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OPTIMIZATION AND EXPERIMENTAL VERIFICATION OF A VARIABLE RATIO FLOW DIVIDER VALVE

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Abstract

A common problem in fluid power control systems is the synchronization of two loads. A frequent solution to this problem is to use a flow divider valve. Typical flow divider valves can deliver flow to two circuits such that the ratio of flows is independent of the load pressures, but it is not possible to easily change the ratio after manufacturing. In this paper, the design process used to develop a new variable ratio flow divider valve is introduced. As the first step, a preliminary model was used to optimize the physical parameters for a prototype. In the second step, a valve was subsequently constructed and the performance experimentally determined. The prototype constructed exhibits low dynamic and steady state error with low pressure losses in experimental tests. This novel valve shows sufficient feasibility to warrant future study and development for commercialization.

Keywords: flow divider, hydraulic valve, variable ratio, multiobjective optimization, experimental

1 Introduction

Hydraulic flow divider valves are often used to synchronize two loads, such as two cylinders which must extend at the same rate. In a flow divider valve, one inlet flow is divided into two outlets such that the ratio of the outlet flows is constant, usually 50:50. In a typical flow divider valve this ratio is set during manufacture and cannot be easily changed. A great deal of research has been conducted on reducing the flow dividing error in such fixed ratio valves (Chan, 1981; Guo, 1988; Federoff, 1990) but no studies to date have been reported on the development of a variable ratio flow divider valve. This was the motivation for this research.

The object of this paper is to show the design procedures that were used to develop a variable ratio flow divider valve. Based on this new valve concept, a preliminary model was developed and used as a basis for valve design (Wiens, 2004). This paper will demonstrate the use of a multiobjective optimization technique to produce appropriate geometrical parameters for the final design. Based on this optimization, a physical prototype was fabricated and evaluated experimentally. The experimental performance of the design is presented and discussion on its application potential forwarded.

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2 Valve Description

In order to facilitate a description of the variable ratio valve, it is desirable to understand the operation of a fixed ratio (conventional) spool flow divider valve. A conventional spool type flow divider valve is shown in Fig. 1, and includes two main parts: a pair of fixed orifices and a pair of hydrostatic (variable) orifices mounted on one spool. With reference to Fig. 1, the pressures downstream of the fixed orifices, P_{a1} and P_{a2}, act on the spool ends to adjust the hydrostatic orifices to maintain equilibrium. In the ideal case, in which flow forces and friction at equilibrium are neglected, $P_{a1} = P_{a2}$. From the turbulent orifice equation for the fixed orifices, the two inlet flows, Q_{s1} and Q_{s2}, are

$$Q_{s1} = C_{d1} A_{o1} \sqrt{\frac{2}{\rho}(P_s - P_{a1})} \tag{1}$$

$$Q_{s2} = C_{d2} A_{o2} \sqrt{\frac{2}{\rho}(P_s - P_{a2})} \tag{2}$$

where P_s is the inlet pressure, ρ is the fluid density, and C_{d1} and C_{d2} are approximately constant for large Reynolds number flows (Merritt, 1967). If $P_{a1} = P_{a2}$ at equilibrium, and identical orifices are used ($A_{o1} = A_{o2}$), then the flows are equal. If the load changes, the pressure change will force the spool to a new equilibrium position. For example, if Q_{s1} momentarily increases, P_{a1} will decrease relative to P_{a2}. This results in a force imbalance on the spool, which tends to move the spool to the left. This closes the left orifice and opens the right one, thereby decreasing Q_{s1} and increasing Q_{s2}. This continues until P_{a1} again equals P_{a2} and therefore $Q_{s1} = Q_{s2}$. It should be noted that while the magnitudes of the flows may have changed, the ratio has not.

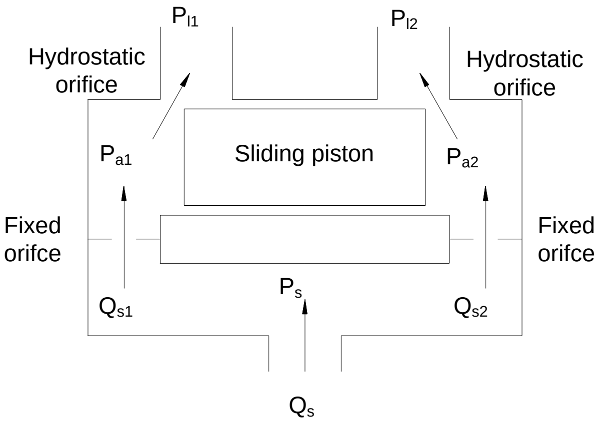

Fig. 1: *A typical fixed-ratio flow divider. In the ideal case, the spool compensates for changes in load pressure such that P_{a1}=P_{a2}, maintaining the ratio of Q_{s1} to Q_{s2}*

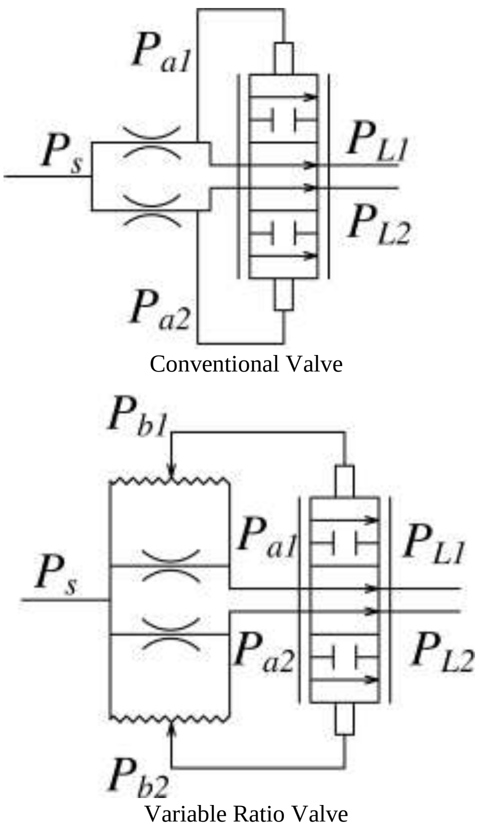

Fig. 2: *A conventional flow divider (top) and the variable ratio flow divider (bottom). Rather than connecting the intermediate pressures P_{a1} and P_{a2} directly to the spool ends, the variable ratio valve uses two hydraulic bridges in parallel with the fixed orifices to modify the pressures in order to control the ratio of outlet flows*

In the ideal case, the ratio of outlet flows is equal to the ratio of $C_{d1}A_{o1}$ to $C_{d2}A_{o2}$, which is generally equal to the ratio of the areas of the fixed orifices. Unless a second spool is used to adjust the relative areas of the "fixed" orifices, the divider can only be used to divide the flow at a fixed ratio after the valve has been manufactured. The accuracy of the division ratio is also limited by the accuracy of the orifice sizes.

The adjustable-ratio flow divider valve proposed in this study works on a similar principle to that of a conventional spool flow divider. However, the controlling pressures acting on the spool ends are modified by the pilot stage, as shown in Fig. 2. In a conventional flow divider spool, the pressures acting on the spool are ported directly downstream of the fixed orifices. In the proposed valve, two hydraulic bridges are connected in parallel with the fixed orifices to modify these driving pressures. A hydraulic bridge is the fluid equivalent of an electrical rheostat. Whereas the output of a rheostat is an electrical potential that is linearly dependent on voltage tap position, a hydraulic bridge taps a pressure that is linearly dependent on the position of the pressure tap and the pressure differential across the bridge. In the design the two bridges are physically connected (to be discussed) such that when one pressure tap moves toward P_s the other moves further away (toward P_{a2} for instance)

The basic physical construction of the valve is shown in Fig. 3 and 4; the concept takes advantage of the "2-D" concept introduced by Ruan et al. (2001) and Cui (1991) to allow both the pilot stage and main stage to exist on the same spool. The spool has two motions: linear (sliding along the sleeve axis) and rotary (the complete spool rotates), hence the label "2-D". The linear motion provides the flow dividing function (as in a traditional valve). The rotary motion is used to set the position of the hydraulic bridges of the pilot stage, producing control pressures which act on the outermost lands of the spool to move it linearly to compensate for load changes.

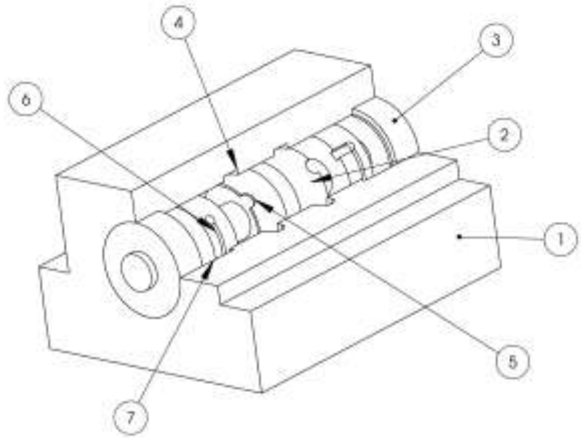

Fig. 3*:* *Cutaway view of the varible ratio flow divider, showing casing (1), spool (2), and seal glands (3). The main stage consists of grooves in the casing (4) and semicircular notches in the spool (5), while the pilot stage consists of circumferential grooves in the spool (6) and axial grooves in the casing (7)*

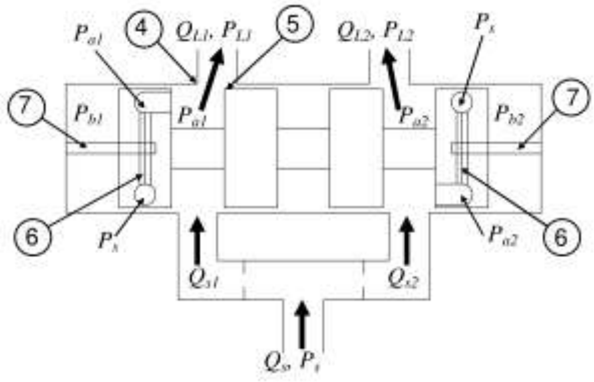

Fig. 4: *Schematic of the varible ratio flow divider, using same labels as Fig. 3*

On each of the outer lands, there is a circumferential groove, labeled 6 in Fig. 3 and 4, the ends of which are connected to the source pressure (P_s) and the intermediate pressure for each branch (P_{a1} or P_{a2}). Partway along the length of this circumferential groove is a perpendicular groove (the sensing channel, 7, in Fig. 3 and 4) in the valve casing which intersects the first groove and hence “taps” the output pressure. This output is connected to the spool end chamber at pressure P_{b1} or P_{b2}. The position of the circumferential groove relative to the sensing channel is determined by the angular position of the spool. At the extreme positive angle, (that is, the groove 7 is lined up with the top slots on both sides of the end spool lands) $P_{b1} = P_{a1}$ and $P_{b2} = P_s$. At the zero angle position (that is, the groove is lined up with the bottom slots of the end spool lands) $P_{b1} = P_s$ and $P_{b2} = P_{a2}$, while intermediate positions give pressures that vary approximately linearly between the P_s and P_b values. Thus, by rotating the spool with a stepper motor (or indeed, manually), the spool end pressures can be adjusted, which, in turn, can be used to set the dividing ratio.

In order to determine how this configuration actually results in a variable ratio flow divider valve, it is necessary to examine an ideal steady state model of the valve.

3 Ideal Steady State Model

Consider Fig. 2. Under the ideal steady state conditions of no friction and no flow forces, P_{b1} must be equal to P_{b2}. However, P_{a1} will not be equal to P_{a2} if the spool is rotated away from the centre position (for unequal flow division). Assume the valve spool has been rotated to some “noncentred” position to create a non equal flow division situation. From a steady state position, if the load changes such that the flow in branch 1 (the top branch in Fig. 2) increases, the intermediate pressure, P_{a1}, will decrease. Since one of the ends of the hydraulic bridge is connected to P_{a1}, P_{b1} will also decrease (although not to the same extent as P_{a1} since the other end of the hydraulic bridge is connected to the source pressure). As there is now a force imbalance on the spool (P_{b1} is less than P_{b2}), the spool will move to the top, closing the orifice in branch 1 and opening that in branch 2. This tends to reduce the flow in branch 1 (increasing P_{a1} and hence P_{b1}) and to increase the flow in branch 2 (decreasing P_{a2} and P_{b2}). P_{a1} now increases and P_{a2} decreases until force equilibrium is again achieved when P_{b1} equals P_{b2} and the flows are in the original ratio. This ratio is determined by how much a change in P_{a1} effects P_{b1} which, in turn, is determined by the position of the hydraulic bridges (or the spool’s rotary position).

If the inlet pressure is at P_s, the turbulent orifice equation (Eq. 1 and 2) can be used to calculate the flows Q_{s1} and Q_{s2} in each branch (Note: it is assumed that the discharge coefficients and fluid properties are constant). In the absence of leakage across the piston lands and for small pilot flows, at steady state these flows are equal to the outlet load flows, Q_{L1} and Q_{L2}.

As discussed above, the major difference between a conventional flow divider and the proposed 2D valve is the rotary pilot stage, consisting of two hydraulic bridges, each of which contains a circumferential groove on the spool, intersected by a perpendicular sensing channel in the casing. It is assumed that the resistance of the bridge is sufficiently large enough that by-pass flow is small compared to the main flows and hence flow is laminar (linear pressure distribution within the bridge). On the left hand side of the spool, the clockwise end of the bridge is connected to P_{a1} and the counterclockwise end is connected to P_s. The pressure along this groove varies approximately linearly between P_{a1} and P_s if there are no leakage or entrance effects (Ruan et al., 2001). On the right hand side, P_s is at the clockwise end and P_{a2} is at the counterclockwise end, so

$$P_{b1} = P_s - \left(\frac{\phi}{\phi_0}\right)\left(P_s - P_{a1}\right) \tag{3}$$

$$P_{b2} = P_{a2} - \left(\frac{\phi}{\phi_0}\right)\left(P_{a2} - P_s\right) \tag{4}$$

where ϕ is the angular position of the spool and ϕ_0 is the maximum angle, at which the sensing channel reaches the end of the groove.

In the absence of flow forces and friction and if the spool end areas are equal, the spool moves to an equilibrium position such that

$$P_{b1} = P_{b2} \tag{5}$$

By solving the system Eq. 1, 2, 3, 4 and 5, the ratio of steady state load flows, Q_{L1}/Q_{L2}, is found to be

$$\frac{Q_{L1}}{Q_{L2}} = \frac{C_{d1}A_{b1}}{C_{d2}A_{b2}}\sqrt{\frac{\phi_0}{\phi} - 1} \tag{6}$$

This equation demonstrates that ideally, the flow ratio can be changed by simply adjusting the angular position of the spool with respect to the angular length of the groove.

4 Optimization

A comprehensive dynamic model used in the design of the variable ratio flow divider was developed in Wiens (2004) and consisted of ordinary differential equations expressing the outlet flows and internal pressures, given inputs of source flow, spool rotary position, and load orifice areas. It was an objective to use this model to optimize critical geometrical parameters before a prototype could be constructed.

The first step toward the goal of optimizing a design was to quantify the "goodness" of the parameter set. The valve presented in this paper should have a number of properties:

- The flow division error should be low. This error has both dynamic and steady state components, as defined later in this section.
- The pressure losses across the valve should be low.
- There should be no signs of instability.
- The design should be inexpensive and easy to manufacture.

In order to evaluate the performance of the simulated valve against these criteria, a standard test was defined. For the model developed in Wiens (2004), there were inputs of source flow, Q_s, spool rotary position, ϕ, and the load orifice areas A_{L1}, and A_{L2}. As a standard test, the rotary position of the spool was constant at the centred position ($\phi = \phi_0/2$) and a constant flow of $Q_s = 1\times10^{-3}$ m^3/s (60 L/min) was applied. The orifice areas were initially set so that the steady state load pressures were both 10 MPa. The simulation was allowed to reach steady state; then the orifice area of one branch (herein referred to as branch 1) was decreased such that the steady state load pressure increased to 15 MPa (while the other area was held constant).

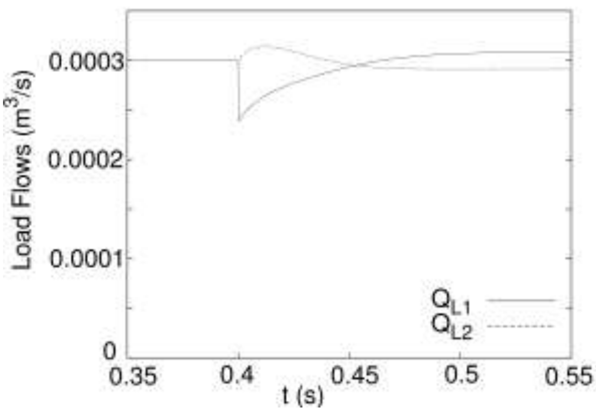

Fig 5: *Simulated valve outlet flows for some sample parameters. A step of 5 MPa was applied to one load pressure at t=0.4 s*

In an industrial setting, these values of flow and pressure would typically be matched to the desired application. However, as this was a research valve, the values were selected to be compatible with the equipment and instrumentation available in the university laboratory. Figure 5 shows the resultant outlet flows when the simulation was run with some sample (non optimized) parameters. This non-optimized simulated result shows that both a dynamic and steady state error does exist even for the special case of 50-50 flow division.

In order to evaluate the performance of a valve such as this, some quantitative accuracy measurements were required. The most common measure of error is steady state error, defined for a conventional flow divider as

$$\%E_{ss} = \frac{Q_{L1} - Q_{L2}}{Q_s} \times 100\% \tag{7}$$

However, this assumes that the ratio of output flows is 50-50. As the novel feature of the proposed valve is the ability to change the flow division ratio, a new definition of steady state error is required. The following definition allowed for changing division ratios:

$$\%E_{ss} = \frac{|Q_{L1} - Q_{L1d}| + |Q_{L2} - Q_{L2d}|}{Q_s} \times 100\% \tag{8}$$

where the "*d*" subscript denotes the desired flow, calculated using Eq. 6.

Guo (1988) noticed that when a flow divider valve is used to synchronize two hydraulic actuators, there can be a significant positional error accumulated before the system reaches steady state. This is due to the large temporary flow dividing error that occurs before the valve can compensate. Guo defined this cumulative error for a 50:50 divider ratio as

$$E_{CU} = \int_0^{t_{cum}} \left(Q_{L2} - Q_{L1}\right) dt \tag{9}$$

where t_{cum} is the test time. In this study, a modified version is proposed which takes into account the possibility of a changing divider ratio:

$$E_{CU} = \int_0^{t_{cum}} \left[\left(Q_{L1} - Q_{L1d}\right) - \left(Q_{L2} - Q_{L2d}\right)\right] dt \tag{10}$$

It can be informative to separate the total cumulative error into dynamic and steady state portions, as shown in Fig. 6. The cumulative steady state error is

$$E_{CUSS} = \left[\left(Q_{L1ss} - Q_{L1d}\right) - \left(Q_{L2ss} - Q_{L2d}\right)\right] t_{cum} \tag{11}$$

where Q_{Lss} is the steady state flow. The cumulative dynamic error is then

$$E_{CUDY} = \int_0^{t_s} \left[\left(Q_{L1} - Q_{L1ss}\right) - \left(Q_{L2} - Q_{L2ss}\right)\right] dt \tag{12}$$

where ts is the settling time. These can be expressed as a percentage by comparing them to the total flow over the cumulative time:

$$\%E_{CUDY} = \frac{\int_0^{t_s} \left[\left(Q_{L1} - Q_{L1ss}\right) - \left(Q_{L2} - Q_{L2ss}\right)\right] dt}{Q_s t_{cum}} 100\% \tag{13}$$

$$\%E_{CUSS} = \frac{\left|\left(Q_{L1ss} - Q_{L1d}\right) - \left(Q_{L2ss} - Q_{L2d}\right)\right|}{Q_s} 100\% \tag{14}$$

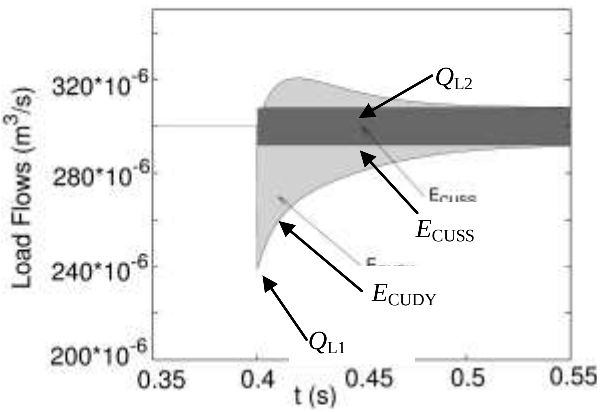

Fig. 6: *A sample flow response to a step in one load pressure. The cumulative steady state error (E_{CUSS}) and accumulated dynamic error (E_{CUDY}) are shaded*

Thought must be put into the selection of the test length, t_{cum}, when comparing cumulative errors, as this determines the weighting of the dynamic and steady state contributions to the total cumulative error. For example, if a test time close to the settling time is chosen, the steady state portion is less important than the dynamic part. However, as the test time increases, the contribution of the steady state error grows, but the dynamic portion stays constant, decreasing its relative importance. The intended application of the valve should be used to determine t_{cum}; for example, if the valve was to be used to drive conveyors, a very large time would be chosen, as the load pressure is likely constant over periods of minutes or hours. Conversely, if the valve was to be used to control fuel flow to an engine, the load pressure changes occur on the order of milliseconds. In this study, the test time was selected as 0.2s, which should be suitable for many mobile industrial applications.

The first two qualities required of the valve, steady state and dynamic accuracy, have been considered. The third quality, pressure loss, needs a few words of explanation. The purpose of the flow divider valve is to maintain a set ratio of flows by controlling the pressure drop across the adjustable orifices to compensate for the difference in load pressure. Therefore, it is impossible to reduce the pressure drop across one branch of the valve beyond this load pressure difference. However, the pressure drop in the other branch can theoretically go to zero, and should be reduced as much as possible. The valve with the best efficiency (and heat production traits) is one that causes the lowest pressure drop in the branch with the greatest load pressure. For the rest of this paper, when the pressure drop across the valve is discussed, this is the intended meaning.

The fourth quality, instability, could be determined, for example, by linearizing the model equations and applying the Routh-Hurwitz criterion (Phillips and Harbor, 2000). For a complex nonlinear system, it can be difficult to ensure stability, as the calculation must be repeated for all possible operating points. Since a numerical simulation is available, the results of a commercial simulation package such as Matlab®/Octave® can be used to determine possible regions of instability.

The final criteria, ease and cost of manufacturing, is perhaps too complex to leave to a computational algorithm to determine an accurate value, as finding a single function that can estimate manufacturing time and cost requires years of experience and expertise beyond that of the authors. However, it is possible to qualitatively evaluate certain aspects of the design. For example, it is more difficult and expensive to manufacture a valve with very small clearances (requiring tight tolerances) than one with larger clearances and looser tolerances. Since no way of automatically calculating a quantitative value for this item was available, it will only appear in the analysis in a qualitative form.

The next step in optimizing the parameter set is deciding which parameters strongly affect the performance of the valve, so that less important parameters can be ignored, simplifying the optimization. The effect of parameters on these qualities can be displayed in tabular form using the "House of Quality" (Andersson, 2001), shown in Table 1. In this table, a higher number indicates that the parameter has a strong effect on the objective, while a blank or low number shows a weak relationship. The table also includes a column ranking the importance of each objective (higher numbers mean increasing importance and a "D" stands for a demand requirement; for example, the valve must be stable, but in this application there is no advantage to being "more" stable).

This table was compiled by systematically varying each parameter and examining the result of the output of the simulation (refer to Wiens (2004) for details). From this table one can see that the most important qualities are the spool radius, r, the width of the hydraulic bridge grooves, b, the spool-to-casing clearance, δ the total pressure drop across the valve, ΔP_t, the ratio of the pressure drop across the fixed orifices to the total drop, P_o/P_t, the fractional opening of the variable orifices when the spool is centred, F_O, and the orifice rim thickness, trim.

Table 1: House of Quality

	weight	***r***	***b***	$\boldsymbol{\delta}$	$\boldsymbol{\Delta P_t}$	$\boldsymbol{P_o/P_t}$	$\boldsymbol{F_O}$	**trim**
steady state error	3	4	3	4	4	2	1	4
dynamic error	3	3	3	1	4	2	3	1
pressure losses	1	0	1	0	4	2	3	0
stability	D	4	2	1	3	1	1	1
cost	2	2	2	4	0	0	0	3

Since there are a number of objectives that work in opposition to each other, this problem is classified as a "multiobjective optimization problem". One approach to this problem is to combine the objectives into one aggregate number, which is referred to by a number of terms such as the fitness or cost function. The major drawback of this approach is that it is often difficult to quantitatively compare the value of two different qualities, for example, accuracy and pressure drop. The goal attainment method (Matlab®, 2002) was used to partially overcome this problem.

Based on the goal attainment method, a general aggregate objective function was calculated by minimizing the maximum value of the absolute value of matrix λ which was composed of elements

$$\lambda_{\mathrm{i}} = \frac{F_{\mathrm{i}}\left(\boldsymbol{X}\right) - F_{\mathrm{i}}^{*}}{F_{\mathrm{i}}^{*}} w_{\mathrm{i}} \tag{15}$$

where $F_{\mathrm{i}}(\boldsymbol{X})$ is the i^{th} objective for the parameter vector $\boldsymbol{X}$, F_{i}^{*} is the goal for that objective, and w_{i} is the weight assigned to that objective (from the House of Quality).

The House of Quality used for the optimization is shown in Table 2. In this case, the spool clearance and rim thickness do not appear as they are set at $\delta = 3\ \mu$m and t_{rim} = 0.8 mm, based on an estimate of the best values reasonably attainable, supplied by the intended manufacturer of the prototype. The objectives ease and cost of manufacture also do not appear, as they are difficult to quantify.

Table 2: Simplified House of Quality

	weight	***r***	***b***	ΔP_{t}	Po/P_{t}	F_{O}
steady state error	3	4	3	4	2	1
dynamic error	3	3	3	4	2	3
pressure losses	1	0	1	4	2	3
stability	D	4	2	3	1	1

Following the procedure introduced by Andersson (2001), a genetic algorithm and the complex method were used in parallel to minimize the above function. The resultant parameter set (shown in Table 3) was then used to manufacture the first prototype. An experimental evaluation of this prototype is presented in the next section.

Table 3: Optimized Parameters

r	13.3 mm
b	1.35 mm
ΔP_{t}	7.91 MPa
$P_{\mathrm{o}}/P_{\mathrm{t}}$	0.886
F_{O}	0.421

5 Experimental Verification

A prototype valve was constructed using the physical parameters found by the optimization strategy outlined above. The valve is shown in Fig. 7.

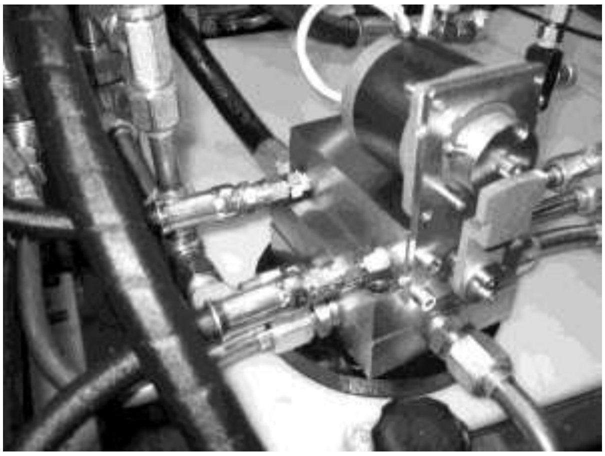

Fig. 7: *Photo of the prototype valve. The valve body is approximately 100 mm long*

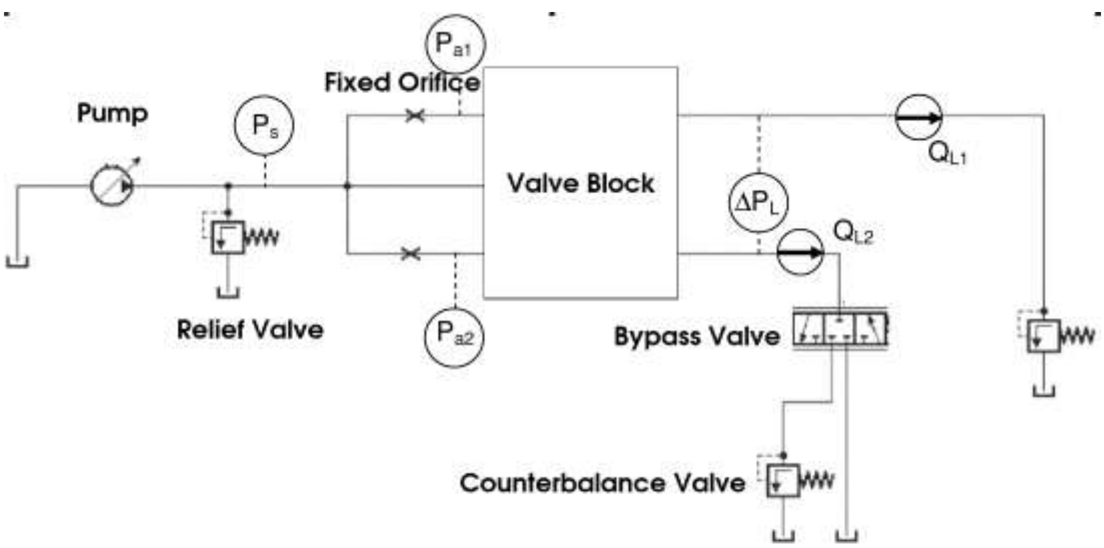

Fig. 8: *The hydraulic circuit used to experimentally test the valve*

in this paper. Instead, the experimental program was intended to verify the trends that the simulation pre

The hydraulic circuit used to test the valve is shown in Fig. 8. Two counterbalance valves were used as loads, with a three-way valve connected so that the flow could bypass one load. This allowed the load pressure to be quickly reduced, approximating a step in load pressure. As this valve needed to be easily modified for research purposes, it was designed to be modular, with external (adjustable) fixed orifices and external plumbing. This greatly increased the involved volumes (slowing performance) and introduced pressure losses (reducing the range of division ratios), but the extent of these effects could be estimated using the numerical simulation. Also, many ports were drilled in the casing to facilitate pressure measurements, which also increased the volumes.

The objective of the first test performed was to verify that the valve could compensate for changes in load flow. The valve was connected as described above and the response of the flow to a step change in load pressure was recorded (the counterbalance valves took some time to respond, and as such, the input was not an ideal step). The two outlet flows are shown in Fig. 9. This test was performed with a total flow of 4.99×10^{-4} m^3/s (29.9 L/min), a 427 kPa pressure drop across the fixed orifices and the spool's rotary position set for approximately equal flow division. The load pressures were initially set to be approximately equal and at the step the pressure in branch number 1 dropped so that the load pressure difference was 1640 kPa. The pressure measurements recorded are shown in Fig. 10.

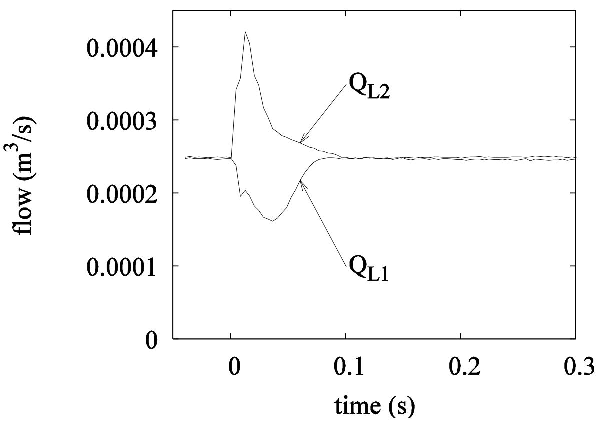

Fig. 9: *Experimental response of load flows to a step change in load pressure*

The valve behaved as expected. With reference to Fig. 9, before the step in load pressure difference, the flows were nearly equal, and the valve attempted to equalize them after the load disturbance. There was some steady state division error due to flow forces and static friction, in this case 1.23%. There was also an accumulated flow error as the valve could not respond instantaneously. For the test shown below, the accumulated error was −0.55 L. With reference to Fig. 10, notice that the load pressure difference is not a perfect step, as the counterbalance valves take some time to respond. The intermediate pressures, P_{a1} and P_{a2}, do behave as expected; they are nearly equal before the step, have some error as the valve responds, but then converge to a relatively small steady state error as time progresses.

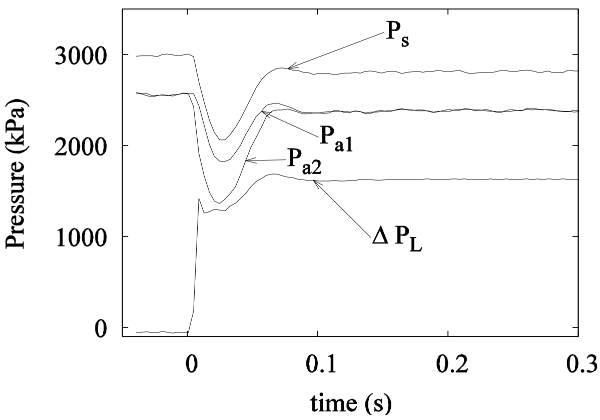

Fig. 10: *Pressures for the same response as in Fig. 9. Notice that ΔP_L is not a true step*

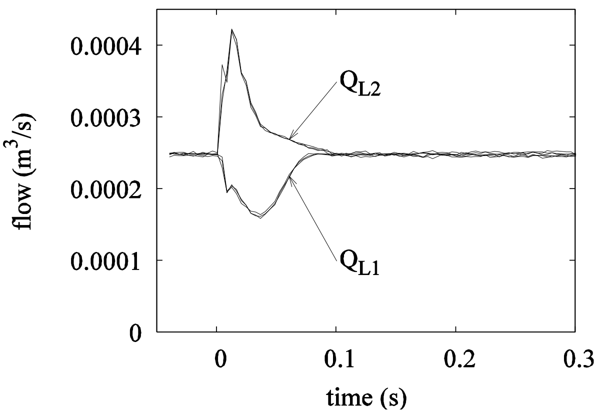

Fig. 11: *Three tests superimposed, showing repeatability of the test*

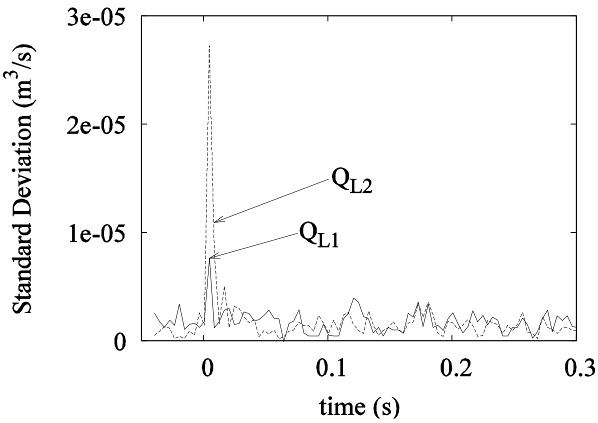

Fig. 12*: Standard deviation of the tests shown in Fig. 11*

It was necessary to verify that valve performance was repeatable at an arbitrary typical operating point. Three step responses were recorded at similar operating conditions. The responses are superimposed over each other in Fig. 11. These tests were performed at the same operating conditions as above. The standard deviation of the multiple tests is plotted in Fig. 12. Notice that the difference between tests was greatest in the time period just after the step, but decreased as time progressed. This suggests that valve steady state performance was quite stable, but the transient response varied from test to test (perhaps in response to variables such as operating temperature).

One of the objectives of the experimental program was to verify the trends predicted by the pre-design analysis. The first trend to be examined was that the steady state and cumulative dynamic error increased as the pressure drop across the fixed orifice decreased. What this implies is that improved accuracy can be obtained, but at the cost of efficiency. Numerous tests were performed at approximately the same operating conditions, while varying the size of the fixed orifices. These tests were performed with $Q_s = 4.99 \times 10^{-4}$ m^3/s (29.9 L/min) and ΔP_L = 1630 kPa. The results are shown in Table 4, and behave as predicted: the steady state error decreased from 1.23% to 0.73% as the pressure drop increased from 154 kPa to 1460 kPa and the accumulated dynamic error decreased from −0.71 to −0.39 L.

Table 4: The effect of the pressure drop across the fixed orifices on the cumulative dynamic error. These data are the average of a number of trials with a mean $Q_s = 4.99 \times 10^{-4}$ m^3/s (29.9 L/min) and ΔP_L = 1630 kPa.

ΔP_{fixed} (kPa)	$\%E_{SS}$ (%)	E_{CUDY} (L)
154	1.23	-0.712
430	0.87	-0.549
1459	0.73	-0.386

A second test was conducted to verify that the steady state error increased with load pressure difference. This would show the effect of flow forces on the valve. A number of step responses were recorded at varying load pressure differences and the error was calculated. The steady state results, shown in Table 5, also agree with the theoretical prediction.

Table 5: The effect of the load pressure difference on the steady state and dynamic error. These data are an average of a number of tests taken with a mean $Q_s = 4.96 \times 10^{-4}$ m^3/s (29.8 L/min) and ΔP_{fixed} = 534 kPa.

ΔP_L (kPa)	$\%E_{SS}$ (%)	E_{CUDY} (L)
679	0.070	-0.348
1649	0.734	-0.430
3350	1.257	-0.425

All of the above tests were performed with the spool rotary position centred (for equal flow division). The purpose of the final test was to verify the valve's ability to adjust the ratio of flows by changing the spool rotary position. Pursuant to this purpose, a number of step responses were recorded with the spool in two rotary positions: centred and fully counter-clockwise at the end of the gear travel (the spool could be rotated further if a gear was used with more teeth). The resultant flow responses are shown in Fig. 13. It is evident that the ratio of outlet flows can be adjusted, as designed.

The range of this particular valve is smaller than a production valve for two reasons; the gear used didn't have a large enough rotary range and any pressure losses in the long pilot line connected to P_s caused the ratio to t_{end} toward 50:50 (if similar fixed orifices are used).

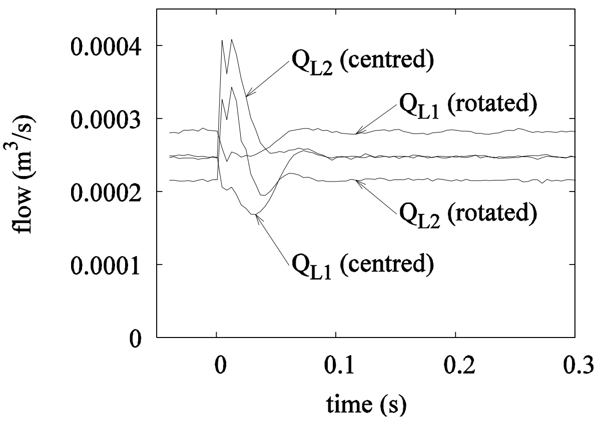

Fig. 13: *Flow responses, showing the effect of rotating the spool. Each curve is the average of a number of tests taken with a mean $Q_s = 4.96 \times 10^{-4}$ m^3/s (29.8 L/min), a mean ΔP_{fixed} = 553 kPa, and a mean ΔP_L = 1649 kPa*

6 Conclusions

In this paper, the optimization and design of a new type of a variable ratio flow divider valve was presented: Although flow divider valves exist commercially, none permit the user to change the ratio of outlet flows without requiring remachining. Furthermore, the valve has only one moving part and is therefore more reliable and less expensive to produce than a valve with separate pilot and main stages. A multiobjective optimization strategy was used to optimize the physical parameters for a prototype, which was subsequently tested experimentally. The prototype constructed exhibits low dynamic and steady state error with low pressure losses in experimental tests. This novel valve shows sufficient feasibility to warrant future study and development for commercialization.

Nomenclature

$\%E_{ss}$	Steady state error	[%]
$\%E_{CU}$	Cumulative error	[%]
$\%E_{CUSS}$	Cumulative steady state error	[%]
$\%E_{CUDY}$	Cumulative dynamic error	[%]
δ	Radial clearance between spool and casing	[m]
ΔP_{fixed}	Pressure drop across fixed orifices	[Pa]
ΔP_L	Load pressure difference	[Pa]
ΔP_t	Total pressure drop across the valve	[Pa]
ΔP_o	Pressure drop across fixed orifices	[Pa]
λ	Optimization matrix	
ϕ	Angular position of spool	[rad]
ϕ_o	Maximum angular position of spool	[rad]
ρ	Fluid density	[kg/m^3]
A_L	Area of load orifice	[m^2]
A_v	Area of variable orifice	[m^2]
A_o	Area of fixed orifice	[m^2]

b	Width of pilot groove	[m]
C_d	Discharge Coefficient	
E_{CUDY}	Cumulative dynamic error	[m^3]
E_{CUSS}	Cumulative steady state error	[m^3]
E_{CU}	Cumulative total error	[m^3]
F	Objective matrix	
F^*	Goal matrix	
F_O	Fractional opening of the variable orifices when the spool is centred	
P_a	Intermediate pressure	[Pa]
P_b	Pressure acting on spool	[Pa]
P_L	Load pressure	[Pa]
P_s	Source pressure	[Pa]
Q_L	Flow to load from valve	[m^3/s]
Q_{Lo}	Flow through load orifice	[m^3/s]
Q_{Lxd}	Desired load flow	[m^3/s]
Q_{Lxss}	Steady state load flow	[m^3/s]
Q_S	Source flow	[m^3/s]
r	Spool radius	[m]
t	Time	[s]
t_s	Settling time	[s]
t_{cum}	Time used when calculating cumulative error	[s]
w	Weights for optimization	

References

Andersson J. 2001. *Multiobjective Optimization in Engineering Design*. PhD thesis, Linkoping Universitet, Sweden.

Chan, R., **Schoenau, G.** and **Burton, R.** 1981. A Simple Design Modification for Improved Accuracy of Piston Type Flow Divider Valves. *Proceedings of the 35th Conference on Fluid Power*, October, Chicago.

Cui, P. 1991. *Development of a high-speed on/off valve*. Master's thesis, University of Saskatchewan.

Fedoroff M. 1990. *Analysis of steady state and dynamic characteristics of an auto-regulator valve*. Master's thesis, University of Saskatchewan.

Guo, Q., **Burton, R.** and **Schoenau, G.** 1988. Dynamic Response of Flow Divider Valves. *Journal of Fluid Control*, Vol 19(1), pp. 20–42.

Matlab. 2002. *Matlab Help*. The Mathworks, Natick, MA.

Merritt, H. E. 1967. *Hydraulic Control Systems*. John Wiley & Sons, New York,

Phillips C. and **Harbor, R.** 2000. *Feedback Control Systems*. Prentice Hall, New Jersey.

Ruan, J., **Burton, R.**, **Ukrainetz, P.** and **Xu, Y.** 2001. 2-D pressure control valve. *Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science*, Vol. 215(9), pp 1031–1038.

Wiens, T. 2004. *An Adjustable-Ratio Flow Dividing Hydraulic Valve*. Master's thesis, University of Saskatchewan.

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