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Practical Study for Investigation Proportional Pressure Relief Valve (DBETR-1X/25) Performance

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Abstract

The trend in hydraulic power applications is to improve efficiency and performance of hydraulic systems parts. This paper examines the performances of direct operated proportional pressure relief valve type (DBETR-1X/25). There are limitations to the closed loop performance of the valve when it is included in a valve-controlled electro hydraulic system.

Proportional pressure relief valves are one of fundamental important elements for modern hydraulic control systems, which used in protecting all circuit parts from damage. These valves decreases costs and can be employed to obtain many difference pressures by using only one proportional pressure relief valve, which plays a vital role in automation process. Practical results obtained from this work were satisfactory and acceptable. The experimental and theoretical resullts are compared with data sheet from manufacture companies. A small dissimilarity has been found, but it can not be negligible. The results are very important to the designers and engineers, who are working in hydraulic proportional systems field, in helping them to calculate losses accuracy and design applicable circuits to arrive actual ideal performance for hydraulic proportional systems and not to absolutely rely on the company ideal results.

The model is simulated using MATLAB\ SIMULINK model (R2008a) and the theoretical results are compared to those obtained practically.

Keywords: pressure control, proportional valve, relief valve, modeling, Simulink

دراسة عملية للتحقق من اداء صمام تناسبي لحد الضغط نوع (DBETR-1X/25)

الخلاصة

الشائع في قدرة التطبيقات الهيدروليكية هو لتحسين الاداء والكفاءة لاجزاء المنظومات الهيدروليكية ، حيث تم في هذا المقال اجراء اختبارات عملية لاداء صمام حد الضغط المباشر التشغيل تناسبي نوع (DBETR-1X/25). ومحددات الاداء المغلق لهذا الصمام تتظمن سيطرة كهروهيدروليكية للنظام.

تعتبر صمامات الضغط التناسبية من العناصر الاساسية والمهمة جدا في الدوائرالهيدروليكية الحديثة وذلك للمحافظة على كل اجزاء الدائرة من التلف والتقليل من الكلف والحصول على ضغوط مختلفة باستخدام صمام ضغط واحد فقط لاداء مجموعة من العمليات الاوتوماتيكية. النتائج العملية التي حصلنا عليها من هذا البحث والموضحة في المخططات ادناه ومقارنتها مع المخططات الخاصة بالشركات المنتجة لمثل هذه الصمامات وجد فارق لايستهان به ولايمكن اهماله.تكون النتائج مهمة جدا للمصممين والمهندسين العاملين في مجال الدوائر الهيدروليكية

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التناسبية مع الاخذ بنظر الاعتبار الناحية العملية واختلافها مع الناحية التصميمية النظرية وعدم الاعتماد على مخططات الشركات المنتجة بصورة مطلقة وذلك كون هذه المخططات مثالية ولكي يتمكنوا من تصميم دوائر دقيقة التحكم وكذلك التخلص من الخسائر في الكلف والوصول الى الاداء الحقيقي للمنظومات الهيدروليكية التناسبية.

تم تمثيل ألآنموذج الرياضي باستخدام (MATLAB\ SIMULINK) (R2008a) وتم مقارنة النتائج النظرية بنظيرتها العملية.

1. Introduction

Power at any point in a hydraulic system can be determined by multiplying the fluid flow, Q, by the pressure drop, p, across a section of the machine. Flow is produced by an appropriate hydraulic pump. Pressure is the result of restriction in the system, caused by fluid viscosity, system geometry, and power output. Because the hydraulic pumps that are used in fluid power systems are of the positive displacement type, pressure will develop up to a preset regulated value. This desired pressure value is controlled by an appropriately designed pressure regulating valve.

Hydraulic drives, thanks to their high power intensity, are low in weight and require a minimum of mounting space. The facilitate fast and accurate control of very high energies and forces [1]. The hydraulic cylinder represents a cost–effective and simply constructed linear drive. The combination of these advantages opens up a wide range of applications for hydraulics in mechanical engineering, vehicle construction and aviation. The increase in automation makes it ever more necessary for pressure, flow rate and flow directional in hydraulic systems to be controlled by means of an electrical control system. The obvious choices for this are hydraulic proportional valves as in interface between controller and hydraulic system [2].

The direct operated proportional pressure relief valve type DBETR used in this work is a valve with 30 bar pressure rating. This means that the setting range of the system pressure to be controlled can be varied only between the minimum setting pressure and 25 bar. For pressure that is greater than 25 bar, the valve is at its maximum. That means that the ratio of forces between the system pressure and the spring force on the valve poppet resulting from the position control can no longer be brought into equilibrium, so that the reduction of the control land necessary to increase the system pressure [3,4].

2. Experimental Work

The proportional valve shown in figure (1) is a direct acting pressure relief valve of poppet construction which controls the pressure proportionally to the electrical signal value.

Proportional pressure relief valve type DBETR is a remote control valve. In design terms it is a direct operated pressure relief valve of poppet design. This valve regulates pressure in proportion to the electrical command value. The valve consists basically of a housing (1), proportional solenoid (2) with inductive positional transducer (3), valve seat (4) and valve poppet (5). Pressure is set by adjusting the command value potentiometer (0-9) Volts.

Adjusting the command value causes tensioning of the compression spring (2) via the electronic controls and the proportional solenoid (6). Tensioning of the compression spring (6), i.e. the position of the spring plate (7), is determined by the inductive positional transducer (3). Any deviations from the command value are corrected by the closed loop positional control. The use of this principle eliminates the effect of solenoid friction Advantages: – Low hysterics, Good repeatability. If the command value is zero or in the event of a power failure to the proportional solenoid or cable breakage at the positional transducer the lowest possible setting pressure will be set [5, 6, 7, 8].

To ensure optimum valve function, bleeding must be carried out at the commissioning stage:

– Remove item 8,

– Pour pressure fluid into open screw hole at item 8,

– When no further bubbles appear screw in item 8.

– Emptying of the tank lines to be avoided. With the appropriate installation conditions, a back pressure valve is to be installed (back pressure approx. 2 bar) [6].

2.1 Hydraulic Circuit

The object of this paper is to draw the characteristic curve for signal value voltage/setting and the characteristic curve for flow rate/minimum setting pressure. Because of the flow rate dependency of the direct operated proportional pressure relief valve, select four characteristic curves from the complete group (signal value voltage/setting pressure). In order to maintain a constant flow rate through the complete pressure range, a flow control valve must be installed before the pressure relief valve.

Construct the circuit on test rig (Hydro-prop.2 figure -2) as shown in figure (3) in such a way a constant flow rate passes through the proportional pressure relief valve. It should be able to measure the factors for computing the flow rate. Safeguard the system from pressure overload with an additional pressure relief valve. To adjust system pressures include a shut-off valve in the circuit and direct this pressure to a pressure gauge. It should be possible to read the setting pressure of the proportional pressure relief valve from the fine pressure gauge.

2.2 Electrical Circuit

In the following drawn schematically, first, the terminal connections of the proportional amplifier VT5003 as shown in figure(4), and then the circuit diagram with the outputs and inputs on the electrical test rig as shown in figure (5). Set up the electrical circuit, regarding both the terminal connections and circuit diagram, so that he corresponding controllable signal value (0-9) Volts applied to the valve. It should be possible to measure this signal value voltage, which has a value of (0-6) Volts at the signal value measuring sockets, using the digital multi-meter.

3. Modeling System Using Simulink

The practical model, to check the pressure-flow characteristic of a pressure-relief valve, has been built using MATLAB/SIMULINK (R2008) as shown in Fig.(10) [12].

The relief-valve model is a subsystem built of an orifice with round holes, translational converter, preloaded spring, and a hard stop. The following figure shows the typical dependency between the valve passage area A and the pressure differential p across the valve

The valve remains closed while pressure at the valve inlet is lower than the valve preset pressure. When the preset pressure is reached, the value control member (spool, ball, poppet, etc.) is forced off its seat, thus creating a passage between the inlet and outlet. Some fluid is diverted to a tank through this orifice, thus reducing the pressure at the inlet. If this flow rate is not enough and pressure continues to rise, the area is further increased until the control member reaches its maximum. At this moment, the maximum flow rate is passing through the valve. The pressure increase over the preset level is frequently referred to as valve steady state error, or regulation range. The valve maximum area and regulation range are the key parameters of the block.

In addition to the maximum area, the leakage area is also required to characterize the valve. The main purpose of the parameter is not to account for possible leakage, even though this is also important, but to maintain numerical integrity of the circuit by preventing a portion of the system from getting isolated after the valve is completely closed. An isolated or "hanging" part of the system could affect computational efficiency and even cause failure of computation. Theoretically, the parameter can be set to zero, but it is not recommended.

The model accounts for the laminar and turbulent flow regimes by monitoring the Reynolds number (Re) and comparing its value with the critical Reynolds number (Recr). The flow rate is determined according to the following equations:

$$q = \begin{cases} C_D \cdot A \sqrt{\frac{2}{\rho}} \sqrt{|p|}\, sign(p) & for\ R_e > R_{cr} \\ 2 C_{DL} \cdot A \frac{D_H}{\nu \cdot \rho} p & for\ R_e < R_{cr} \end{cases} \qquad (1)$$

$$A(p) = \begin{cases} A_{leak} & for\ p \le p_{set} \\ A_{leak} + k(p - p_{set}) & for\ p_{set} < p < p_{max} \\ A_{max} & for\ p \ge p_{max} \end{cases} \qquad (2)$$

$$k = \frac{A_{max}}{p_{reg}} \qquad (3)$$

$$p = p_A - p_B \qquad (4)$$

$$R_e = \frac{q.D_H}{A(p).v} \qquad (5)$$

$$C_{DL} = \left(\frac{C_D}{\sqrt{R_{ecr}}} \right)^2 \qquad (6)$$

$$D_H = \sqrt{\frac{4\,A(\,p\,)}{p}} \qquad (7)$$

Where

q	Flow rate through the valve
p	Pressure differential across the valve
p_A, p_B	Gauge pressures at the block terminals
CD	Flow discharge coefficient
$A(\,p\,)$	Instantaneous orifice passage area
A_{max}	Fully open valve passage area
A_{leak}	Closed valve leakage area
p_{reg}	Regulation range
p_{set}	Valve preset pressure
p_{max}	Valve pressure at maximum opening
D_H	Instantaneous orifice hydraulic diameter
r	Fluid density
v	Fluid kinematics viscosity

Since the relief valve in the present SIMULINK Package is not supplied by pressure-setting solenoid, therefore, one can use dialogue parameter block to apply the required setting of pressure. The model is based on the following assumptions:

- Valve opening is linearly proportional to the pressure differential.
- No loading on the valve, such as inertia, friction, spring, and so on, is considered.
- The transition between laminar and turbulent regimes is assumed to be sharp and taking place exactly at $R_e = R_{ecr}$.

The pump, which is simulated with an ideal flow rate source, delivers fluid to a system through an initially-opened variable orifice. As the orifice gets closed, pressure gradually builds up and eventually reaches the setting of the pressure-relief valve. At this pressure, the valve starts diverting flow to a tank and maintains preset pressure at the pump outlet. As the orifice is opened again, the pressure-relief valve is closed and the entire flow passes through the variable orifice. The PS-Simulink and converter blocks convert a physical signal into a Simulink output signal, while Simulink-PS Converter block converts the input Simulink signal into a physical signal. One can use Simulink-PS block to connect outputs of a Physical Network diagram to Simulink scopes or other Simulink blocks.

The block represents a variable orifice of any type as a data-sheet-based model. Depending on data listed in the manufacturer's catalogs or data sheets for your particular orifice, you can choose one of the following model parameterization options.

Fig (11) In the first case, the passage area is assumed to be linearly dependent on the control member displacement, that is, the orifice is assumed to be closed at the initial position of the control member (zero displacement), and the maximum opening takes place at the maximum displacement. In the second case, the passage area is determined by one-dimensional interpolation from the table A=A (h). In both cases, a small leakage area is assumed to exist even after the orifice is completely closed. Physically, it represents a possible clearance in the closed valve, but the main purpose of the parameter is to maintain numerical integrity of the circuit by preventing a portion of the system from getting isolated after the valve is completely closed. An isolated or "hanging" part of the system could affect computational efficiency and even cause failure of computation.

4 Result & Discussion

All the worked experimental data listed in Table (1) and the characteristic curves have been measured at n = 41 mm2/s and T = 50 °C with approximate zero back pressure at port T. Figure (6) shows the relationship between setting pressure and signal voltage obtained experimentally at pressure rating of 25 bar, while these shown in Figure (8-a) and (9-a) explains the documented results, (references [6] and [7]) at rate of 30 and 25 bar, respectively. The difference between Figure (6) and Figure (7-a) is due to the difference in pressure rate.

Figure (7) explain the relationship between flow rate and minimum setting pressure obtained experimentally and at pressure stage 25 bar, while those shown in Figure (8-b) and Figure (9-b) obtained from document [6] and [7] and at stage pressure of 30 and 25 bar respectively.

When one compares minimum setting pressure shown in Figure (5) and that in Figure (8-a) and (9-a), it is clear that this minimum setting found in Figure (6) is higher than that obtained from documents counterparts. The reason for this is attributed to the additional flow resistances of hoses, pipes and quick-release couplings, which do not allow the pressure to reach 0 bar to exist directly after the proportional pressure relief valve.

Figure (12) shows the relation between the flow rate and pressure setting of the relieve-valve. One can easily deduce that the characteristic curve is considerably compatible to that obtained in practical case.

There are small discrepancies between the results from practical and simulated cases. This difference is expected due to temperature rise in each measuring case, which may in turn lead to change the system viscosity. Also, the losses in the pipe lines of the system are not taken into account.

Figure (13) describes the relation between input signal (volt) with pressure setting of the relieve-valve with different setting of flow valve. The change in flow valve setting would change the relief valve sensitivity to pressure setting of relieve valve. In other words, for fixed pressure setting of the relieve valve, solenoid voltage should be increased proportionally with amount of flow rate.

5. Conclusions

The setting pressure of a proportional pressure relief direct operated valve increases as the flow rate increases. The reason for this lies in the stroke control of the valve. With increasing flow rate, up to a certain limit the force of the pressure spring is increased since the valve poppet is raised progressively with increasing flow rate and the stroke control requires maintaining a certain armature position. As the spring force becomes greater, the setting pressure increases with increasing flow rate and constant signal value.

A low pressure Pmin (setting pressure) corresponding to the immediate flow rate might occur. The reason is due to 0 V signal value and then a power failure for the proportional solenoids would result.

These valves however can be used only for very small flow rates, as a higher pressure rating requires both design alterations (control land) and also a higher force from the proportional solenoid. This means that a limit set at this side necessitate the presence of pilot-operated proportional pressure relief valve.

References

[1] Massimo Rundo & Nicola Nervegna "Geometry Assessment of Variable Displacement Vans Pumps" ASME Vol.129 JULY 2007

[2] W. Kmmetmuller, S. Muller & A. Kugi "Mathematical Modeling and Nonlinear Controller Design for a Novel Electro hydraulic Power Steering System" IEEE/ASME Trans, On Mechatronics, Vol. 12 No.1 Feb.2007, pp.85-97

[3] Merrit H. E., "Hydraulic Control System" Wilegand Sons 1967

[4] Arno Schmitt "Proportional Valves, Component Technology" The Hydraulic Trainer volume (2) by Mannesmann Rexroth GmbH 1986

[5] Mannesmann Rexroth GmbH staff "Hydraulic & Electronic Components for Proportional and Servo Systems" 1988

[6] HU Yanping el, "Research on the Relief Valve with G-p Bridge Pilot Hydraulic Resistances Network" 2000

[7] Arthur Akers, Max Gassman, Richard Smith "Hydraulic Power system Aalysis" by Taylor & Francis Group, LLC 2006

[8] James A. Sullivan "Fluid Power:Theory and Applications" Fourth Edition by Prentice- Hall, Inc 1998

[9] Rexroth Bosch Group "Proportional Pressure Relief Valve, Direct Operated, falling characteristic curve" RE 18139-05/2005-2006

[10] Rexroth Bosch Group "Proportional pressure relief valve direct operated, without/with integral control electronics" RE 29162/2005-2006

[11] D. Scholz "Proportional Hydraulics Basic Level", by Festo GmbH, 2002.

[12] The Math Works "Simhydralic Notes", 2009, The Math Works Inc.

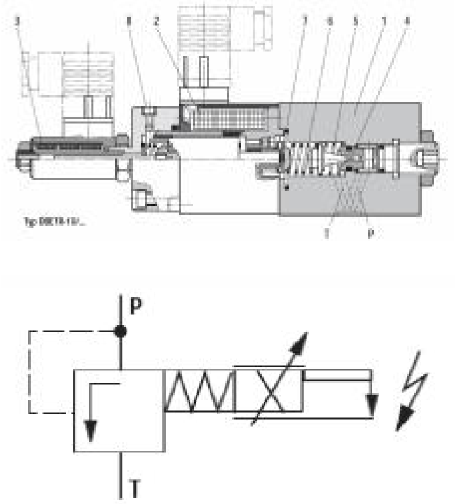

Figure (1) Direct operated proportional pressure relief valve type (DBETR-1X/25) with positional feedback of spring pre-loading [2, 7]

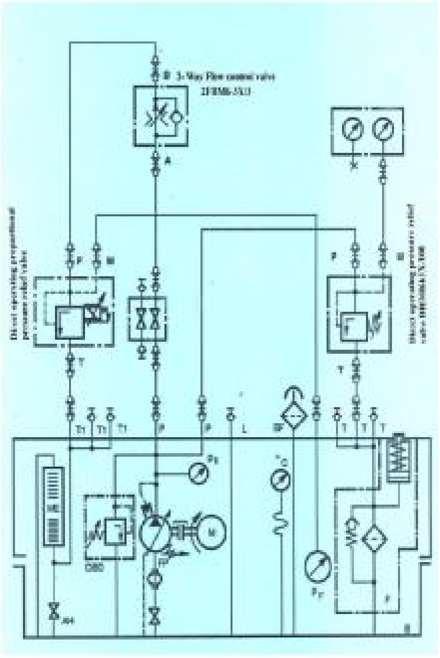

Figure (3) Connection Electro-hydraulic control circuit

Figure (2) Photograph of electro-hydraulic control unit test bench

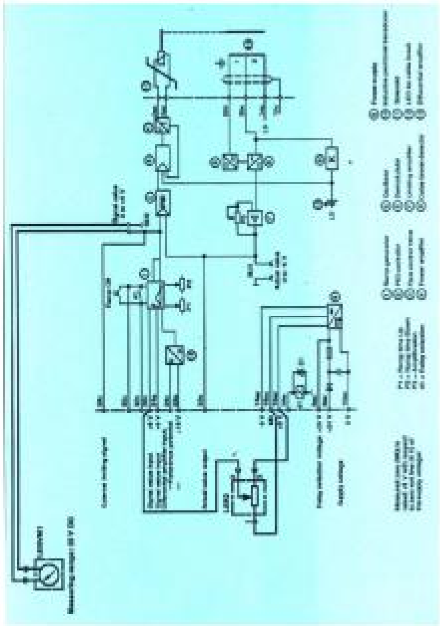

Figure (4) Electronic terminal for direct operated proportional pressure relief valve

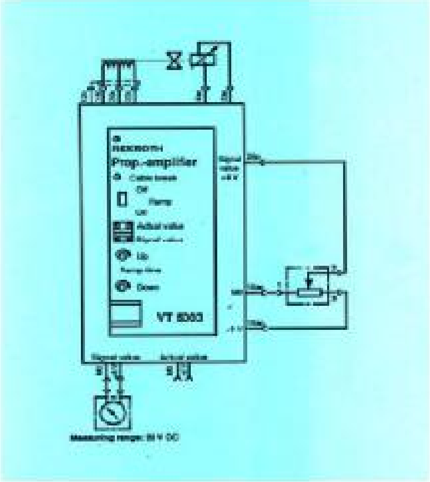

Figure (5) Circuit diagram (electrical circuit)

Table (1) Experimental data

Signal value voltage (volt)	0	2	3	4	4.5	5	5.5	6
2-way flow control valve (2FRM6) (at setting 4) [illegible] Q=1.5 L/min								
Setting pressure (bar)	0.2	1.0	1.3	1.4	2.0	3.2	5.7	12.8
2-way flow control valve (2FRM6) (at setting 6) [illegible] Q=4 L/min								
Setting pressure (bar)	1.8	2	2.4	4.1	6.2	9.1	12.5	17
2-way flow control valve (2FRM6) (at setting 8) [illegible] Q=6 L/min								
Setting pressure (bar)	3.2	4	5.2	8.6	11.1	14.5	18	22.8
2-way flow control valve (2FRM6) (at setting 10) [illegible] Q=7.5 L/min								
Setting pressure (bar)	5.6	7.4	9.6	12.8	17	18.6	22.1	25.5

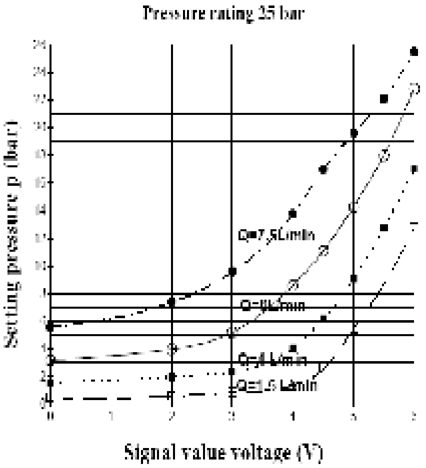

Figure (6) Characteristic curve signal voltage/setting pressure

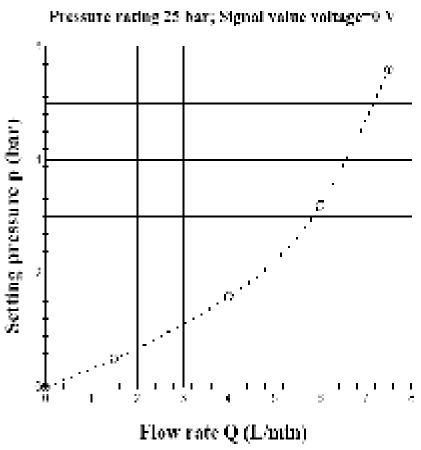

Figure (7) Characteristic curve-flow rate/minimum setting pressure

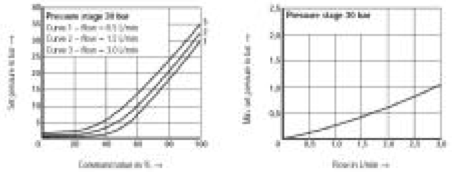

Figure (8) Operating curves for proportional valve type (DBETR-1X/25) [6, 7]

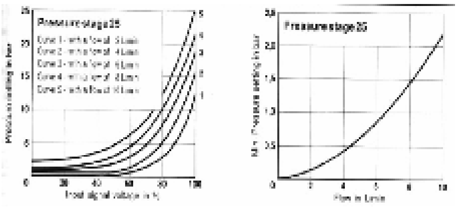

Figure (9) Operating curves for proportional valve type (DBETR-1X/25) [2]

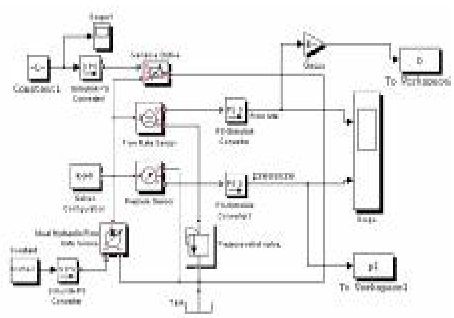

Figure (10) SIMULINK-simulated block diagram of operating pressure relief-valve

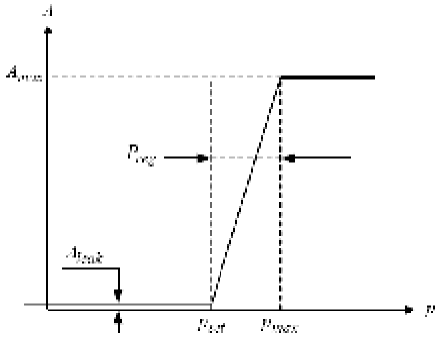

Figure (11) The relation between the valve passage areas
& the pressure differential across the valve

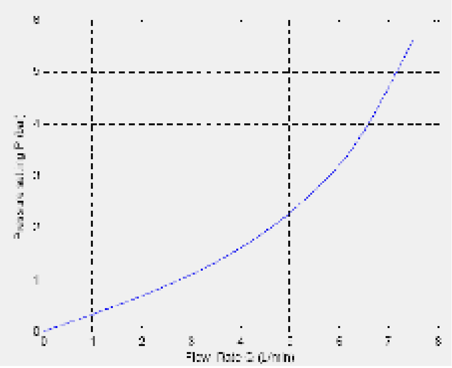

Figure (12) characteristic curve flow rate/ min. setting Pressure by simulatio*n*

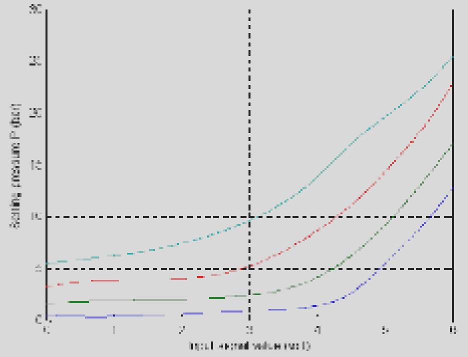

Figure (13) characteristic curve signal voltage/setting Pressure by simulation