{-1}$ ； $p_c$  为无弹簧变量机构柱塞缸无杆腔油液压力，Pa。

Note:  $p_L$  is the load pressure, Pa;  $p_{L1}$  is the pressure of oil chamber with low pressure control of the flow control valve, Pa;  $p_0$  is the return oil pressure, Pa;  $p_s$  is the variable pump outlet pressure, Pa;  $q_s$  is the variable pump outlet flow,  $m^3 \cdot s^{-1}$ ;  $p_c$  is the rodless chamber oil pressure of the plunger cylinder without spring variable mechanism.

图 1 负载敏感变量泵液压原理图

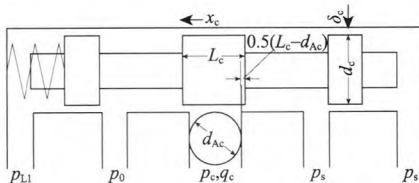
Fig.1 Hydraulic schematic diagram of load-sensing variable pump

## 1.2 负载敏感变量泵数学模型的建立

### 1.2.1 流量控制阀数学模型

#### 1) 主阀口的压力-流量方程

流量控制阀为正重叠双边滑阀，即有 2 个控制节流口，其内部结构简图如图 2 所示，假定流入无弹簧变量机构柱塞缸的流量为负，则：



注： $x_c$  为流量控制阀阀芯位移，m； $L_c$  为主阀芯控制台肩宽度，m； $d_{Ac}$  为流量控制阀通往无弹簧变量机构柱塞缸的油孔直径，m； $\delta_c$  为阀芯与阀套的径向配合间隙，m； $d_c$  为流量控制阀阀芯直径，m； $q_c$  为变量泵流量控制阀的流量， $m^3 \cdot s^{-1}$ ；其他符号含义见图 1。

Note:  $x_c$  is the spool displacement of flow control valve, m;  $L_c$  is the width of main spool land, m;  $d_{Ac}$  is the oil hole diameter from flow control valve to plunger cylinder without spring variable mechanism, m;  $\delta_c$  is the radial clearance between valve spool and valve sleeve, m;  $d_c$  is the spool diameter of flow control valve, m;  $q_c$  is the flow of variable pump flow control valve,  $m^3 \cdot s^{-1}$ ; the meanings of other symbols are as shown in Fig.1.

图 2 流量控制阀内部结构简图

Fig.2 Internal structure diagram of flow control valve

$$q_c = \begin{cases} C_{dc} \cdot A_c(-x_c) \cdot \sqrt{\frac{2}{\rho}(p_c - p_0)}, & x_c < -\frac{L_c - d_{Ac}}{2} \\ 0, & -\frac{L_c - d_{Ac}}{2} \leq x_c \leq \frac{L_c - d_{Ac}}{2} \\ -C_{dc} \cdot A_c(x_c) \cdot \sqrt{\frac{2}{\rho}(p_s - p_c)}, & x_c > \frac{L_c - d_{Ac}}{2} \end{cases} \quad (1)$$

万方数据

式中  $q_c$  为变量泵流量控制阀的流量， $m^3/s$ ； $C_{dc}$  为流量控制阀节流口流量系数； $\rho$  为油液密度， $kg/m^3$ ； $d_{Ac}$  为流量控制阀通往无弹簧变量机构柱塞缸的油孔直径，m； $L_c$  为主阀芯控制台肩宽度，m； $x_c$  为流量控制阀阀芯位移，向左为正方向，原点取重叠区中点，m； $x_{cm1}$ 、 $x_{cm2}$  分别为阀芯左、右最大位移量，m； $p_s$  为变量泵出口压力，Pa； $p_c$  为无弹簧变量机构柱塞缸无杆腔油液压力，Pa； $p_0$  为回油压力，Pa； $A_c(x_c)$  为流量控制阀节流口通流截面积， $m^2$ 。

节流口由阀芯对称布置的 2 个圆形孔构成，如图 3 所示，其通流截面积及其对阀芯位移  $x_c$  的导数按下式计算：

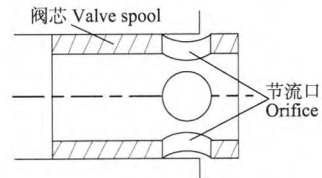


图 3 弓形节流口示意图

Fig.3 Schematic diagram of arcuate orifice

$$\begin{cases} A_c(x_c) = \frac{1}{2} d_{Ac}^2 \left[ \arccos \left( 1 - 2 \frac{x_c - (L_c - d_{Ac})/2}{d_{Ac}} \right) - 2 \left( 1 - 2 \frac{x_c - (L_c - d_{Ac})/2}{d_{Ac}} \right) \sqrt{\frac{x_c - (L_c - d_{Ac})/2}{d_{Ac}} - \left( \frac{x_c - (L_c - d_{Ac})/2}{d_{Ac}} \right)^2} \right] \\ \frac{dA_c(x_c)}{dx_c} = 4d_{Ac} \sqrt{\frac{x_c - (L_c - d_{Ac})/2}{d_{Ac}} - \left( \frac{x_c - (L_c - d_{Ac})/2}{d_{Ac}} \right)^2} \end{cases} \quad (2)$$

#### 2) 流量控制阀阀芯的力平衡方程

流量控制阀阀芯受力包括：阀芯两端油液压力产生的驱动力、惯性力、黏性阻尼力、弹簧力、稳态液动力（始终指向使流量控制阀阀口趋于关闭的方向）、瞬态液动力（油液流入流量控制阀阀腔时，瞬态液动力起正阻尼作用，流出流量控制阀阀腔时，油液起负阻尼作用）、干摩擦力（由于在阀芯上开有多条均压槽，液压卡紧力很小，干摩擦力可忽略不计）等。

$$\frac{\pi d_c^2}{4} (p_s - p_{L1}) = m_c \cdot \ddot{x}_c + B_c \cdot \dot{x}_c + F_{cs} +$$

$$F_{ct} + k_c(x_c + x_{c0}), -x_{cm1} < x_c < x_{cm2} \quad (3)$$

式中  $d_c$  为流量控制阀阀芯直径，m； $x_{c0}$  为流量控制阀弹簧的预压缩量，m； $p_{L1}$  为流量控制阀低压控制油腔压力，Pa； $m_c$  为流量控制阀阀芯质量，kg； $k_c$  为流量控制阀弹簧刚度，N/m； $B_c$  为阀芯运动阻尼系数，N·s/m； $F_{cs}$  为阀芯所受稳态液动力，N； $F_{ct}$  为阀芯所受瞬态液动力，N。

$$B_c = \frac{\pi \mu \cdot L_{cv} \cdot d_c}{\delta_c} \quad (4)$$

式中  $\mu$  为液压油动力黏度, Pa·s;  $L_{cv}$  为流量控制阀阀芯密封长度, m;  $\delta_c$  为阀芯与阀套的径向配合间隙, m。

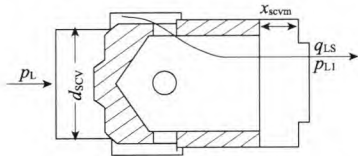
$$F_{cs} = \begin{cases} -2C_{cv} \cdot C_{dc} \cdot A_c(-x_c) \cdot (p_c - p_0) \cos \theta_c, & x_c \leq -\frac{L_c - d_{Ac}}{2} \\ 0, & -\frac{L_c - d_{Ac}}{2} \leq x_c \leq \frac{L_c - d_{Ac}}{2} \\ 2C_{cv} \cdot C_{dc} \cdot A_c(x_c) \cdot (p_s - p_c) \cos \theta_c, & x_c > \frac{L_c - d_{Ac}}{2} \end{cases} \quad (5)$$

式中  $\theta_c$  为阀芯节流口出射角度, (°);  $C_{cv}$  为节流口流速系数。

$$F_{ct} = \begin{cases} -L_{ct} \cdot C_{dc} \frac{dA_c(-x_c)}{dx_c} \sqrt{2\rho(p_c - p_0)} \dot{x}_c, & x_c \leq -\frac{L_c - d_{Ac}}{2} \\ 0, & -\frac{L_c - d_{Ac}}{2} \leq x_c \leq \frac{L_c - d_{Ac}}{2} \\ L_{ct} \cdot C_{dc} \frac{dA_c(x_c)}{dx_c} \sqrt{2\rho(p_s - p_c)} \dot{x}_c, & x_c > \frac{L_c - d_{Ac}}{2} \end{cases} \quad (6)$$

式中  $L_{ct}$  为流量控制阀的阻尼长度, m。

3) 负载反馈单向阀压力-流量特性方程  
负载反馈单向阀内部结构如图 4 所示。



注:  $x_{scvm}$  为单向阀阀芯最大位移量, m;  $d_{scv}$  为单向阀锥阀式阀座孔直径, m;  $q_{LS}$  为负载反馈单向阀流量,  $m^3 \cdot s^{-1}$ ; 其他符号含义见图 1。  
Note:  $x_{scvm}$  is the maximum displacement of check valve spool, m;  $d_{scv}$  is the cone-type valve seat hole diameter of check valve, m;  $q_{LS}$  is the flow of load-sensing check valve,  $m^3 \cdot s^{-1}$ ; the meanings of other symbols are as shown in Fig. 1.

图 4 负载反馈单向阀内部结构示意图

Fig.4 Internal structure of load-sensing check valve

$$q_{LS} = \begin{cases} \pi C_{dscv} \cdot d_{scv} \cdot x_{scvm} \cdot \sin \theta_{scv} \cdot \sqrt{\frac{2}{\rho}(p_L - p_{L1})}, & p_L \geq p_{L1} \\ 0, & p_L < p_{L1} \end{cases} \quad (7)$$

式中  $q_{LS}$  为负载反馈单向阀流量,  $m^3/s$ ;  $C_{dscv}$  为单向阀节流口流量系数;  $d_{scv}$  为单向阀锥阀式阀座孔直径, m;  $x_{scvm}$  为单向阀阀芯最大位移量, m;  $p_L$  为负载压力, Pa;  $\theta_{scv}$  为负载反馈单向阀阀芯半锥角, rad。

4) 负载敏感变量泵泄压阻尼孔流量-压力方程

$$q_{c3} = \frac{\pi d_{03}^2}{4} \cdot C_{d03} \cdot \sqrt{\frac{2}{\rho}(p_{L1} - p_0)} \quad (8)$$

式中  $q_{c3}$  为泄压阻尼孔流量,  $m^3/s$ ;  $C_{d03}$  为泄压阻尼孔流量系数;  $d_{03}$  为泄压阻尼孔直径, m。

5) 负载反馈单向阀至变量泵泄压阻尼孔间油腔流量连续性方程

忽略流量控制阀阀芯与阀套配合间隙处的泄漏, 流入负载反馈单向阀至负载敏感变量泵泄压阻尼孔间油腔

的流量, 一部分补偿油腔内油液压缩量, 一部分补偿流量控制阀阀芯运动引起的油腔容积变化量, 其余部分经泄压阻尼孔流回油箱。

$$q_{LS} - q_{c3} = \frac{V_{L1} - \frac{\pi d_c^2}{4} x_c}{\beta_e} \dot{p}_{L1} - \frac{\pi d_c^2}{4} \dot{x}_c \quad (9)$$

式中  $\beta_e$  为油液体积弹性模量, Pa;  $V_{L1}$  为流量控制阀阀芯处于原点时, 负载反馈单向阀至泄压阻尼孔间油腔容积,  $m^3$ 。

1.2.2 负载敏感变量泵变量控制机构数学模型

1) 变量控制机构油腔流量连续性方程

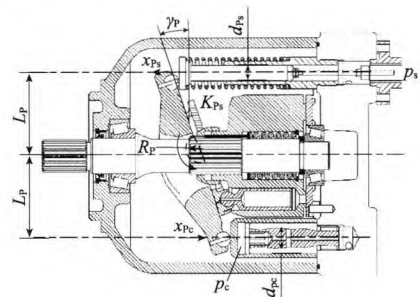
流入无弹簧变量机构柱塞缸控制油腔的流量除推动柱塞运动外, 还用来补偿油液压缩量以及通过减震阻尼孔漏入泵腔的流量。

$$q_c = \frac{\pi d_{pc}^2}{4} \dot{x}_{pc} - \left( \frac{\pi^3 d_{pp}^8}{12544 \rho \cdot \mu \cdot l_{pp}} \right)^{1/3} (p_c - p_0)^{2/3} - \frac{V_c}{\beta_e} \cdot \dot{p}_c \quad (10)$$

式中  $V_c$  为无弹簧柱塞缸及油道的总容积,  $m^3$ ;  $d_{pc}$  为无弹簧柱塞缸柱塞直径, m;  $x_{pc}$  为无弹簧柱塞缸柱塞位移量, m;  $l_{pp}$  为减震阻尼孔长度, m;  $d_{pp}$  为减震阻尼孔直径, m。

2) 变量控制机构的力矩平衡方程

变量泵结构如图 5 所示。由图 5 可知, 通过无弹簧和有弹簧变量机构柱塞缸控制变量泵斜盘倾角, 其中有弹簧变量机构柱塞缸无杆腔与变量泵出口油路相通, 其作用力指向使斜盘倾角增大的方向; 无弹簧变量机构柱塞缸无杆腔经压力控制阀右位与流量控制阀的控制节流口相通, 其作用力指向使斜盘倾角减小的方向, 变量泵斜盘受力分析图如图 6 所示。



注:  $L_p$  为有弹簧、无弹簧变量机构柱塞缸轴线至斜盘转动中心的垂直距离, m;  $\gamma_p$  为变量泵斜盘倾角, rad;  $x_{ps}$  为有弹簧变量机构柱塞缸柱塞位移, m;  $x_{pc}$  为无弹簧变量机构柱塞缸柱塞位移, m;  $d_{ps}$  为有弹簧变量机构柱塞缸柱塞直径, m;  $d_{pc}$  为无弹簧变量机构柱塞缸柱塞直径, m;  $K_{ps}$  为变量机构压紧弹簧刚度,  $N \cdot m^{-1}$ ;  $R_p$  为变量泵斜盘支撑轴颈的半径, m; 其他符号含义见图 1。

Note:  $L_p$  is the vertical distance from the piston cylinder axis to the swash plate center of rotation, m;  $\gamma_p$  is the swash plate angle of the variable pump, rad;  $x_{ps}$  is the plunger displacement in the plunger type cylinder with a spring, m;  $x_{pc}$  is the plunger displacement in the plunger type cylinder without a spring, m;  $d_{ps}$  is the plunger diameter in the plunger type cylinder with a spring, m;  $d_{pc}$  is the plunger diameter in the plunger type cylinder without a spring, m;  $K_{ps}$  is the stiffness of the compressed spring in the variable displacement mechanism,  $N \cdot m^{-1}$ ;  $R_p$  is the radius of the supporting shaft neck, m; the meanings of other symbols are as shown in Fig. 1.

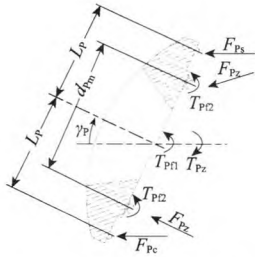
图 5 变量泵结构原理图

Fig.5 Schematic diagram of variable pump structure

负载敏感变量泵变量机构力矩平衡方程:

$$L_P \left( \frac{\pi d_{Ps}^2}{4} p_s - \frac{\pi d_{Pc}^2}{4} p_c \right) = J_P \cdot \ddot{\gamma}_P + T_{Pz} + \text{sign}(\dot{\gamma}_P) \cdot (T_{Pf1} + T_{Pf2}) + L_P [m_{Ps} \cdot \ddot{x}_{Ps} + m_{Pc} \cdot \ddot{x}_{Pc} + \frac{\pi \mu \cdot L_{Psv} \cdot d_{Ps}}{\delta_{Ps}} \dot{x}_{Ps} + \frac{\pi \mu \cdot L_{Pcv} \cdot d_{Pc}}{\delta_{Pc}} \dot{x}_{Pc} - K_{Ps} (x_{Ps0} + L_P \cdot \gamma_{Pmax} - x_{Ps})], \gamma_{Pmin} < \gamma_P < \gamma_{Pmax} \quad (11)$$

式中  $L_P$  为有、无弹簧变量机构柱塞缸轴线至斜盘转动中心的垂直距离, m;  $d_{Ps}$  为有弹簧柱塞缸柱塞直径, m;  $J_P$  为变量泵斜盘和回程盘对旋转中心的转动惯量,  $\text{kg} \cdot \text{m}^2$ ;  $\gamma_P$  为变量泵斜盘倾角, rad;  $T_{Pz}$  为柱塞组轴向惯性力转矩,  $\text{N} \cdot \text{m}$ ;  $T_{Pf1}$ 、 $T_{Pf2}$  分别为变量泵斜盘轴颈处的摩擦转矩、柱塞球铰上的摩擦转矩,  $\text{N} \cdot \text{m}$ ;  $m_{Ps}$ 、 $m_{Pc}$  分别为有、无弹簧柱塞缸质量, kg;  $\gamma_{Pmin}$ 、 $\gamma_{Pmax}$  分别为变量泵斜盘的最小、最大倾角, rad;  $K_{Ps}$  为变量机构压紧弹簧刚度,  $\text{N/m}$ ;  $x_{Ps}$  为有弹簧变量机构柱塞缸柱塞位移,  $x_{Ps} = L_P \cdot \tan(\gamma_P - \gamma_{Pmax})$ , m;  $x_{Ps0}$  为斜盘在最大倾角时变量机构弹簧预压缩量, m;  $L_{Psv}$ 、 $L_{Pcv}$  分别为有、无弹簧柱塞缸的密封长度, m;  $\delta_{Ps}$ 、 $\delta_{Pc}$  分别为有、无弹簧柱塞缸内孔与柱塞的配合间隙, m。



注:  $d_{pm}$  为柱塞分布圆直径, m;  $F_{Pc}$  为无弹簧变量机构柱塞缸对斜盘的支反力, N;  $F_{Ps}$  为有弹簧变量机构柱塞缸对斜盘的支反力, N;  $F_{Pz}$  为柱塞球铰支反力, N;  $T_{Pf1}$  为改变斜盘倾角时变量泵斜盘轴颈处的摩擦转矩,  $\text{N} \cdot \text{m}$ ;  $T_{Pf2}$  为改变斜盘倾角时柱塞球铰上的摩擦转矩,  $\text{N} \cdot \text{m}$ ;  $T_{Pz}$  为变量泵柱塞组轴向惯性力所产生的转矩,  $\text{N} \cdot \text{m}$ ; 其他符号含义见图 5。

Note:  $d_{pm}$  is the plunger distribution diameter, m;  $F_{Pc}$  is the support reaction between the swash plate and the plunger type cylinder without a spring, N;  $F_{Ps}$  is the support reaction between the swash plate and the plunger type cylinder with a spring, N;  $F_{Pz}$  is the support reaction of the plunger spherical hinge, N;  $T_{Pf1}$  is the frictional torque of the swash plate shaft neck,  $\text{N} \cdot \text{m}$ ;  $T_{Pf2}$  is the frictional torque of the plunger spherical hinge,  $\text{N} \cdot \text{m}$ ;  $T_{Pz}$  is the torque of the axial inertia force of the plunger group,  $\text{N} \cdot \text{m}$ ; the meanings of other symbols are as shown in Fig. 5.

图 6 变量泵斜盘受力分析图

Fig.6 Force analysis of variable pump swash plate

$$\begin{cases} T_{Pf1} = k_{Pf1} \cdot R_P \left( \frac{\pi d_P^2 \cdot z_P}{8 \cos \gamma_P} p_s + \frac{\pi d_{Ps}^2}{4} p_s + \frac{\pi d_{Pc}^2}{4} p_c + K_{Ps} (x_{Ps0} + L_P \cdot \gamma_{Pmax} - L_P \cdot \gamma_P) \right) \\ T_{Pf2} = k_{Pf2} \frac{\pi d_P^2 \cdot z_P \cdot r_{Pz} (z_P + 1)}{8 \cos \gamma_P} p_s \end{cases} \quad (12)$$

式中  $R_P$  为变量泵斜盘支撑轴颈的半径, m;  $r_{Pz}$  为柱塞铰接球头半径, m;  $k_{Pf1}$ 、 $k_{Pf2}$  分别为变量泵斜盘轴颈处、变量泵柱塞球铰处摩擦系数;  $z_P$  为变量泵柱塞数,  $z_P=9$ , 假定泵工作中平均有一半柱塞处于泵油状态;  $d_P$  为变量泵的柱塞直径, m。

$$T_{Pz} = - \frac{z_P \cdot m_{Pz} \cdot d_{Pm}^2}{8 \cos^3 \gamma_P} \left[ (2\pi n_P)^2 \sin \gamma_P - \frac{2 \sin \gamma_P}{\cos^2 \gamma_P} \dot{\gamma}_P^2 - \frac{1}{\cos \gamma_P} \ddot{\gamma}_P \right] \quad (13)$$

式中  $m_{Pz}$  为单个柱塞及其滑履的质量, kg;  $n_P$  为变量泵的转速, r/s;  $d_{Pm}$  为柱塞分布圆直径, m。

将式 (12) ~ (13) 代入式 (11) 可得:

$$L_P (A_{Ps} \cdot p_s - A_{Pc} \cdot p_c) = J_{Pe} \cdot \ddot{\gamma}_P + B_{Pel} \cdot \dot{\gamma}_P + B_{Pe2} \cdot \dot{\gamma}_P^2 + G_{Pel} \cdot \gamma_P - T_{Pe10} + \text{sign}(\dot{\gamma}_P) \cdot T_{Pf} (p_c, p_s, \gamma_P) \quad (14)$$

其中:

$$\begin{cases} T_{Pf} (p_c, p_s, \gamma_P) = K_{Pfs} \cdot p_s + K_{Pfc} \cdot p_c - G_{Pe2} \cdot \gamma_P + T_{Pe20} \\ A_{Ps} = \frac{\pi d_{Ps}^2}{4} & A_{Pc} = \frac{\pi d_{Pc}^2}{4} \\ G_{Pel} = L_P^2 \cdot K_{Ps} & G_{Pe2} = k_{Pf1} \cdot R_P \cdot K_{Ps} \cdot L_P \\ K_{Pfs} = k_{Pf1} \cdot R_P \left( \frac{\pi d_P^2 \cdot z_P}{8 \cos \gamma_P} + \frac{\pi d_{Ps}^2}{4} \right) + \\ & k_{Pf2} \frac{\pi d_P^2 \cdot z_P \cdot r_{Pz} (z_P + 1)}{8 \cos \gamma_P} \\ K_{Pfc} = k_{Pf1} \cdot R_P \cdot A_{Pc} \\ J_{Pe} = J_P + L_P^2 (m_{Ps} + m_{Pc}) + \frac{z_P \cdot m_{Pz} \cdot d_{Pm}^2}{8 \cos^4 \gamma_P} \\ B_{Pel} = \pi \mu \cdot L_P^2 \left( \frac{L_{Psv} \cdot d_{Ps}}{\delta_{Ps}} + \frac{L_{Pcv} \cdot d_{Pc}}{\delta_{Pc}} \right) \\ B_{Pe2} = \frac{z_P \cdot m_{Pz} \cdot d_{Pm}^2 \sin \gamma_P}{4 \cos^5 \gamma_P} \\ T_{Pe10} = L_P \cdot K_{Ps} (x_{Ps0} + L_P \cdot \gamma_{Pmax}) + \\ & \frac{z_P \cdot m_{Pz} \cdot d_{Pm}^2 \pi^2 \sin \gamma_P}{2 \cos^3 \gamma_P} n_P^2 \\ T_{Pe20} = k_{Pf1} \cdot R_P \cdot K_{Ps} (x_{Ps0} + L_P \cdot \gamma_{Pmax}) \end{cases} \quad (15)$$

### 1. 2. 3 液压系腔流量连续性方程

负载敏感变量泵腔活塞排出的流量, 除了用于负载流量输出外, 一部分用于推动有弹簧柱塞缸运动, 一部分通过缸体的泄漏流回油箱, 还有一部分用于补偿变量泵排油腔及其连接管道内油液的压缩量。

$$q_s = \begin{cases} V_P \cdot n_P - A_{Ps} \cdot \dot{x}_{Ps} - C_{IP} (p_s - p_0) - \frac{V_s}{\beta_e} \dot{p}_s, & x_c \leq \frac{L_c - d_{Ac}}{2} \\ V_P \cdot n_P - C_{dc} \cdot A_c (x_c) \sqrt{\frac{2}{\rho} (p_s - p_c)} - \\ & A_{Ps} \cdot \dot{x}_{Ps} - C_{IP} (p_s - p_0) - \frac{V_s}{\beta_e} \dot{p}_s, & x_c > \frac{L_c - d_{Ac}}{2} \end{cases} \quad (16)$$

式中  $q_s$  为变量泵出口流量,  $\text{m}^3/\text{s}$ ;  $V_P$  为变量泵的排量,  $\text{m}^3/\text{r}$ ;  $C_{IP}$  为变量泵的泄漏系数,  $\text{m}^3/(\text{Pa} \cdot \text{s})$ ;  $V_s$  为变量泵排油腔及连接管道总容积,  $\text{m}^3$ 。

$$V_P = \frac{\pi d_P^2}{4} \cdot z_P \cdot d_{Pm} \cdot \tan \gamma_P \quad (17)$$

### 1. 2. 4 负载敏感变量泵状态方程

根据前面建立的数学模型, 选取状态量:  $x_1 = p_c$ ;

$$x_2 = p_s; x_3 = x_c; x_4 = \dot{x}_c; x_5 = p_{L1}; x_6 = \gamma_P; x_7 = \dot{\gamma}_P。$$

建立如下状态方程组:

$$\dot{x}_1 = \begin{cases} \frac{\beta_e}{V_c} \left[ C_{dc} \cdot A_c(-x_3) \sqrt{\frac{2}{\rho}(x_1 - p_0)} + \frac{\pi d_{pc}^2 \cdot L_P}{4} x_7 \right. \\ \left. - \left( \frac{\pi^3 d_{pp}^8}{12544 \rho \cdot \mu \cdot l_{pp}} \right)^{1/3} (x_1 - p_0)^{2/3} \right], & x_3 < -\frac{L_c - d_{Ac}}{2} \\ \frac{\beta_e}{V_c} \left[ \frac{\pi d_{pc}^2 \cdot L_P}{4} x_7 - \left( \frac{\pi^3 d_{pp}^8}{12544 \rho \cdot \mu \cdot l_{pp}} \right)^{1/3} (x_1 - p_0)^{2/3} \right], & \frac{L_c - d_{Ac}}{2} \leq x_3 \leq \frac{L_c - d_{Ac}}{2} \\ \frac{\beta_e}{V_c} \left[ -C_{dc} \cdot A_c(x_3) \sqrt{\frac{2}{\rho}(x_2 - x_1)} + \frac{\pi d_{pc}^2 \cdot L_P}{4} x_7 - \right. \\ \left. \left( \frac{\pi^3 d_{pp}^8}{12544 \rho \cdot \mu \cdot l_{pp}} \right)^{1/3} (x_1 - p_0)^{2/3} \right], & x_3 > \frac{L_c - d_{Ac}}{2} \end{cases}$$

$$\dot{x}_2 = \begin{cases} \frac{\beta_e}{V_s} \left[ \frac{\pi d_p^2}{4} \cdot z_p \cdot n_p \cdot d_{pm} \tan x_6 - q_s \right. \\ \left. - A_{ps} \cdot L_P x_7 - C_{IP}(p_s - p_0) \right], & x_3 \leq -\frac{L_c - d_{Ac}}{2} \\ \frac{\beta_e}{V_s} \left[ \frac{\pi d_p^2}{4} \cdot z_p \cdot n_p \cdot d_{pm} \tan x_6 - q_s - C_{dc} \cdot \right. \\ \left. A_c(x_3) \sqrt{\frac{2}{\rho}(x_2 - x_1)} - A_{ps} \cdot L_P \cdot x_7 - \right. \\ \left. C_{IP}(p_s - p_0) \right], & x_3 > \frac{L_c - d_{Ac}}{2} \end{cases}$$

$$\dot{x}_3 = \begin{cases} 0, & x_3 = -x_{cm1} \text{ 且 } \frac{\pi d_c^2}{4} (x_2 - x_5) \leq k_c (x_{c0} - x_{cm1}), \\ \text{或 } x_3 = x_{cm2} \text{ 且 } \frac{\pi d_c^2}{4} (x_2 - x_5) \geq k_c (x_{c0} + x_{cm2}), \\ \text{或 } x_3 = x_{cm2} \text{ 且 } \frac{\pi d_c^2}{4} (x_2 - x_5) \geq k_c (x_{c0} + x_{cm2}) \\ x_4, & \text{其他} \end{cases}$$

$$\dot{x}_4 = \begin{cases} 0, & x_3 = -x_{cm1} \text{ 且 } \frac{\pi d_c^2}{4} (x_2 - x_5) \leq k_c (x_{c0} - x_{cm1}), \text{ 或 } x_3 = x_{cm2} \text{ 且 } \frac{\pi d_c^2}{4} (x_2 - x_5) \geq k_c (x_{c0} + x_{cm2}) \\ \frac{1}{m_c} \left[ \frac{\pi d_c^2}{4} (x_2 - x_5) - \left( \frac{\pi \mu \cdot L_{cv} \cdot d_c}{\delta_c} - L_{ct} \cdot C_{dc} \frac{dA_c(-x_3)}{dx_3} \sqrt{\frac{2}{\rho}(x_1 - p_0)} \right) x_4 - k_c (x_3 + x_{c0}) + 2C_{cv} \cdot C_{dc} \cdot A_c(-x_3) \cdot (x_1 - p_0) \cos \theta_c \right], & x_3 = -x_{cm1} \text{ 且 } \frac{\pi d_c^2}{4} (x_2 - x_5) > k_c (x_{c0} - x_{cm1}), \text{ 或 } -x_{cm1} < x_3 < -\frac{L_c - d_{Ac}}{2} \\ \frac{1}{m_c} \left[ \frac{\pi d_c^2}{4} (x_2 - x_5) - \frac{\pi \mu \cdot L_{cv} \cdot d_c}{\delta_c} x_4 - K_c (x_3 + x_{c0}) \right], & -\frac{L_c - d_{Ac}}{2} \leq x_3 \leq \frac{L_c - d_{Ac}}{2} \\ \frac{1}{m_c} \left[ \frac{\pi d_c^2}{4} (x_2 - x_5) - \left( \frac{\pi \mu \cdot L_{cv} \cdot d_c}{\delta_c} + L_{ct} \cdot C_{dc} \frac{dA_c(x_3)}{dx_3} \sqrt{\frac{2}{\rho}(x_2 - x_1)} \right) x_4 - k_c (x_3 + x_{c0}) - 2C_{cv} \cdot C_{dc} \cdot A_c(x_3) \cdot (x_2 - x_1) \cos \theta_c \right], & x_3 = x_{cm2} \text{ 且 } \frac{\pi d_c^2}{4} (x_2 - x_5) < k_c (x_{c0} + x_{cm2}), \text{ 或 } \frac{L_c - d_{Ac}}{2} < x_3 < x_{cm2} \end{cases}$$

$$\dot{x}_5 = \begin{cases} \frac{\beta_e}{V_{L1} - \frac{\pi d_c^2}{4} x_3} \left[ \frac{\pi d_c^2}{4} x_4 + \pi d_{scv} \cdot C_{dscv} \cdot x_{scvm} \sin \theta_{scv} \cdot \sqrt{\frac{2}{\rho}(p_L - x_5)} - C_{d03} \frac{\pi d_{03}^2}{4} \sqrt{\frac{2}{\rho}(x_5 - p_0)} \right], & p_L \geq x_5 \\ \frac{\beta_e}{V_{L1} - \frac{\pi d_c^2}{4} x_3} \left[ \frac{\pi d_c^2}{4} x_4 - C_{d03} \frac{\pi d_{03}^2}{4} \sqrt{\frac{2}{\rho}(x_5 - p_0)} \right], & p_L < x_5 \end{cases}$$

$$\dot{x}_6 = \begin{cases} \left. \begin{aligned} & x_6 = \gamma_{Pmin} \text{ 且 } L_P(A_{ps} \cdot x_2 - A_{pc} \cdot x_1) + T_{Pe10} - G_{Pel} \cdot \gamma_{Pmin} > 0; \\ & \text{或 } x_6 = \gamma_{Pmax} \text{ 且 } L_P(A_{ps} \cdot x_2 - A_{pc} \cdot x_1) + T_{Pe10} - G_{Pel} \cdot \gamma_{Pmax} < 0; \\ & \text{或 } \gamma_{Pmin} < x_6 < \gamma_{Pmax} \end{aligned} \right\} \\ 0, & \text{其他} \end{cases}$$

$$\dot{x}_7 = \begin{cases} J_{Pe}^{-1} \left[ L_P(A_{ps} \cdot x_2 - A_{pc} \cdot x_1) + T_{Pe10} - B_{Pe1} \cdot x_7 - B_{Pe2} \cdot x_7^2 - G_{Pel} \cdot x_6 - \text{sign}(x_7) \cdot T_{Pf}(x_1, x_2, x_6) \right], \\ \left. \begin{aligned} & x_6 = \gamma_{Pmin} \text{ 且 } L_P(A_{ps} \cdot x_2 - A_{pc} \cdot x_1) + T_{Pe10} - G_{Pel} \cdot \gamma_{Pmin} > 0; \\ & \text{或 } x_6 = \gamma_{Pmax} \text{ 且 } L_P(A_{ps} \cdot x_2 - A_{pc} \cdot x_1) + T_{Pe10} - G_{Pel} \cdot \gamma_{Pmax} < 0, \text{ 或 } \gamma_{Pmin} < x_6 < \gamma_{Pmax}; \end{aligned} \right\} \\ 0, & \text{其他} \end{cases}$$

边界条件:

如果  $x_3 < -x_{cm1}$ , 则  $x_3 = -x_{cm1}$ ; 如果  $x_3 > x_{cm2}$ , 则

$$x_3 = x_{cm2};$$

如果  $x_6 < \gamma_{Pmin}$ , 则  $x_6 = \gamma_{Pmin}$ ; 如果  $x_6 > \gamma_{Pmax}$ , 则  $x_6 =$

万方数据

$\gamma_{Pmax}$ ;

如果  $x_3 = -x_{cm1}$  且  $x_4 < 0$ , 或  $x_3 = x_{cm2}$  且  $x_4 > 0$ , 则  $x_4 = 0$ ;

如果  $x_6 = \gamma_{Pmax}$  且  $x_7 > 0$ , 或  $x_6 = \gamma_{Pmin}$  且  $x_7 < 0$ , 则  $x_7 = 0$ 。

## 2 负载敏感变量泵仿真分析

### 2.1 负载敏感变量泵稳态特性仿真分析

基于 MATLAB/Simulink 建立了负载敏感变量泵仿真模型, 其主要参数值如表 1 所示, 给定负载反馈压力 10 MPa, 系统流量 30 L/min, 仿真时间为 2 s, 得到负载敏感变量泵出口压力和斜盘倾角稳态响应特性曲线分别如图 7a、7b 所示。

表 1 负载敏感变量泵主要参数

Table 1 Main parameters of load-sensing variable pump

参数 Parameters	数值 Value	参数 Parameters	数值 Value	参数 Parameters	数值 Value
流量控制阀 节流口流量 系数 $C_{dc}$	0.61	节流口流速 系数 $C_{cv}$	0.98	阀芯与阀套径向 配合间隙 $\delta_c/mm$	0.01
油液密度 $\rho/(kg \cdot m^{-3})$	900	油液体积弹 性模量 $\beta_e/Pa$	$9 \times 10^8$	流量控制阀阻 尼长度 $L_{cv}/mm$	10
流量控制阀 至无弹簧变 量柱塞缸油 孔直径 $d_{Ac}/mm$	4.3	无弹簧柱塞 缸柱塞直径 $d_p/m$	0.023	有弹簧柱塞缸柱 塞直径 $d_{ps}/m$	0.025
流量控制阀 阀芯直径 $d_c/mm$	8.2	主阀芯控制 台肩宽度 $L_c/mm$	4.5	变量泵柱塞直径 $d_p/m$	0.017
油液动力黏 度 $\mu/Pa \cdot s$	$4.14 \times 10^{-2}$	流量控制阀 弹簧刚度 $k_c/(N \cdot m^{-1})$	$3.5 \times 10^4$	变量泵斜盘和回 程盘对旋转中心 转动惯量 $J_p/(kg \cdot m^2)$	$6.2 \times 10^{-3}$
流量控制阀 阀芯密封长 度 $L_{cv}/mm$	15	变量机构压 紧弹簧刚度 $K_{ps}/(N \cdot m^{-1})$	$2.45 \times 10^4$	变量泵泄露系数 $C_{1P}/(m^3 \cdot s^{-1} \cdot Pa^{-1})$	$1.92 \times 10^{-14}$

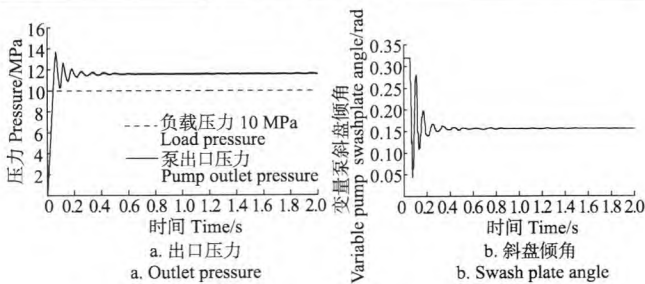


图 7 变量泵出口压力和斜盘倾角稳态响应特性

Fig.7 Steady state response characteristics of outlet pressure and swashplate angle of variable pump

由图 7a 可知, 负载敏感变量泵出口压力经小幅振荡后稳定在 11.6 MPa 左右, 补偿压力约为 1.6 MPa, 出口压力的超调量约为 17.2%, 调整时间约为 0.5 s, 该补偿压力可通过流量控制阀调压弹簧设定, 出口压力建立时间约为 0.06 s, 动态响应性能良好。由图 7b 可知, 负载敏感变量泵斜盘倾角经 0.5 s 左右振荡后稳定在 0.16 rad 左右, 结合变量泵输出流量理论公式可知, 变量泵输出流量与给定系统输入流量相符。

万方数据

### 2.2 负载敏感变量泵动态特性仿真分析

给定系统流量为 30 L/min, 仿真时间为 4 s, 得到负载敏感变量泵出口压力和斜盘倾角在负载反馈压力由 10~15 MPa 阶跃变化时的动态响应特性曲线分别如图 8a、8b 所示。

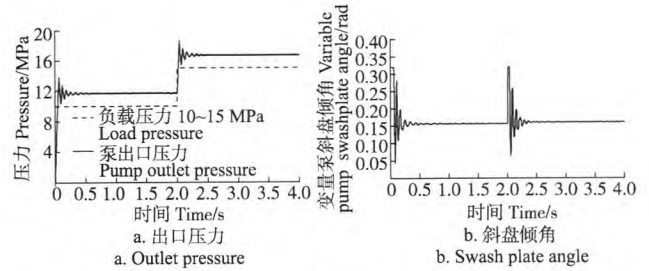


图 8 负载阶跃输入变量泵出口压力和斜盘倾角响应特性

Fig.8 Response characteristics of outlet pressure and swashplate angle of variable pump with load step input

由图 8a 可知, 变量泵出口压力建压时间约为 0.4 s, 稳定后达到 11.6 MPa 左右, 当负载反馈压力阶跃变化到 15 MPa 后, 变量泵出口压力随之增加, 建压时间约为 0.4 s, 超调量约为 12%, 稳定后达到 16.6 MPa 左右, 补偿压力约为 1.6 MPa, 负载敏感变量泵的动态压力补偿特性良好。

由图 8b 可知, 初始状态下变量泵斜盘倾角经一段时间振荡后, 稳定在 0.16 rad 左右, 调整时间约为 0.5 s, 当负载反馈压力阶跃变化时, 变量泵斜盘倾角迅速振荡后稳定在 0.16 rad 左右, 调整时间约为 0.5 s, 系统流量不受负载阶跃变化的影响, 变量泵具有良好的稳态流量输出特性。

给定负载反馈压力 10 MPa, 仿真时间为 4 s, 得到负载敏感变量泵出口压力和斜盘倾角在系统流量由 30~60 L/min 阶跃变化时的动态响应特性曲线分别如图 9a、9b 所示。

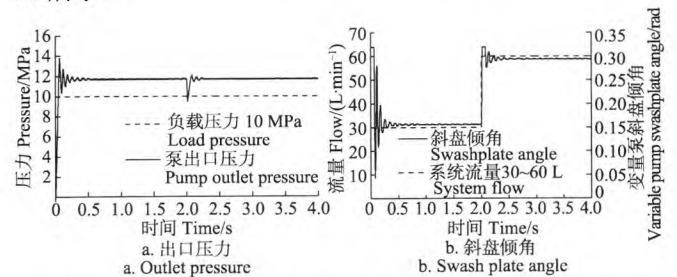


图 9 流量阶跃输入变量泵出口压力和斜盘倾角响应特性

Fig.9 Response characteristics of outlet pressure and swashplate angle of variable pump with flow step input

由图 9a 可知, 初始状态下变量泵出口压力的建压时间约为 0.5 s, 稳定后达到 11.6 MPa 左右; 当系统流量阶跃变化到 60 L/min 后, 变量泵出口压力迅速减小, 随即快速上升, 经一段时间振荡后稳定在 11.6 MPa 左右, 超调量约为 17.2%, 调整时间约为 0.4 s, 补偿压力约为 1.6 MPa, 变量泵出口压力不受系统流量阶跃变化的影响, 具有良好的稳压特性。

由图 9b 可知, 初始状态下变量泵斜盘倾角经一段时

间振荡后稳定在 0.16 rad 左右, 调整时间约为 0.5 s; 当系统流量阶跃变化到 60 L/min 后, 变量泵斜盘倾角迅速增大, 经一段时间振荡后稳定在 0.29 rad 左右, 调整时间约为 0.5 s, 变量泵动态流量输出特性良好。

综上仿真分析可知, 负载敏感变量泵可根据负载提供其所需的流量和压力, 有效降低了系统功率损失。

### 3 负载敏感变量泵试验

为了验证负载敏感变量泵数学模型和仿真分析的正确性, 搭建了闭心式负载敏感液压系统室内试验平台, 如图 10 所示。

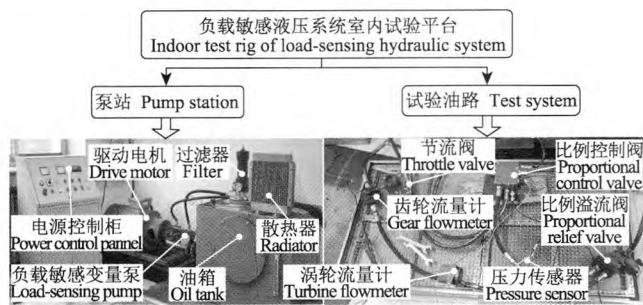


图 10 负载敏感液压系统室内试验平台

Fig.10 Indoor test rig of load-sensing hydraulic system

油压传感器采用德国米科 MIK-P300 型压力传感器, 其技术参数如表 2 所示, 响应时间约为 20 ms, 而变量泵液压系统压力控制响应时间约为 0.5 s, 可满足系统油液压力动态测量需求。智能变送仪用于实时显示油压传感器所采集的油压数值, 流量传感器采用 TLW-15G 型涡轮流量传感器, 压力范围 0~25 MPa, 量程为 0~100 L/min, 24 V 电源供电, 输出信号为 4~20 mA 电流信号。压力和流量信号可通过 NI 采集卡实时传输到 PC 机中, 并通过 LabVIEW 程序界面实时显示传感器输出信号变化曲线。

比例溢流阀选用华德液压生产的 DBEM2-30B/315YM 型锥阀式先导比例溢流阀, 通径为 25 mm, 允许通过的最大流量为 600 L/min, 可提供的最大开启压力为 31.5 MPa。可根据 VT-2000BS40G 型电液比例控制器无级调节比例溢流阀的开启压力。比例阀控制放大器采用 9~32 V 电源供电, 输入电压范围为 2.5~5 V, 输出比例线圈驱动电流范围为 0~1.2 A, 最大输出电流为 2 A。

表 2 油压传感器技术参数

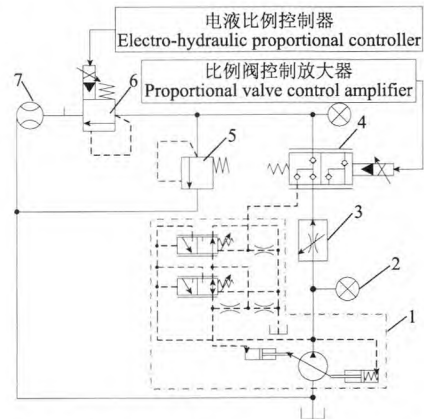
Table 2 Technical parameters of pressure sensor

参数 Parameters	数值 Value
量程 Range/MPa	0~30
激励电压 Excitation Voltage/V	24
输出信号 Output signal/mA	4~20
工作温度 Working temperature/°C	-45~85
精度等级 Accuracy class	±0.5%FS
过载等级 Overload level	200%FS
零点漂移 Zero drift	±0.1%FS

注: 表中 FS 表示满量程。  
Note: FS meanings full scale.

### 3.1 试验方案

图 11 为负载敏感变量泵性能试验方案原理图, 节流阀与负载敏感变量泵出口相连, 用于调节液压系统流量。在节流阀出口位置并联比例溢流阀可为比例控制阀提供所需的负载压力。同时, 负载压力经节流阀出口可反馈至泵流量控制阀进行压力补偿, 安全溢流阀用于液压系统过载保护, 开启压力为 20 MPa。其中, 比例溢流阀开启压力和比例控制阀阀芯开度可分别通过电液比例控制器和比例阀控制放大器进行实时控制。



1. 负载敏感变量泵 2. 油压传感器 3. 节流阀 4. 两位三通比例换向阀 5. 安全溢流阀 6. 比例溢流阀 7. 流量传感器  
1. Load-sensing variable pump 2. Oil pressure sensor 3. Throttle valve 4. Two position three-way proportional directional valve 5. Relief valve 6. Proportional relief valve 7. Flow sensor

图 11 负载敏感变量泵性能试验方案原理图

Fig.11 Schematic diagram of performance test of load-sensing variable pump

### 3.2 试验结果分析

设定比例控制阀输入电压为 4.7 V, 保持其阀口开度不变, 由电液比例控制器控制比例溢流阀开启压力在 10 s 时由 10 MPa 阶跃变化到 15 MPa, 得到变量泵出口压力、负载压力以及变量泵出口压力仿真值得动态响应特性曲线如图 12 所示。

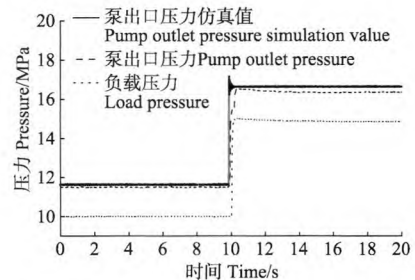


图 12 负载阶跃时变量泵出口压力响应特性曲线

Fig.12 Pressure curves of variable pump with step change in load

由图 12 可知, 负载压力在 10~15 MPa 之间阶跃变化时, 负载敏感变量泵建压时间约为 0.5 s, 变量泵出口压力与负载压力几乎同步变化, 由 11.5 MPa 阶跃变化到 16.5 MPa, 补偿压力约为 1.5 MPa, 与变量泵出口压力仿真曲线对比可知, 补偿压力的稳态误差约为 0.1 MPa, 主要受负载敏感液压系统试验管路油液压缩性及压力损失

的影响，负载敏感变量泵动态压力补偿特性良好，验证了负载敏感变量泵非线性数学模型的正确性。

设定比例溢流阀开启压力为 5 MPa，比例控制阀输入电压在 10 s 时由 4.3 V 阶跃变化到 4.7 V，得到变量泵出口压力、负载压力以及系统流量的动态响应特性曲线如图 13 所示。

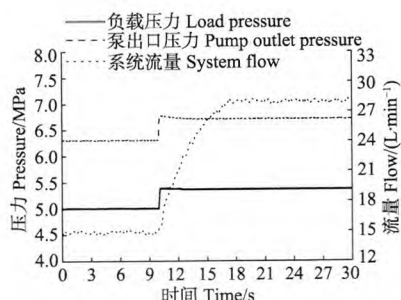


图 13 阀芯阶跃时变量泵出口压力及流量响应特性曲线

Fig.13 Pressure and flow rate curves of variable pump with step change in spool displacement

由图 13 可知，比例控制阀输入电压在 4.3~4.7 V 之间阶跃变化时，由于受到系统流量变化及液压系统试验管路压力损失的影响，回油管路背压略微升高，由比例溢流阀设定的负载压力会出现小幅阶跃变化，变量泵出口压力也跟随变化，补偿压力平均值约为 1.5 MPa，系统流量由 15 L/min 阶跃变化到 28 L/min，由于负载敏感变量泵机构惯性、内部泄漏、液压系统试验管路油液压缩性等因素影响，液压系统整体惯性增加，导致系统流量响应变慢，调整时间约为 7 s，变量泵在保持一定补偿压力时可根据比例控制阀阀芯开度变化为其提供相应流量。

## 4 结论

1) 充分考虑负载敏感变量泵内部液压元件的有效行程范围，建立了基于边界条件的负载敏感变量泵非线性数学模型，该模型更为准确地描述了负载敏感变量泵的工作特性。

2) 搭建了负载敏感变量泵 Simulink 仿真模型，仿真分析结果表明：变量泵可根据负载所需压力和流量实时调整斜盘倾角大小，进而实现压力-流量补偿功能，补偿压力约为 1.6 MPa，负载压力和流量阶跃变化时，变量泵具有良好的动态补偿特性。

3) 试验研究结果表明：在给定比例控制阀阀口开度不变的情况下，负载压力阶跃变化时，负载敏感变量泵通过流量控制阀调整斜盘倾角为比例控制阀提供所需流量，此外，变量泵出口压力始终高出负载压力 1.5 MPa，通过与仿真结果对比可知，补偿压力稳态误差约为 0.1 MPa，主要与液压系统试验管路压力损失有关，验证了负载敏感变量泵非线性数学模型的正确性。当比例控制阀阀芯阶跃变化时，负载敏感变量泵输出其所需流量，满足现代农机装备液压系统对负载敏感变量泵压力-流量补偿功能的需求。

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## Structural modeling and performance analysis of load-sensing variable pump

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**Abstract:** In recent years, to meet the development requirements of green production and energy efficiency of modern agriculture, the closed-center load-sensing hydraulic system has been widely used in the domestic and overseas agricultural machinery because of its low energy consumption and strong adaptability. As a key power component of the hydraulic system of the modern agricultural equipment, the performance of the load-sensing variable pump plays an important role in the performance of the whole hydraulic system. At present, the domestic scholars have mainly concentrated in the optimization and characteristics analysis of hydraulic valve, while have done little about the hydraulic pump. Therefore, this paper focused on the mathematical modeling, performance simulation and testing of the load-sensing variable pump, which was composed of swashplate piston pump, plunger cylinder, pressure control valve, flow control valve, load feedback check valve, and so on. In order to better analyze the performance of the pump, the mathematical models of these parts were established by using the pressure-flow equation, flow continuity equation, force balance equation, and so on. Taking into account the effective travel range of the moving parts such as valve spool and swashplate, the nonlinear mathematical model of the load-sensing variable pump based on the boundary conditions was constructed by using the state space method. Then, the simulation model of the pump was established using MATLAB/Simulink based on the mathematical model. By using the fourth order Runge-Kutta algorithm, the steady-state and dynamic performances of the pump were simulated. At the steady state, the compensating pressure of the variable pump was about 1.6 MPa. And in the dynamic state, the outlet pressure and swashplate angle of the pump could respectively follow the step changes of the load and system flow very well, the adjustment time of which was about 0.4 and 0.5 s, respectively. The simulation results showed that the load-sensing variable pump could adjust the inclination angle of the swashplate in real time according to the required flow and pressure of the load, and thus had a good pressure-flow compensation characteristic. Finally, to verify the correctness of the mathematical model and the simulation analysis, an indoor test platform of the load-sensing hydraulic system was built, including hydraulic pump station, proportional control valve, proportional relief valve, flowmeter, pressure sensor, and so on. The dynamic performance test of the load-sensing variable pump was carried out by using the platform. Among them, the step change of the load pressure could be achieved by controlling the proportional valve's opening pressure, while the step change in flow could be achieved by controlling the spool opening of the proportional relief valve. The test results showed that the load-sensitive pump could output its required flow when the proportional valve spool encountered a step change. And when the load made a step change, the outlet pressure of the variable pump changed almost synchronously with the load pressure. Compared with the simulation results, the steady-state error of the compensation pressure was about 0.1 MPa, which verified the correctness of the nonlinear mathematical model of the load-sensing variable pump. It can be seen that the output flow and pressure of the load-sensing variable pump can be adapted to the load in real time, thus effectively improving the efficiency of the hydraulic system and reducing the system heat, which meets the field work demands of the modern agricultural equipment operation unit.

**Keywords:** pumps; computer simulation; agricultural equipment; load-sensing; variable pump; mathematical model; boundary condition