

## Behavioural prediction of hydraulic step-up switching converters

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In this paper the fundamental principles of energy-conservative hydraulic control based on the fluid inertance principle are discussed and a detailed analysis of step-up switched-inertance control is presented. A non-loss system comprising an inertance tube and switching valve is modelled and its operational curves are presented as a reference for an ideal behaviour. Considering the load loss at both the tube and the PWM switched valve, a linear mathematical model for the step-up switched-inertance hydraulic system is presented which describes the pressure response as a function of the PWM duty cycle. Mathematical expressions of the flow rates through the tube and the supply and return ports as well as the system efficiency are also presented. A system prototype is evaluated on a test rig and the experimental data compared with the theoretical results, demonstrating the model accuracy. The proposed model simplifies the analysis process for step-up switching converters and thus their restrictions and potential can be investigated more quickly.

**Keywords:** digital hydraulics; hydraulic switching converter; hydraulic valve; PWM switched valve

### 1. Introduction

In most power hydraulic systems the speed and/or force of the load are controlled using valves to throttle the flow and limit or reduce the hydraulic pressure or reduce the flow rate. Since this is an energy dissipative process, the hydraulic circuit efficiency is often lower than 50%. Furthermore, a valve can only serve to reduce, and not increase, the flow or pressure. Other alternatives using variable displacement pump or motor, fixed pump driven by variable speed electrical motor, and hydro-mechanical transformer are available technologies and applicable depending on the working cycle and equipment cost constraints.

An alternative design, named switched-reactance hydraulics, was presented by F. Brown in the 1980s as being an energy-conservative control (Brown 1984, Brown 1987, and Brown *et al.*, 1988). The hydraulic control principle is based on the cyclical acceleration and deceleration of fluid using pulse-width modulation. As mentioned by Brown (1987) the switched-reactance hydraulic system was developed independently of electrical switched power converters, but the concepts are the same and a direct analogy exists between them.

As discussed by Scheidl *et al.* (2008), the hydraulic switching control was used in the late 18th century. Last decade this subject became focus of research again as an alternative for energy efficiency increase as can be seen in Manhartgruber *et al.* (2005), Kogler and Scheidl (2008), Hettrich *et al.* (2009), and Johnston (2009), among others.

Using the nomenclature from the electrical area, basic types of switching devices have been studied. These are the step-up or boost circuit, the step-down or

buck circuit, and the Cuk-converter which can step up or step down (Brown 1987, Kogler and Scheidl 2008). Brown (1987) also proposed another concept named the switching gyrator which was not normally associated with electrical design.

Several aspects can influence the switching fluid inertance including pressure wave propagation, control orifice switching time, non-linear load losses, leakages, and fluid capacitances. Despite of that, it is shown in this paper the well-known linear modelling of buck and boost electrical circuits is applicable for hydraulic circuit prediction.

This paper is organized as follows. In Section 2, the step-up converter is described and in Section 3 this system is modelled. An ideal device is analysed as well as a realistic one including the resistance associated with tube and switching orifices. Section 4 presents the experimental setup and the system parameters. In Section 5 experimental and theoretical results are compared confirming the model validity and Section 6 exemplify the model use predicting the system performance according to the switching frequency. Finally, Section 7 provides the main conclusions.

### 2. Step-Up PWM Valves

From the electrical point of view the function of switching-converter circuits is to convert an unregulated DC input into a regulated DC output. The efficiency is usually up to 98% and these devices can provide an output that is greater than the input.

The step-down or buck converter is used to convert a DC voltage to a lower DC voltage of the same polarity.

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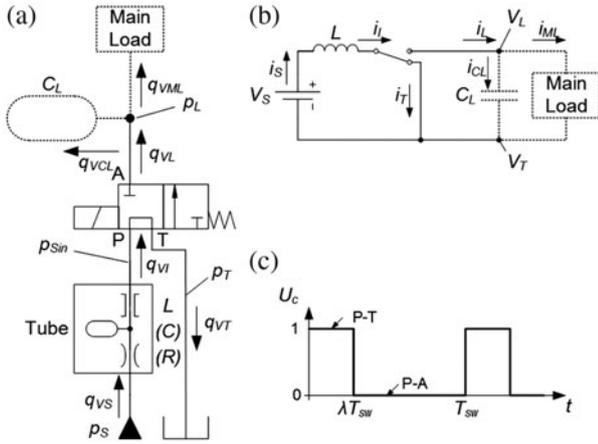


Figure 1. Step-up circuit: (a) Hydraulic circuit; (b) Electrical circuit; (c) PWM signal.

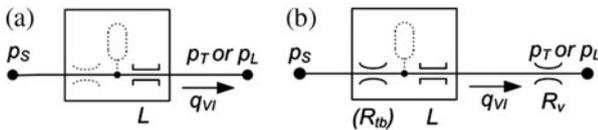


Figure 2. Step-up fundamental hydraulic circuit: a) Ideal system; b) System with resistances.

In the step-up or boost regulator the output voltage is higher than the input voltage. In an electric-hydraulic analogy, an equivalent hydraulic system will be working as a pressure control valve. A constant supply pressure is presupposed to be available. On the other hand, in the switching gyrator proposed by Brown (1987) the output current is proportional to the input voltage, and

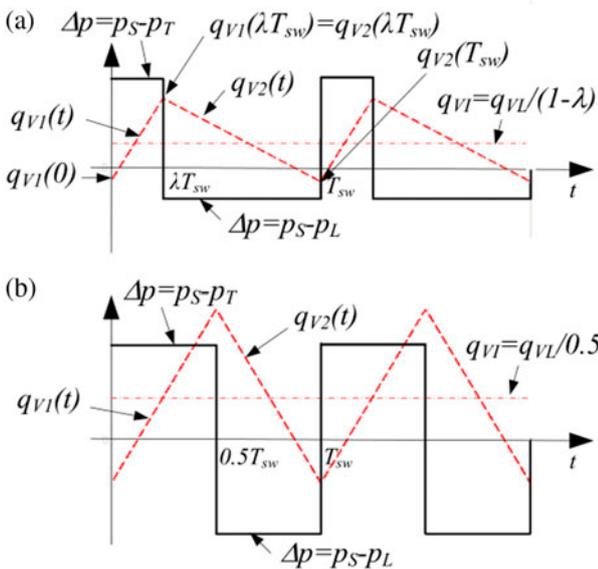


Figure 3. Inertance tube response for a square wave (ideal system): (a) General response; (b) Response for  $\lambda = 0.5$ .

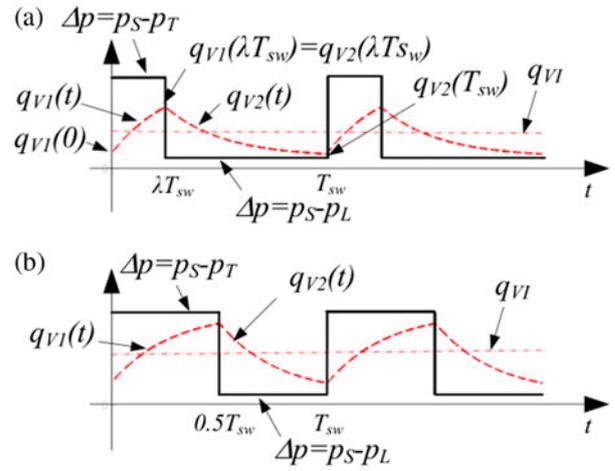


Figure 4. Inertance tube response for a square wave (system with resistance): (a) General response; (b) Response for  $\lambda = 0.5$ .

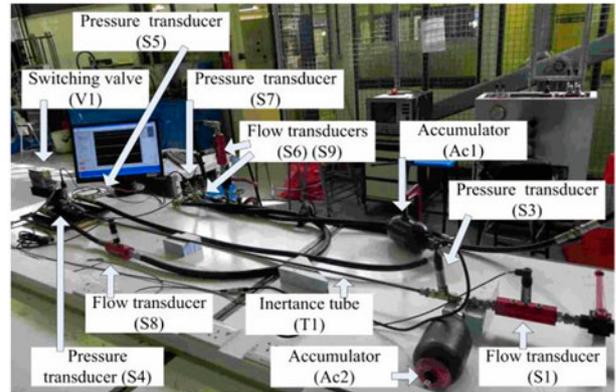


Figure 5. Test rig at the Centre for Power Transmission and Motion Control.

vice-versa. Therefore, the corresponding hydraulic system operates as a flow control valve.

The fundamental circuit of step-up switching converter and its electrical counterpart are shown in Figure 1. In this system, the directional valve is driven by a pulse-width modulated (PWM) signal such that it switches cyclically and rapidly, modulating the time in which each flow path remains active. The other major element is the hydraulic tube which ideally has only inertance effect ( $L$ ), but hydraulic resistance ( $R$ ) and capacitance ( $C$ ) effects are also present in practical systems, as represented in this Figure 1.

When the flow path P-T is active the internal pressure ( $p_{Sin}$ ) is reduced and the fluid is accelerated through the tube. As the valve is switched to the other position (P-A and T blocked), the fluid momentum in the tube causes the internal pressure to increase and, consequently, the load pressure ( $p_L$ ) to increase. In the next time period the load port will be blocked again while the fluid is accelerated. Under a theoretical point

of view, with a duty cycle ( $\lambda$ ) of 100% (P–T and A blocked) the load pressure tends to infinity and with a duty cycle of 0% (P–A and T blocked)  $p_L$  and  $p_{Sin}$  are equal to  $p_S$ . Therefore, ideally the load pressure ( $p_L$ ) can be modulated from the supply pressure value to infinity achieving twice the supply pressure for  $\lambda = 50\%$ .

The basic assumption for the operation of this system is that there is a capacitance ( $C_L$ ) associated with the load circuit such that it absorbs the flow variation produced by the switching process and the load pressure remains constant for a specific duty cycle value. Therefore, a system comprised of the inertance and capacitance, whose natural frequency is given by Equation (1), must filter the switching frequency.

$$\omega_n = \frac{1}{\sqrt{LC_L}} \quad (1)$$

A good design rule is to choose a load capacitance value such that the natural frequency is less than a tenth of the switching frequency (Brown, 1987).

Since the step-up circuit is a pressure regulator it must deliver an average flow rate equal to the flow rate consumed by the main load. Consequently, the average load pressure will remain constant and theoretically proportional to the duty cycle.

### 3. Step-Up PWM valve modelling

#### 3.1. Ideal Valve Model

Considering the step-up circuit presented in Figure 1 and assuming that the load capacitance is high enough for the load pressure to be considered constant, the switching circuit can be analysed separately from the loading circuit. Therefore, one can perform the behavioural analysis based on the fundamental circuit shown in Figure 2(a), where the pressure difference ( $\Delta p$ ) through the inertance tube is considered as the input and the inertance tube flow rate ( $q_{VI}$ ) as the output.

In the step-up circuit the pressure at tube upstream end is constant and equal to  $p_S$  while the pressure at the downstream end is being switched between the return pressure ( $p_T$ ) and the load pressure ( $p_L$ ).

The ideal inertance tube behaviour can be expressed by

$$\Delta p = L \frac{dq_{VI}}{dt} \quad (2)$$

where  $\Delta p = p_S - p_T$  for  $0 \leq t \leq \lambda T_{sw}$ ,  $\Delta p = p_S - p_L$  for  $\lambda T_{sw} \leq t \leq T_{sw}$ ,  $\lambda \in \mathfrak{R}$  in  $[0, 1]$  defines the duty cycle, and  $T_{sw}$  is the PWM signal period.

Therefore, the inertance tube corresponds to an integrator such that the flow rate behaviour for successive pressure steps is as shown in Figure 3. The tube flow rate can be expressed by

$$q_{VI}(t) = q_{VI}(0) + \frac{p_S - p_T}{L} t \quad \text{for } 0 \leq t \leq \lambda T_{sw} \quad (3)$$

and

$$q_{VI}(t) = q_{VI}(\lambda T_{sw}) + \frac{p_S - p_L}{L} (t - \lambda T_{sw}) \quad \text{for } \lambda T_{sw} \leq t \leq T_{sw} \quad (4)$$

Substituting  $t = \lambda T_{sw}$  in Equation (3) and  $t = T_{sw}$  in Equation (4), the tube flow amplitude can be written as

$$\Delta q_{VI} = \frac{p_S - p_T}{L} \lambda T_{sw} = -\frac{p_S - p_L}{L} (1 - \lambda) T_{sw} \quad (5)$$

Consequently,

$$p_L = \frac{p_S - p_T \lambda}{1 - \lambda} \quad (6)$$

Observing Figure 1 and 3(a), one concludes that the average load flow rate ( $q_{VL}$ ) can be calculated by integration of Equation (4) through the interval  $\lambda T_{sw} \leq t \leq T_{sw}$  and dividing by  $T_{sw}$ . Consequently, being  $q_{VI}(0) = q_{VI}(T_{sw})$ ,

$$q_{VI}(t) = \frac{q_{VL}}{1 - \lambda} + \frac{g}{L} \frac{1}{2} T_{sw} (1 - \lambda) + \frac{f}{L} t \quad \text{for } 0 \leq t \leq \lambda T_{sw} \quad (7)$$

and

$$q_{VI}(t) = \frac{q_{VL}}{1 - \lambda} - \frac{g}{L} \frac{1}{2} T_{sw} (1 - \lambda) + \frac{g}{L} (t - \lambda T_{sw}) \quad \text{for } \lambda T_{sw} \leq t \leq T_{sw} \quad (8)$$

where  $f = p_S - p_T$  and  $g = p_S - p_L$ , the average supply flow rate ( $q_{VS}$ ) is equal to the average flow rate through the inertance tube ( $q_{VI}$ ) and can be obtained by integration of the sum of equations Equations (7) and (8) through the entire interval and dividing by  $T_{sw}$ , resulting in

$$q_{VS} = q_{VI} = \frac{q_{VL}}{1 - \lambda} \quad (9)$$

The average return flow rate ( $q_{VT}$ ) can be obtained by integration of Equation (7) through the interval  $0 \leq t \leq \lambda T$  and dividing by  $T_{sw}$ , resulting in

$$q_{VT} = \frac{q_{VL} \lambda}{1 - \lambda} \quad (10)$$

The step-up efficiency can be determined through the following equation where the use of Equations (6), (9) and (10) equations provides an efficiency of 100% as expected for an ideal system.

$$\eta = \frac{p_L q_{VL}}{p_S q_{VS} - p_T q_{VT}} \quad (11)$$

#### 3.2. Model Including Linear Resistance

In this section the influence of the valve and tube resistances is included, as shown in Figure 2(b). As previously stated, in the step-up circuit there are two different valve flow paths that are switched alternately, consequently connecting the tube downstream end to the load

or the reservoir. Assuming that both flow paths have the same resistance and adding the tube resistance, the circuit model can be described by

$$\frac{L}{R} \frac{dq_{VI}}{dt} + q_{VI} = \frac{1}{R} \Delta p \quad (12)$$

where  $\Delta p = p_S - p_T$  for  $0 \leq t \leq \lambda T_{sw}$ ,  $\Delta p = p_S - p_L$  for  $\lambda T_{sw} \leq t \leq T_{sw}$ , and  $R = R_v + R_{tb}$ .

Aiming to achieve the time response of this system for a square wave input the approach presented by Millman & Taub (1965) for electric circuits is used. Therefore, since the steady-state output corresponds to the steady-state gain multiplied by the step magnitude, the flow rate through the hydraulic step-up circuit can be expressed by

$$q_{V1}(t) = \frac{1}{R} f + (q_{V1}(0) - \frac{1}{R} f) e^{-t/\tau} \quad (13)$$

for  $0 \leq t \leq \lambda T_{sw}$

and

$$q_{V2}(t) = \frac{1}{R} g + (q_{V2}(\lambda T_{sw}) - \frac{1}{R} g) e^{-(t-\lambda T_{sw})/\tau} \quad (14)$$

for  $\lambda T_{sw} \leq t \leq T_{sw}$

where  $f = p_S - p_T$ ,  $g = p_S - p_L$ , and  $\tau = L/R$ .

Figure 4(a) shows the graphic representation for these functions as well as identifies their specific values at instants zero,  $\lambda T_{sw}$ , and  $T_{sw}$ . As demonstrated in Millman and Taub (1965), the average output value ( $q_{VI}$ ) is equal to the average input value (multiplied by the steady-state gain) for an entire period. Figure 4(b) presents a specific condition where the duty cycle is equal to 50%.

Combining Equation (13) for  $t = \lambda T_{sw}$  and Equation (14) for  $t = T_{sw}$  one can obtain the maximum flow rate through the inertance tube ( $q_{VImax}$ ), that is,

$$q_{V1}(\lambda T_{sw}) = \frac{1}{1 - e^{-\lambda T_{sw}/\tau}} \left\{ \frac{1}{R} f (1 - e^{-\lambda T_{sw}/\tau}) + \frac{1}{R} g (e^{-\lambda T_{sw}/\tau} - e^{-T_{sw}/\tau}) \right\} \quad (15)$$

and  $q_{V2}(\lambda T_{sw}) = q_{V1}(\lambda T_{sw})$

and the minimum flow rate through the inertance tube ( $q_{VImin}$ ), that is,

$$q_{V2}(T_{sw}) = \frac{1}{1 - e^{-T_{sw}/\tau}} \left\{ \frac{1}{R} g (1 - e^{-(1-\lambda)T_{sw}/\tau}) + \frac{1}{R} f (e^{-(1-\lambda)T_{sw}/\tau} - e^{-T_{sw}/\tau}) \right\} \quad (16)$$

and  $q_{V1}(0) = q_{V2}(T_{sw})$

The amplitude of the flow rate wave can be calculated by subtracting Equation (16) from Equation (15) such that

$$\Delta q_{VI} = \frac{(1 - e^{-(1-\lambda)T_{sw}/\tau} - e^{-\lambda T_{sw}/\tau} + e^{-T_{sw}/\tau})}{(1 - e^{-T_{sw}/\tau})} \frac{1}{R} (f - g) \quad (17)$$

The average load flow rate ( $q_{VL}$ ) corresponds to the integral of Equation (14) through the interval  $\lambda T_{sw} \leq t \leq T_{sw}$  divided by  $T_{sw}$ . The result using Equation (15) is

$$q_{VL} = \frac{1}{(1 - e^{-T_{sw}/\tau}) RT_{sw}} (gy + fx) \quad (18)$$

where

$$x = \tau(1 - e^{-(1-\lambda)T_{sw}/\tau})(1 - e^{-\lambda T_{sw}/\tau}) \quad (19)$$

and

$$y = (T_{sw}(1 - \lambda) + \tau(e^{-(1-\lambda)T_{sw}/\tau} - 1))(1 - e^{-T_{sw}/\tau}) + \tau(1 - e^{-(1-\lambda)T_{sw}/\tau})^2 e^{-\lambda T_{sw}/\tau} \quad (20)$$

The average supply flow rate ( $q_{VS}$ ) is equal to the average flow rate through the inertance tube ( $q_{VI}$ ) and can be obtained by integration of the sum of equations Equation (13) and (14) through the entire interval and dividing by  $T_{sw}$ , resulting in

$$q_{VS} = q_{VI} = \frac{1}{(1 - e^{-T_{sw}/\tau}) RT_{sw}} (g(x + y) + f(x + z)) \quad (21)$$

where

$$z = (\lambda T_{sw} + \tau(e^{-\lambda T_{sw}/\tau} - 1))(1 - e^{-T_{sw}/\tau}) + \tau(1 - e^{-\lambda T_{sw}/\tau})^2 e^{-(1-\lambda)T_{sw}/\tau} \quad (22)$$

The average return flow rate ( $q_{VT}$ ) is obtained by integration of Equation (13) through the interval  $0 \leq t \leq \lambda T_{sw}$  and dividing by  $T_{sw}$ , resulting in

$$q_{VT} = \frac{1}{(1 - e^{-\lambda T_{sw}/\tau}) RT_{sw}} (gx + fz) \quad (23)$$

Substituting  $f = p_S - p_T$  and  $g = p_S - p_L$  in Equation (18), the load pressure can be written as a function of the supply pressure, return pressure, average load flow rate, duty cycle, and switching period as

$$p_L = \frac{p_S(x + y) - p_T x - q_{VL}(1 - e^{-T_{sw}/\tau}) RT_{sw}}{y} \quad (24)$$

The step-up valve efficiency is determined by Equation (11), presented above.

The equations presented above are used straightforwardly in Sections 5 and 6 to calculate the average values of flow rates and load pressure as function of both load flow rate and duty cycle. These theoretical results are validated by comparison with experimental results as shown in Section 5.

#### 4. Experimental Setup

Figure 5 shows the test rig where the hydraulic circuit shown in Figure 7 was implemented. A directional proportional valve (Parker D1FPE50MA9NB01) driven by a PWM signal was used to accomplish the function of the directional valve shown in Figure 1(a). Table 1 presents the valve parameters where the equivalent resistance of the valve corresponds to the average of the linear coefficients obtained from the experimental curves shown in Figure 6.

The inertance tube has an internal diameter ( $d_i$ ) of 7.1 mm and a length ( $l_i$ ) of 1.7 m. The hydraulic fluid



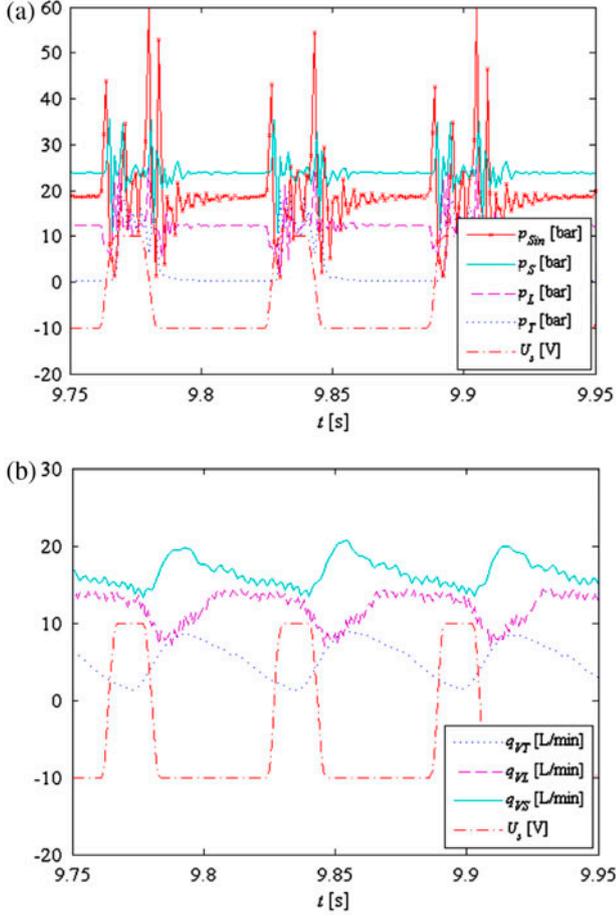


Figure 8. Experimental time response for  $f_{sw} = 16$  Hz,  $q_{vL} = 0.2$  L/s (12 L/min), and  $\lambda = 0.25$ : (a) Pressure curves; (b) Flow curves.

flow rates. As one can see the summing of the tube and valve load losses has a great influence on the system performance and thus the expected pressure booster effect (ideal model) is not reached as the load flow rate increases. In this figure, and also in those which follow, the lines correspond to theoretical results according to the equations presented previously. The points relate to average experimental values.

Figures 10 and 11 present the average supply flow rate and the average return flow rate, respectively, where the experimental points confirm the model prediction. However, one can observe in Figure 9 that the effective operational range of the step-up valve is limited for load pressure values greater than zero. Therefore, only a range of the flow theoretical values is reached under real conditions and these limits are defined by the duty cycle values at  $p_L = 0$  Pa.

The system efficiency calculated according to Equation (11) is shown in Figure 12. The difference between the experimental values and the theoretical prediction is intensified as a consequence of the small errors observed in the load pressure and supply and return flow rates.

Table 2. Average return pressures.

Load flow rate	Average return pressure	
	For $T_{sw} = 62.5ms$	For $T_{sw} = 125ms$
0 L/s	0.33 MPa	0.20 MPa
0.1L/s	0.37 MPa	0.35 MPa
0.2 L/s	0.35 MPa	0.36 MPa
0.3 L/s	0.28 MPa	0.36 MPa

### 5.3. Case 2: Switching Period of 125 ms

Experiments and calculations were carried out under the same conditions described in the previous section but using a switching frequency ( $f_{sw}$ ) of 8 Hz ( $T_{sw} = 125$  ms). As observed in Figure 13, the model gives a good prediction of the valve performance despite the valve load loss being described by a linear coefficient ( $R_e$ ). Equivalent results to figures 11 to 13 were also obtained operating at 8 Hz.

## 6. Performance of step-up Converters

As presented on previous sections, the model including linear resistance gives a very good description of the average response of a hydraulic step-up switching converter. Therefore, it is expected that this model can be used to predict the valve performance under both different operational conditions and with different values for parameters such as inductance, resistance, switching frequency, and supply pressure. So the model can be used to help design a system which has maximum efficiency.

As an example, the influence of the switching frequency is analysed in Figures 14 and 15. In this case the same inductance and supply pressure as used in the previous section are being considered however the return pressure is assumed to be zero. The valve load loss is neglected such that the total resistance corresponds to

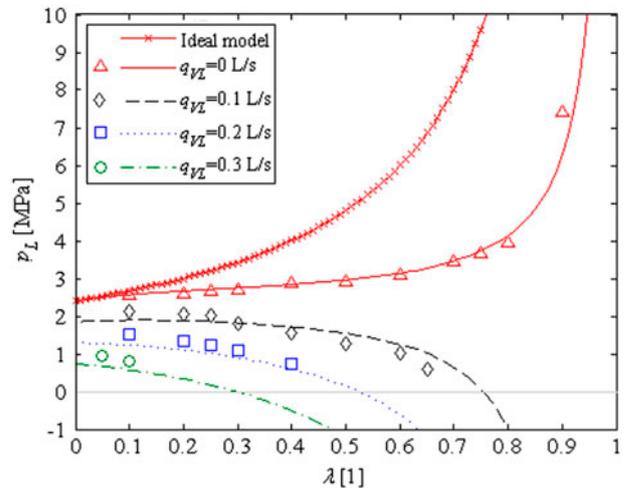


Figure 9. Load pressure versus duty cycle for 16 Hz.

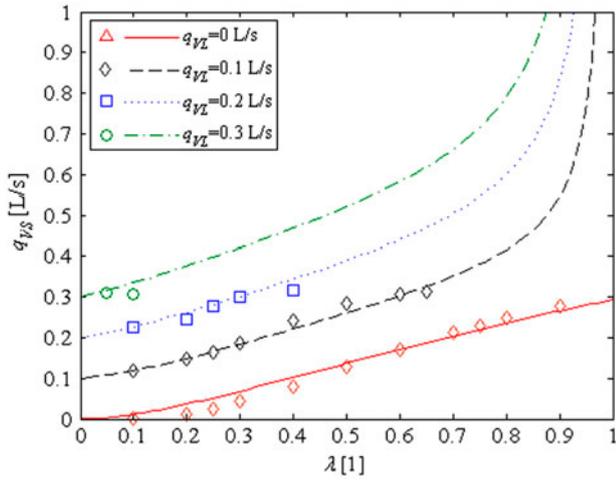


Figure 10. Supply flow rate versus duty cycle for 16 Hz.

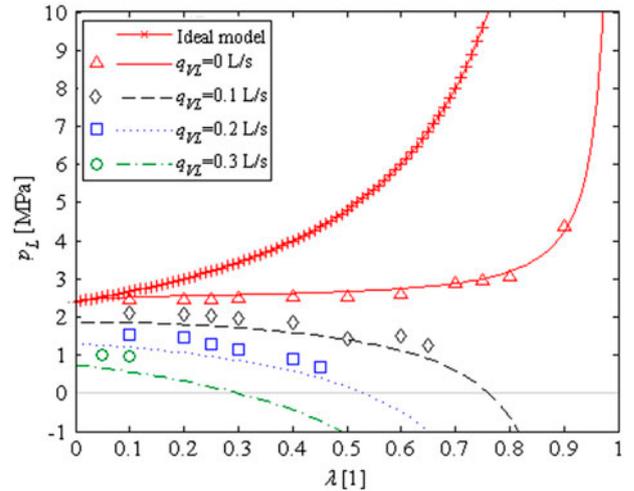


Figure 13. Load pressure versus duty cycle for 8 Hz.

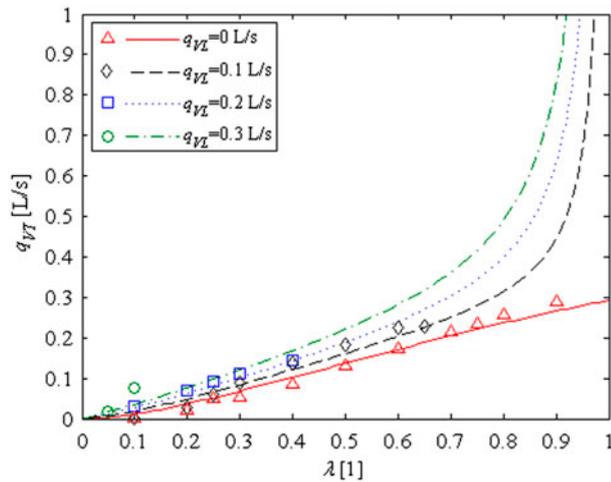


Figure 11. Return flow rate versus duty cycle for 16 Hz.

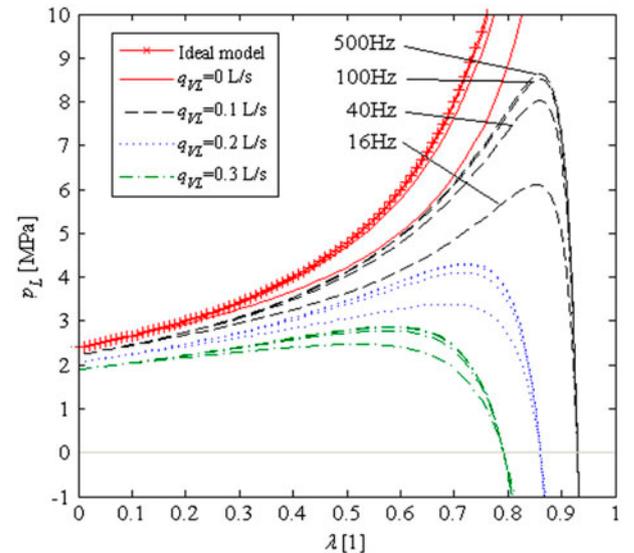


Figure 14. Predicted influence of the switching frequency on the load pressure.

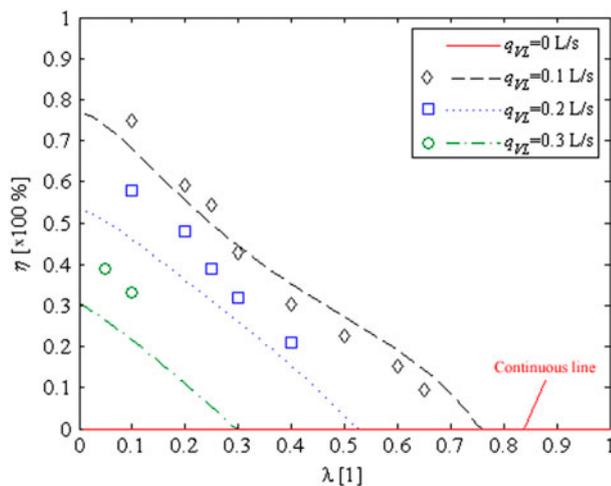


Figure 12. Efficiency versus duty cycle for 16 Hz.

the tube resistance presented before. As can be observed, the efficiency increases as the switching frequency increases but small influence is observed for frequencies higher than 100 Hz. Therefore, for this system a fast switching valve with 1 ms of settling time will be enough to carry out duty cycles between 10% and 90%. New valve designs as presented by Murrenhoff (2003), Winkler *et al.* (2008), Uusitalo *et al.* (2010), and Winkler *et al.* (2010) are potential solutions to reach this requirement.

It is important to observe that for  $f_{sw} = 100$  Hz and  $q_{yZ} = 0.1$  L/s (6 L/min) the efficiency is higher than 75% for duty cycle lower than 0.7, controlling the pressure up to approximately 6 MPa (2.5 times greater than the supply pressure). Based on this example, it can be seen

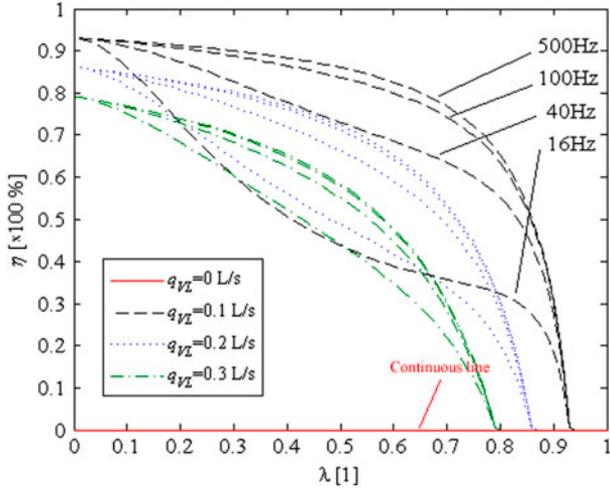


Figure 15. Predicted influence of the switching frequency on the valve efficiency.

that the definition of the maximum flow rate for a specific step-up converter and the consequent tube and switching device sizing is imperative for achieving an efficient device.

The responses at 16 Hz shown in Figures 14 and 15 can be compared with Figures 9 and 12 respectively, where the significant influence of the load loss can be observed closely. Switching valves as reported in Winkler *et al.* (2010) are potential solutions for achieving reduced load loss. Furthermore, the tube load loss can be reduced by increasing the tube length and diameter or working with lower viscosity fluids as, for example, on water hydraulic systems.

## 7. Conclusions

Two detailed models for step-up PWM valves which are of interest for the analysis and design of new systems using the switched inductance principle were presented. Firstly, an ideal valve was studied as a reference for the analysis of real valves. Using the model including linear resistance the theoretical and experimental results presented show that it is possible to predict the average value of the controlled pressure and the flow rates in several parts of the system.

The modelling also gives the minimal and maximum values of flow rate through the inductance tube and thus an effective idea of the valve operation can be achieved including a prediction of reverse flow rates.

Therefore, despite both flow-pressure non-linearity and limited time response of the switching valve as well as the pressure wave propagation in the inductance tube, the presented linear model describes the global behaviour of step-up switching converters very well.

The time responses presented are complex. Several dynamic phenomena occur, such as fluid compressibility

in the internal chambers and wave propagation through the tube. Determining the system performance and the effectiveness of designs based only on this kind of information is difficult. In this context, the model presented in this paper can be used for the preliminary design of switching converters and a time or frequency analysis can be used for system optimization.

According to the presented equations, the step-up converter performance depends on the PWM signal period, resistance, inductance, as well as the average load flow rate. Therefore, using this model it is possible find the ideal parameter combination for maximum efficiency.

The same modelling approach was applied by the authors for step-down converters and validated on a test rig. As in the case of step-up converters, the results are very promising and the model represents a very good approach for the analysis and design of inductance switching valves.

## Nomenclature

$C$	Hydraulic capacitance	[m <sup>3</sup> /Pa]
$C_L$	Load capacitance	[m <sup>3</sup> /Pa]
$f_{sw}$	Switching frequency	[Hz]
$f, g$	Auxiliary variables	[Pa]
$L$	Tube inductance	[kg/m <sup>4</sup> ]
$R$	Hydraulic resistance	[Pa.s/m <sup>3</sup> ]
$p_{Ain}$	Internal working pressure	[Pa]
$p_L$	Load pressure	[Pa]
$p_S$	Supply pressure	[Pa]
$p_{Sin}$	Internal supply pressure	[Pa]
$p_T$	Return pressure	[Pa]
$q_{VI}$	Tube flow rate	[m <sup>3</sup> /s]
$q_{VImax}$	Maximum tube flow rate	[m <sup>3</sup> /s]
$q_{Vimin}$	Minimum tube flow rate	[m <sup>3</sup> /s]
$q_{VL}$	Load flow rate	[m <sup>3</sup> /s]
$q_{Vn}$	Nominal flow rate	[m <sup>3</sup> /s]
$q_{VML}$	Main load flow rate	[m <sup>3</sup> /s]
$q_{VS}$	Supply flow rate	[m <sup>3</sup> /s]
$q_{VT}$	Return flow rate	[m <sup>3</sup> /s]
$R$	Hydraulic resistance	[Pa.s/m <sup>3</sup> ]
$Re$	Equivalent resistance	[Pa.s/m <sup>3</sup> ]
$T_{sw}$	Switching period	[s]
$t$	Time	[s]
$t_s$	Settling time	[s]
$U_c$	Command signal	[1]
$U_s$	Spool position voltage	[V]
$x, y, z$	Auxiliary variables	[s]
$\Delta p$	Pressure difference	[Pa]
$\Delta q_{VI}$	Tube flow amplitude	[m <sup>3</sup> /s]
$\lambda$	Duty cycle	[1]
$\eta$	Efficiency	[1]
$\omega_n$	Natural frequency	[rad/s]
$\tau$	Time constant	[s]

## Funding

This work was supported by the Engineering and Physical Sciences Research Council (EPSRC-UK) [grant number EP/H024190] and the Coordenação de Aperfeiçoamento de Pessoal de Nível Superior (CAPES-Brazil) [grant number BEX 3659/09-7].

## Notes on contributors



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