Scientific Research

Experimental Verification of Fault Predictions in High Pressure Hydraulic Systems

P. Athanasatos, D. Koulocheris, Th. Costopoulos[*], V. Spitas

Mechanical Engineering Department, National Technical University of Athens, Zografos Campus, Athens, Greece
Email: [*]cost@central.ntua.gr

Abstract

In this paper a model of a high pressure hydraulic system was developed to simulate the effect of increased internal leakages inside the hydraulic cylinder and the 4/2 way directional control valve and to calculate the main parameters of the hydraulic system under various loads through the use of leakage-simulating throttle valves. After the completion of modeling, the throttle valves that simulate the internal leakages were calibrated and a number of test runs were performed for the cases of normal operation and the operation with increased internal leakages. The theoretical predictions were compared against the experimental results from an actual hydraulic test platform installed in the laboratory. In all cases, modeling and experimental data curves correlate very well in form, magnitude and response times for all the system's main parameters. This proves that the present modeling can be used to accurately predict various faults in hydraulic systems, and can thus be used for proactive fault finding in many cases, especially when the defective component is not easily detected and obvious at first sight.

Keywords

Hydraulic Systems, Cylinder, Directional Control Valve, Fault Prediction, Internal Leakage

1. Introduction

The bond graph method is used to create the model of a high pressure hydraulic system and to simulate various degrees of internal leakage in its 4/2 way directional control valve and hydraulic cylinder under various loads.

---

[*]Corresponding author.

After the model was created a series of test runs, simulating normal operation and increased internal leakage for the cylinder and the 4/2 way directional control valve, were conducted and their results were compared with the measurements from an actual high pressure hydraulic system with the same degrees of internal leakage. The goal is to experimentally verify whether bond graph modeling can be used as a tool for proactive fault finding in high pressure hydraulic systems, by accurately simulating various operating conditions of the system. In this paper, the model is compared to the actual system during operation when there is a gradually increasing internal leakage inside the cylinder and the 4/2 way directional control valve.

A useful insight into the scope, operation and components of high pressure hydraulic systems can be found in Kaliafetis and Costopoulos [1], Hehn [2], Meritt [3], Mitchell and Pippenger [4], and Rabie [5], whereas Totten *et al.* [6] deal with the troubleshooting principles of hydraulic systems.

For the modeling of hydraulic systems through the bond graph method an in depth research has been done by Athanasatos and Costopoulos [7], Athanasatos *et al.* [8], Barnard and Dransfield [9], Barnard [10], Dransfield [11], and Dransfield and Steki [12].

2. Hydraulic System Modeling

2.1. Description of Hydraulic System

The line diagram of the hydraulic system used as the test platform can be seen in **Figure 1** and its main components and measuring instruments are explained in **Table 1**. The system can be used to simulate a variety of operating conditions. In this paper though, we will focus exclusively in the internal leakage of the hydraulic cylinder (26) and the 4/2 way directional control valve (11) under various loads. The internal leakage on the hydraulic cylinder is simulated via throttle valve (*H*) which, when open, allows some amount of flow to bypass the
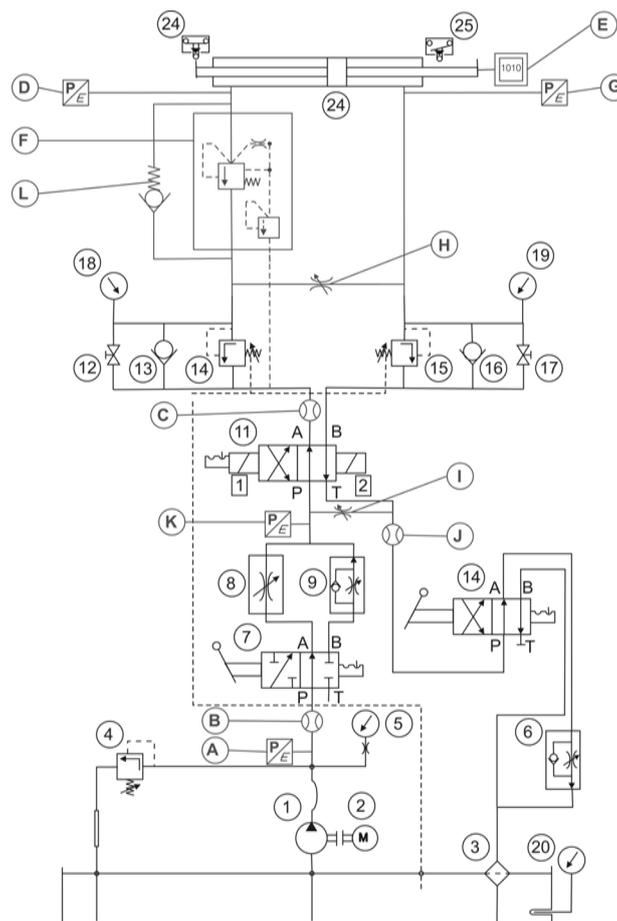

Figure 1. Line diagram of the hydraulic system to be modeled.

Table 1. Components of hydraulic system.

Number/symbol	Component
1	Vane pump
2	Electric motor
3	Return line oil filter
4	Main relief valve
5, 18, 19	Pressure gauges
6, 9	Flow control valves
7	4/2 way valve
8	Flow control valve
10	4/2 way valve
11	4/2 way direction control valve
12, 17	Shutoff switches
13, 16	Check valves
20	Thermometer
24, 25	Terminal switch
26	Hydraulic cylinder
14, 15	Manual load valves
A	Pressure transducer
B	Flowmeter
C	Flowmeter
D	Pressure transducer
E	Linear encoder
F	Two-stage electrically operated load control valve
G	Pressure transducer
H	Flow control valve for internal leakage simulation
I	Flow control valve for internal leakage simulation
J	Flowmeter
K	Pressure transducer
L	Check valve

cylinder. Similarly, the internal leakage in the 4/2 way valve is simulated via throttle valve (*I*) which, when open, allows some flow directly from port "*P*" of the valve to port "*T*".

2.2. Modeling Procedure of the Hydraulic System

To ease the modeling procedure, each main component of the system, was modeled separately. Due to the focus on the hydraulic cylinder and the directional control valve more detailed models are used for these two components and simpler ones were used for the rest of the components, in order to avoid creating an overly complex model.

2.3. Symbol Nomenclature

The main symbols that appear in the bond graph components and the resulting equations are the following:

ω = Angular velocity of electric motor and pump shaft

Q_p = Flow rate at pump outlet

V_p = Geometric displacement per radian rotation of the pump

T_p = Torque at electric motor shaft

P_p = Pressure at pump outlet

C_p = Hydraulic capacitance in pump outlet

Q_{lp} = Flow loss due to pump internal leakage

R_{lp} = Hydraulic resistance coefficient in the pump internal leakage flow path

Q_{rv} = Flow rate through pressure relief valve

R_{rv} = Hydraulic resistance coefficient in the main relief valve flow path

P_{set} = Main relief valve opening pressure

Q_i = Input flow

Q_o = Output flow

P_i = Input pressure

P_o = Output pressure

PR_{pv} = Pressure drop from pump outlet to 4/2 valve inlet

Q_A = Flow rate from 4/2 way valve "*A*" port

Q_B = Flow rate from 4/2 way valve "*B*" port

Q_P = Flow rate from 4/2 way valve "*P*" port

Q_T = Flow rate from 4/2 way valve "*T*" port

Q_{PA} = Flow rate through P-$>A$ flow path

Q_{PB} = Flow rate through P-$>B$ flow path

Q_{AT} = Flow rate through A-$>T$ flow path

Q_{BT} = Flow rate through B-$>T$ flow path

P_A = Pressure in 4/2 way valve "*A*" port

P_B = Pressure in 4/2 way valve "*B*" port

P_P = Pressure in 4/2 way valve "*P*" port

P_T = Pressure in 4/2 way valve "*T*" port

Q_{PA} = Flow through the P-$>A$ path in 4/2 way direction control valve

Q_{PB} = Flow through the P-$>B$ path in 4/2 way direction control valve

Q_{AT} = Flow through the A-$>T$ path in 4/2 way direction control valve

Q_{BT} = Flow through the B-$>T$ path in 4/2 way direction control valve

Q_{lv} = 4/2 way valve internal leakage flow rate

R_{PA} = Hydraulic resistance coefficient in the P-$>A$ flow path in 4/2 way direction control valve

R_{PB} = Hydraulic resistance coefficient in the P-$>B$ flow path in 4/2 way direction control valve

R_{AT} = Hydraulic resistance coefficient in the A-$>T$ flow path in 4/2 way direction control valve

R_{BT} = Hydraulic resistance coefficient in the B-$>T$ flow path in 4/2 way direction control valve

K_{PA} = Flow coefficient of P-$>A$ path in 4/2 way direction control valve

K_{PB} = Flow coefficient of P-$>B$ path in 4/2 way direction control valve

K_{AT} = Flow coefficient of A-$>T$ path in 4/2 way direction control valve

K_{BT} = Flow coefficient of B-$>T$ path in 4/2 way direction control valve

N = Valve spool displacement to maximum valve spool displacement ratio

X_i = Valve spool displacement ($X_i > 0$ when flow paths P-$>A$ and B-$>T$ are enabled)

W = Maximum valve spool displacement

D_v = 4/2 way valve spool diameter

c_v = Radial clearance between 4/2 valve spool and body

μ = Absolute fluid viscosity

PR_{vt} = Pressure drop between 4/2 valve T port and tank

x = Hydraulic cylinder piston displacement

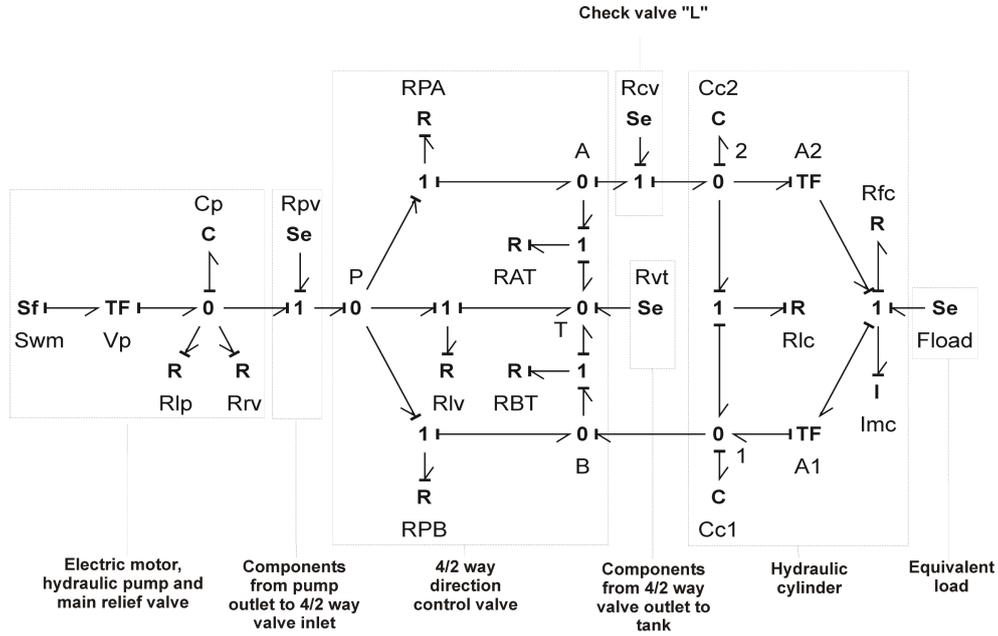$\dot{x}$ = Hydraulic cylinder piston velocity

Q_1 = Flow rate entering the hydraulic cylinder right chamber
Q_2 = Flow rate entering the hydraulic cylinder left chamber
A_1 = Hydraulic cylinder piston surface (right side)
A_2 = Hydraulic cylinder piston surface (left side)
F_{fc} = Friction force during the motion of the hydraulic cylinder piston
F_{mc} = Force due to mass inertia in the hydraulic cylinder moving parts
SF_{load} = Equivalent load
t = Elapsed time
m_c = Mass of cylinder moving parts
P_1 = Pressure in hydraulic cylinder right chamber
P_2 = Pressure in hydraulic cylinder left chamber
F = Force exerted in hydraulic cylinder piston
C_{c1} = Hydraulic capacitance in cylinder right chamber
C_{c2} = Hydraulic capacitance in cylinder left chamber
Q_{lc} = Internal leakage through hydraulic cylinder piston
D_c = Hydraulic cylinder piston diameter
L = Length of hydraulic cylinder piston
F_u = Coulomb friction force
K_f = Viscosity coefficient
$PRcv$ = Pressure drop through check valve "*L*"

2.4. Hydraulic System Bond Graph Model

From the models of the individual components, the entire hydraulic system bond graph model is created. In **Figure 2** we see the bond graph model of the system during the motion of the hydraulic cylinder piston to the left, while in **Figure 3** we see the bond graph model of the system during the motion of the piston to the right.

2.5. Hydraulic System Bond Graph Model Equation Layout

The equations for the components of the bond graph are the following:
   - **Electric motor, pump and main relief valve**

Figure 2. Hydraulic system bond graph model for the motion of the piston to the left.

Figure 3. Hydraulic system bond graph model for the motion of the piston to the right.

$$Swm: \quad \omega = \text{const} \tag{1}$$

$$TFV_p: \quad Q_p = \omega \cdot V_p \tag{2}$$

$$T_p = \frac{1}{V_p} \cdot P_p \tag{3}$$

$$C_p: \quad P_p = P_p(0) + \frac{1}{C_p} \cdot \int \Delta Q_p \cdot dt \tag{4}$$

$$R_{lp}: \quad Q_{lp} = \frac{P_p}{R_{lp}} \tag{5}$$

$$R_{rv}: \quad Q_{rv} = 0 \text{ if } P_p < P_{set}$$

$$Q_{rv} = \frac{(P_p - P_{set})}{R_{rv}} \text{ if } P_p \geq P_{set} \tag{6}$$

$$\text{0-junction:} \quad \Delta Q_p = Q_{ip} - Q_{lp} - Q_{rv} - Q_{op} \tag{7}$$

- **Section from pump outlet to 4/2 way valve inlet**

$$R_{pv}: \quad PR_{pv} = -\Delta P_{pv} = \text{const} \tag{8}$$

$$\text{1-junction:} \quad \Delta PR_{pv} = P_i - P_o \tag{9}$$

- 4/2 way direction control valve
0-junctions

$$Q_P = Q_{PA} + Q_{PB} + Q_{lv} \tag{10}$$

$$Q_T = Q_{AT} + Q_{BT} + Q_{lv} \tag{11}$$

$$Q_A = Q_{PA} - Q_{AT} \tag{12}$$

$$Q_B = Q_{PB} - Q_{BT} \tag{13}$$

1-junctions

$$\Delta P_{PA} = P_P - P_A \tag{14}$$

$$\Delta P_{PB} = P_P - P_B \tag{15}$$

$$\Delta P_{AT} = P_A - P_T \tag{16}$$

$$\Delta P_{BT} = P_B - P_T \tag{17}$$

$$\Delta P_{PT} = P_P - P_T \tag{18}$$

$$R_{PA}: \ Q_{PA} = 0 \ \text{if} \ N = 0$$

$$Q_{PA} = K_{PA} \cdot |N| \cdot \sqrt{\Delta P_{PA}} \ \text{if} \ -1 \le N < 0 \tag{19}$$

$$Q_{PA} = 0 \ \text{if} \ N > 0$$

$$R_{PB}: \ Q_{PB} = 0 \ \text{if} \ N = 0 \quad Q_{PB} = K_{PB} \cdot N \cdot \sqrt{\Delta P_{PB}} \ \text{if} \ 0 < N \le 1 \tag{20}$$

$$Q_{PB} = 0 \ \text{if} \ -1 \le N < 0$$

$$R_{AT}: \ Q_{AT} = 0 \ \text{if} \ N = 0$$

$$Q_{AT} = K_{AT} \cdot N \cdot \sqrt{\Delta P_{AT}} \ \text{if} \ 0 < N \le 1 \tag{21}$$

$$Q_{AT} = 0 \ \text{if} \ -1 \le N < 0$$

$$R_{BT}: \ Q_{BT} = 0 \ \text{if} \ N = 0$$

$$Q_{BT} = 0 \ \text{if} \ 0 < N \le 1 \tag{22}$$

$$Q_{BT} = K_{BT} \cdot |N| \cdot \sqrt{\Delta P_{BT}} \ \text{if} \ -1 \le N < 0$$

$$R_{lv}: \ Q_{lv} = 2 \cdot \frac{\pi \cdot D_v \cdot c_v^3}{12 \cdot \mu \cdot |X_i|} \cdot \Delta P, -W \le X_i \le W \tag{23}$$

- **Section from valve outlet to tank**

$$S_e: \ PR_{vt} = -\Delta P_{vt} = \text{const} \tag{24}$$

- **Hydraulic cylinder and equivalent load**

A) Piston movement to the left

0-junctions

$$\Delta Q_1 = Q_1 - A_1 \cdot \dot{x} - Q_{lc} \tag{25}$$

$$\Delta Q_2 = A_2 \cdot \dot{x} + Q_{lc} - Q_2 \tag{26}$$

1-junctions

$$\Delta P_{12} = P_1 - P_2 \tag{27}$$

$$\Delta F = P_1 \cdot A_1 - P_2 \cdot A_2 - F_{fc} - F_{mc} + SF_{\text{load}} \tag{28}$$

A_1 transformer

$$F = P_1 \cdot A_1 \tag{29}$$

$$\dot{x} = \frac{Q_1}{A_1} \tag{30}$$

A_2 transformer

$$Q_2 = A_2 \cdot \dot{x} \tag{31}$$

$$P_2 = \frac{F}{A_2} \tag{32}$$

C_{c1} capacitance

$$P_1 = P_1(0) + \frac{1}{C_{c1}} \cdot \int \Delta Q_1 \cdot \mathrm{d}t \tag{33}$$

C_{c2} capacitance

$$P_2 = P_2(0) + \frac{1}{C_{c2}} \cdot \int \Delta Q_2 \cdot \mathrm{d}t \tag{34}$$

I_{mc} inertia phenomenon

$$\dot{x} = \dot{x}(0) + \frac{1}{m_c} \cdot \int F_{mc} \cdot \mathrm{d}t \tag{35}$$

R_{lc} resistance

$$Q_{lc} = \frac{\pi \cdot D_c \cdot c^3}{12 \cdot \mu \cdot L} \cdot \Delta P_{12} + \frac{1}{2} \cdot \pi \cdot \dot{x} \cdot D \cdot c \tag{36}$$

R_{fc} resistance

$$F_{fc} = \mathrm{sgn}(\dot{x}) \cdot F_u + K_f \cdot \dot{x} \tag{37}$$

SeF_{load}

$$SeF_{\mathrm{load}} = F_{\mathrm{load}} = \mathrm{const} \tag{38}$$

B) Piston movement to the right
0-junctions

$$\Delta Q_2 = Q_2 - A_2 \cdot \dot{x} - Q_{lc} \tag{39}$$

$$\Delta Q_1 = A_1 \cdot \dot{x} + Q_{lc} - Q_1 \tag{40}$$

1-junctions

$$\Delta P_{21} = P_2 - P_1 \tag{41}$$

$$\Delta F = P_2 \cdot A_2 - P_1 \cdot A_1 - F_{fc} - F_{mc} + SF_{\mathrm{load}} \tag{42}$$

A_2 transformer

$$\dot{x} = \frac{Q_2 \cdot}{A_2} \tag{43}$$

$$F = P_2 \cdot A_2 \tag{44}$$

A_1 transformer

$$P_1 = \frac{F}{A_1} \tag{45}$$

$$Q_1 = \dot{x} \cdot A_1 \tag{46}$$

C_{c1} capacitance

$$P_1 = P_1(0) + \frac{1}{C_{a1}} \cdot \int \Delta Q_1 \cdot \mathrm{d}t \tag{47}$$

C_{c2} capacitance

$$P_2 = P_2(0) + \frac{1}{C_{a2}} \cdot \int \Delta Q_2 \cdot \mathrm{d}t \tag{48}$$

I_{mc} inertial phenomenon

$$\dot{x} = \dot{x}(0) + \frac{1}{m_c} \cdot \int F_{mc} \cdot \mathrm{d}t \tag{49}$$

R_{lc} resistance

$$Q_{lc} = \frac{\pi \cdot D \cdot c_c^3}{12 \cdot \mu \cdot L} \cdot \Delta P_{21} + \frac{1}{2} \cdot \pi \cdot |\dot{x}| \cdot D \cdot c \tag{50}$$

R_{fc} resistance

$$F_{fc} = \mathrm{sgn}(\dot{x}) \cdot F_u + K_f \cdot \dot{x} \tag{51}$$

SeF_{load}:

$$SeF_{\text{load}} = F_{\text{load}} = 0 \tag{52}$$

-   Check valve "L"
1-junction:

$$\Delta P_{Rcv} = P_i - P_o \tag{53}$$

SeR_{cv}:

$$PR_{cv} = -\Delta P_{Rcv} = \mathrm{const} \ \ \text{if} \ \ Q_{PA} > 0$$
$$PR_{cv} = 0 \ \ \text{if} \ \ Q_{PA} \le 0 \tag{54}$$

2.6. Constants and Initial Conditions Definition

Based on manufacturer data and measurements performed, the following constants and initial conditions were determined.

$$\omega = 1600 \ \text{rpm} = 167.55 \ \text{rad/s}$$

$$V_p = 1.31e-5 \ \text{m}^3/\text{rev} = \frac{1.375e-5}{2\pi} \ \text{m}^3/\text{rad}$$

$$C_p = 4.48e^{-12} \ \frac{\text{m}^3}{\text{Pa}}$$

$$Q_p = \omega \cdot V_p = 167.55 \cdot \frac{1.375e^{-5}}{2\pi} = 3.67e^{-4} \ \text{m}^3/\text{s}$$

$$R_{lp} = 6e-12 \ \frac{\text{m}^3}{\text{s} \cdot \text{Pa}}$$

$$P_{set} = 7.0e6 \ \text{Pa}$$

$$R_{rv} = 2.5e-9 \ \frac{\text{m}^3}{\text{s} \cdot \text{Pa}}$$

$$PR_{pv} = \Delta P_{pv} = -9.537e4 \ \text{Pa}$$

$$K_{PA} = K_{PB} = K_{AT} = K_{BT} = 5.185e-7 \ \frac{\text{m}^3}{\text{Pa}^{1/2} \cdot \text{s}}$$

$$W = 0.02 \ \text{m}$$

$$D_v = 23.5 \ \text{mm}$$

$$L_v = 5 \ \text{mm}$$

$$\mu = 0.0261 \ \text{Ns/m}^2$$

$$c_v = 34 \ \mu\text{m}$$

$$A_1 = A_2 = 0.0021 \text{ m}^2$$

$$C_{c1} = \frac{5.30 \cdot e - 4 + 2.1 \cdot e - 3 \cdot x}{1.5 \cdot e9} \left(\frac{\text{m}^3}{\text{Pa}} \right)$$

$$C_{c2} = \frac{5.85 \cdot e - 4 + 2.1 \cdot e - 3 \cdot (0.5 - x)}{1.5 \cdot e9} \left(\frac{\text{m}^3}{\text{Pa}} \right)$$

$$D_c = 60 \text{ mm}$$

$$L_c = 30 \text{ mm}$$

$$c_c = 83 \text{ µm}$$

$$F_u = 163.2 \text{ N}$$

$$K_f = 1.49e^3 \text{ N} \cdot \text{s/m}$$

$$m_c = 7.32 \text{ kg}$$

$$\Delta PR_{cv} = 0.1e6 \text{ Pa}$$

$$PR_{vt} = -1.46e5 \text{ Pa}$$

$$P_1(0) = P_2(0) = 0 \text{ Pa}$$

$$\dot{x}(0) = v(0) = 0 \text{ m/s}$$

$$x(0) = 0 \text{ m}$$

2.7. Calibration of Throttle Valves "*H*" and "*I*" with Calculation of "Equivalent Clearances" for Increased Internal Leakage in Hydraulic Cylinder and 4/2 Way Valve

To calibrate throttle valves "*H*" and "*I*" that control flow bypass through the cylinder and valve and thus simulate internal leakage, the "equivalent clearances" for the hydraulic cylinder and 4/2 way valve corresponding to various opening degrees of the throttle valves were calculated. The "equivalent clearance" is defined as the internal clearance between the hydraulic cylinder piston and body (the same goes for the internal clearance of the 4/2 way valve spool and body) that would cause the same amount of flow to bypass the cylinder (or the 4/2 way valve) as the flow bypassing it due to the gradual opening of the respective throttle valve plus the flow already bypassing it due to the existing clearance. To calculate the "equivalent clearances" of the hydraulic cylinder, firstly, the total flow rate exiting the cylinder during the working phase was measured by flow meter "*C*". Then, the actual "motion flow rate" used to move the cylinder piston was calculated from the piston velocity. Then, the "motion flow rate" was subtracted from the total flow rate in order to calculate the "equivalent internal leakage" flow rate bypassing the cylinder. Finally, Equation (36) was solved for clearance c_c by using analytical methods, in order to calculate the equivalent clearances. In total, 4 tests were performed, each by gradually opening throttle valve "*H*" by half a turn. The equivalent leakages calculated can be seen in **Table 2**, where the initial mea-

Table 2. Equivalent clearances of hydraulic cylinder according to opening of throttle valve "*H*".

Throttle valve "*H*" opening	Bypass flow rate Ql_c (m³/s)	Equivalent clearance c_c (µm)
Closed	1.80e−5	83
Open by 1/2 turn	3.15e−5	101
Open by 1 turn	7.00e−5	132
Open by 1 1/2 turn	8.20e−5	139
Open by 2 turns	1.05e−4	151

surement with the throttle valve completely shut corresponds to the existing internal clearance between the piston and the cylinder body.

In the same manner, the "equivalent clearances" of the 4/2 way valve were calculated. The total flow rate that is headed to the valve is measured by flow meter "B", while the flow that actually goes through the valve ports is measured by flow meter "C". By subtracting the flow rate measured from flow meter "C" from the flow rate measured by flow meter "B", we calculate the "equivalent internal leakage" flow rate bypassing the valve. Finally, Equation (23) was solved for clearance c_v in order to calculate the equivalent clearances. In total, 4 tests were performed, each by gradually opening throttle valve "I" by half a turn. The equivalent leakages calculated can be seen in **Table 3**, where the initial measurement with the throttle valve completely shut corresponds to the existing internal clearance between the spool and the valve body. All measurements were performed with the load control valve set to create a backpressure of 10 bar, equivalent to a 2.1 kN load.

3. Experimental Verification of Fault Predictions

Using throttle valves "H" and "I", a number of test runs was performed in the high pressure hydraulic system simulating internal leakage in the hydraulic cylinder and 4/2 way valve respectively. These test runs were replicated in the bond graph model of the system using the "equivalent clearances" of the cylinder and valve calculated in paragraph 2.7. The results of the system and the model were compared and the comparison results are discussed in the following paragraphs.

3.1. Comparison of Model and System during Operation with Increased Internal Leakage of the Hydraulic Cylinder

In total, 3 test runs were conducted, with the load control valve being set to create a backpressure of 10, 20 and 30 bars, which correspond to 2.1 kN, 4.2 kN and 6.3 kN of equivalent load respectively. Each test run started with throttle valve "H" completely shut off, and when one operating cycle was completed, it was gradually opened by half a turn at a time. The results of the model were compared to the ones of the actual system in terms of the equivalent internal leakage flow rate in the working phase and the ratio of the average speed in the working phase to the average speed in the return phase. The results are shown in the following figures. In **Figure 4**, the comparison of equivalent internal leakage values of the model and the actual system for an equivalent load of 2.1 kN is shown. As seen, there is a very good correlation between the experimental and the measurement data, with the correlation coefficient being $r = 0.994$. Also, the regression curves for both the experimental (continuous line) and the model data (dashed line) have the form of a 3rd degree polynomial equation, as expected by Equation (36). In **Figure 5**, we see the comparison of equivalent internal leakage values of the model and the actual system for an equivalent load of 4.2 kN. Again, the correlation of experimental and model data is very high ($r = 0.998$) and the regression curves for both the experimental (continuous line) and the model data (dashed line) have the form of a 3rd degree polynomial equation. Finally, the same goes for **Figure 6**, where the comparison of equivalent internal leakage values for the model and the actual system for an equivalent load of 6.3 kN is shown. The correlation coefficient of experimental and model data is $r = 0.995$, with the regression curves for both the experimental (continuous line) and the model data (dashed line) again having the form of a 3rd degree polynomial equation.

In **Figure 7**, the comparison of working phase to return phase piston velocity ratio values of the model and the actual system for an equivalent load of 2.1 kN is shown. The correlation coefficient is $r = 0.997$ and the re-

Table 3. Equivalent clearances of 4/2 way valve according to opening of throttle valve "I".

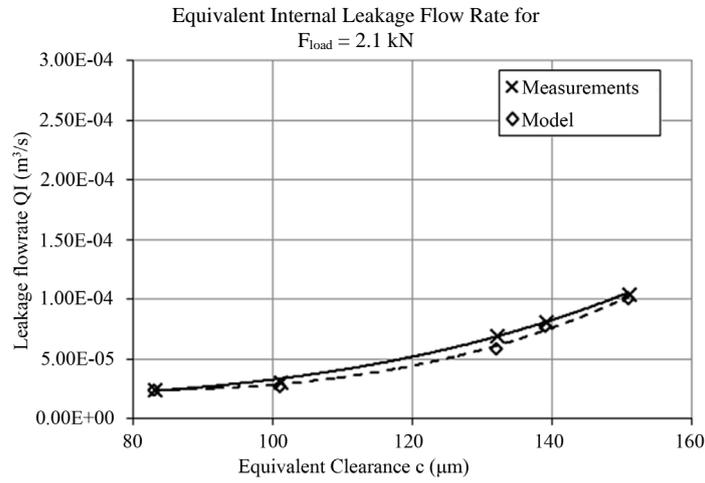| Throttle valve "H" opening	Bypass flow rate Ql_c (m³/s)	Equivalent clearance c_c (µm)
Closed	8.6e−6	34
Open by 1/2 turn	1.10e−5	37
Open by 1 turn	2.70e−5	49
Open by 1 1/2 turn	5.40e−5	62
Open by 2 turns	8.70e−5	73

Figure 4. Comparison of equivalent leakage flow rate between experimental and model data for $F_{load} = 2.1$ kN.
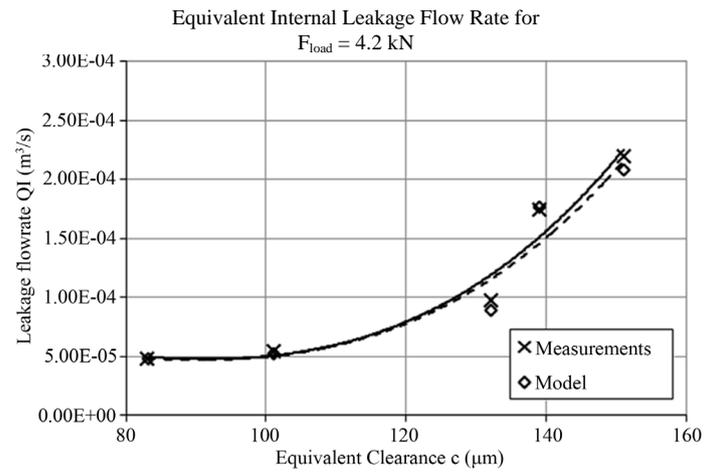

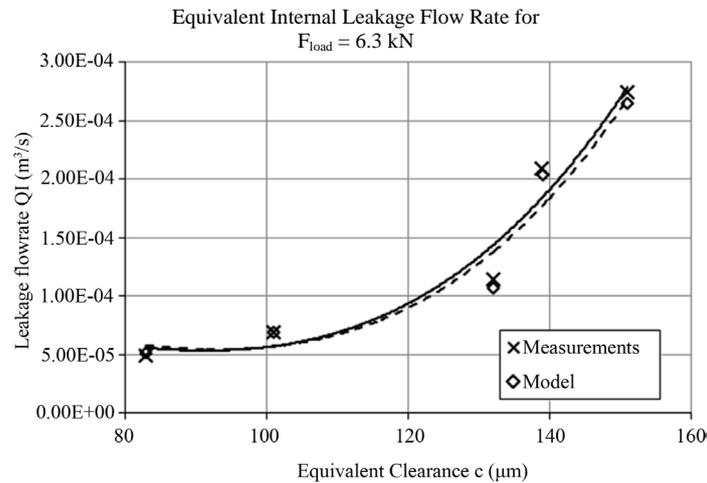**Figure 5.** Comparison of equivalent leakage flow rate between experimental and model data for $F_{load} = 4.2$ kN.

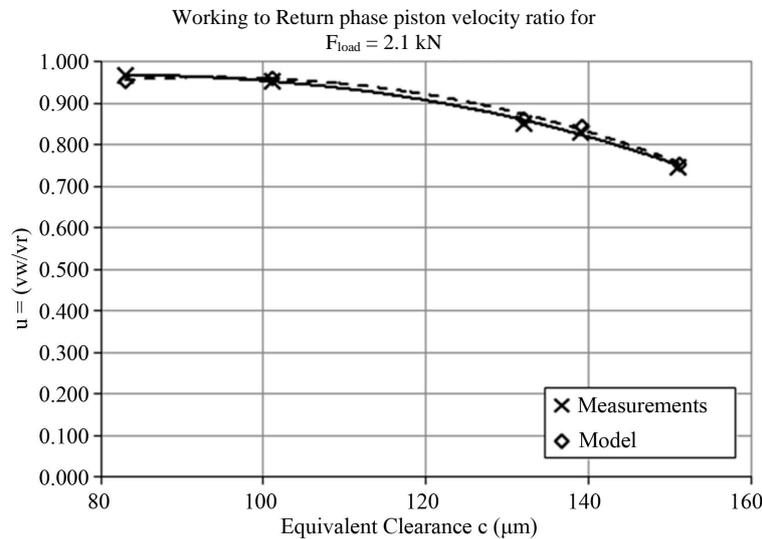**Figure 6.** Comparison of equivalent leakage flow rate between experimental and model data for $F_{load} = 6.3$ kN.

Working to Return phase piston velocity ratio for
$F_{load} = 2.1$ kN

Figure 7. Comparison of working to return piston velocity ratio between experimental and model data for $F_{load} = 2.1$ kN.

gression curves for both the experimental (continuous line) and the model data (dashed line) have the form of a 3rd degree polynomial equation. Similarly, in **Figure 8** we see the same values for the test run with an equivalent load of 4.2 kN. The correlation coefficient between the experimental and the model data is $r = 0.999$ and again regression curves for both the experimental and the model data have the form of a 3rd degree polynomial equation, as expected. Finally, the same goes for **Figure 9**, where the comparison of working phase to return phase piston velocity ratio values for the model and the actual system for an equivalent load of 6.3 kN is shown. Here, the correlation coefficient between the experimental and the model data is $r = 0.996$, and the regression curves retain the form of a 3rd degree polynomial equation.

3.2. Comparison of Model and System during Operation with Increased Internal Leakage of the 4/2 Way Direction Control Valve

Again, the values of equivalent clearances for the 4/2 way valve calculated in Paragraph 2.7 were used in the model and its results were compared with the ones of the actual system. In **Figure 10**, the comparison of equivalent internal leakage values in the working phase of the model and the actual system for an equivalent load of 2.1 kN is shown. There is very good correlation between the experimental and the measurement data, with the correlation coefficient being $r = 0.997$. Also, the regression curves for both the experimental (continuous line) and the model data (dashed line) have the form of a 3rd degree polynomial equation, as expected by Equation (23).

In **Figure 11**, we see the comparison of equivalent internal leakage values of the model and the actual system for an equivalent load of 4.2 kN. Again, the correlation of experimental and model data is very high ($r = 0.999$) and the regression curves for both the experimental (continuous line) and the model data (dashed line) have the form of a 3rd degree polynomial equation. Finally, the same goes for **Figure 12**, where the comparison of equivalent internal leakage values for the model and the actual system for an equivalent load of 6.3 kN is shown. The correlation coefficient of experimental and model data is again very high ($r = 0.998$), with the regression curves for both the experimental (continuous line) and the model data (dashed line) having the form of a 3rd degree polynomial equation.

In **Figure 13**, the comparison of working phase to return phase piston velocity ratio values of the model and the actual system for an equivalent load of 2.1 kN is shown. The correlation coefficient is $r = 0.997$ and the regression curves for both the experimental (continuous line) and the model data (dashed line) have the form of a 3rd degree polynomial equation. Similarly, in **Figure 14** we see the same values for the test run with an equivalent load of 4.2 kN. The correlation coefficient between the experimental and the model data is $r = 0.999$ and again regression curves for both the experimental and the model data have the form of a 3rd degree polynomial equation, as expected. Finally, the same goes for **Figure 15**, where the comparison of working phase to return
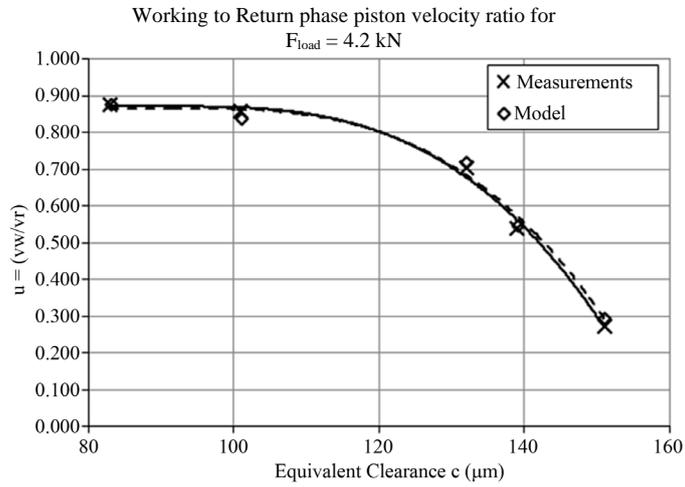
Working to Return phase piston velocity ratio for
$F_{load} = 4.2$ kN

Figure 8. Comparison of working to return piston velocity ratio between experimental and model data for $F_{load} = 4.2$ kN.
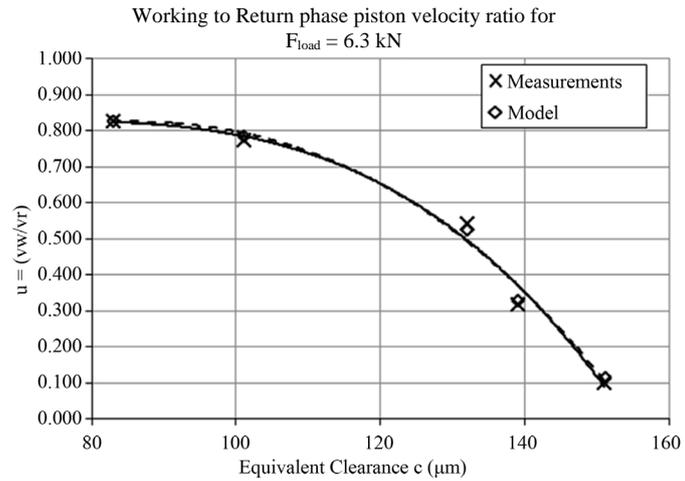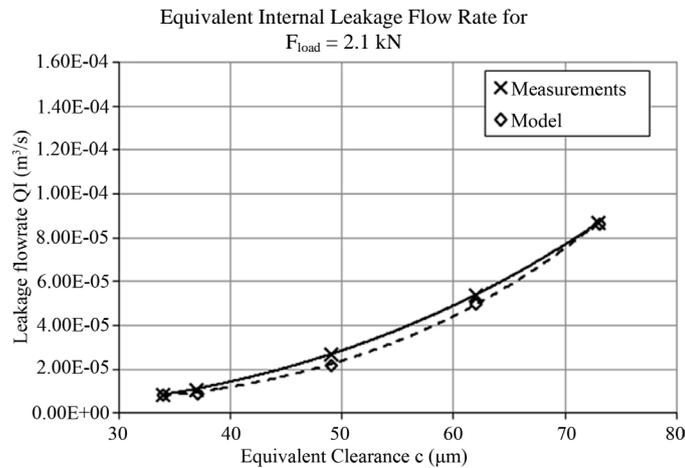
Working to Return phase piston velocity ratio for
$F_{load} = 6.3$ kN

Figure 9. Comparison of working to return piston velocity ratio between experimental and model data for $F_{load} = 6.3$ kN.

Equivalent Internal Leakage Flow Rate for
$F_{load} = 2.1$ kN

Figure 10. Comparison of equivalent leakage flow rate between experimental and model data for $F_{load} = 2.1$ kN.
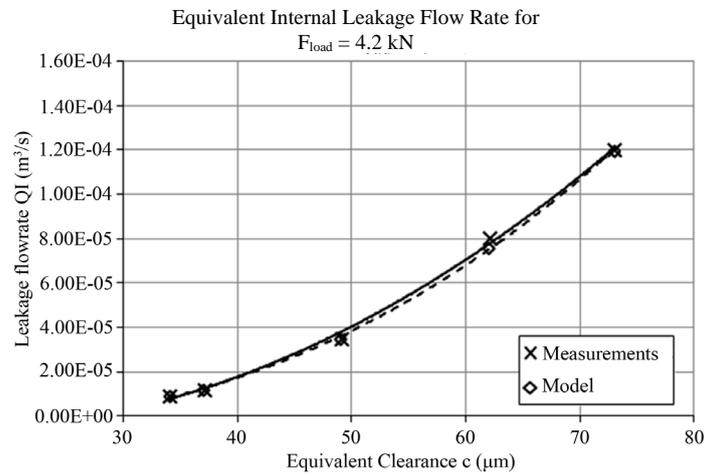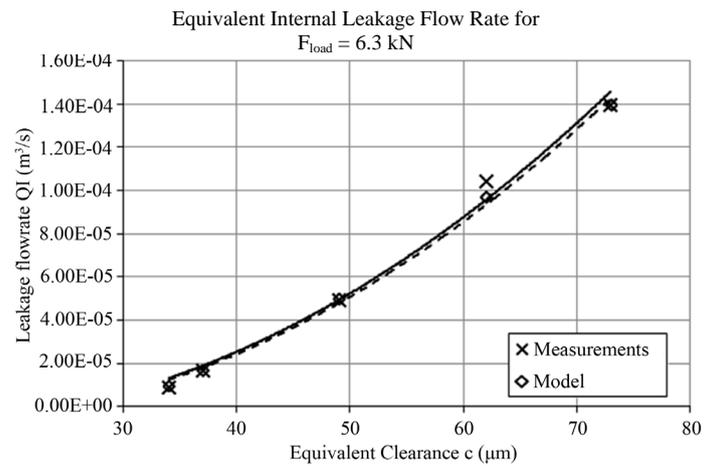
Equivalent Internal Leakage Flow Rate for
F_{load} = 4.2 kN

Figure 11. Comparison of equivalent leakage flow rate between experimental and model data for F_{load} = 4.2 kN.

Equivalent Internal Leakage Flow Rate for
F_{load} = 6.3 kN

Figure 12. Comparison of equivalent leakage flow rate between experimental and model data for F_{load} = 6.3 kN.

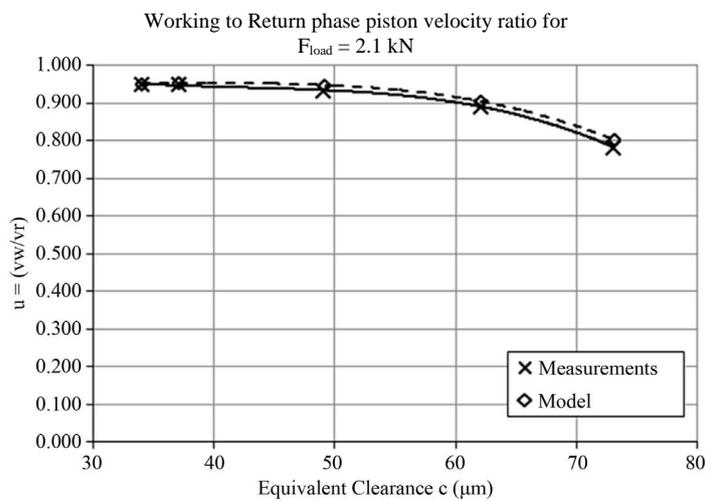Working to Return phase piston velocity ratio for
F_{load} = 2.1 kN

Figure 13. Comparison of working to return piston velocity ratio between experimental and model data for F_{load} = 2.1 kN.

Working to Return phase piston velocity ratio for
$F_{load} = 4.2$ kN

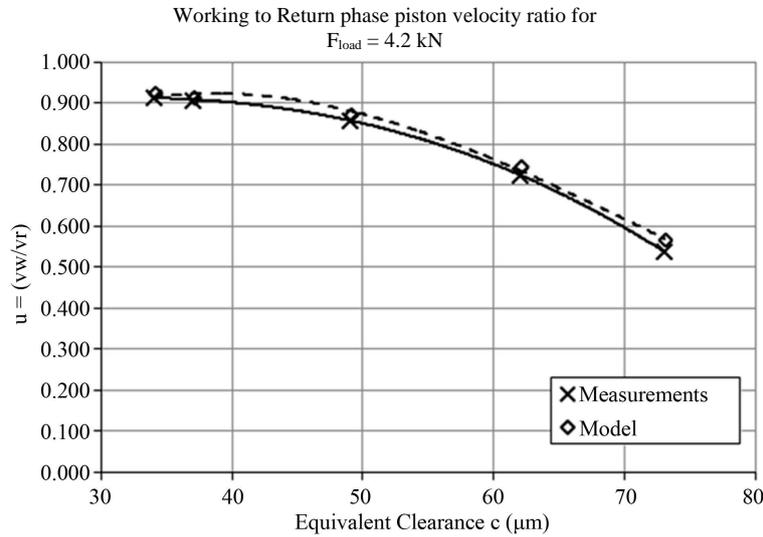**Figure 14.** Comparison of working to return piston velocity ratio between experimental and model data for $F_{load} = 4.2$ kN.

Working to Return phase piston velocity ratio for
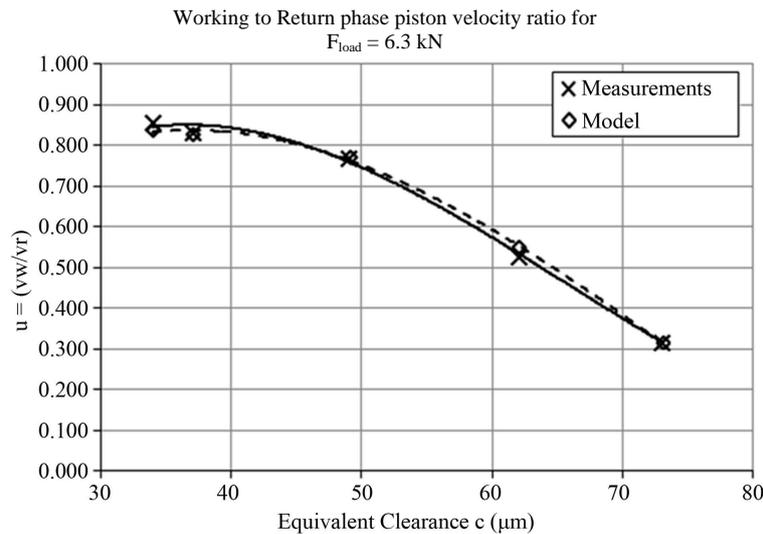$F_{load} = 6.3$ kN

Figure 15. Comparison of working to return piston velocity ratio between experimental and model data for $F_{load} = 6.3$ kN.

phase piston velocity ratio values for the model and the actual system for an equivalent load of 6.3 kN is shown. Here, the correlation coefficient between the experimental and the model data is $r = 0.998$, and the regression curves retain the form of a 3rd degree polynomial equation.

4. Conclusion

In this paper, the fault predictions of a model of an actual high pressure hydraulic system were verified by comparison with the results of an actual hydraulic system. The results of the model were compared to the results provided by an actual hydraulic system, modified in order to simulate various loads and degrees of internal leakage in the hydraulic cylinder and the 4/2 way valve. Comparisons were made during simulation of increased internal leakage in the cylinder and the valve. In all cases, the results of the model correlate very well to the data provided by the actual system both in shape and in minima and maxima of the curves. The equivalent internal leakage in the hydraulic cylinder and the 4/2 way valve is a function of the equivalent internal clearance that has the form of a 3rd degree polynomial equation, as evidenced by the regression curves in both the experimental

and modeling data. Additionally, the ratio of the average velocity of the cylinder piston in the working phase to the average velocity in the return phase is also a function of the equivalent internal clearance of the hydraulic cylinder and the 4/2 way valve in the form of a 3rd degree polynomial equation. This also means that this ratio can be used as an index for the assessment of the internal clearance of the cylinder and the valve, given that it is fairly easy to calculate, unlike the more complex procedure and instrumentation required to measure the internal leakage. This could prove useful in real life applications of high pressure hydraulic systems where it is obvious that a problem exists, but the source of it cannot be easily traced. In similar situations the application of this modeling technique could also help the troubleshooting procedure even further, by allowing the simulation of "fault scenarios" in various components of the system, in order to locate the source of the problem.

References

[1] Kaliafetis, P. and Costopoulos, Th. (1994) Modelling and Simulation of an Axial Piston Variable Displacement Pump with Pressure Control. *Mechanism and Machine Theory*, **30**, 599-612. http://dx.doi.org/10.1016/0094-114X(94)00057-R

[2] Hehn, A.H. (1994) Fluid Power Troubleshooting: Second Edition, Revised and Expanded. CRC Press, Boca Raton.

[3] Meritt, H.E. (1967) Hydraulic Control Systems. John Wiley and Sons, Inc., Hoboken.

[4] Mitchell, R.J. and Pippenger, J.J. (1997) Fluid Power Maintenance Basics and Troubleshooting. Marcel Dekker Inc., New York.

[5] Rabie, G.M. (2009) Fluid Power Engineering. McGraw-Hill, New York.

[6] Totten, G.E., Wills, D.K. and Feldmann, D.K. (2001) Hydraulic Failure Analysis: Fluids, Components and System Effects. ASTM. http://dx.doi.org/10.1520/STP1339-EB

[7] Athanasatos, P. and Costopoulos, T. (2011) Proactive Fault Finding in a 4/3-Way Direction Control Valve of a High Pressure Hydraulic System Using the Bond Graph Method with Digital Simulation. *Mechanism and Machine Theory*, **50**, 64-89. http://dx.doi.org/10.1016/j.mechmachtheory.2011.11.018

[8] Athanasatos, P., Koulocheris, D., Costopoulos, T. and Raptis, K. (2013) Operational Fault Diagnosis in Industrial Hydraulic Systems through Modeling the Internal Leakage of Its Components. *American Journal of Applied Sciences*, **10**, 1648-1659. http://dx.doi.org/10.3844/ajassp.2013.1648.1659

[9] Barnard, B.W. and Dransfield, P. (1977) Predicting Response of a Proposed Hydraulic Control System Using Bond Graphs. *Journal of Dynamic Systems*, *Measurement*, *and Control*, **99**, 1-8. http://dx.doi.org/10.1115/1.3427068

[10] Barnard, B.W. (1973) Predicting the Dynamic Response of a Hydraulic System Using Power Bond Graphs. M.Sc Thesis in Mechanical Engineering, Monash University, Melbourne (Unpublished).

[11] Dransfield, P. (1981) Hydraulic Control Systems—Design and Analysis of Their Dynamics. Lecture Notes in Control and Information Sciences. Springer-Verlag, Berlin. http://dx.doi.org/10.1007/BFb0005627

[12] Dransfield, P. and Steki, J.S. (1991) Bond Graph Influenced Research and Learning in Fluid Power. *Journal of the Franklin Institute*, **328**, 535-545. http://dx.doi.org/10.1016/0016-0032(91)90042-2