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Performance Improvement of Pressure Compensated Flow Control Valve Using Virtual Engineering

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Abstract.

This paper addresses the performance challenges associated with pressure compensated flow control valves, particularly in maintaining effective pressure compensation at higher flow rates. The primary objective is to investigate how various design parameters influence the valve ability to maintain consistent flow under pressure variations. Given the complexities and resource demands of physical testing across numerous design variants, a more efficient approach is required. This study uses virtual engineering techniques to create a simulation model of the valve, enabling a comprehensive analysis of the impact of different design parameters on its performance.

Keywords. Pressure compensation, Flow control valve, Flow force, CFD, Mathematical model.

1. Introduction

Hydraulic systems are widely used in both industrial and mobile applications because of their high-power density and ability to generate large output forces [1]. Actuator motion control is required in most hydraulic applications, which can be achieved by regulating the flow rate of hydraulic oil. Table 1 summarizes the different components used to control flow rate in hydraulic systems.

Components	Remarks
Fixed displacement pump	- Requires a mechanism to regulate pump speed. - Cannot provide independent speed control for multiple actuators.
Variable displacement pump	- More expensive compared to fixed displacement pumps. - Cannot provide independent speed control for multiple actuators.
Flow control valves	- Allow independent speed control of multiple actuators.
Closed loop control with servo system	- Provides precise control of flow rate. - Involves higher complexity and cost.

Table 1: Overview of different components for flow rate control.

Flow control valves regulate the flow rate by varying the throttle area. Such valves are commonly referred to as throttle valves. They are available from several manufacturers, including F valves (Bosch Rexroth), T valves (Yuken), and Q valves (Atos). The main limitation of throttle valves is that the regulated flow rate depends on the pressure drop across the valve. To maintain a constant flow independent of pressure variations, pressure compensation is employed in valves. Pressure compensated flow control valves (referred as PCFCV here) are offered by several manufacturers, including FRM valves (Bosch Rexroth), F valves (Yuken), QV valves (Atos).

A new design of a pressure-compensated flow control valve for water hydraulics was developed [2]. The valve dimensions were determined by solving the simplified equations in MATLAB. Experimental investigations were then carried out on the prototype valve, which revealed the following key findings:

- The pressure–flow characteristics indicated that an increase in pressure led to a slight decrease in flow rate.
- Incorporation of flow-force compensation reduced the rate of flow reduction with increasing pressure.
- Flow-force compensation using an O-ring further improved the pressure–flow characteristics; however, it also introduced a negative impact on hysteresis.

[3] analysed such a valve using (i) steady-state analysis with a nonlinear model and (ii) dynamic response analysis with a linearized model. The nonlinear model was used to extract parameters for the frequency response model, from which flow-gain and pressure-coefficient transfer functions were derived. The study also examined the effect of various design parameters on valve dynamics, including critically compensated, over-compensated, and under-compensated conditions. Similar investigations have been reported in [4], [5], [6], focusing on the analysis and performance evaluation of pressure-compensated flow control valves.

Performance improvement of pressure compensated flow control valve can be done by improving flow control accuracy, effective pressure compensation and output flow stability. In reference [7] shows influence of main orifice shape on accuracy of flow control. They identified two shapes (triangle and circular) for proportional flow control valve. They created characteristic curves for valve using CFD data in conjunction with MATLAB/SIMULINK. [8] performed dynamic analysis of two stage poppet valve for flow control. The effect of flow forces on force equilibrium to maintain constant outflow is studied.

The present study aims to develop a methodology for predicting the performance of pressure-compensated flow control valves (PCFCVs) using CFD. For this purpose, 2FRM6 valve from Bosch Rexroth is used. Furthermore, the study seeks to enhance valve performance by investigating the influence of various design parameters on its dynamic and steady-state behaviour

2. Flow Control Valve: 2FRM6

2.1. Working of 2FRM6

The 2FRM6 is a pressure- and temperature-compensated two-way flow control valve. Figure 2.1 illustrates its schematic layout. The valve housing (1) accommodates a main sleeve (3) and a compensator sleeve (9). The main sleeve contains a main orifice (5) of triangular shape, the opening of which is regulated by the throttle screw (8). This screw moves axially within the main sleeve and is externally adjusted through the control knob (2).

The compensator spool (4), located within the compensator sleeve (9), governs the pressure-compensation mechanism. Flow from the main orifice discharges through circular holes in both the compensator spool and sleeve. The compensator spool is free to move axially to an equilibrium position determined by the balance of forces acting on it. Specifically, fluid pressure acts on its lower face via the nozzle (7) and on its upper face via the main orifice (5). The resulting position of the spool adjusts the throttling area, thereby compensating for pressure variations. Additionally, the spring (6) applies a restoring force on the compensator spool, which varies with the relative positions of the throttle screw and spool.

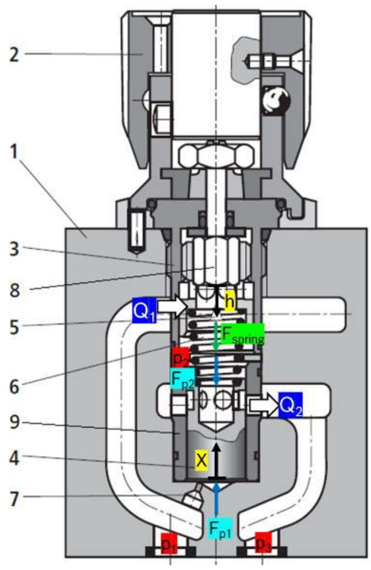


Figure 2.1: Schematic representation of 2FRM6: 1-Housing, 2-Adjustment knob, 3-Main sleeve, 4- Compensator spool, 5- Main orifice, 6- Spring, 7- Nozzle, 8- Throttle screw, 9- Compensator sleeve.

Subsequent section presents the approach used to estimate the performance of the 2FRM6 valve using CFD simulations. Before detailing the CFD methodology, a mathematical description of the valve operation is provided to illustrate how the equilibrium position of the compensator spool governs the flow characteristics. This formulation establishes the foundation for the subsequent numerical analysis by linking hydraulic forces and spring forces to the valve's operating points.

2.2. Mathematical Description of 2FRM6

Flow through main orifice is described by standard orifice equation as:

$$Q_1 = c_{d1} A_{main} \sqrt{\frac{2(p_1 - p_2)}{\rho}} \tag{2.1}$$

Similarly, the flow through compensator orifice can be described as:

$$Q_2 = c_{d2} A_{compensator} \sqrt{\frac{2(p_2 - p_3)}{\rho}} \tag{2.2}$$

where, c_{d1} and c_{d2} are discharge coefficients of main and compensator orifices, respectively, and A_{main} and $A_{compensator}$ are the effective flow areas of main orifice and compensator orifice. For simplicity, the downstream pressure p_3 can be assumed equal to tank pressure, which is taken as zero.

In valve performance evaluation, the main throttle area A_{main} is fixed according to the control knob setting. The discharge coefficients c_{d1}, c_{d2} are generally unknown and can be determined experimentally or through CFD simulations.

At steady state, the flow through both orifices must be equal ($Q_1 = Q_2$). By equating Eqs. (2.1) and (2.2), the intermediate pressure p_2 can be expressed as:

$$p_2 = \frac{c^2}{1 + c^2} p_1 \tag{2.3}$$

where, $c = \frac{c_{d1} A_{main}}{c_{d2} A_{compensator}}$.

The position of the compensator spool is determined from the equilibrium between the hydraulic and spring forces:

$$F_{spring} = F_{hyd} \tag{2.4}$$

The spring (6) is compressed between the throttle screw and compensator spool, and its force can be written as:

$$F_{spring} = F_0 + kx \tag{2.5}$$

where F_0 is the initial compression force dependent on the throttle screw position ($F_0 = f(h)$), k is the spring stiffness, and x is the axial displacement of the compensator spool. F_{hyd} consists of all type of forces acting on compensator, i.e., force due to pressure, flow, and friction. Considering only pressure force for simplicity one can express it as:

$$F_0 + kx = \frac{p_1 A}{1 + c^2} \tag{2.6}$$

A is area of compensator where pressure acts. Since c depends on $A_{compensator}$, which in turn depends on the spool displacement x, Eq. (2.6) is nonlinear and can be solved iteratively to determine the compensator position for a given inlet pressure p_1. Once the equilibrium position is known, the corresponding flow rate can be calculated using Eq. (2.1).

3. PERFORMANCE EVALUATION OF 2FRM6 USING CFD

The mathematical model provides a framework for understanding the steady-state behaviour of the 2FRM6. However, the model has inherent limitations. The discharge coefficients of the main and compensator orifices (c_{d1} and c_{d2}) are generally unknown and cannot be predicted analytically accurately without detailed flow information. Furthermore, while the model accounts for the pressure forces acting on the compensator spool, it neglects the effects of flow-induced forces and other secondary effects such as friction. These simplifications limit the predictive capability of the mathematical model for precise performance evaluation and hence, CFD simulations are employed. By combining the mathematical understanding of spool force balance with CFD-predicted pressure distributions, the equilibrium position of the compensator spool and the corresponding flow–pressure characteristics of the valve can be reliably evaluated.

3.1. Simulation Strategy

For each simulation, the main throttle screw was fixed at a prescribed position, while multiple axial positions of the compensator spool were considered to capture the variation in pressure and flow. This procedure can be repeated for different main spool positions to cover the full operational range of the valve. In the present study, the main spool position, h, is set to 2.8 mm, and five different compensator positions are considered.

For each compensator spool configuration, CFD simulations were performed to obtain the detailed pressure and velocity distributions. The resulting pressure fields were then combined with the spring force acting on the compensator spool to evaluate the equilibrium position using the force-balance relation. This procedure allowed the determination of the operating point corresponding to each spool configuration, from which the flow–pressure characteristics of the valve were obtained.

The computational domain was discretized using a hybrid mesh generated with commercial software. Mesh was refined in the vicinity of the orifices and the compensator spool to accurately resolve the pressure gradients. A representative mesh snapshot is shown in Figure 3.1, illustrating the resolution in critical regions.

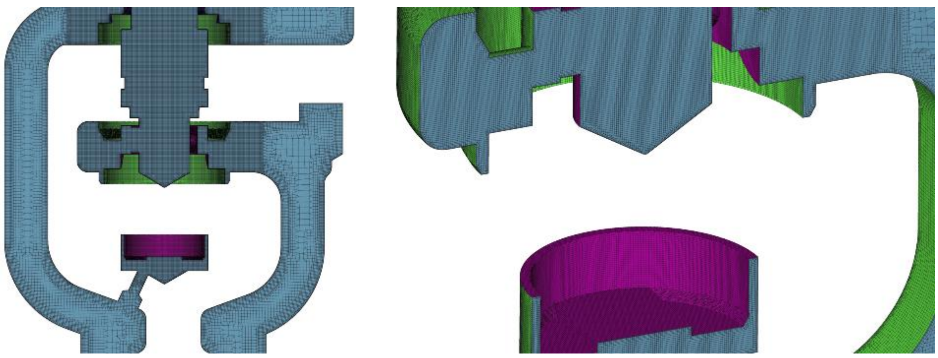


Figure 3.2: Snapshot of discretized domain.

3.2. CFD Model Setup and Parameters

Steady-state CFD simulations were performed using commercial software to resolve the pressure and velocity fields within the 2FRM6 valve. The fluid was treated as incompressible and isothermal, with properties corresponding to HLP46 hydraulic oil at 40 °C. Boundary conditions were applied as follows: a prescribed pressure at the inlet (upstream of the main orifice) and zero-gauge pressure at the outlet (tank). No-slip conditions were applied at all solid surfaces, including the valve housing and spool.

Software provides a wide range of turbulence modelling options. Among them realizable $k - \epsilon$ model [9] was selected, as it is widely regarded as a suitable choice for simulating turbulent flow in hydraulic valves and has been successfully employed in several previous studies [10], [11]. The boundary values of turbulent kinetic energy (k) and dissipation rate (ϵ) were specified using standard relations:

$$k = \frac{3}{2}\left(\frac{V}{TI}\right)^2 \tag{2.7}$$

$$\epsilon = k^{3/2}/L \tag{2.8}$$

Here, V is free stream velocity estimated using orifice equation and using a discharge coefficient of 0.7. TI is turbulence intensity assumed as 3%. L is the turbulence length scale taken as port radius.

4. Results

This section presents the results obtained from the CFD simulations described in the previous section. The subsequent section will focus on the validation of these CFD results through comparison with experimental data.

4.1. Experimental Validation

Figure 4.1 shows the experimental test bench used to measure the performance of the 2FRM6 valve. The throttle screw of the valve was set to the fully open condition (scale position 10), which corresponds to the maximum flow capacity. CFD simulations were carried out for the same configuration, and the figure also presents a comparison between experimental measurements and CFD predictions. The results demonstrate that the CFD predictions are in very good agreement with the experimental data. Next section, provide in detail discussion on valve performance.

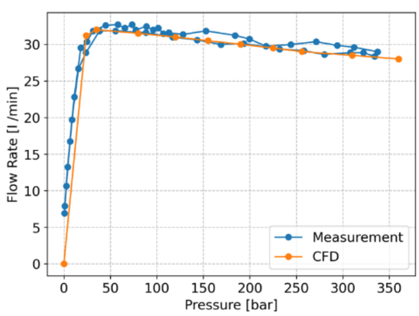


Figure 4.1: Experimental test bench and comparison of CFD and measurement data.

4.2. Discussion on Valve Performance

In a pressure-compensated flow control valve, the performance curve exhibits a distinct characteristic of maintaining nearly constant flow irrespective of pressure variations. At low flow rates, the pressure drop increases gradually, governed primarily by the resistance of the valve geometry. As the flow continues to rise, the compensator mechanism becomes active, adjusting its position in response to the pressure differential across it. Once the compensator engages, any further increase in flow is restricted because the compensator movement effectively reduces the available flow area. This behaviour is represented graphically by a plateau region in the performance curve, where the flow rate remains constant despite variations in pressure. After the plateau region, where the valve maintains nearly constant flow, the performance curve may exhibit a slight fall in the flow rate. This decline occurs due to nonlinear effects that influence the valve dynamics. Maximum flow capacity of 2FRM6 valve is 32 l/min, with the decline in flow rate being highest at this maximum and lower at reduced rates. The force balance can be disturbed by factors like friction in the compensator mechanism and the flow forces acting on the spool. It is important to note that when the inlet pressure is the same, this flow force is higher at increased flow rates and lower at reduced flow rates. These nonlinearities reduce the effective control of the compensator, causing a gradual decrease in flow rate even though the pressure differential remains high. The width of the plateau depends on flow rate and the extent of nonlinearities.

CFD simulations predict the decline in flow even without modeling effects such as friction, indicating that flow forces are the dominant source of nonlinearity in this case. Figure 4.2 illustrates the forces acting on the compensator at a flow rate of 32 l/min, along with the corresponding spring force. As the stroke increases (i.e., as the flow area decreases), the flow force rises sharply compared to the spring force. This rapid increase in flow force causes the compensator to move further than required for ideal pressure compensation, resulting in an excessive reduction in flow area. Consequently, the flow rate decreases at higher pressures.

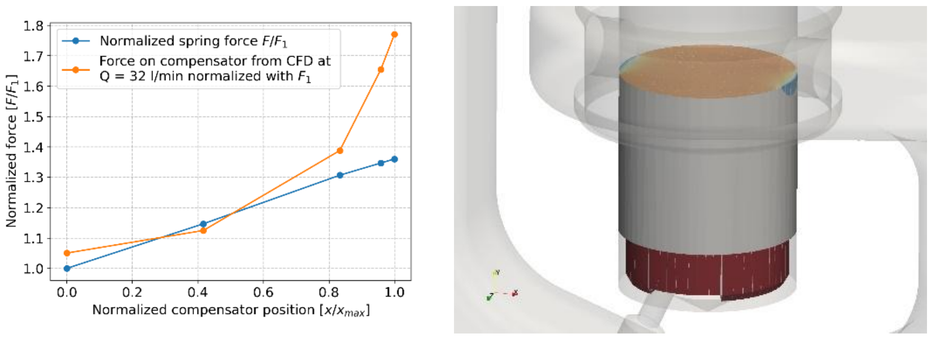


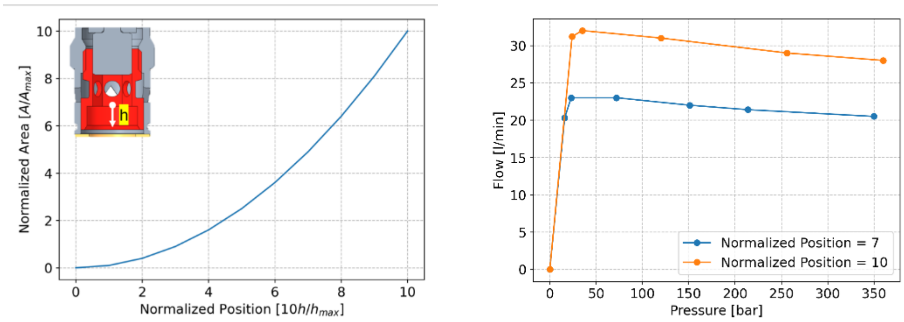
Figure 4.2: Flow force from CFD and pressure profile on compensator surface.

4.3. Influence of Design Parameters

The performance of a pressure-compensated valve is strongly affected by its design parameters. Key elements such as the main orifice, spring, compensator design determine how the valve responds to flow and pressure variations. Understanding the influence of these parameters is essential for optimizing valve behaviour, achieving the desired flow control, and ensuring stable operation under different operating conditions. This section presents the influence of these design parameters on the valve performance.

4.3.1. Main Orifice Area

The main orifice (5) shown in Figure 2.1 plays a crucial role in the valve operation. While the orifice area determines the set flow rate through the valve, the orifice shape controls how this area changes with the movement of the throttle screw. Triangular-shaped orifices are commonly preferred to achieve a gradual increase in flow area. Figure 4.3 illustrates the variation of orifice area with different throttle screw positions and presents the characteristic curves of the valve for two set positions. It can be observed that changing the throttle screw position affects only the set flow, without altering the overall valve behaviour.

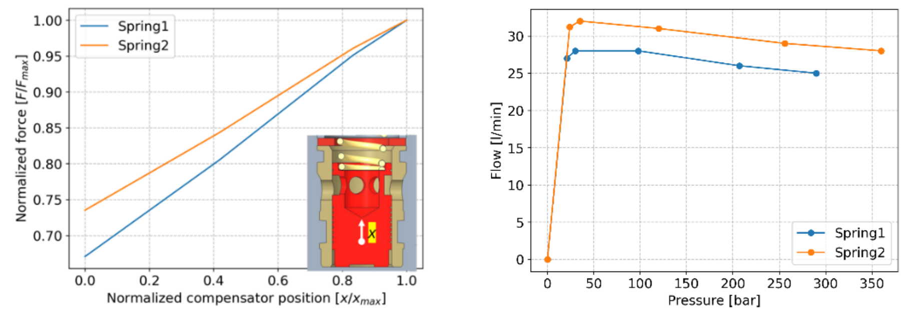


(a) (b)

Figure 4.3: (a) Variation of the main orifice area with the throttle screw position, (b) Performance curves corresponding to two different set positions of the throttle screw.

4.3.2. Influence of Spring

The spring in a pressure-compensated valve plays a crucial role in determining its operational characteristics. The regulation of flow rate under varying pressure conditions is primarily governed by the force balance between the spring and the compensator. The spring's initial compression force (F_1), its rate, and any nonlinearity directly influence the valve's response.

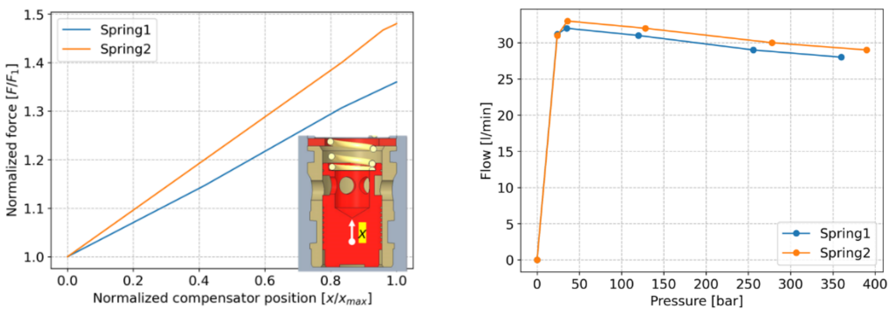


(a) (b)

Figure 4.4: (a) Normalized spring force with normalized compensator position, (b) Performance curves corresponding to two different springs.

Figure 4.4 illustrates the influence of the initial spring compression force on valve performance. Figure 4.4(a) shows the normalized spring force plotted against the compensator position. In this study, the initial compression of the same spring was increased while maintaining the same spring rate. As the spring rate remains unchanged, the force at $x/x_{max} = 1$ also increases, and this value is used for normalization. Two characteristic curves corresponding to different initial forces are presented. Increasing the initial compression force raises the valve's set pressure. As shown in the figure, with spring 1 the flow is set to 27 l/min at 21 bar, whereas with spring 2 it increases to 31 l/min at 24 bar.

Figure 4.5 shows influence of spring rate on performance curve. Figure 4.5(a) shows normalized force for spring1 and spring2 for different compensator position. Spring force is normalized by initial compression of the spring. It can be observed that with changing spring rate there is minor change in set flow, however, it does not influence the remaining characteristic of the flow.



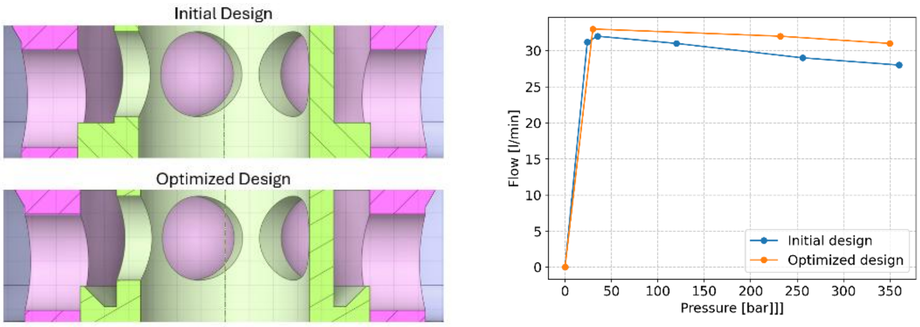
(a) (b)

Figure 4.5: (a) Normalized spring force with normalized compensator position, (b) Performance curves corresponding to two different springs.

The variation of flow rate with inlet pressure is governed by the force balance between the compensator and the spring. As discussed earlier (see Figure 4.2), the nonlinear nature of the flow-induced force acting on the compensator plays a decisive role in defining the valve's pressure–flow characteristics. To achieve a stable response, the spring must generate a corresponding nonlinear force. This can be realized through a conical spring with variable stiffness.

4.3.3. Compensator Design

The design of the compensator plays an important role in determining the overall performance of a pressure-compensated flow control valve, as the flow-induced forces act directly on its surface. By optimizing the compensator geometry—particularly the profile of the surface exposed to flow—the magnitude of these forces can be effectively reduced. Incorporating sharper edges can limit the area over which the pressure distribution acts, thereby decreasing the effect of flow forces on compensator. Such geometric refinements improve the valve's ability to maintain a consistent flow rate under varying pressure conditions.



(a) (b)

Figure 4.6: (a) Design changes of compensator, (b) Performance curves corresponding to two different compensator designs.

5. CONCLUSION

A comprehensive methodology has been developed for predicting the performance of a pressure-compensated flow control valve. The approach enables accurate estimation of pressure–flow characteristics and internal forces under different operating conditions. The predicted results have been validated against experimental measurements, showing good agreement and demonstrating the reliability of the developed simulation methodology. Using this validated model, the influence of key design parameters—such as the main orifice geometry, spring characteristics, and compensator land dimensions—has been systematically investigated. The study highlights how each parameter governs the valve's set pressure and compensating behaviour. The findings emphasize the importance of proper selection and optimization of these design parameters to achieve desired performance and ensure consistent pressure compensation across the operating range.

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Biographies

Harsh Maniar received the bachelor's degree in mechanical engineering from Gujarat Technological University in 2013, the master's degree in mechanical engineering from Indian Institute of Technology, Dhanbad in 2017, respectively. He is currently working as Computational Fluid Dynamics engineer at the Bosch Rexroth India. He has broad experience in macroscopic particle method, multiphase flow and transient simulation. His research interests include flow-induced forces and pressure drop analysis.

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