HYSTERESIS REDUCTION IN HYDRAULIC PROPORTIONAL VALVE CONTROL

TUNING PARAMETERS IN DITHER COMPENSATION METHOD

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Subject Hysteresis reduction in hydraulic proportional valve control: Tuning parameters in dither compensation method

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ABSTRACT

The purpose of this thesis was to examine the phenomenon of hysteresis occurring in hydraulic proportional valve control and to find out a parameter tuning method in dither compensation to achieve an optimal reduction of hysteresis.

The existence of hysteresis causes nonlinearity in proportional valve resulting the controlled system cannot achieve the desired control effect. The main reason that causes hysteresis is static friction or stiction between the spool and the valve body. To eliminate the negative effects of stiction in a valve, the dither compensation method has been found in practice. A dither signal is superimposed on the PWM command signal with a specific amplitude and frequency to always keep the spool valve in micro-movement. Further, it has been discovered that a low frequency PWM can efficiently reduce hysteresis.

To examine the valve dynamic and the dither effect on it, a model of a proportional valve and several dynamic friction models were executed in this thesis project. Based on the relationship between dither parameters, stiction compensation and valve performance, a parameter tuning method was proposed to obtain optimal hysteresis without causing any other problems. A series of simulations and practical experiments were conducted to verify the effectiveness of dithering and the correctness of the parameters tuning method. The simulation and the measurement results showed that optimal hysteresis could be achieved with a proper set of PWM frequency, dither frequency and dither amplitude.

Keywords Hysteresis, hydraulic drives, hydraulic proportional valves, dither compensation.

Pages 55 pages including appendices 15 pages

CONTENTS

1 INTRODUCTION

In hydraulic and pneumatic drives, nonlinear valve problems are the main factor restricting the control loop performance. From a systems theory point of view, the less nonlinearity, the simpler a controller is good enough (Karvonen 2017). One of the main reasons for the nonlinearity of the control valve is the friction between the valve spool and body. The static friction inside the valve causes hysteresis and a dead zone which negatively impact on the accuracy and stability of the valve control system output. (Wang 2016.) Hysteresis is the lag on the valve performance between the generation and removal of the input commands. It makes the precise position and velocity control of the proportional system difficult to obtain.

Metso Paper, or currently Valmet, is utilizing digital hydraulics instead of proportional valves and one major reason why they have chosen this technology is zero-hysteresis (Valmet 2016). Hydraulic control valves are a fundamental part of pulp and paper machinery. They allow moving large heavy equipment accurately and safely. For example, proportional valves have been widely used to adjust the fluid flow into cylinders inside massive steel rollers which generate individual pressure for consistent paper web thickness (Rexroth 2003). Handling with paper thickness (commonly 0.06 – 0.3 mm for printing paper) means the required pressure and velocity control is extremely precise. And hysteresis is one of the major problems that inhibit valve control. It is, therefore, valuable to find a method to eliminate or to reduce hysteresis inside the proportional control valves.

In hydraulic proportional control, hysteresis mainly caused by stiction (or static friction), is minimized by using a high-frequency dither signal, superimposed on the PWM command signal. The combination signal allows the valve spool in a high-frequency micro-movement to eliminate the static friction between the spool and the valve body. Besides, it has been found that not only the dither parameters but also PWM parameters directly impact on the effectiveness of the method. How to find good parameters is still a big concern.

In this thesis, a literature survey was conducted to examine the phenomenon of hysteresis and the dither compensation method to reduce hysteresis occurring in hydraulic proportional valves. The literature survey consists of seven chapters, from Chapter 2 to Chapter 6. There is a review of the fundamentals and state-of-the-art of proportional valve control in hydraulic applications. Definitions of related terms and concepts, such as friction, stiction, hysteresis, PWM and superimposed dither signal, and so on, are also explained here. In this survey, the dither compensation method is also discussed, and a parameter tuning method is subsequently proposed with a guidance flowchart. Chapter 7 presents the mathematic

model of a proportional valve and the simulation with different settings. To verify the parameter tuning method, several practical measurements were taken. Chapter 8 shows the results of the measurements. A conclusion and recommendations from the author are given in Chapter 9.

2 HYDRAULIC PROPORTIONAL VALVE

2.1 Hydraulic control valves

Hydraulic control valves are used within hydraulic control systems to direct and regulate the fluid flow from the hydraulic power supply to the load output devices. In control circuits, control valves receive the command signal from the human operator or from the electrical control unit (ECU) to drive the actuator units to the required performance.

Understanding of control valves is imperative for design and implement an efficient, cost effective and energy saving hydraulic systems. Hydraulic control valves can be classified in-to different ways, such as construction type, operation type, function type and so on (Manring 2005, 169).

The construction classification is based on the number of flow lines. For example, a two-way valve has two flow lines given by a single input and a single output. A three-way valve often consists one supply input line, one output line, and one return line which leads the fluid back to a reservoir.

Within operation type, control valves are divide into manual actuation, mechanical actuation, electromagnetic actuation, hydraulic piloted actuation, and some combination of those types. Manual control valves are operated by manual actuators, for example push button, pedals and hand lever. Mechanically actuated valves include also auxiliary components as spring, plunger, roller, or detent. Electromagnetic valves are functioned by a solenoid which generates an electromagnetic force on the valve spool. Hydraulic piloted valves are piloted by another valve by means of hydraulic pressure in case its required force is too large for a solenoid or manual.

In function classification, there are essentially three main types of valve: directional valves, proportional control valves, and servo valves. Directional control valves are also called finite position valves which consist spools displacing in a finite position. They simply allow fluid passing through the valve or change the direction of the fluid from a supply source to an actuator. However, they cannot be used to continuous control the flow or pressure of the fluid. To solve the above issue, proportional valves have been developed. Proportional valves contain variable positioning spools which commonly actuated by proportional solenoid. Proportional valves provide infinitely adjustable set of control edges. The servo valve is an alternative solution for adjustable flow/speed passing through valve. A

servo valve is actuated by toque motor and is equipped with a feedback unit which can provide precise control of spool position. However, servo valves are much more expensive, have zero or negative overlapped spools, are sensitive to thermal chocks and have high requirement for oil cleanliness. Therefore, proportional valves are widely used in most hydraulic systems which do not require absolute accuracy such as mobile hydraulics, metal handling machines, process automation system and so on.

Another type of control valves is the so-called digital valve which is considered as the state of the art control valve and it “will change the concept of hydraulics” (Linjama 2017). The principle of digital hydraulic valves is to use a serial of parallel connected on/off-valves together with intelligent control algorithms to produce a digital counterpart to the analogue proportional or servo valve (Karvonen 2016). The digital valves are programmable which can behave different performance with the same hardware by using different control codes. In addition, independent metering in/out, zero hysteresis, and ideal piecewise linearity make them comparable to other type of hydraulic control valves. However, the size of the whole digital valve system is not a competitive ad-vantage when compared to proportional or servo valves (2-3 times bigger and heavier) (Linjama et al. 2003). The total cost effectiveness is also difficult to affirmative which depends strongly in application. Besides, a currently limited application range due to the lack of suitable valves is considered as a main drawback of digital hydraulic valves.

Figures 1, 2, and 3 show cross-sectional drawings of typical directional valves, proportional valves, and servo valves.

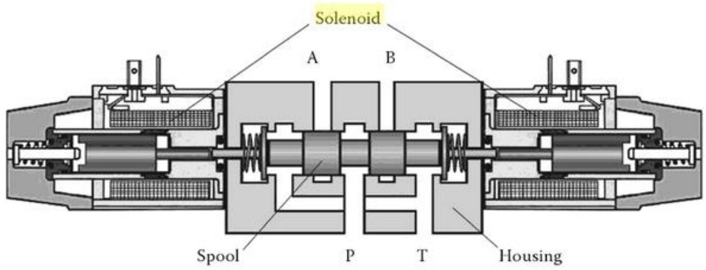

Figure 1. 4/3 Directional control valve, directly controlled by two solenoids (Linsingen 2008, as cited in Totten 2011, 37)

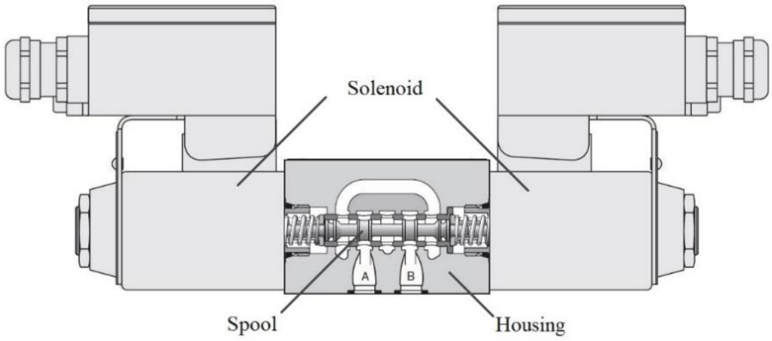

Figure 2. Cross-sectional view of a proportional valve structure (Parker Hannifin Corp 2016, 8)

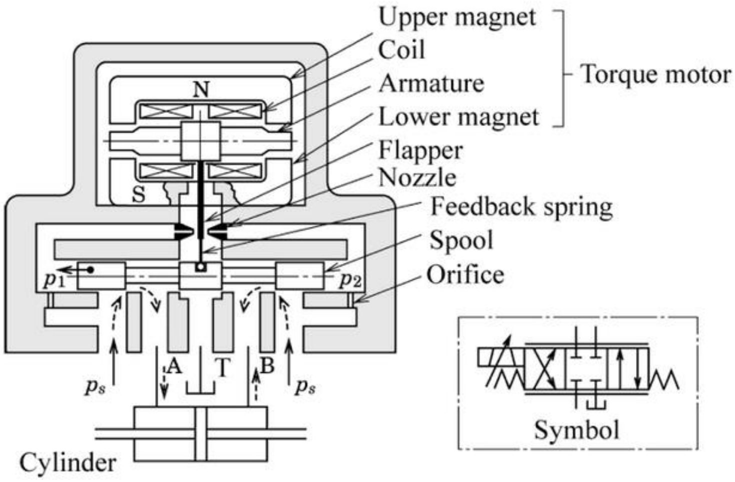

Figure 3. Flapper nozzle type servo valves (Konami & Nishiumi 2016, 159)

2.2 Proportional control valves

Proportional valves are suitable for applications where the precision of a servo valve is not needed, but operation at variable velocity is needed. Directional on/off valves provide only one speed for actuators which is related to the load if pressure compensator is not used. Proportional valves, however, generate a proportional control of the output pressure and flow by varying the electrical input signal. Figure 4 shows the symbol of a direct-acting proportional valve actuated by solenoid.

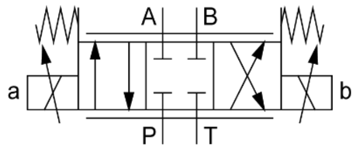

Figure 4. Direct-acting proportional valve symbol 3-position actuated by solenoid

A proportional control valve is often actuated by proportional solenoids. The electrical current passes through the coil of the solenoids and generates a magnetic field. The magnetic field develops a force which acts on the armature of the solenoid. This force is used to shift the armature which is connected to the valve spool. By varying the input current feeding to the coil, the spool valve position can be adjusted anywhere between 0% to 100% full stroke. To give a predicable response the solenoid must produce a force which is dependent solely on the current and not on the spool position. In other words, the force for a given current must be constant over the full stroke range (Parr 2011, p.97.). Figure 5 demonstrates the relationship between coil current and valve stroke.

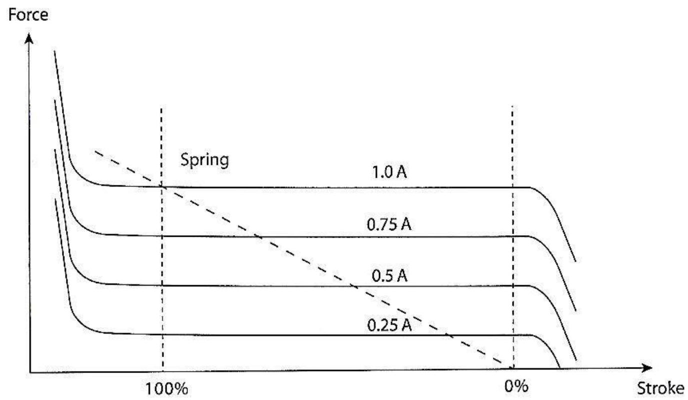

Figure 5. The relationship between coil current force and stroke of a proportional valve (Parr 2011, 98)

According to Festo Textbook TP 701 (2002), signal flow (as shown in Figure 6) in typical proportional hydraulic consists of the following states:

- A controller generates an electrical voltage acting upon an electrical amplifier.
- The amplifier converts the voltage (input signal) into a current (output signal).
- The current is applied to the proportional solenoid.
- The proportional solenoid actuates the valve.
- The valve controls the power flow to the hydraulic drive.
- The drive converts the energy into kinetic energy.
- The electrical voltage can be infinitely adjusted, and the speed and force can be infinitely adjusted on the drive accordingly.

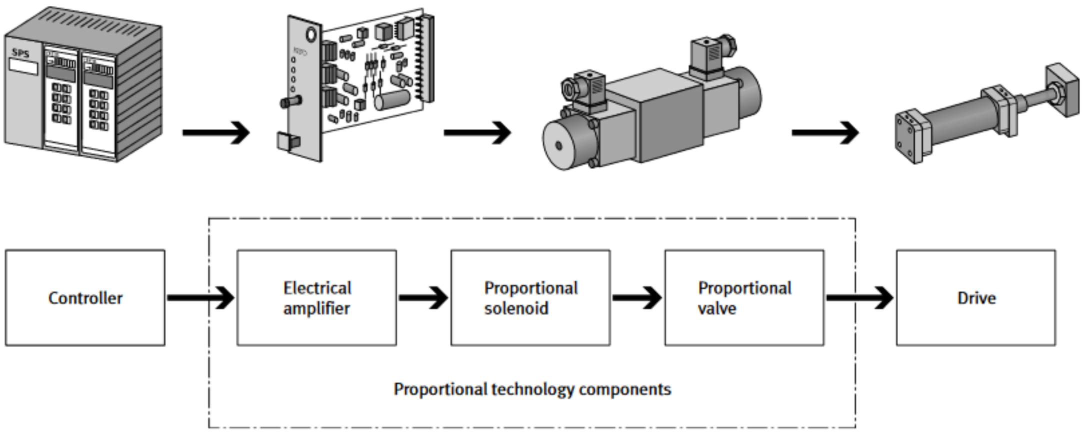

Figure 6. Signal flow in proportional hydraulics (Festo 2002)

3 FRICTION AND STICTION WITH PROPORTIONAL VALVES

3.1 Friction and stiction

Friction is the resistance to motion of an object contacting or moving relative to another. It can occur everywhere, between all types of matter: solids, liquids, and gases. Friction plays an important part in nature and has many practical uses such as breaking system, anti-slipping, or simply keeping a nail on wall. It also negatively influences on systems such as causing wear on mechanical parts or wasting energy by creating heat.

Friction consist of two type of terms that describe the state of objects: static friction and dynamics friction or known as kinetic friction or sometimes as sliding friction. Static friction or shorted as stiction is the friction force between objects that prevent one from sliding on the other's surface and it occurs when the objects are stationary but forced to start moving. In other words, it describes the friction force at rest (Olsson 1996). When the object is sliding or rolling with respect to the other, the friction turns into the term of dynamic or kinetic friction. Dynamic friction describes the characteristic of friction force at moving condition. Stiction is generally greater than dynamic friction.

In hydraulic systems, stiction is one of the most commonly found valve problem. It causes non-linearities and limits control loop performance. As cited in Modelling valve stiction (Choudhury et al. 2004), Ruel (2000) reported ''Stiction can keep the stem from moving for small control input changes, and then the stem moves when there is enough force to free it. The result of stiction is that the force required to get the stem to move is more than is required to go to the desired stem position." It happens very same in proportional valves. The electro-magnetic force which is applied to move the valve spool, is increased until it exceeds the stiction level. After the valve stem starts to move, the friction transcends from static to dynamic friction level which is a lower value. The applied force then is decreased until the force is insufficient to sustain the motion to the desired position. If the force is not quickly decreased, the valve will overshoot its target position, and therefore, will cause the process to overshoot its setpoint. After this, the valve spool stops and sticks in the new position.

This issue was also described by Horch (2000) that "The control valve is stuck in a certain position due to high static friction. The (integrating) controller then increases the set point to the valve until the static friction can be overcome. Then the valve breaks off and over to a new position (slip phase) where it sticks again." These phenomena produce nonlinearities and causes sticky behaviour of the valve. The valve control is then much more complicated than in an ideally linear system. To investigate the valve dynamic and the effects of stiction on it, it is essential to represent friction characteristics by suitable friction models.

3.2 Friction models

When considering a suitable friction model to employ in the valve dynamic study, an overview of some existing models is essential. In this chapter, the Coulomb model, the viscous friction model, the Stribek model and the LuGre model are discussed.

The earliest physical model of friction is the Coulomb friction model based on the work of Davinci (1519), Amontons (1699) and Coulomb (1785). It is often simplified as

$$F_f = F_c \, sign(v) \tag{1}$$

where F_f is the friction force, $v = \dot{x}$ is the velocity of motion and F_c is the Coulomb sliding friction $F_c = \mu N$. As can be seen in Figure 7, the model describes a constant friction force independent on velocity of motion.

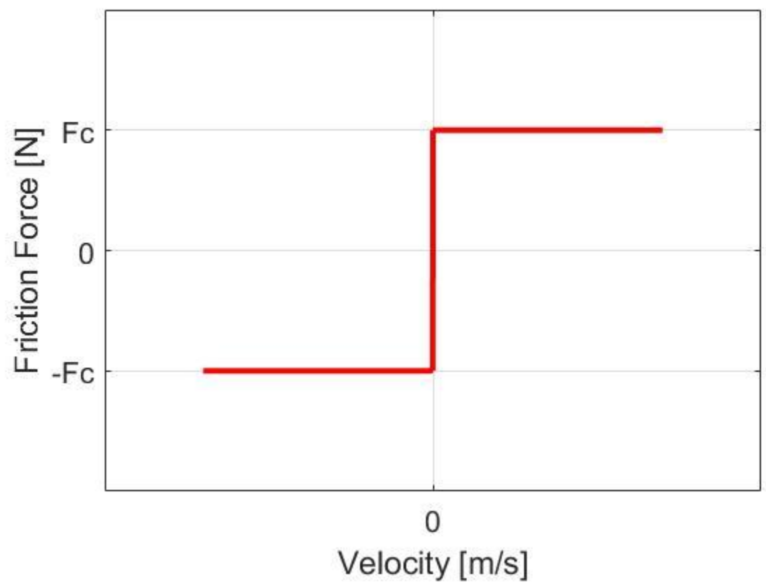

Figure 7. The Coulomb friction model illustrates a constant friction force which is independent on velocity

Another model known as viscous friction model was developed by Reynold (1866). The model can be formulated as

$$F_f = k_v \, v \tag{2}$$

where F_f is the friction force, $v = \dot{x}$ is the velocity of motion and k_v is the viscous coefficient. The viscous friction model is illustrated in Figure 8 where the friction force is proportional to velocity, and it is zero when velocity goes close to zero.

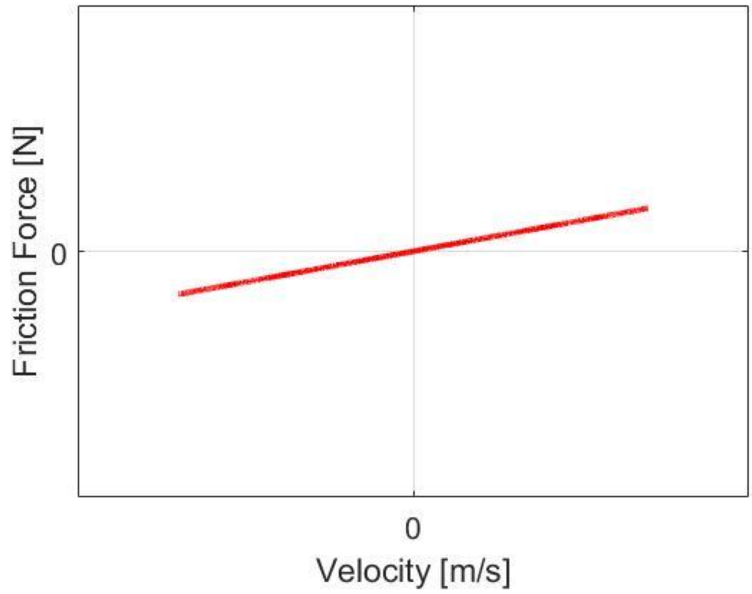

Figure 8. Viscous friction model demonstrates the proportion between viscous friction and velocity

A combination of the Coulomb model and the viscous model is also a well-known model shown in Figure 9. It is represented by

$$F_f = F_c \, sign(v) + k_v \, v \tag{3}$$

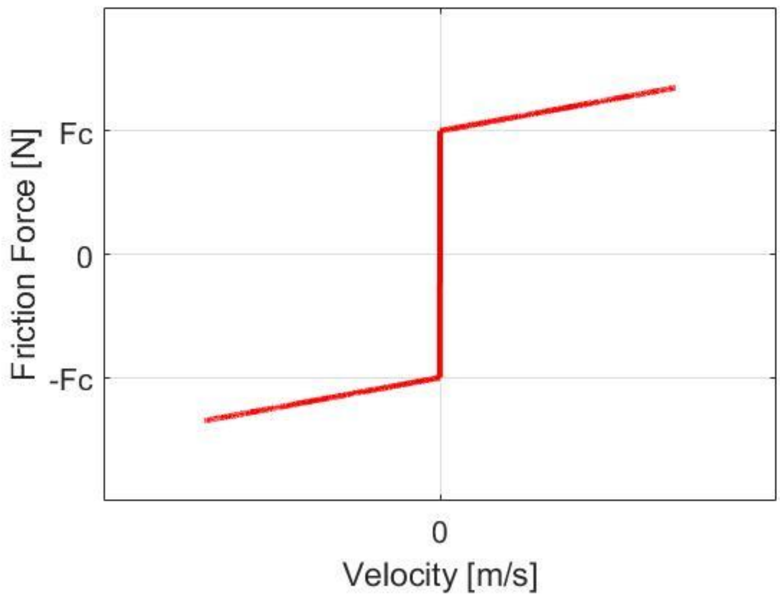

Figure 9. Viscous + Coulomb friction model consists of an independent Coulomb friction and a proportional viscous friction on velocity

Major limitations to the above three models have included a failure to describe static friction and the transition from it to kinetic (Coulomb) friction. In 1902 Stribeck observed and proposed the Stribeck effect as a transition from stiction to kinetic friction. The Stribeck friction model then was developed and evaluated by Armstrong (1993). The model can be formulated as

$$F_f = (F_c + (F_s - F_c)\, e^{-\left(\frac{|v|}{v_s}\right)^i}\, sign(v) + k_v\, v \tag{4}$$

where F_f is the friction force, $v = \dot{x}$ is the velocity of motion, k_v is the viscous coefficient, F_c is the Coulomb sliding friction force, F_s is the maximum static friction force, v_s is the Stribeck velocity, and i is the Stribeck shape factor (i = 2 is usually used in literature).

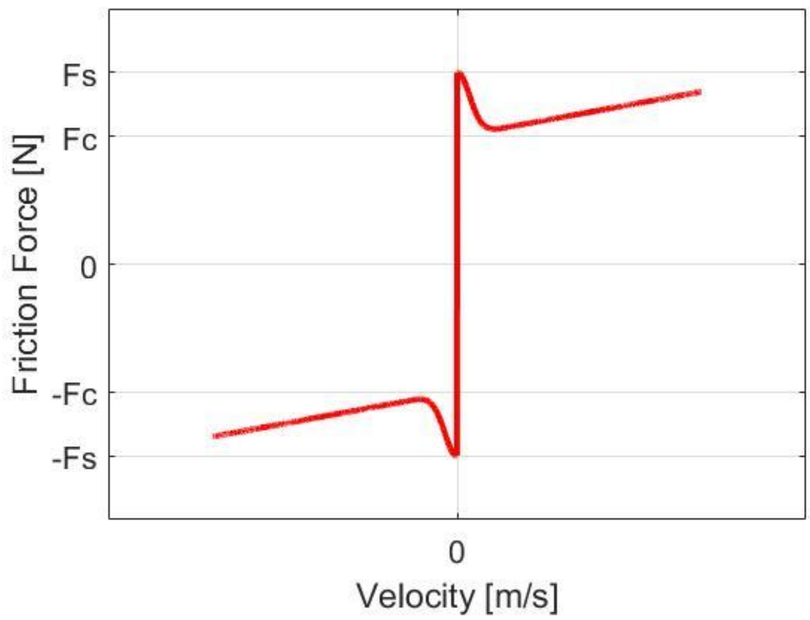

Figure 10. Stribeck friction model describes a stiction force and a transition curve between stiction and kinetic friction

The Stribeck model as illustrated in Figure 10, however, remains the sign function that poorly describes the friction at micro-displacement where the velocity goes close to zero, so called pre-sliding behaviour. The LuGre model was developed by Canudas de Wit, Olsson, Astrom & Lischinsky (1995). It captures both the Stribeck effect and pre-sliding behaviour and is described by

$$F_f = \sigma_0 z + \sigma_1 \dot{z} + k_v v \tag{5}$$

$$\dot{z} = v - \frac{|v|}{g(v)} \tag{6}$$

$$g(v) = \frac{1}{\sigma_0}(F_c + (F_s - F_c)e^{-\left(\frac{|v|}{v_s}\right)^i} \tag{7}$$

where F_f is the friction force, σ_0 and σ_1 are dynamic parameters, which are respectively the frictional stiffness and frictional damping, z is the average deflection of the contacting asperities, $v = \dot{x}$ is the velocity of motion, k_v is the viscous coefficient, F_c is the Coulomb sliding friction force, F_s is the static friction force, v_s is the Stribeck velocity, and i is the Stribeck shape factor.

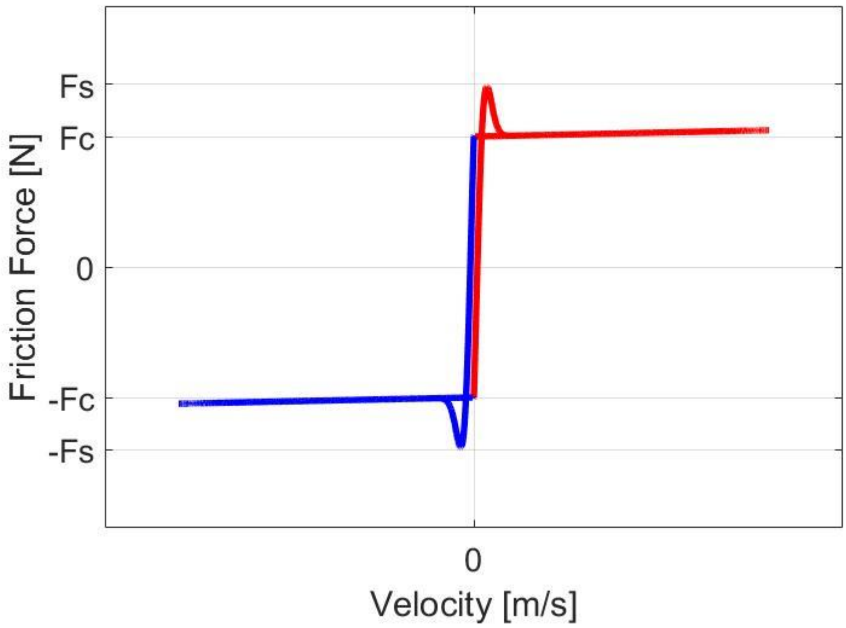

Figure 11. LuGre friction model, friction force vs velocity

Due to the ability of representing the stiction, transition from stiction to dynamic friction and the friction behaviour at low velocity, the LuGre was selected to simulate the valve dynamic under friction.

4 HYSTERESIS WITH PROPORTIONAL VALVES

Hysteresis demonstrates the lag between the generation and the removal of some physical phenomena. It exists in many domains such as electricity and magnetic domain. In hydraulic proportional valves, hysteresis has been considered as a major problem which negatively effects on the precise of the system.

Parker's PV section catalogue HY15-3502/US (2013) defined "Due to various factors, the performance of a proportional valve will show a slightly different performance when the current signal is increasing than it will when the signal is being decreased. This difference is usually expressed as a percentage of total input change and is referred to as the hysteresis of the valve". The different performance can be described as the difference between the output flow rates or the output pressure values or the valve spool displacement positions, depending on the desired output signal.

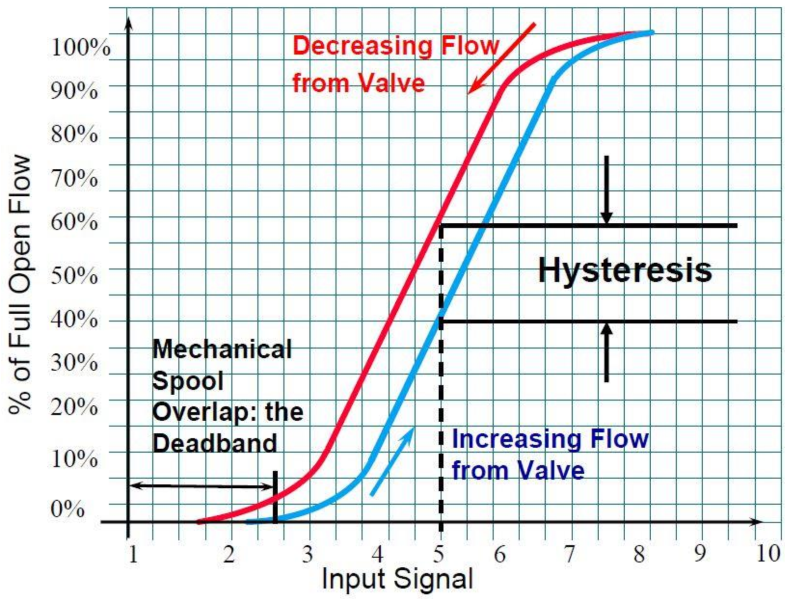

Figure 12. Hysteresis curves (Besch n.d.)

A flow hysteresis map of proportional solenoid valves is represented in Figure 12. The blue curve represents the flow rate (percentage of full open flow) referred as the ascending of the input current signal. The red curve demonstrates the flow rate according to the descending of the input current signal. The largest gap between two curves is the hysteresis and is calculated in percentage of the difference of the valve input values on total input change or in percentage of the difference of the valve output values on total output change.

The existence of hysteresis in proportional valve reduces open-loop accuracy and repeatability by causing nonlinear. The idea of control technologies has been shown that the more linear the more efficiency of

control system. Therefore, it is highly worthy to eliminate or to reduce the hysteresis in hydraulic control, especially in precise hydraulic such as paper technology, robotics.

To answer the question of how to eliminate or to reduce hysteresis, the researcher had raised question about what causes the phenomenon in proportional valve. Approved from many studies hysteresis is mainly caused by static friction or stiction inside the valve. Stiction keeps the valve spool from moving. Therefore, the spool requires an amount of force to change from sticking stage to sliding state. It means a value of energy is lost. That causes the delay in valve response when increasing the command input. The delay happens in both directions due to the spool stopping at desired position.

Recent studies have shown that there has been an amount of research on finding a way at which hysteresis negative impact on the stability and control quality of the system can be eliminated or reduced. For example, many proportional valves use an LVDT (Linear variable differential transformer) spool position sensor with electronic feedback, and the result is a tighter spool position control loop and less hysteresis (less than 1%). Digital hydraulic valve system (DHVS) also had been developed for an alternative technology which has no hysteresis. Digital valve system consists of many on/off valves in one package valve. The amount of opened or closed valves defines the flow passing through the package valve. However, these methods are not cost-efficient. Besides, there are several control methodologies which has been suggested to cancel out the hysteresis effect such as feedforward compensation, impulsive control. In this research, superimposed Dither on PWM command signal is considered.

5 PROPORTIONAL VALVE CONTROL

5.1 Pulse width modulation

Pulse Width Modulation (PWM) is a control technic that creates digital pulsing signals which imitate the continuous variable characteristics of analogue signal. This technic is more effective and produces less heat than using an analogue operational amplifier. PWM has been widely used in proportional valve control and has become the standard for electro-hydraulic valve amplifiers.

PWM uses digital “on/off” pulses to create variable output voltages when applied to a motor or a solenoid. The pulses have a constant amplitude, but the width is varied which is proportional to the desired output amplitude. In a full cycle, the pulses contain an “on” pulse and an “off” pulse. The width of the “on” pulse relative to each “off” pulse is called the PWM duty cycle (D) and refers to the percentage of the cycle. The PWM frequency is to determines how fast a cycle is finished**.** Figure 13 shows waveforms of PWM signals with constant duty cycle of 25%, 50% and 75% with a same frequency.

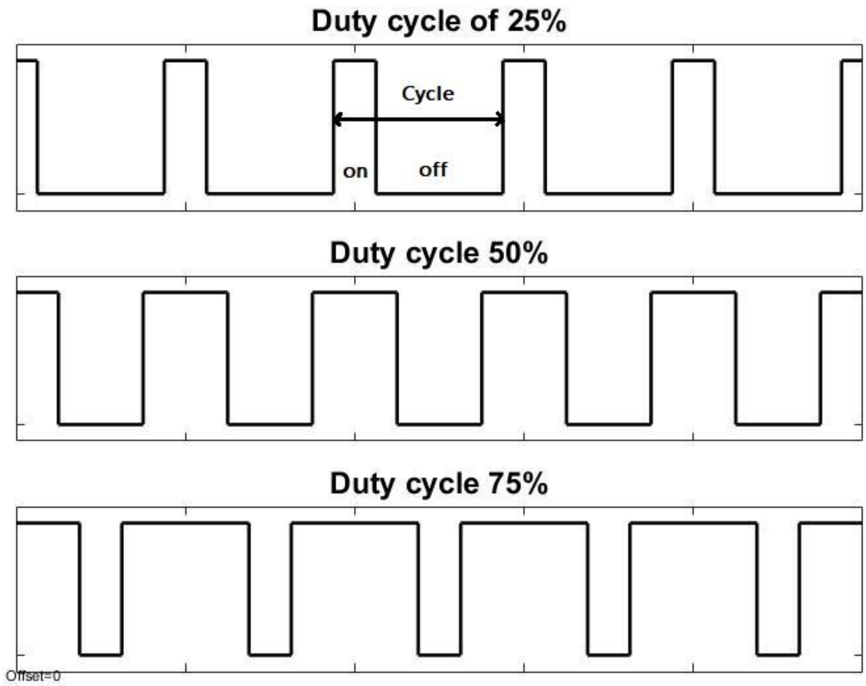

Figure 13. PWM with duty cycle of 25%, 50% and 75%

The average of output voltage seen by the load is proportional to the pulse width or the duty cycle. When varying the duty cycle of the PWM, variable output voltages are obtained. The average voltage can be formulated as

$$V_{avg} = DV_{High} + (1 - D)V_{Low} \tag{8}$$

where the V_{avg} is the average voltage, V_{High} is the high voltage when pulse is "on" or can be considered as the maximum value of the source voltage, V_{Low} is the low voltage when pulse is "off", D is the duty cycle. The low voltage is often set to zero, hence (8) turns into

$$V_{avg} = DV_{Source} \tag{9}$$

For instance, a source voltage 12V is used with PWM at "D" of 50%, the produced voltage has the average value of 6V and at "D" of 25%, the average voltage is 3V.

To generate PWM signals, there have been a number of ways. Using a comparator to compare a command signal with a triangle carrier waveform is commonly used to produces PWM signals. The higher the amplitude of the command signal is, the wider the PWM pulse is. The command signal level can range between the minimum and maximum level of the triangle wave. Figure 14 shows a variable pulse width PWM which is created according to its command sinusoidal signal compared with a triangle waveform.

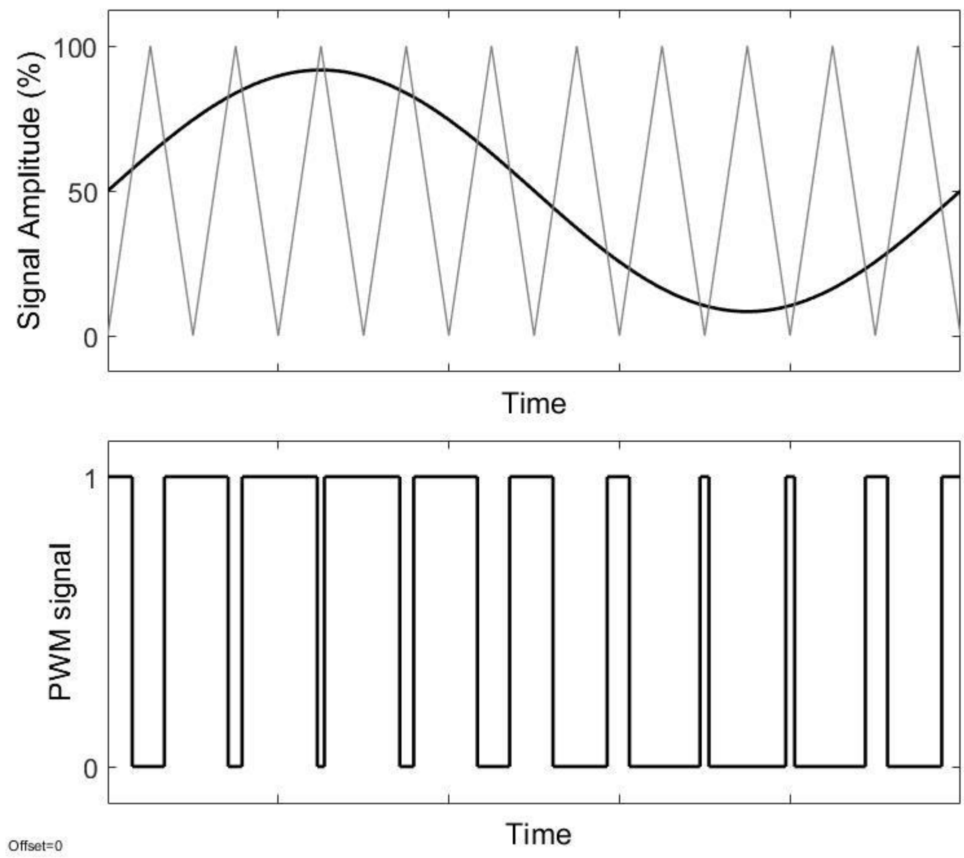

Figure 14. A variable pulse width PWM signal is obtained when comparing a sawtooth waveform with a sinusoidal signal.

When applying a PWM voltage across a solenoid coil, the current produced does not follow the PWM rectangular waveform, but gradually rises and falls due to the effects of inductance (HydraForce 2017). The produced current is proportional to the average voltage of the PWM which is generated according to the command signal. As can be seen in Figure 15, the waveform of the produced current is quite similar to the command signal except the existence of ripples.

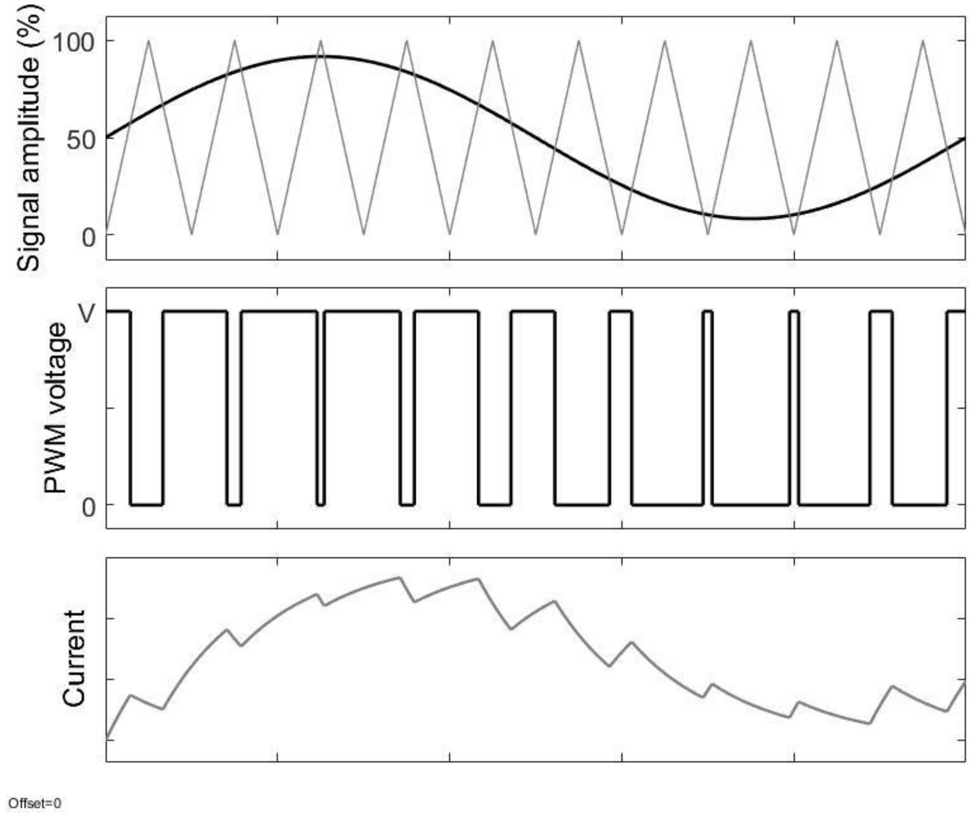

Figure 15. Produced continuous current with ripples of a low frequency PWM voltage according to a sinusoidal command signal.

The ripples are caused due to the rising and falling of current when the pulses turn state between "on" and "off". As can be seen in Figure 18, the amount of ripple is different among current levels because of the varying of duty cycle. In most case, the amount of dither is not adjusted by affecting the PWM duty cycle because the duty cycle is altered for the need of varying average value of voltage. However, the PWM frequency also impacts on the amplitude of ripple. With a high frequency, the current has less time to decay before the next rise begins. (HydraForce 2017.) Hence, to achieve a smooth continuous current, the PWM frequency is increased till an efficiency acceptance. The higher the frequency of the PWM is, the less the ripple is and consequently the smoother the current is. However, too high frequency of the PWM will produce more heat in semiconductors, therefore will cause overheat of controller.

5.2 Dither

Dither is an artificial noise which is intentionally added to a signal to improve the performance of an end overall system (Shome, Prakash, Mukherjee, & Datta 2013). Dithering technique has been employed in many applications such as digital and audio signal processing or video and image handling in computer graphics**.** In proportional valve control, dither has been widely integrated in many amplifiers to enhance the performance of valves.

Dither creates a small vibration of the solenoid current. In chapter 3, stiction is pointed out as a main reason causes hysteresis. It prevents the valve spool from moving with small input current. To eliminate or reduce the effect of hysteresis, it is necessary to counteract the occur of stiction in valve. The small vibration made by dither keeps the spool moving even when the input command is steady.

Low frequency PWM, typically less than 400 Hz, can generate dither or current ripple as a by-product of the PWM process. The amount of dither is not constant and is changed as the average coil current changes (see Figure 16). This can result in too much dither at some current levels and not enough at others. (Beater 2008, 88). Further, this kind of dither is reliant on PWM duty cycle and PWM frequency, and its amplitude and frequency cannot be set independently. It is not ideal for many valve designs that require a specific setting of dither. For high PWM frequency, typically above 5000 Hz, the coil current does not have ripple. No by-product dither is produced (Beater 2008, 88). Therefore, high frequency PWM is often used to eliminate undesirable internal dither, thereafter could carry an adjustable external one.

Figure 16 illustrates the relation between PWM frequency and by-product dither. By increase the PWM frequency, the amount of dither is accordingly decreased. At a high frequency, the ripple effect on the coil current virtually disappears.

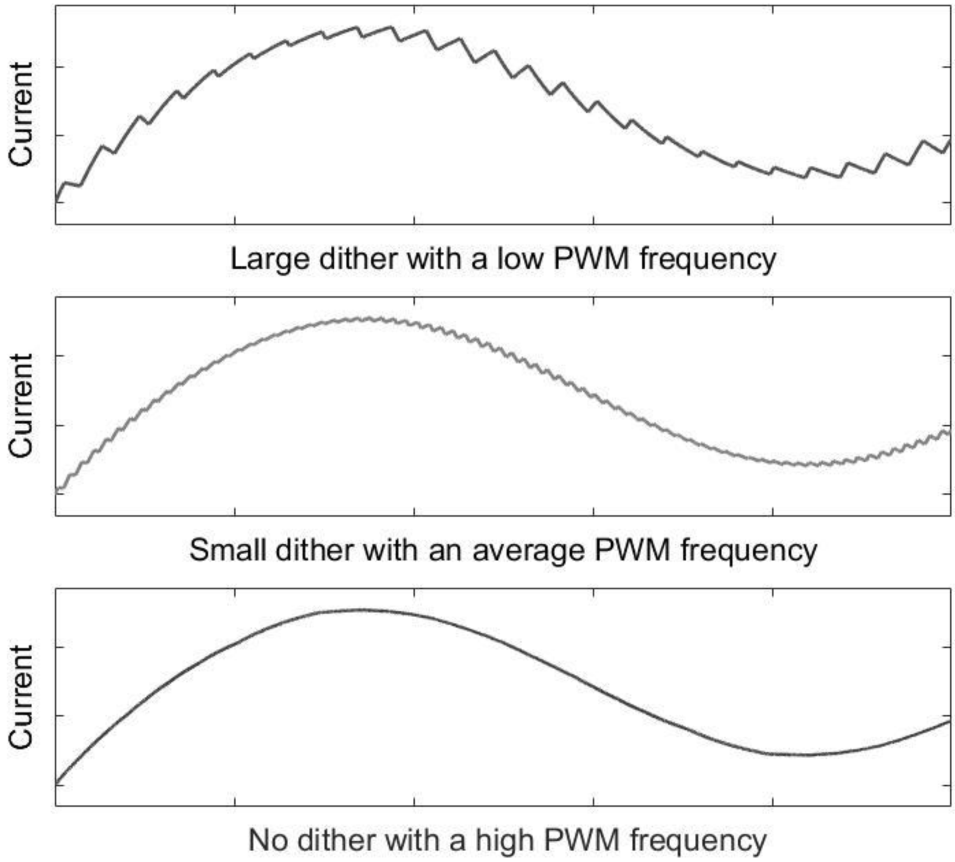

Figure 16. Dither made by PWM frequency

In dither compensation method, an additional high-frequency signal with zero mean (Wang 2016) is superimposed on the control signal of the valve. The additional signal is commonly a sinusoidal waveform. The control signal is a high frequency PWM voltage. In this method, the external dither is capable to be adjusted independently to help the valve spool overcome the maximum static friction even when the derivative of the control signal is zero**.** The valve spool is allowed in a high-frequency micro-movement state around the desired position and continuously experiences dynamic friction. The nonlinear is weakened consequently.

When adding an external dither on the command voltage, accordingly the dither appears on the coil current. Figure 17 illustrates a ramp input current feeding to a valve coil with a high frequency PWM, a low frequency PWM, and a high frequency PWM with external dither. As can been seen from the figure, the amount of external dither is independent on current levels which is different from the amount of dither made by a low frequency PWM. The amplitude of dither made by a low frequency PWM is dependent on the alteration of current and is not constant at different current levels.

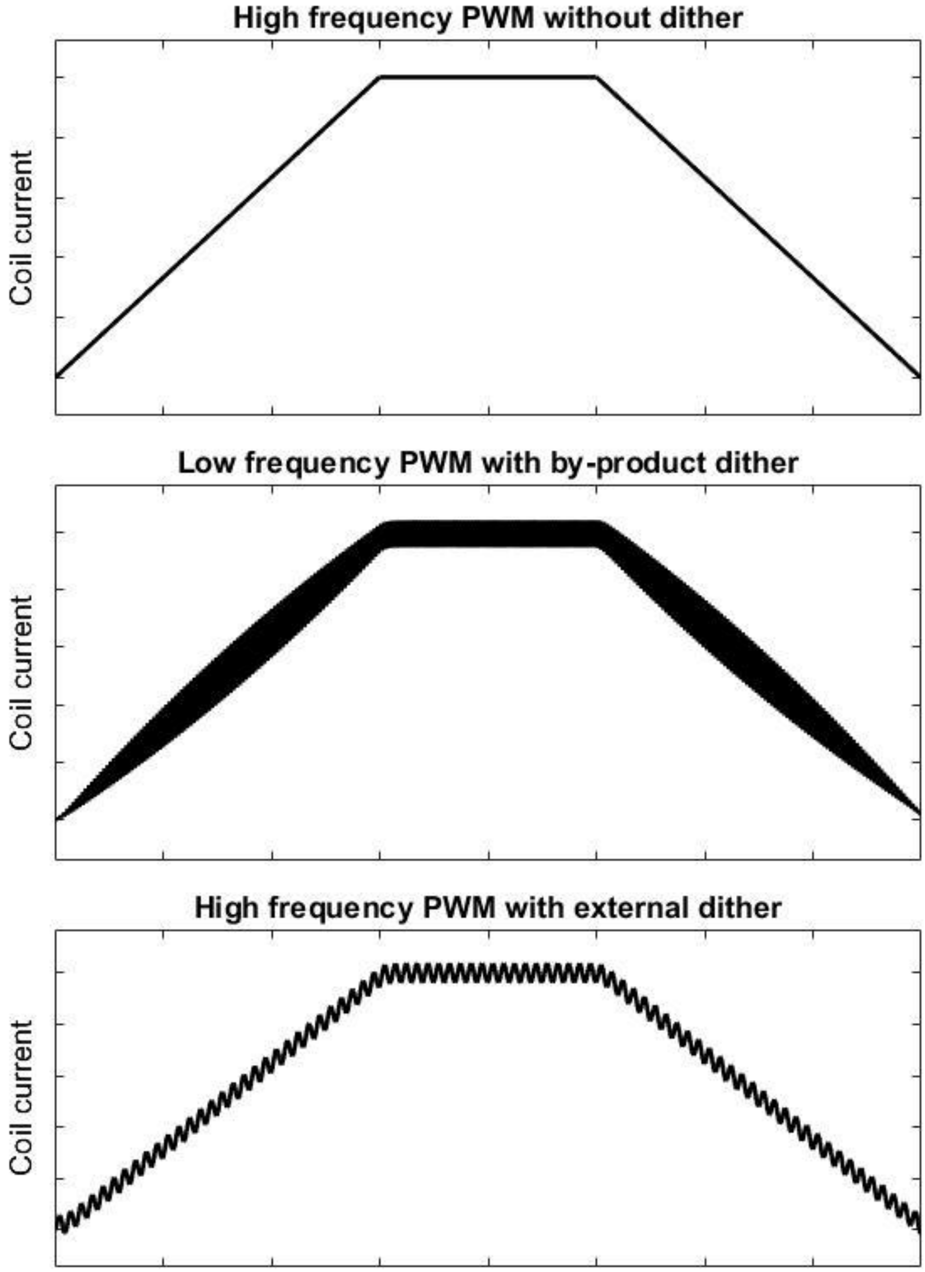

Figure 17. A ramp feeding current signal with a high frequency PWM, a low frequency PWM and a high frequency PWM with external dither.

6 TUNING METHOD OF PARAMETERS – OPTIMAL DITHER

6.1 Tuning principle of parameters

The dither signal allows the valve spool in a high-frequency micro-movement to eliminate the static friction. The amplitude of the dither signal is related to the degree of friction force inside the valve (Wang 2016, 609). The dither amplitude needs to be large enough to help the valve spool overcome the maximum static friction when the control signal derivative is zero. A valve driven with too small amplitude dither cannot afford to move the spool, that is, micro movement does not occur, which leads to a poor hysteresis performance. However, if the amplitude is too large the valve spool will oscillate strongly which will cause the valve output to vibrate.

The dither frequency must be fast enough to avoid valve output vibration. Typical values are between 70 and 350 Hz for the dither frequency (Beater 2007, 88). If the frequency is too low, then the valve may actually be able to follow the signal and it may generate vibrations of the system pressure or flow (PWM controls Inc. FAQ n.d.). However, with too high frequency, the spool responses poorly and even are incapable to response to the dither (because every system is low pass filter)**.** Therefore, the dither has little effect on the valve, although, at high frequencies, there is a slight increase in hysteresis (Price 2011). In other words, when increasing the dither frequency, the dither effects is decreased.

Additional, according to Wang (2016): “The frequency of the dither signal depends on the natural frequency and the open loop cut-off frequency of the valve. The frequency of the dither signal should be away from the natural frequency and less than the open-loop cut-off frequency”. If the dither frequency is equal to the natural frequency of the valve spool, the amplitude of oscillation increases because of the resonance and the system will vibrate aggressively.

HydraForce (2011) suggests the dither frequency range setting for their valve groups: “70 -250 Hz dither frequency on all SP, ZL, and PV valves (flow and directional control valves), and 200-300 Hz dither frequency on TS Valves (proportional pressure control valves)” (David 2011). They also have proportional valve controller which has an adjustable dither setting in the range of 0-10% amplitude and 70-350Hz frequency for different valves.

Continental Hydraulics (2012) published the dither reference guide for their valve family: 60 Hz dither for single frequency power signal (low PWM frequency) and 120 Hz dither for dual frequency (high PWM frequency superimposed with low dither signal).

6.2 Tuning method of parameters

The flow chart below describes the tuning method to obtain the optimal parameters for the dither compensation method.

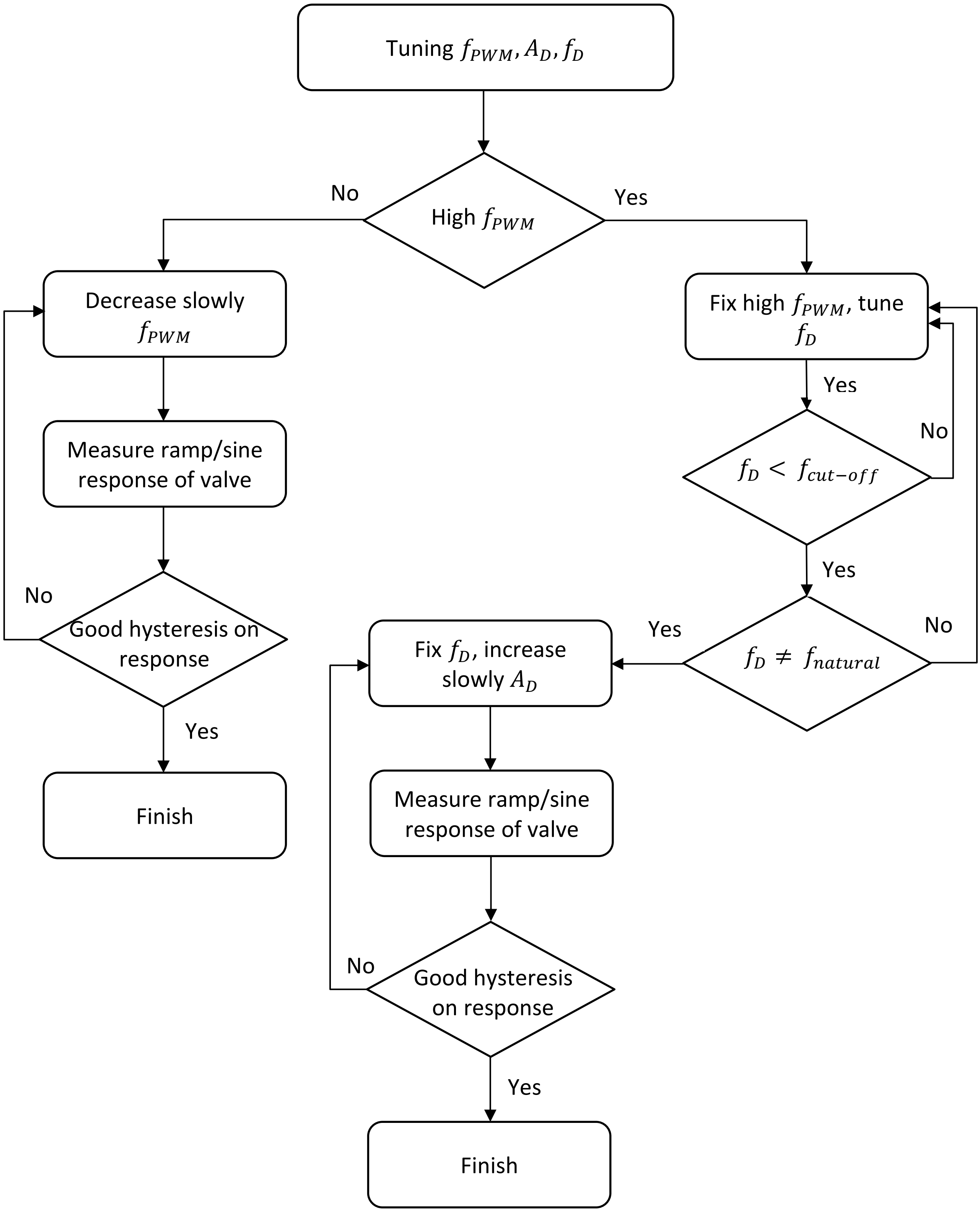

Figure 18. Parameters tuning method flowcharts

7 MODELING AND SIMULATION OF PROPORTIONAL VALVE CONTROL

7.1 Mathematical model of a proportional valve

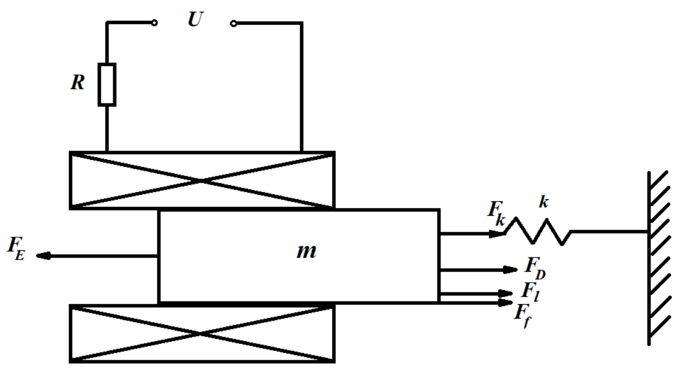

Figure 19. Schematic diagram of a proportional solenoid valve

The force balance equation of the spool is

$$\sum F = F_E - F_k - F_D - F_l - F_f \tag{6}$$

where $\sum F$ is the total force on the spool, F_E is the electromagnetic force, F_k is the spring force, F_D is the viscous damping force, F_f is the friction force, and F_l is the fluid force inside the valve, which is small enough to be ignored (Wang et al. 2016, p.609).

The motion differential equation of the spool determined from Equation (6) is

$$m\ddot{x} = \alpha i - kx - C\dot{x} - F_{friction} \tag{7}$$

where m is the mass of the valve spool, α is the magnetic coupling coefficient, k is the spring rate, C is the viscous damping coefficient, and k_f is the viscous friction coefficient.

Applying Kirchoff's voltage law, the time variation of the electric current through the solenoid coil can be described in the equation:

$$L\frac{di}{dt} + Ri = u - \alpha\frac{dx}{dt} \tag{8}$$

where L and R are the coil inductance and the coil resistance, i is the current in the coil and u is the applied voltage across the coil.

The flow rate of the valve is calculated by

$$Q = C_d A \sqrt{\frac{2\Delta P}{\rho}} \tag{9}$$

where Q is the valve flow rate, C_d is the flow rate coefficient, A is the opening area of the valve, ΔP is the pressure difference, and ρ is the density of fluid. The flow rate Q is proportional to A and A is proportional to the displacement of the spool x. Hence, the flow rate Q is proportional to the displacement x by a gain K_q and can be expressed

$$Q = K_q x \tag{10}$$

Combination of Equation (7), (8) and (10) is the mathematics model of a proportional valve with input voltage u and current i, and output flow rate Q.

7.2 Proportional valve simulation results

Objectives of the simulation are to capture the hysteresis occurring in a proportional valve and to examination the dither compensation effects on reducing the hysteresis. To capture most of the friction characteristics, the LuGre friction model was selected. Figure 20 describes the block diagram of the proportional valve employed in MATLAB Simulink. Table 1 shows the parameters used during the simulations.

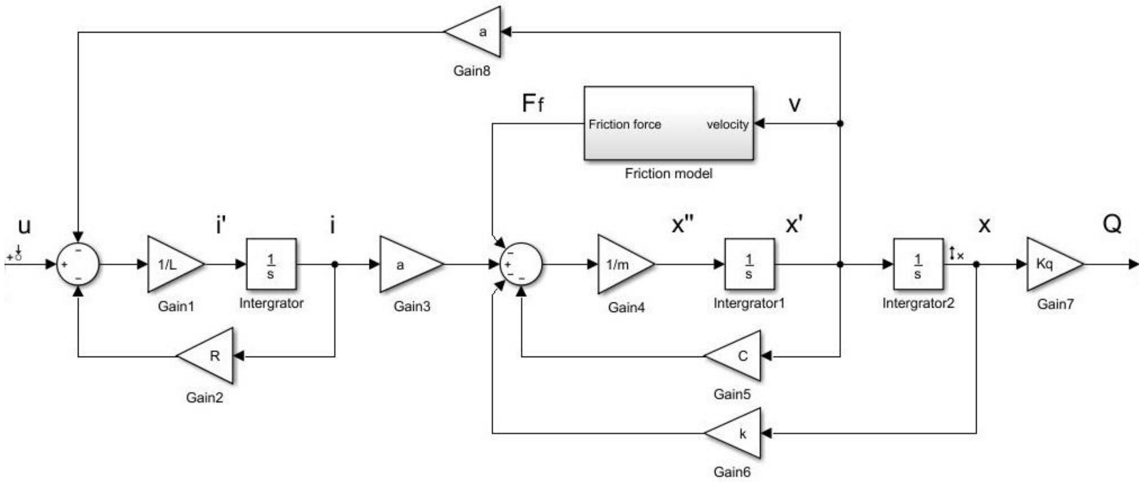

Figure 20. The block diagram of proportional valve

Table 1. Parameters determined for the simulation

Parameter	Value
$m\ (kg)$	0.4
$R\ (\Omega)$	100
$L\ (H)$	0.1
$\alpha\ (V.s.m^{-1})$	20
$k\ (N.m^{-1})$	17000
C	0.2
K_q	10^5
$F_c\ (N)$	0.1
$F_s\ (N)$	0.2
$v_s\ (m.s^{-1})$	0.0001
$\sigma_0\ (N.m^{-1})$	10^5
$\sigma_1\ (N.s.m^{-1})$	470
$\sigma_2\ N.s.m^{-1})$	0.2

7.2.1 Valve performance without dither

The first simulation was carried out to capture the open-loop step response of the valve. The initial value of input step was 0V and the final value were 1V, 2V and 12V. Figure 21 and 22 shows the step response results.

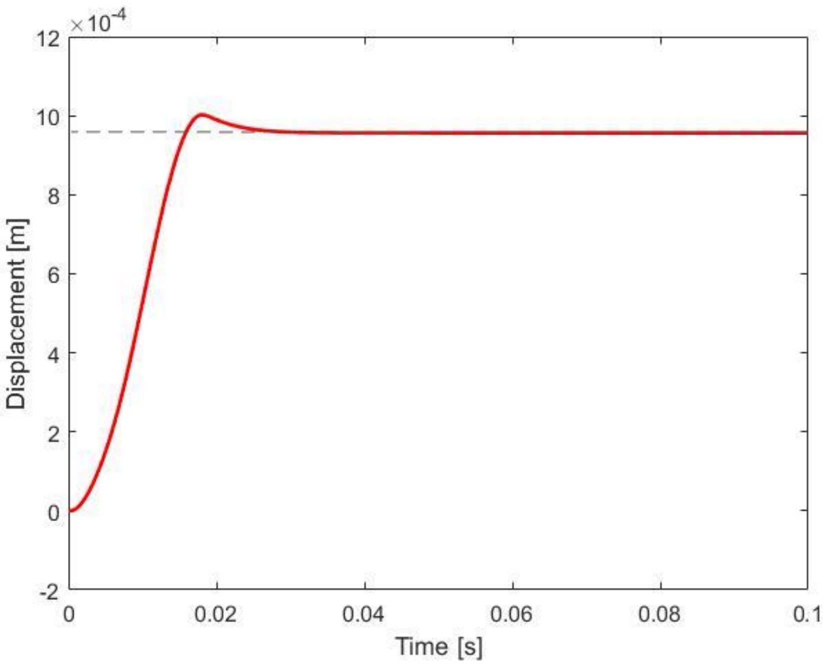

Figure 21. Response of valve at 1V step change.

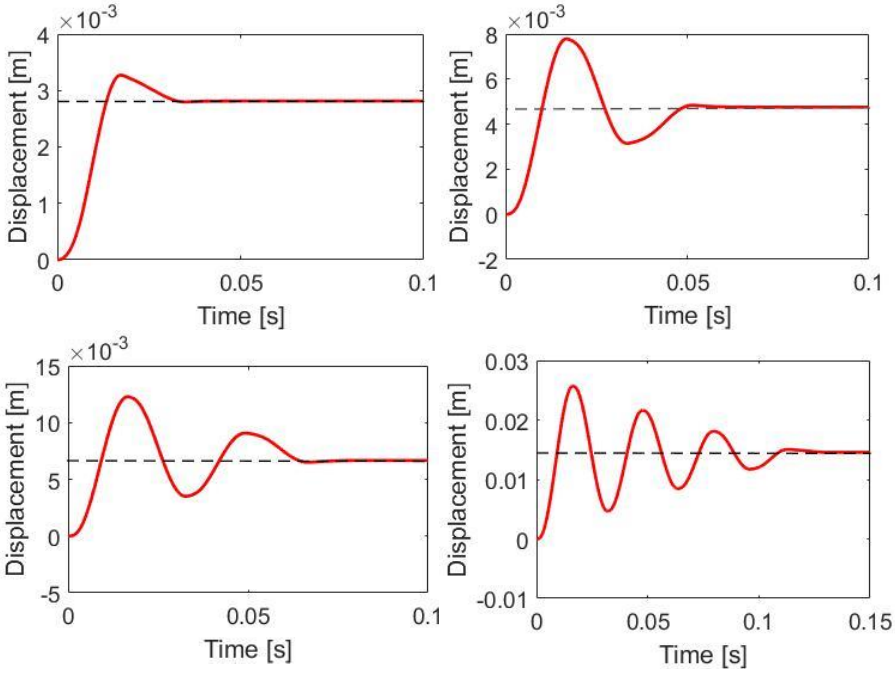

Figure 22. Response of valve at 2V, 4V, 6V and 12V step change

At 2V step change, the rise time of the valve is approximately 10ms and the settling time is about 30ms for the valve to obtain the steady state. Not much overshoot appears on the output response. However, at higher step changes, the valve takes longer to achieve the steady state and produces more overshoots. It seems this is not an optimal valve design, but it is good enough to examine the hysteresis.

To capture the hysteresis existence, a ramp input signal with duration of 15 seconds was used to apply the voltage to the valve. It was constant signal to imitate high frequency PWM. The ramp consists of an up ramp from 0V-12V, a period of steady, a down ramp from 12V-0V and another period of steady. Figure 23 shows the ramp waveform components.

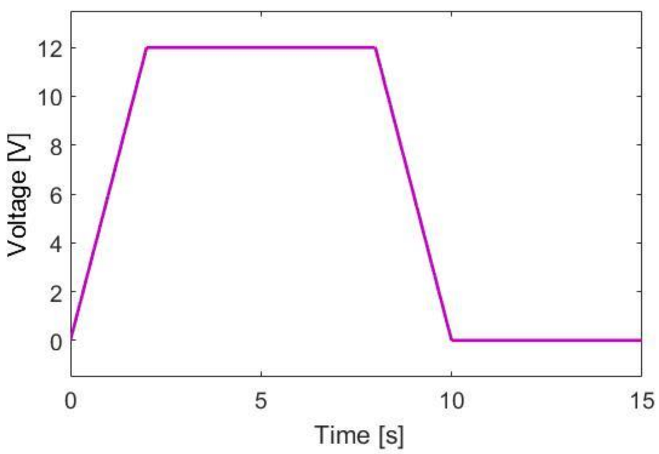

Figure 23. Ramp command with duration of 15 seconds

The result of the ramp simulation verified successfully the existence of hysteresis and another nonlinear characteristic of the valve. Figure 24 shows a staircase shape hysteresis. The hysteresis gap is approximately 8% of total flow rate change of the valve. The staircase shape is caused by stick-slip behaviour which is also a product of stiction force.

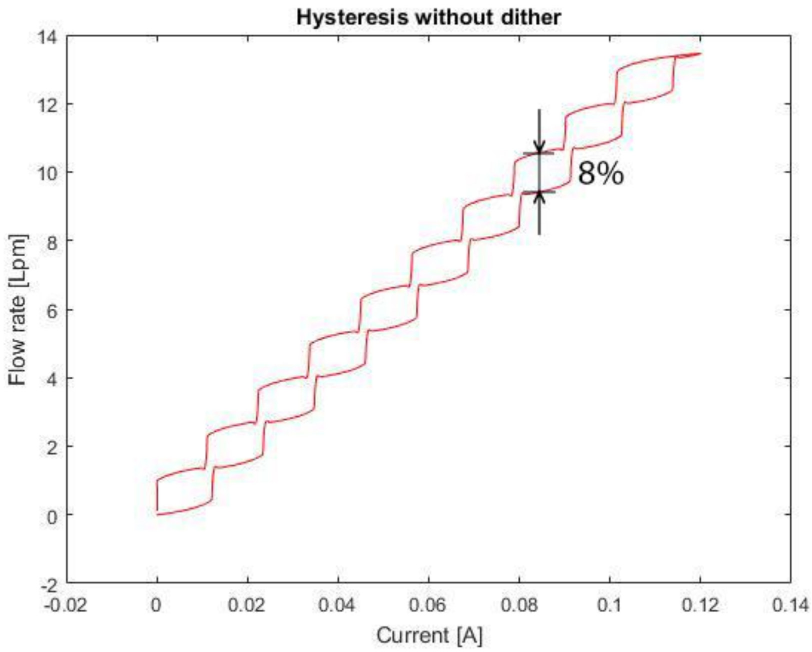

Figure 24. Staircase shape hysteresis and stick-slip behaviour captured by an input ramp of 15s duration

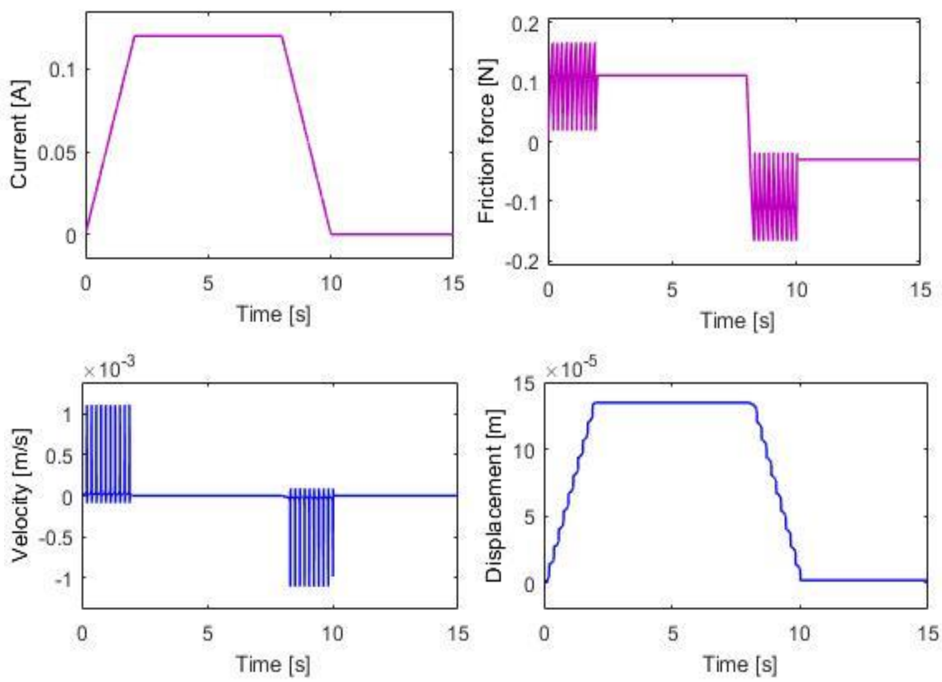

Figure 25. Current, velocity, friction force and spool displacement captured by slow ramp of 15s duration

Figure 25 shows the stick-slip displacement of spool valve according to slight changes of feeding current. The velocity jumps every time the feeding current exceeds a level that creates enough force to overcome stiction. The spool goes to the slipping phase. Otherwise, the velocity drops to zero, the spool valve experiences the sticking phase.

7.2.2 Valve performance with dither

In order to examine the dither effects on hysteresis, a sine wave was superimposed on the ramp command as the additional dither. Several settings of dither signal were employed with different amplitude and frequency. Figure 26 shows how to superimpose a dither signal on command signal.

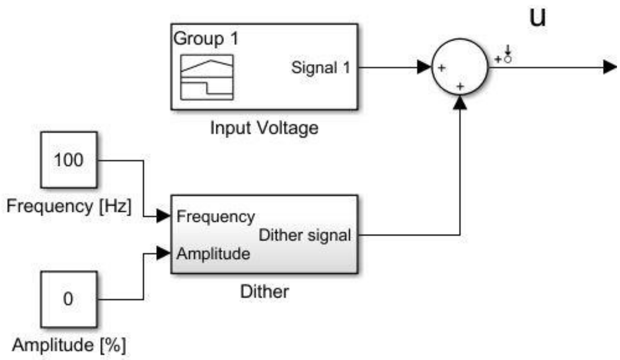

Figure 26. Dither signal is added to the command signal before feeding to the valve

A series of simulation experiments were carried out to test the dither effects on hysteresis. The simulations included hysteresis without dither, with 2% of maximum applied voltage 12V - dither amplitude (0.24V) at 100 Hz frequency, with 5% dither amplitude (0.6V) at 100Hz frequency and with 5% dither amplitude (0.6V) at 200Hz frequency. Figure 27 shows the captured hysteresis without dither and with 2% - 100Hz dither. As can be seen in Figure 27, after adding dither, the hysteresis is significantly reduced. Besides, the stick-slip behavior is virtually eliminated. Figure 28 describes the coil current, the spool velocity, the friction force and the spool displacement after adding 2% - 100Hz dither.

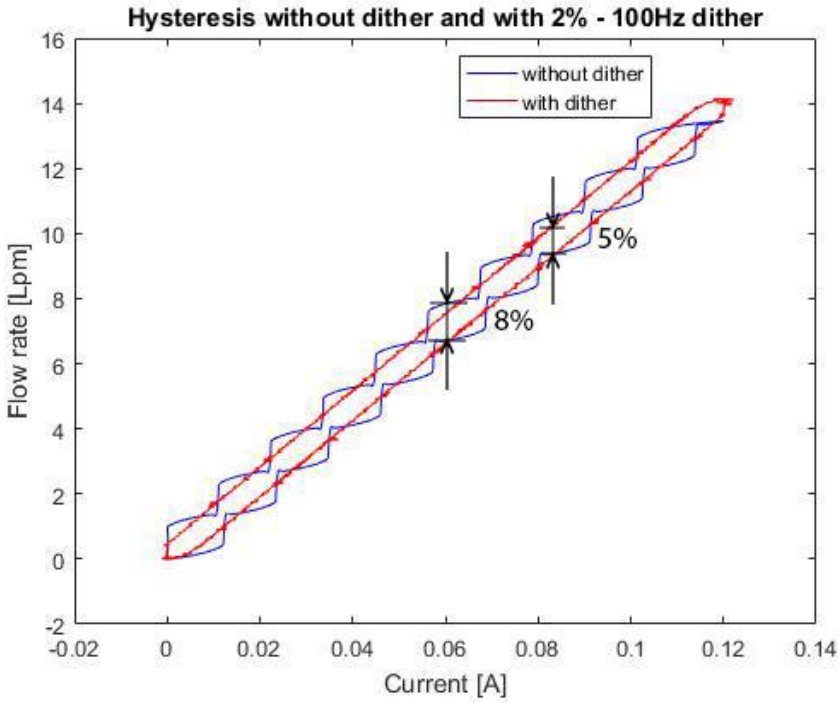

Figure 27. Hysteresis without dither and with 2% - 100Hz dither

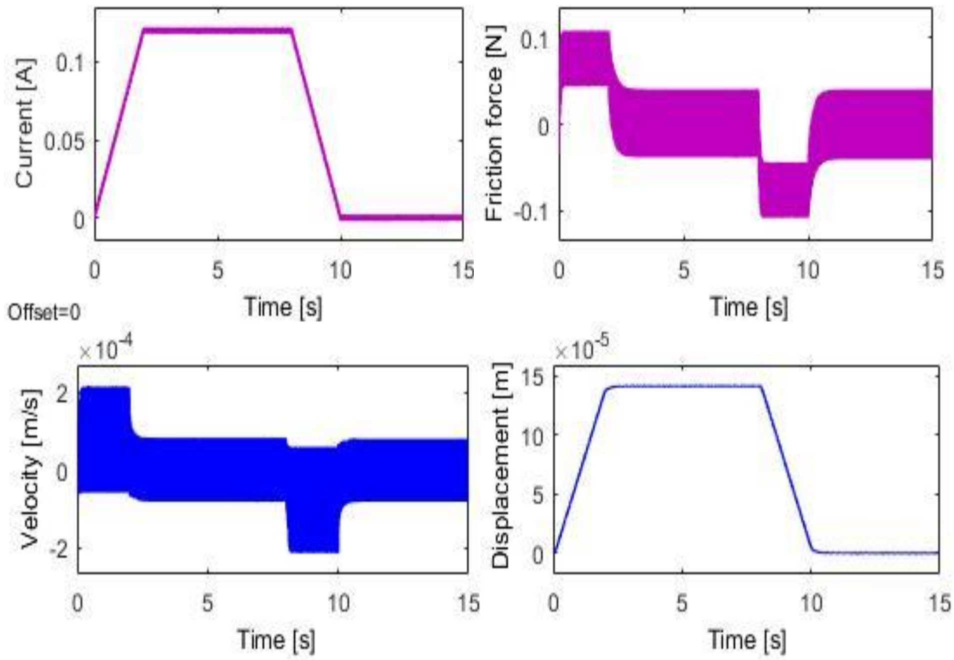

Figure 28. Current, velocity, friction force, displacement captured after adding 2% - 100Hz dither.

As shown in Figure 28, the dither constantly appears on the feeding current. The spool valve velocity also constantly oscillates. As a result, the stiction is eliminated, and the friction force is reduced to lower level. The spool valve also oscillates around its desired positions at micro-displacement scale.

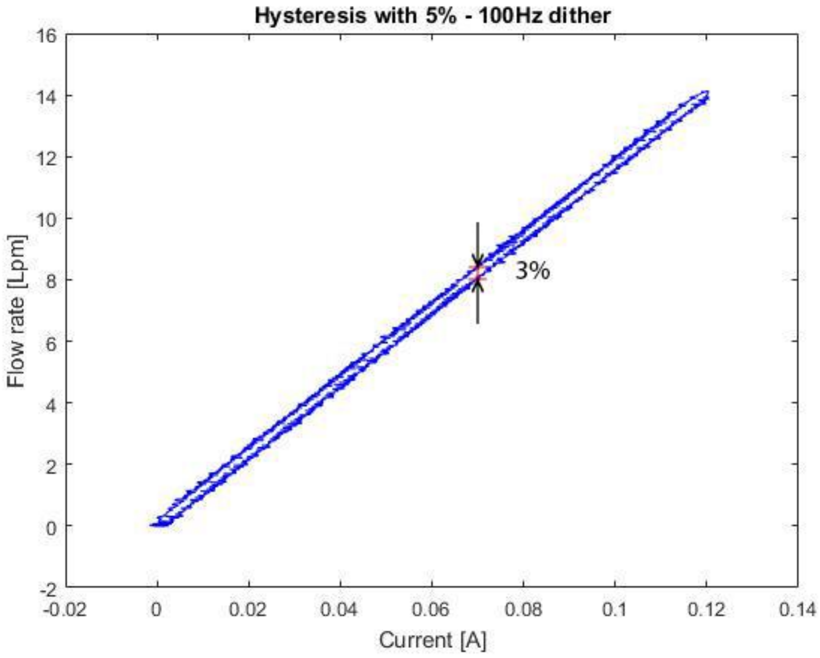

Figure 29. Hysteresis with 5% - 100Hz dither

Figure 29 shows the hysteresis with 5% dither amplitude at 100Hz frequency. Comparing with Figure 27, the hysteresis gap is significantly reduced. It verifies that with the same frequency, dither affects more on hysteresis if its amplitude is increased. However, with the same amplitude, dither affects less on hysteresis if its frequency is increased. Figure 30 shows a lager hysteresis gap than the hysteresis shown in Figure 29. Table 2 shows the hysteresis measured at different dither settings.

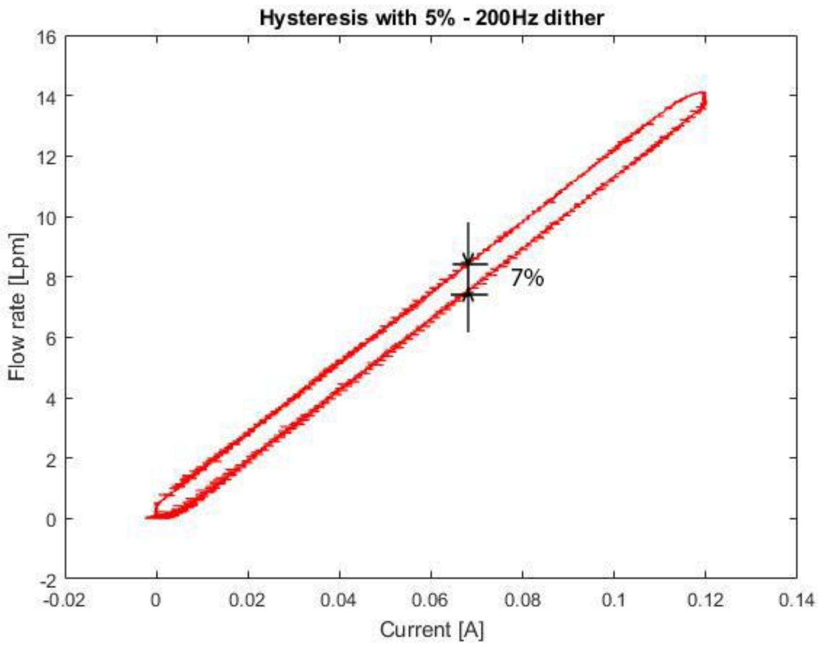

Figure 30. Hysteresis with 5% - 200Hz dither

Table 2. Simulation results

Dither setting	Hysteresis
No dither	8%
2% - 100Hz dither	5%
5% - 100Hz dither	3%
5% - 200Hz dither	7%

8 EXPERIMENTS

In the previous chapter, simulation results successfully verified the dither compensation principle for hysteresis occurring in proportional valves. The relationship between the effects of dither on hysteresis and dither parameters is partially substantiated. However, dither also impacts directly on the valve performance. When tuning the dither parameters, it is vital to get an optimal hysteresis without causing any other problems, such as dramatic vibration. Therefore, a set of experiments were executed to test the validation of the tuning method.

8.1 Experiment setup and test procedure

The experimental system included hydraulic components and an electrical power supply circuit. Proportional relief valve AP04G2YR35CN manufactured by Parker Hannifin was used to release the line pressure by adjusting the input current. A proportional valve amplifier, coded PWD00A-400X from Parker Hannifin, was used to drive the valve and was configured by parameterizing software ProPxD. DASYLab software was to generate input ramp signal to the amplifier. The measurement data was recorded by ServiceMaster plus hardware unit, parsed and analyzed by SensorWin PC analysis and MATLAB software.

Figures 31, 32 and 33 describe the hydraulics components and electrical devices setup.

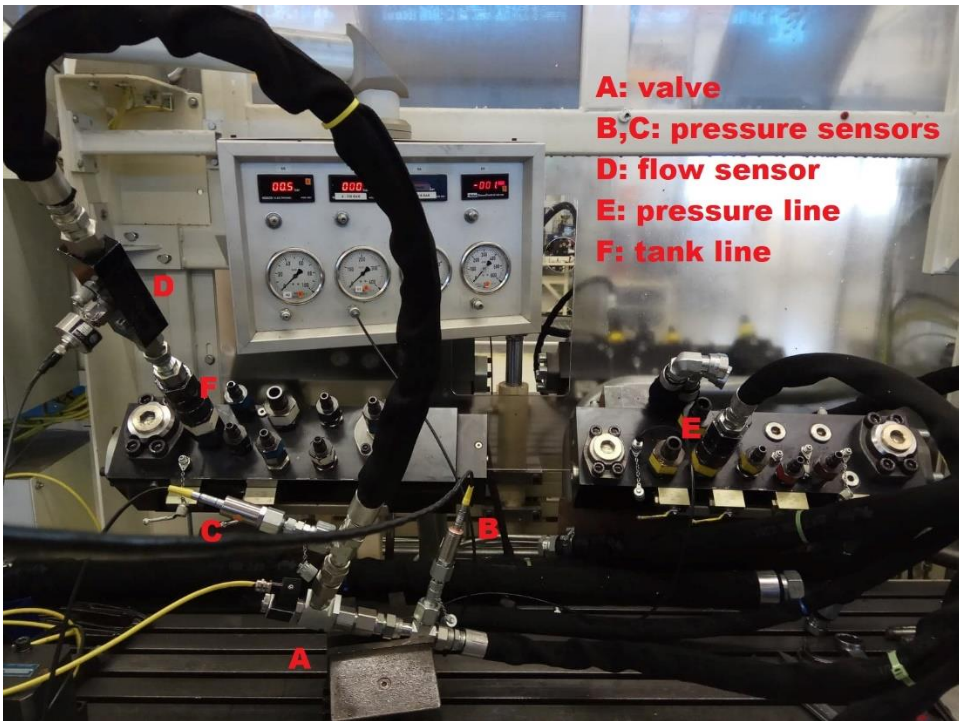

Figure 31. Hydraulic components setup

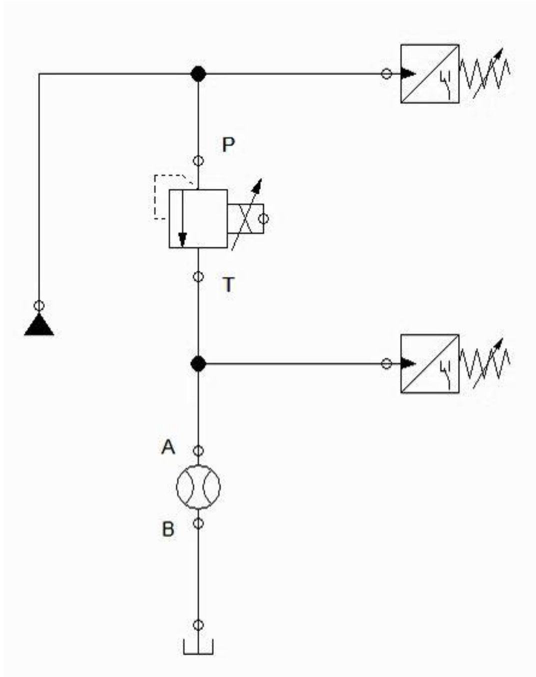

Figure 32. Schematic diagram of hydraulic components

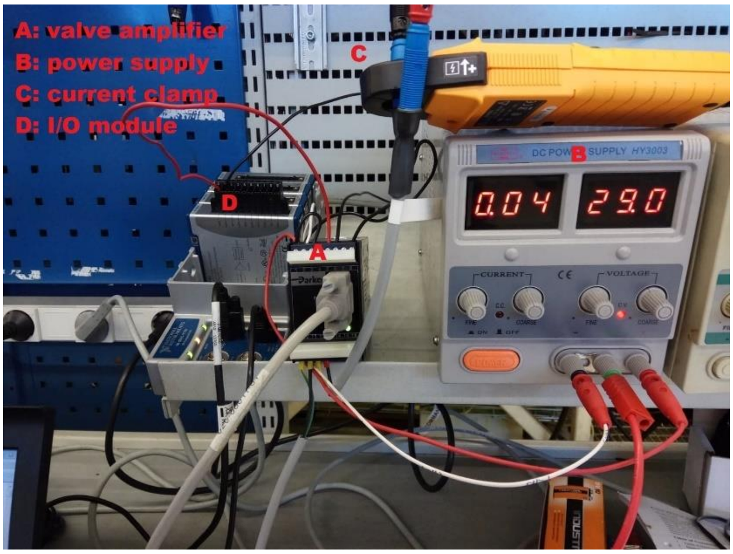

Figure 33. Electronic devices

The experiment process started with measuring the hysteresis with zero dither. The valve was fed by a high PWM ramp voltage command. The ramp components were 5s duration of steady state at 0V, 5s duration of up ramp from 0V to 12V, 10s duration of steady state at 12V, 5s duration of down ramp from 12V to 0V, and 5s duration of steady state at 0V. The next experiment was to test the dither effects and parameters tuning method by measuring inlet pressure, outlet flow rate and observing the valve performance with different dither parameters sets. Each set were measured in three times.

8.2 **Results**

Figure 34 shows the measured hysteresis when testing the valve without using dither. The plot shows a small noise on the valve response, which may due to the dither made by PWM signal. The hysteresis gap is quite wide, approximately 16%.

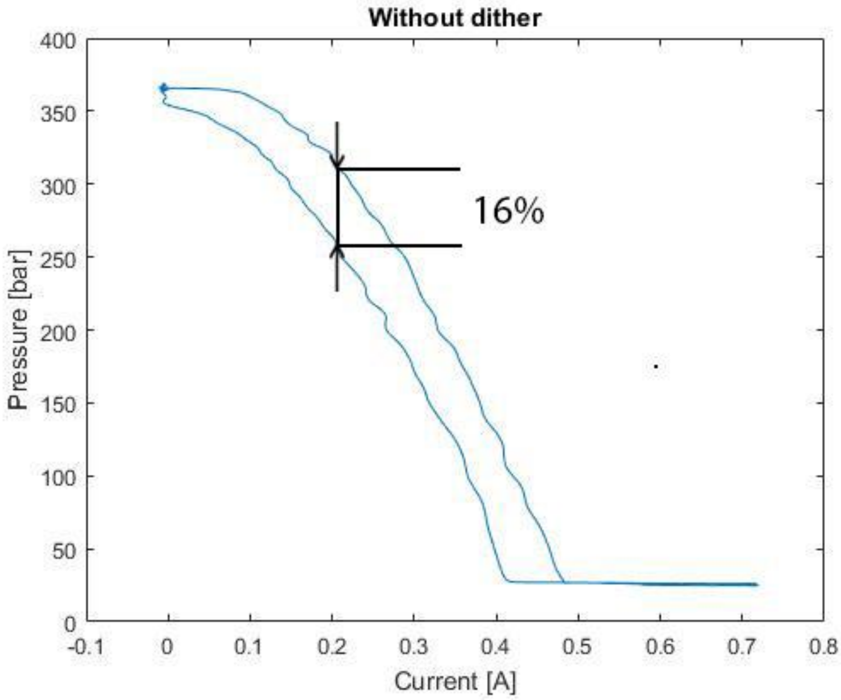

Figure 34. Measured hysteresis without dither

With this dither setting, hysteresis was approximately 14%. At the same frequency of 50Hz, the dither amplitude increased to 4%. Consequently, the hysteresis was reduced to nearly 13%. Figure 35 describes the different measurement results among different dither settings.

When increasing the dither amplitude to 6%, 8% and 10% at the same frequency of 50Hz, the valve vibrated aggressively. The largeness of vibration amplified with higher amplitude of dither. It was same with the theory prediction and simulation results. Figure 36 demonstrates the unstable performance of valve at 8% - 50Hz dither compared with the well performance at 2% - 50Hz dither. Too much dither on the input current causes aggressive oscillation on the output pressure and flow rate. In Figure 36, the pressure is unstable at some current levels at 8% - 50Hz dither.

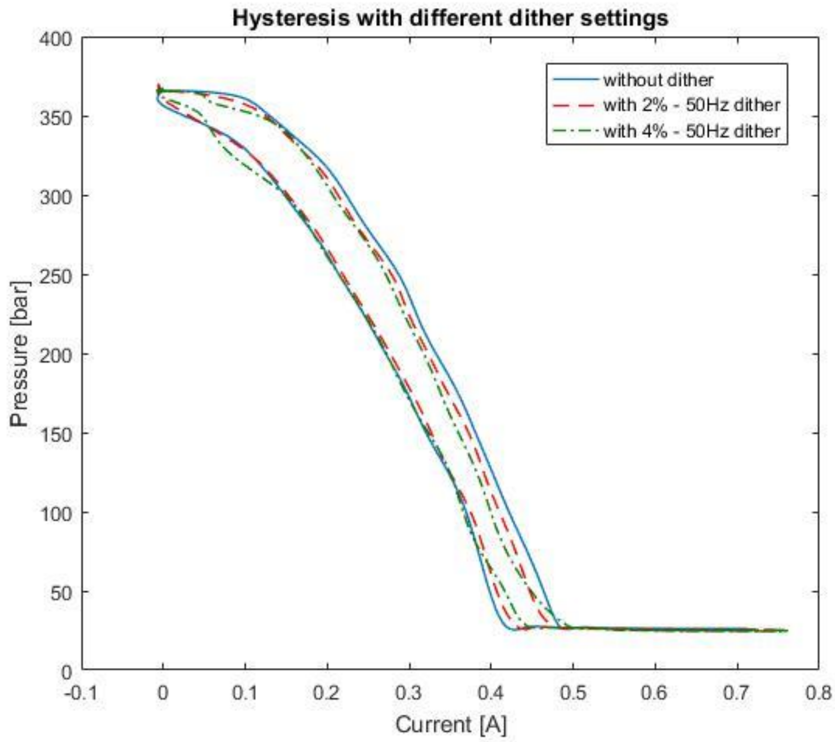

Figure 35. Measured hysteresis with different dither settings

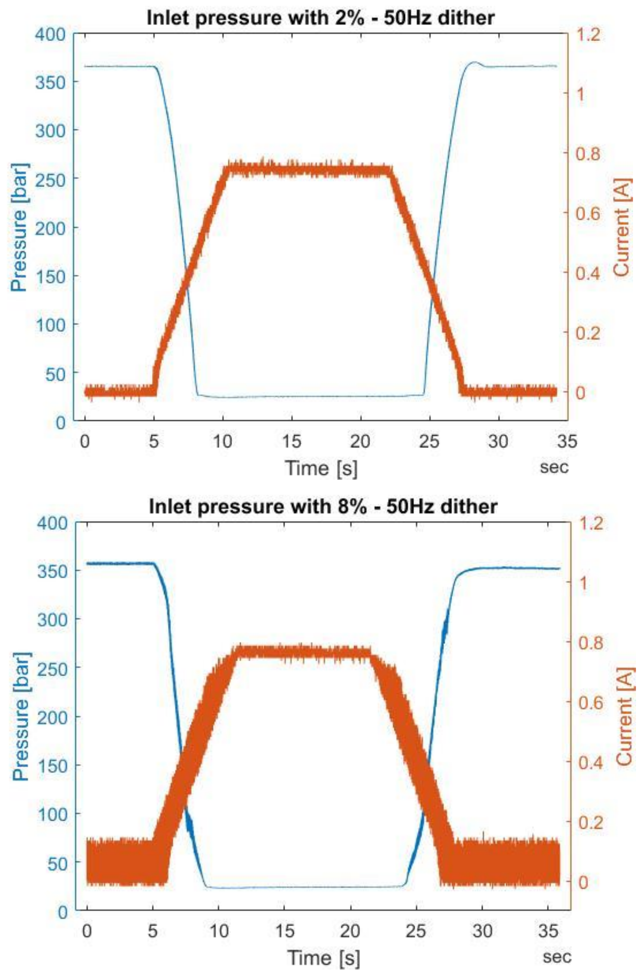

Figure 36. Valve performances at 2% - 50Hz dither and 8%-50Hz dither comparison.

The measurement process continued with other dither parameters sets. Table 3 describes the measurement results. All dither settings marked as minus-sign (-) are unacceptable dither which cause the valve poor stability. Those parameter sets marked as plus-sign (+) are acceptable dither which do not generate much vibration. The arrows directions indicate the tendency of unstable performance.

Table 3. Dither parameters tuning results.

300 Hz	+	+	+	+	+
250 Hz	+	-	-	-	-
200 Hz	+	-	-	-	-
150 Hz	+	-	-	-	-
100 Hz	+	-	-	-	-
50 Hz	+	+	-	-	-
Frequency / Amplitude	2%	4%	6%	8%	10%

To find out the best dither parameters, the measured hysteresis results are examined. The dither settings located near the boundary of unstable area are taken into account to compare their effectiveness on reducing hysteresis. Figure 37 shows different measured hysteresis among potential settings. As can be seen in Figure 37, 2% - 50Hz and 4% - 50Hz dither had the best effect on reducing hysteresis. The others on the other hand, although given a stable performance, however their responses were hardly predicable. The dither effects on hysteresis has been proven. The correctness of the parameters tuning method was also partially verified. To fully verify the tuning method, it is essential to determine the natural frequency and cut-off frequency of the valve which is to be used to find an optimal dither for hysteresis compensation.

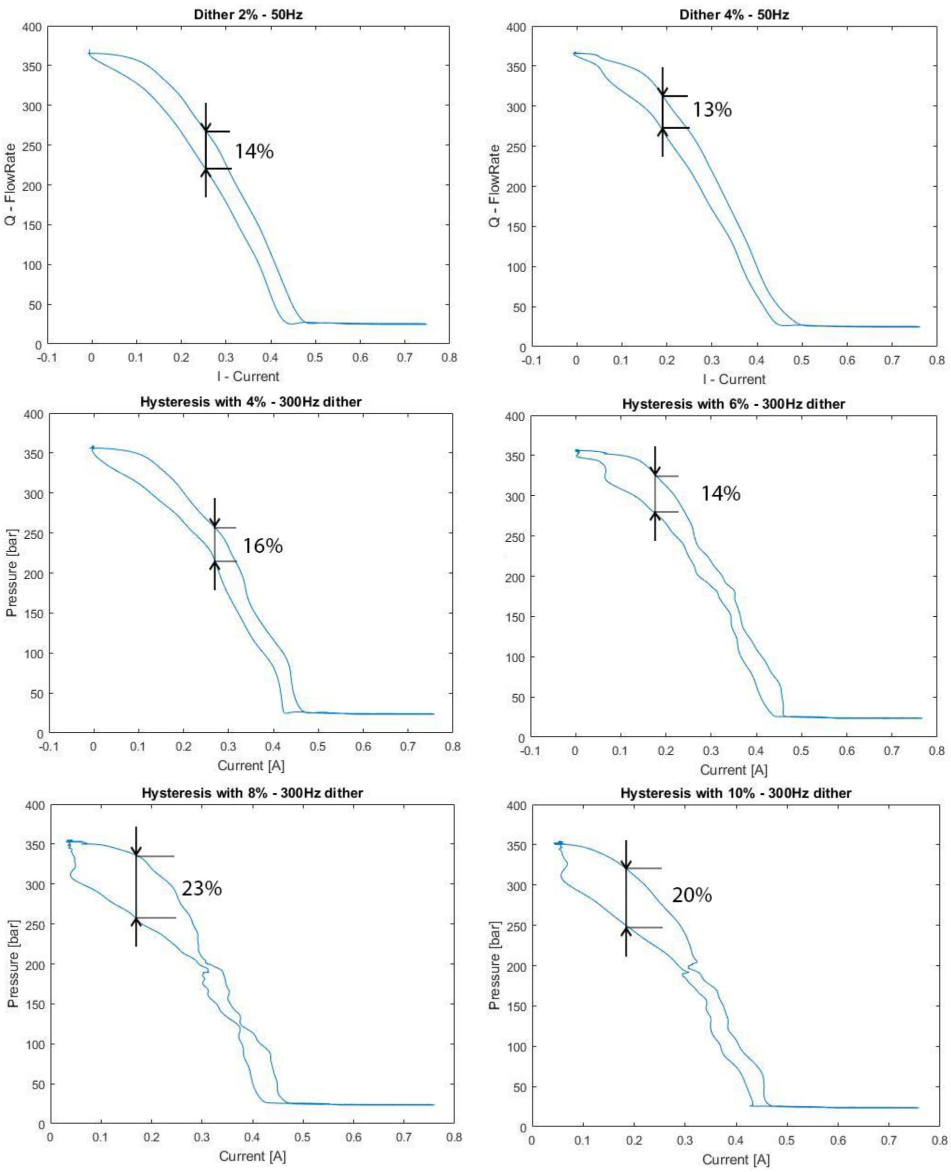

Figure 37. Hysteresis among different dither parameters value sets

9 CONCLUSION

In this thesis project, the existence of hysteresis with hydraulic proportional valves was thoroughly examined as to the reasons causing the phenomenon to the solution for reducing it. Stiction was pointed out as the main reason. To eliminate the nonlinearity caused by stiction, such as hysteresis, dither compensation was accessed with a proposed parameter tuning method to obtain an optimal result. Dithering is a technique in proportional valve control that uses artificial noise to generate small vibration on the command signal. Consequently, the valve spool is forced to move constantly around its desired position. This micro-movement will eliminate stiction, therefore significantly reducing the hysteresis, a by-product of stiction force.

There are two common ways to generate dither on the command signal. Using a low frequency PWM can produce dither. However, the amount of dither cannot be adjusted independently, therefore, it may not effectively reduce hysteresis in some cases. Another method is to superimpose an external dither signal on the command signal. Because the external signal is separately added, therefore its amplitude and frequency can be adjusted freely according to user desires. The dither parameters impact directly on the degree of hysteresis reduction and valve performance. Hence, a parameter tuning method was proposed to obtain the optimal hysteresis without causing other problems.

To examine how dither effects on hysteresis and the correctness of the tuning method, a model of a proportional valve and several models of friction were employed. A set of simulation experiments and practical measurements were carried out. The simulation results predicted a good existence of hysteresis and the effectiveness of dither on reducing hysteresis. The practical measurements were conducted with different sets of dither parameters to verify the parameter tuning method to achieve the best performance of valve with the finest hysteresis. However, an optimal parameter set could not be obtained. The reason was the difficulty in measuring the natural frequency and cut-off frequency of the valve which required high accuracy measurement devices and advanced practical skills. Another limitation was that the measurements were conducted without testing a low frequency PWM. As mentioned above, using a low frequency PWM can generate dither which may significantly compensate the hysteresis.

Further work to fully verify the parameter tuning method can be accomplished such as determining all the valve characteristics, including natural frequency and cut-off frequency or testing valve performance with low frequency of PWM. These are considered as valuable tasks because some manufacturers can apply the method to publish a guidance of selecting parameters for their products. The guidance could powerfully support engineers to significant saving on time on field.

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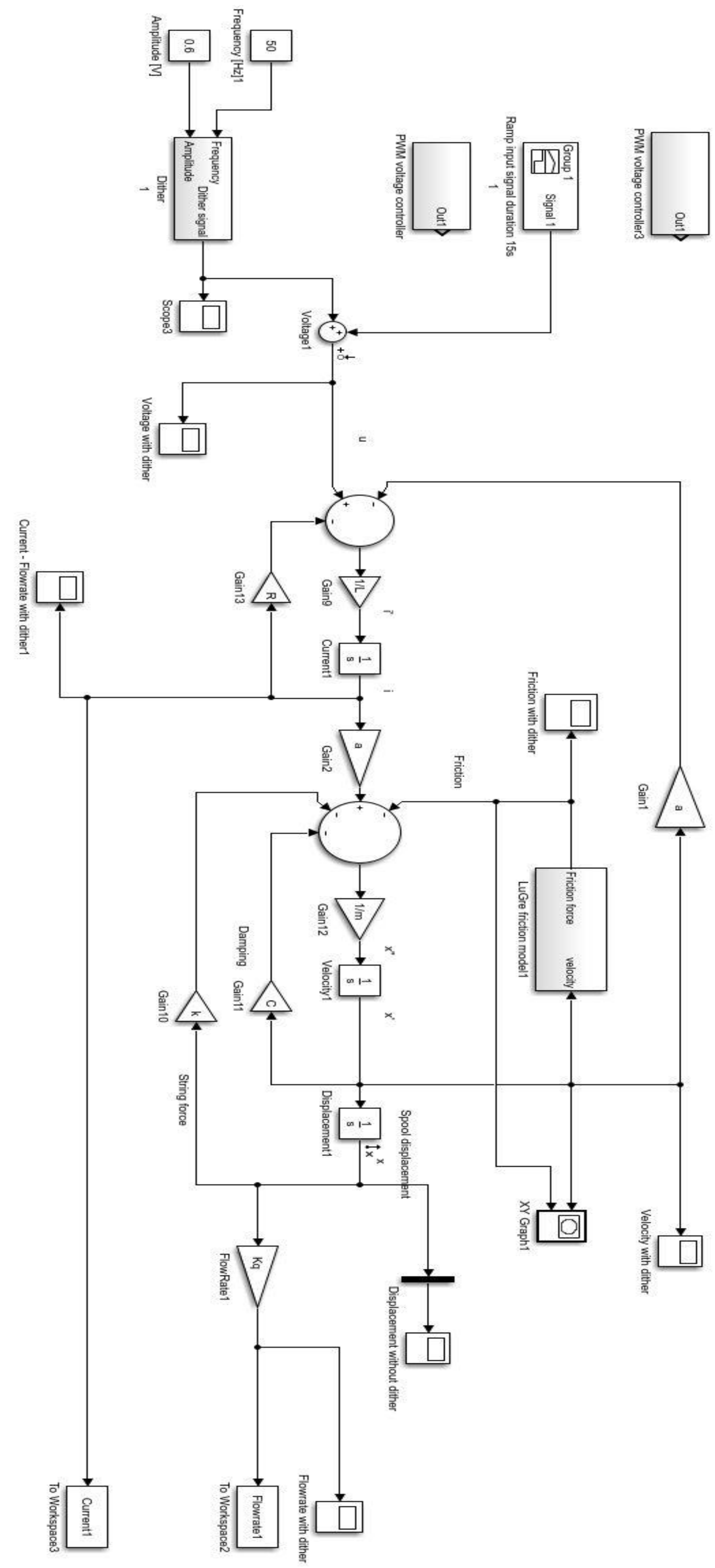
Appendix 1

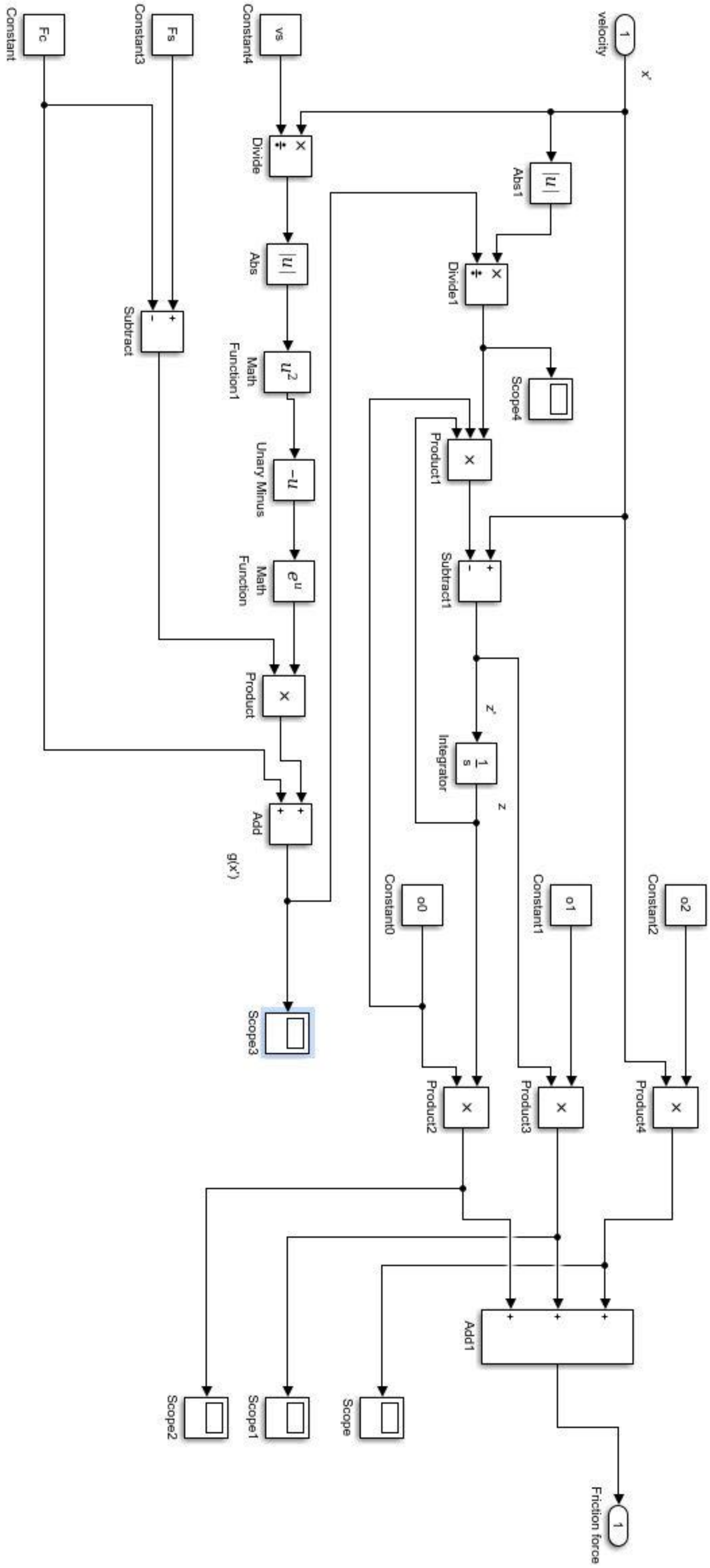
MATLAB scripts – Valve simulation script

```
%Valve parameters
m = 0.4       %Spool mass
R = 100      %Coil resistance
L = 0.1      %Coil inductance
a = 20       %Magnetic coupling coefficient
k = 17000     %Spring rate
C = 0.2      %Viscous damping coeffiction
Kq = 100000     %Flow rage gain

%Friction model parameters
%LuGre model
Fc = 0.1;          %Coulomb friction force
Fs = 0.2;         %Static friction force
vs = 0.0001;       %Stribek velocity
o0 = 10^5          %Contact stiffness
o1 = 470           %Damping coeffiction of bristle
o2 = 0.2;          %Viscous friction coeffiction
j = 2;             %Stribek shape factor (j=2 is often used in
literature)

%Other friction model
Kf = 0.14          %Viscous friction coeffiction
kv = 0.1           %Viscous friction coefiction
u = 0.2;
g = 9.8
FN = u*m*g
i = 0
q = 1.5
kz = 0.00001

%Plot hysteresis curves
set(0,'ShowHiddenHandles','on')
set(gcf,'menubar','figure')
figure(1)
%[B,A] = butter(5,(20)/250);
%a = filtfilt(B,A,Current.signals.values(:,1));
%b = filtfilt(B,A,Flowrate.signals.values(:,1));
%plot(Current.signals.values(:,1),Flowrate.signals.values(:,1),'color'
,'r')
%plot(a,b,'r')
title('Hysteresis with 5% - 200Hz dither')
xlabel('Current [A]');
ylabel('Flow rate [Lpm]');
figure(2)
%[D,C] = butter(5,(20)/1050);
%x = filtfilt(D,C,Current1.signals.values(:,1));
%y = filtfilt(D,C,Flowrate1.signals.values(:,1));
%plot(Current1.signals.values(:,1),Flowrate1.signals.values(:,1),'colo
r','b')
%plot(x,y,'b')
title('Hysteresis with 5% - 50Hz dither')
xlabel('Current [A]');
ylabel('Flow rate [Lpm]');
figure(3)
%plot(a, b,'r',x,y,'b')
title('Hysteresis with dither')
%legend('without dither','with dither')
```

```
xlabel('Current [A]');
ylabel('Flow rate [Lpm]');


%Data parsing and ploting
t = 0:seconds(0.004):seconds(36.932);
I = (MINvalue7  + MAXvalue7 + ACTvalue7)/3;
P = (MINvalue10 + MAXvalue10 + ACTvalue10)/3;
[B,A] = butter(5,(40)/1050);
x = filtfilt(B,A,I);
y = filtfilt(B,A,P);
figure(1)
plot(x,y);
title('Hysteresis with 4% - 300Hz dither')
xlabel('Current [A]');
ylabel('Pressure [bar]');
figure(2)
plot(I,P);
title('Hysteresis with 4% - 300Hz dither')
xlabel('Current [A]');
ylabel('Pressure [bar]');
figure(3)
plot(t,P)
title('Inlet pressure with 4% - 300Hz dither')
xlabel('Time [s]');
ylabel('Pressure [bar]');
figure(4)
plot(t,I)
title('Current with 4% - 300Hz dither')
xlabel('Time [s]');
ylabel('Current [A]');
```

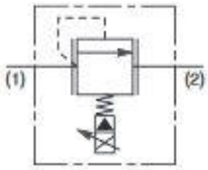


Frequency [Hz]1
50
Amplitude [V]
0.6
Frequency
Amplitude
Dither signal
Dither
1
PWM voltage controller
Out1
Group 1
Signal 1
Ramp input signal duration 15s
1
PWM voltage controller3
Out1
Scope3
Voltage1
Voltage with dither
u
1/L
Gain9
R
Gain13
1
s
Current1
i
Current - Flowrate with dither1
a
Gain2
Friction
Friction with dither
a
Gain1
1/m
Gain12
1
s
Velocity1
Friction force
velocity
LuGre friction model1
C
Damping
Gain11
k
Gain10
String force
1
s
Displacement1
x
Spool displacement
XY Graph1
Velocity with dither
Kq
FlowRate1
Displacement without dither
Flowrate with dither
Current1
To Workspace3
Flowrate1
To Workspace2


1
velocity
x'
vs
Constant4
Fs
Constant3
Fc
Constant
Divide
|u|
Abs1
|u|
Abs
Divide1
Subtract
u^2
Math Function1
Scope4
$-u$
Unary Minus
Product1
e^u
Math Function
Subtract1
Product
z'
$\frac{1}{s}$
Integrator
z
Add
g(x')
o0
Constant0
o1
Constant1
o2
Constant2
Scope3
Product2
Product3
Product4
Add1
Scope2
Scope1
Scope
1
Friction force

General Description

Proportional Relief Valve. Decreasing Pressure With Increasing Current. For additional information see Technical Tips on pages PV1-PV6.



Features

- Pilot operated spool-type design
- Precise setting of factory preset pressure in de-energized mode
- One piece cartridge housing ensures internal concentricity
- Coil: Waterproof, hermetically sealed, requires no O'Rings; Symmetrical coil can be reversed without affecting performance.
- All external parts zinc plated

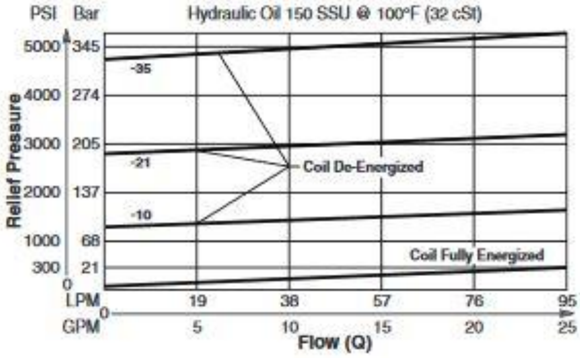
Specifications

Rated Flow (At 300 PSI ΔP) When Coil is Fully Energized	95 LPM (25 GPM)
Factory Set Relief Pressure When Coil De-Energized Measured at 45 LPM (12 GPM)	**10C** 103 Bar (1500 PSI) **21C** 210 Bar (3000 PSI) **35C** 350 Bar (5000 PSI)
Hysteresis @ 250 Hz PWM	< 7% of Maximum Pressure Setting
Response Time At 75% of Nominal Voltage Change (Measured To 90% of Press. Change)	**To Unload** 45ms **To Load** 25ms
Cartridge Material	All parts steel. All operating parts hardened steel.
Operating Temp. Range/Seals	-40°C to +93.3°C (Nitrile) (-40°F to +200°F) -31.7°C to +121.1°C (Fluorocarbon) (-25°F to +250°F)
Fluid Compatibility/ Viscosity	Mineral-based or synthetic with lubricating properties at viscosities of 45 to 2000 SSU (6 to 420 cSt)
Filtration	ISO Code 16/13, SAE Class 4 or better
Approx. Weight	.14 kg (.30 lbs.)
Cavity	C10-2 (See BC Section for more details)

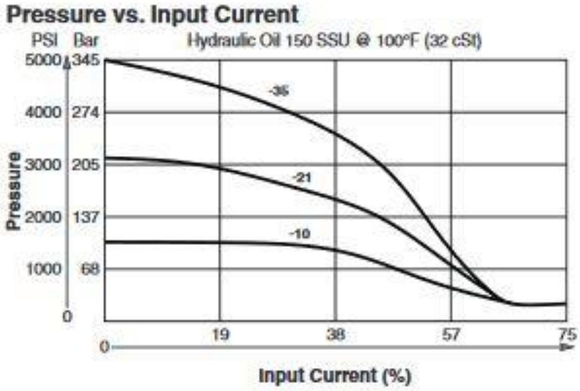
Performance Curves

▲PWM Current Regulator Recommended

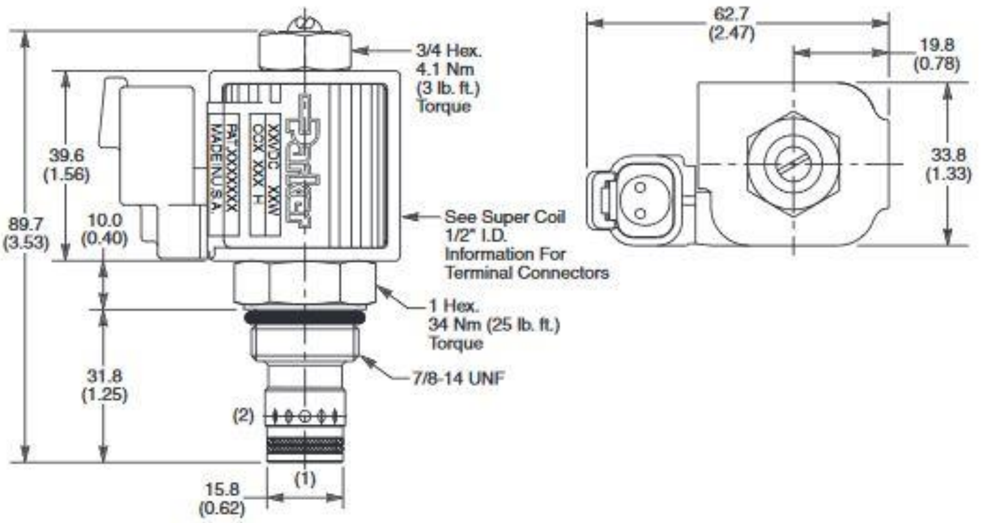
Relief Performance



Pressure vs. Input Current



Dimensions Millimeters (Inches)



THIRD ANGLE PROJECTION

Ordering Information

AP04G2YR						–		
10 Size Proportional Relief Valve	Style	Seals	Coil Type	Coil Voltage	Coil Termination		Body Material	Port Size

Code	Style (Maximum Relief Pressure)
10C	104 Bar (1500 PSI)
21C	210 Bar (3000 PSI)
35C	350 Bar (5000 PSI)
XXC	Custom Pressure Setting Expressed in 1/10 of Bar (Optional)

** Set at .94 LPM (.25 GPM)*

Code	Seals / Kit. No.
N	Nitrile / Buna-N (Std.) (SK30503N-1)
V	Fluorocarbon / (SK30503V-1)

Code	Coil Type
Omit	Without Coil
SP	Super Coil - 19 Watts

Code	Coil Voltage
Omit	Without Coil
D012	12 VDC
D024	24 VDC

Code	Coil Termination
Omit	Without Coil
D	DIN Plug Face
A	Amp Jr. Timer*
L	Dual Lead Wire*
H	Molded Deutsch*

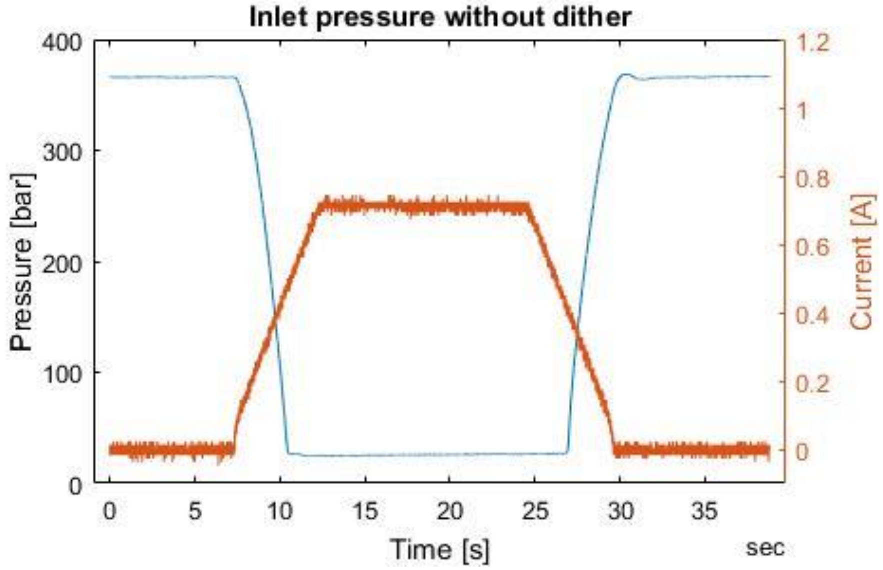
See Super Coil 1/2" I.D.
**DC Only*

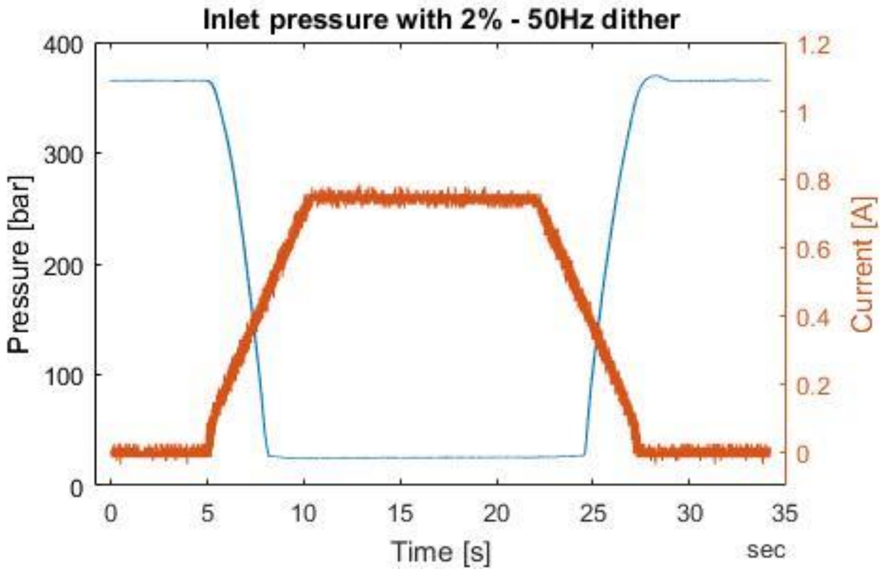
Code	Body Material
Omit	Steel
A	Aluminum

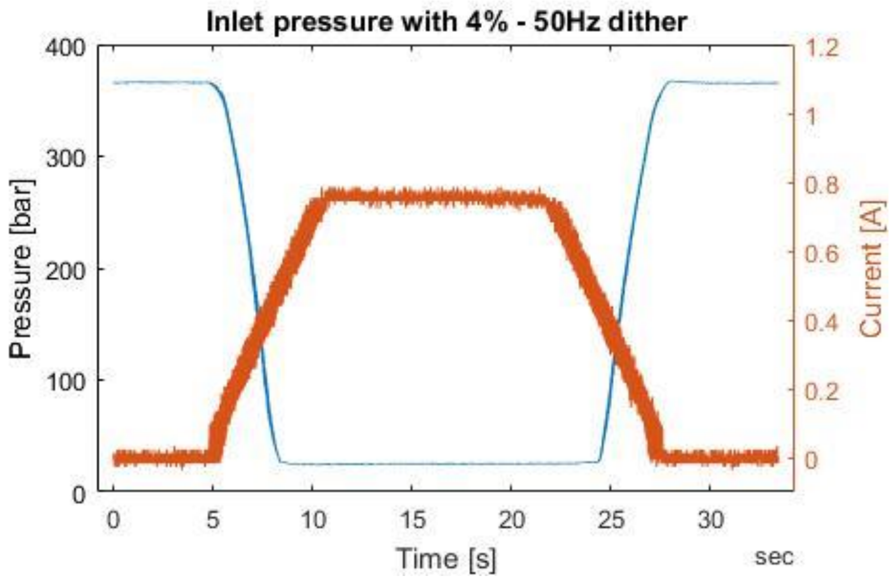
Code	Port Size	Body Part No.
Omit	Cartridge Only	
4P	1/4" NPTF	(B10-2-*4P)
6P	3/8" NPTF	(B10-2-*6P)
8P	1/2" NPTF	(B10-2-*8P)
6T	SAE-6	(B10-2-*6T)
T6T	SAE-6	(B10-2-T6T)†
8T	SAE-8	(B10-2-*8T)
T8T	SAE-8	(B10-2-T8T)†
6B	3/8" BSPG	(B10-2-*6B)

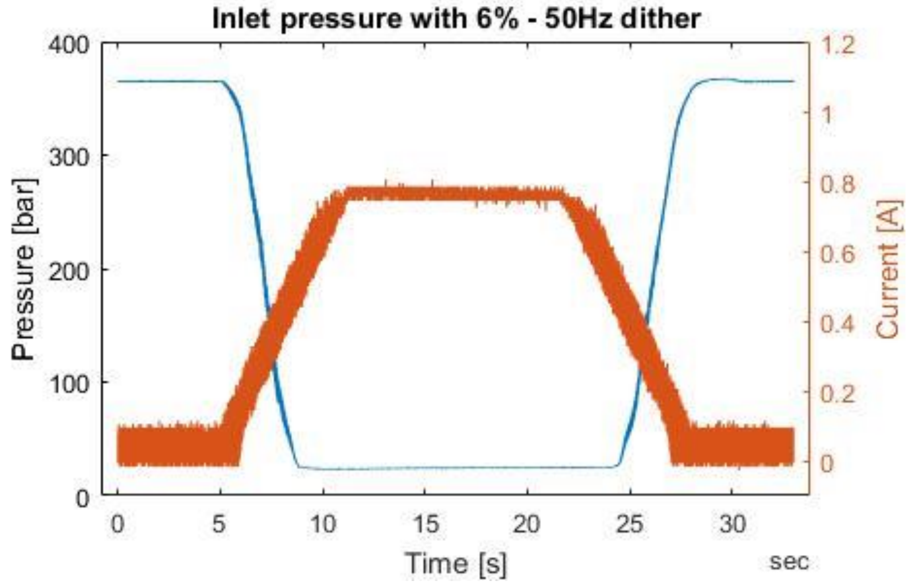
** Add "A" for aluminum, omit for steel.*
† Steel body only.

CV Check Valves
SH Shuttle Valves
LM Load/Motor Controls
FC Flow Controls
PC Pressure Controls
LE Logic Elements
DC Directional Controls
MV Manual Valves
SV Solenoid Valves
PV Proportional Valves
CE Coils & Electronics
BC Bodies & Cavities
TD Technical Data

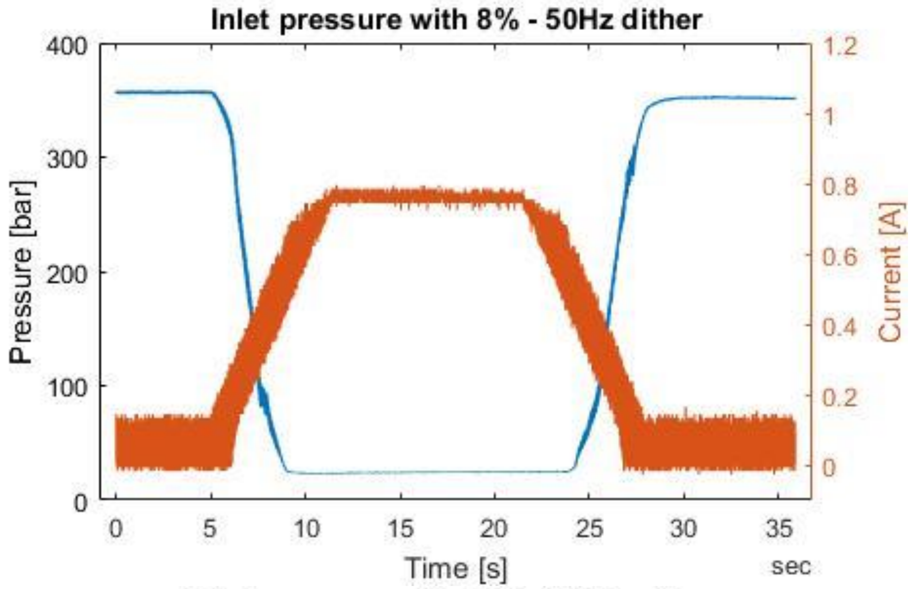

Inlet pressure without dither
Pressure [bar]
Current [A]
Time [s]
sec
400
300
200
100
0
1.2
1
0.8
0.6
0.4
0.2
0
0
5
10
15
20
25
30
35


Inlet pressure with 2% - 50Hz dither
Pressure [bar]
Current [A]
Time [s]
sec
400
300
200
100
0
1.2
1
0.8
0.6
0.4
0.2
0
0
5
10
15
20
25
30
35

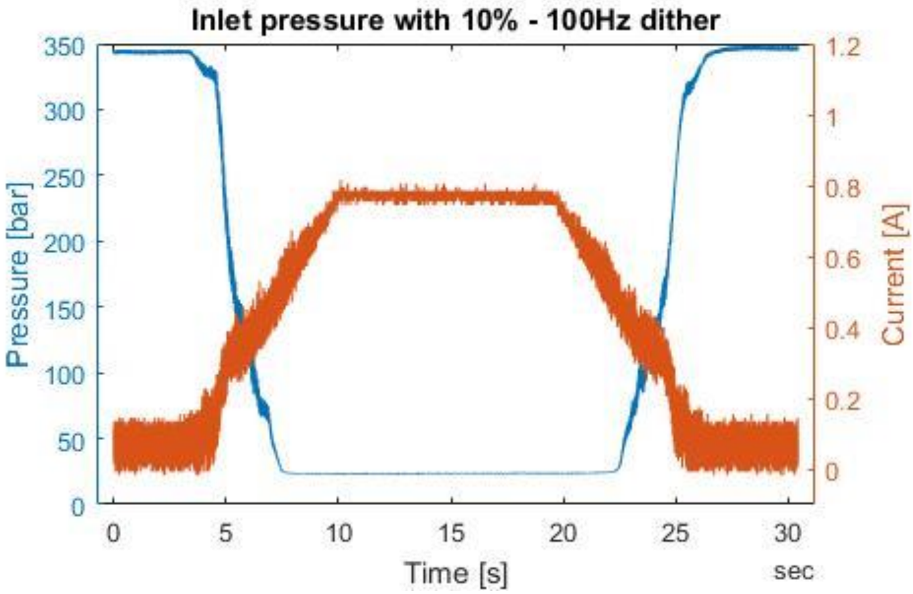

Inlet pressure with 4% - 50Hz dither
Pressure [bar]
Current [A]
Time [s]
sec
400
300
200
100
0
1.2
1
0.8
0.6
0.4
0.2
0
0
5
10
15
20
25
30



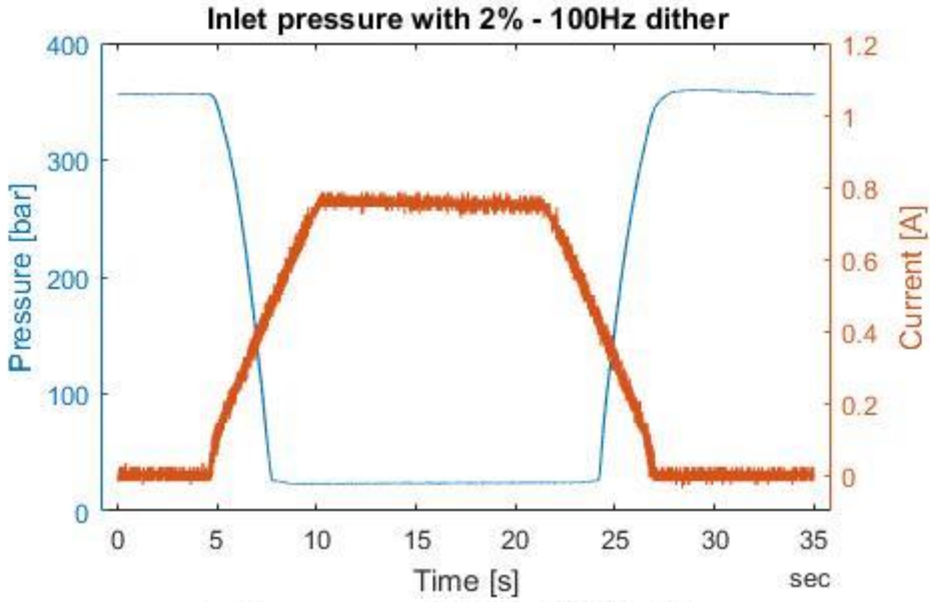
Inlet pressure with 6% - 50Hz dither
Pressure [bar]
Current [A]
Time [s]
sec
0
100
200
300
400
0
0.2
0.4
0.6
0.8
1
1.2
0
5
10
15
20
25
30

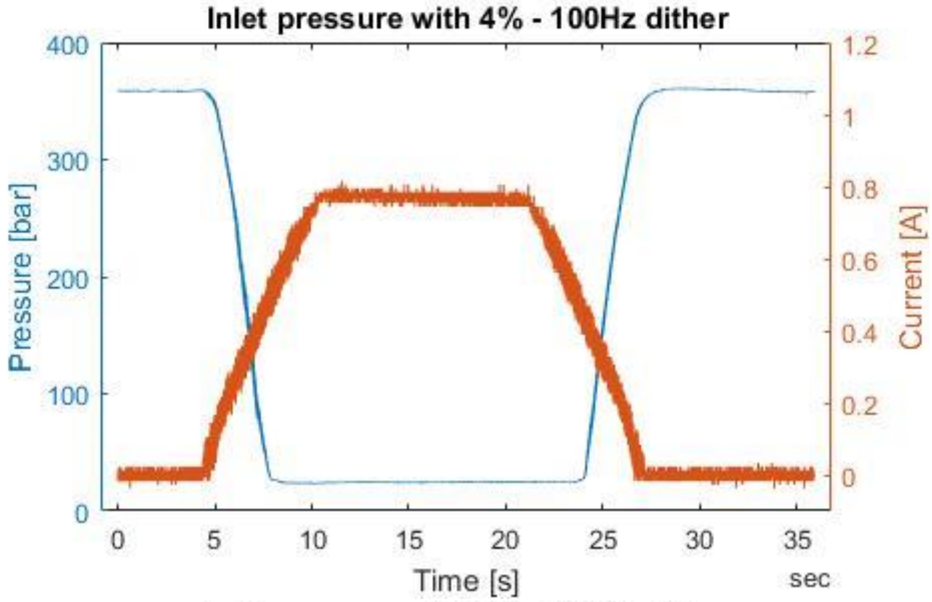


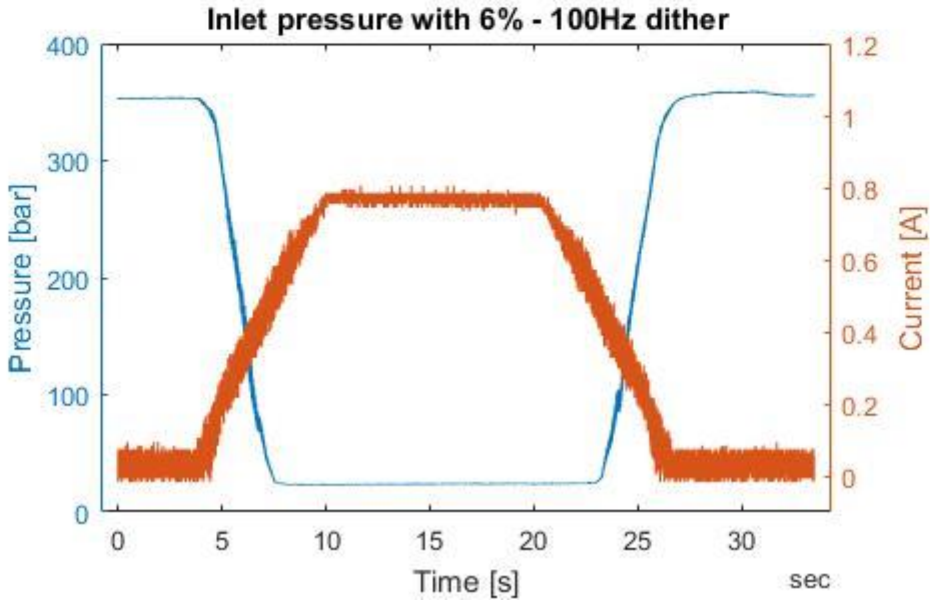
Inlet pressure with 8% - 50Hz dither
Pressure [bar]
Current [A]
Time [s]
sec
0
100
200
300
400
0
0.2
0.4
0.6
0.8
1
1.2
0
5
10
15
20
25
30
35

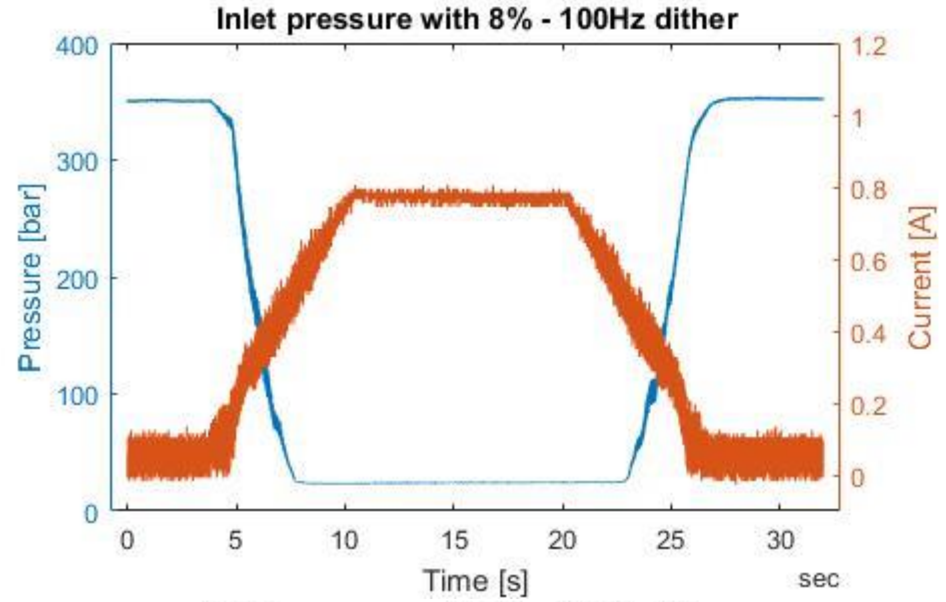


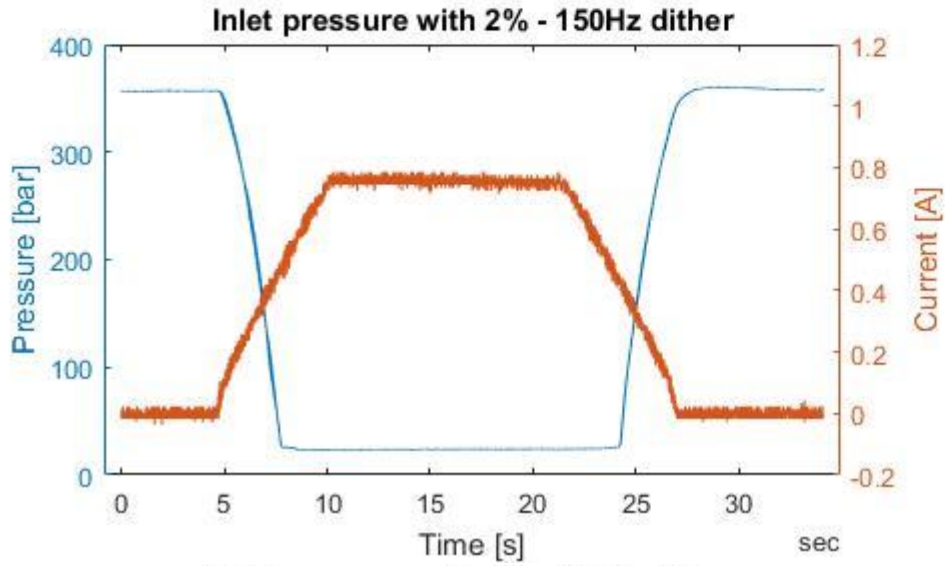
Inlet pressure with 10% - 100Hz dither
Pressure [bar]
Current [A]
Time [s]
sec
0
50
100
150
200
250
300
350
0
0.2
0.4
0.6
0.8
1
1.2
0
5
10
15
20
25
30

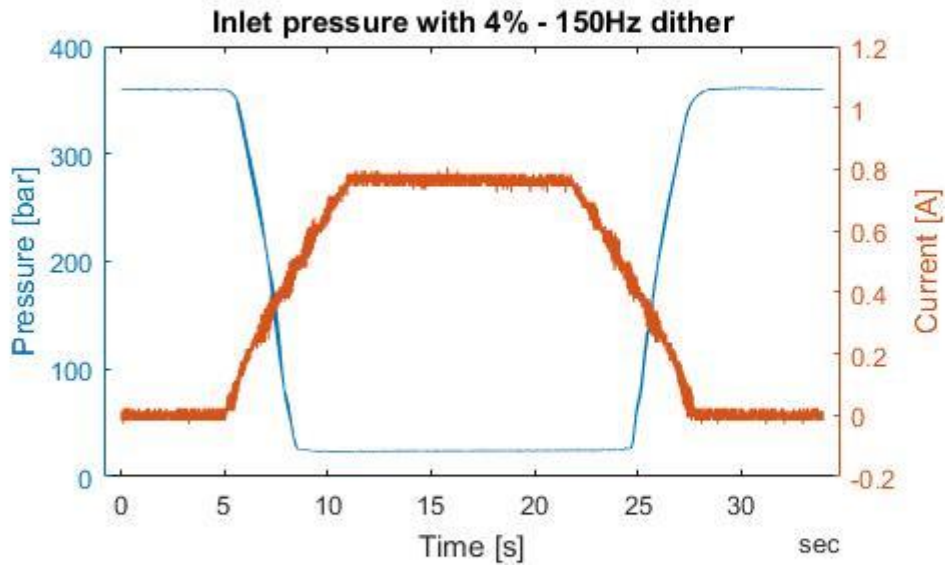

Inlet pressure with 2% - 100Hz dither
Pressure [bar]
Current [A]
Time [s]
sec
0
100
200
300
400
0
0.2
0.4
0.6
0.8
1
1.2
0
5
10
15
20
25
30
35

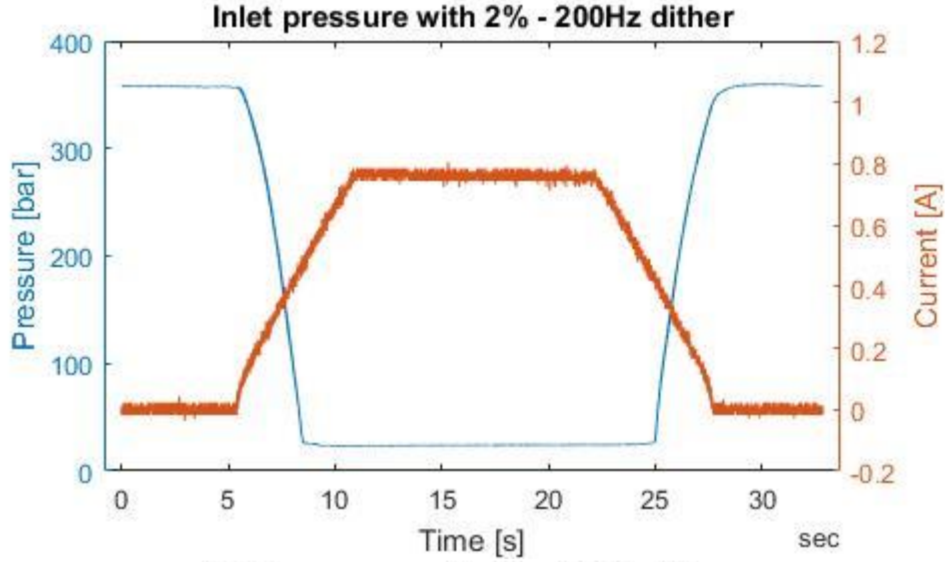

Inlet pressure with 4% - 100Hz dither
Pressure [bar]
Current [A]
Time [s]
sec
0
100
200
300
400
0
0.2
0.4
0.6
0.8
1
1.2
0
5
10
15
20
25
30
35

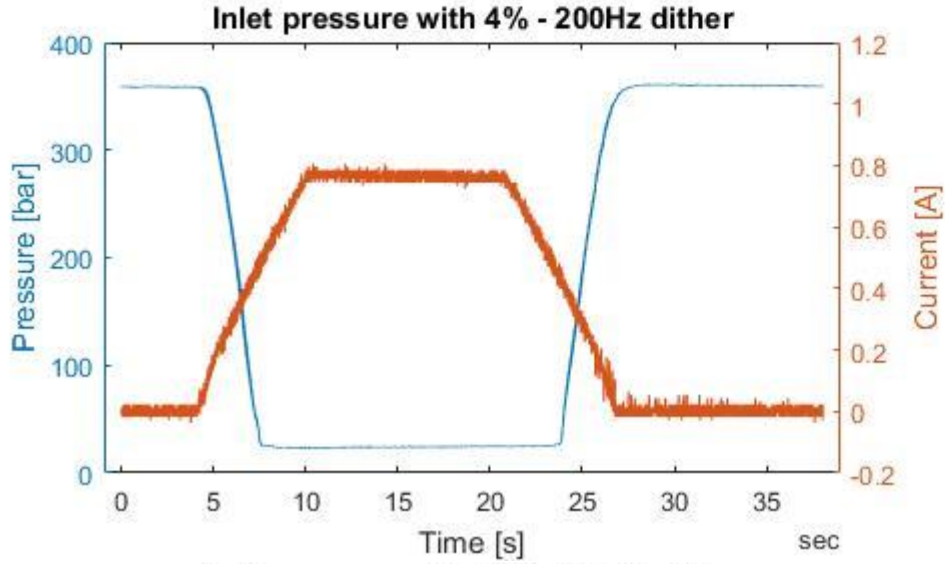

Inlet pressure with 6% - 100Hz dither
Pressure [bar]
Current [A]
Time [s]
sec
0
100
200
300
400
0
0.2
0.4
0.6
0.8
1
1.2
0
5
10
15
20
25
30

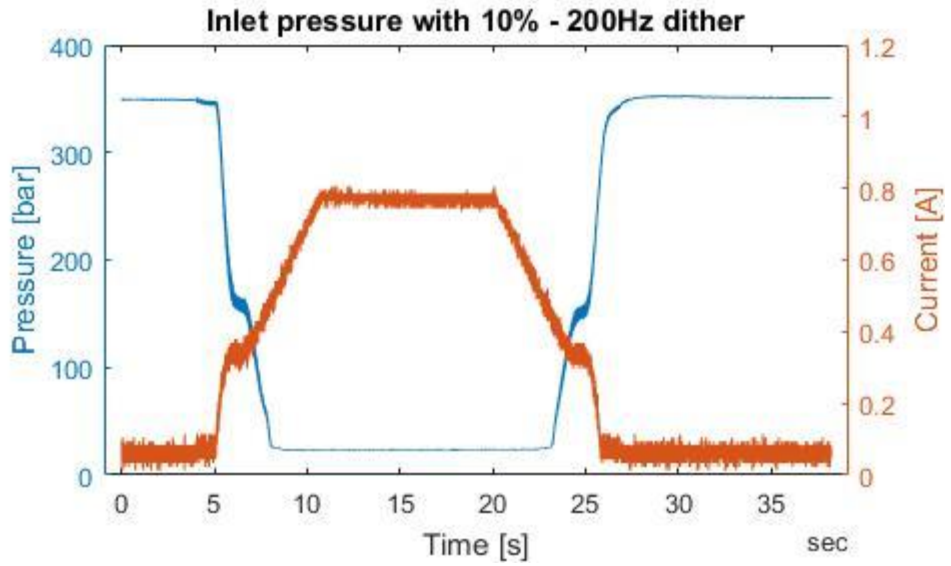

Inlet pressure with 8% - 100Hz dither
Pressure [bar]
Current [A]
Time [s]
sec

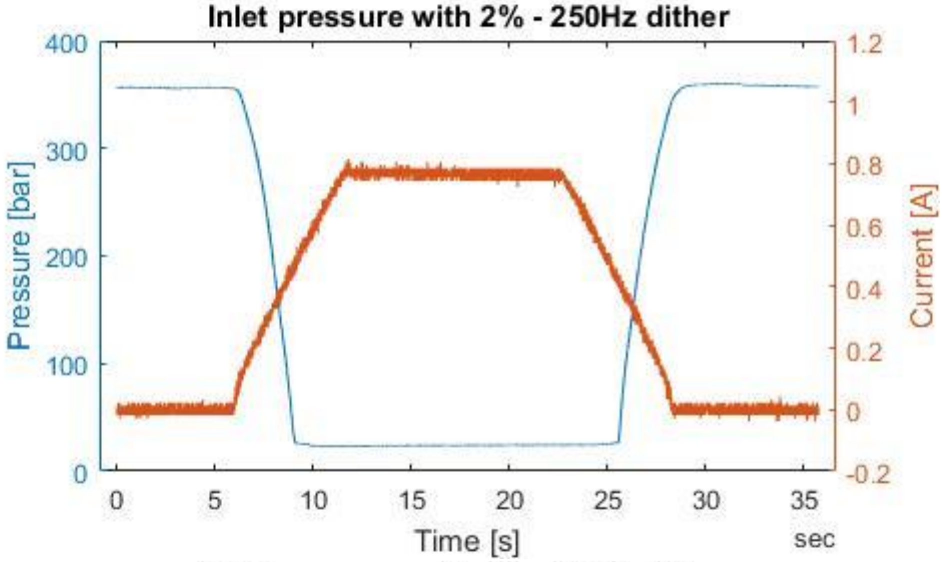

Inlet pressure with 2% - 150Hz dither
Pressure [bar]
Current [A]
Time [s]
sec

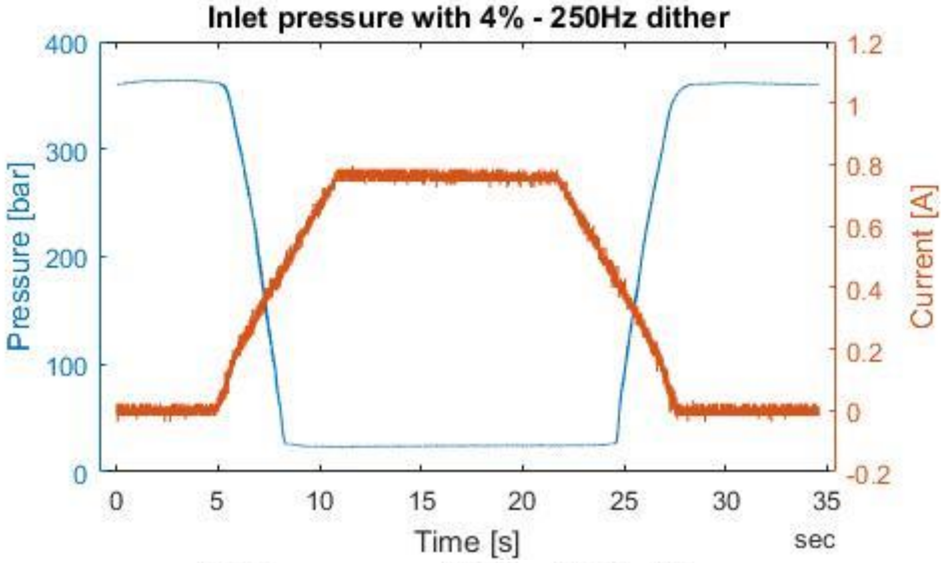

Inlet pressure with 4% - 150Hz dither
Pressure [bar]
Current [A]
Time [s]
sec

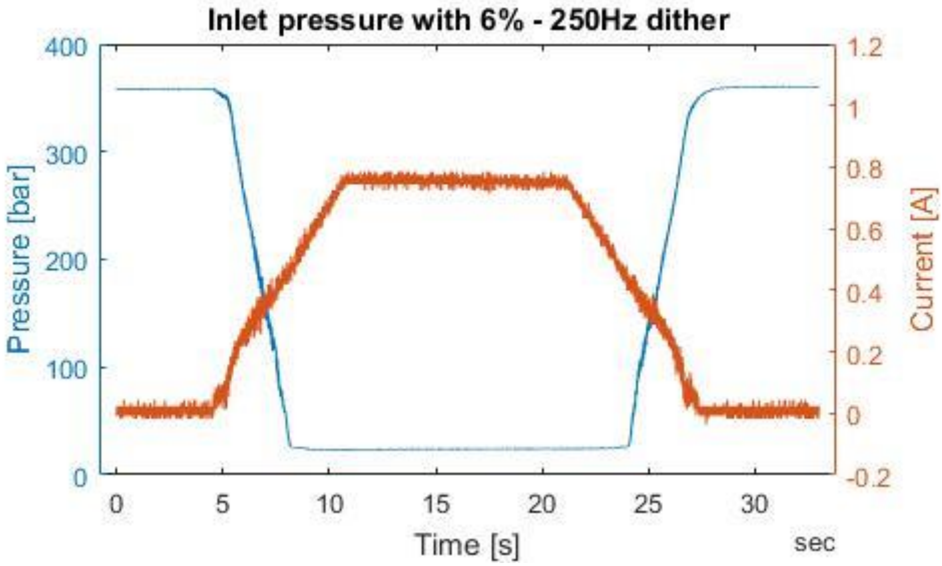

Inlet pressure with 2% - 200Hz dither
Pressure [bar]
Current [A]
Time [s]
sec

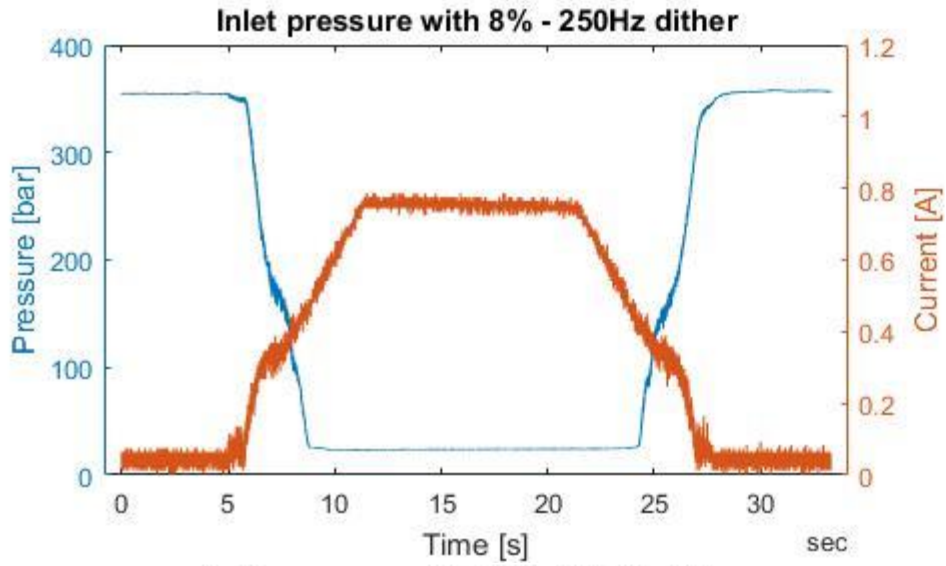

Inlet pressure with 4% - 200Hz dither
Pressure [bar]
Current [A]
Time [s]
sec

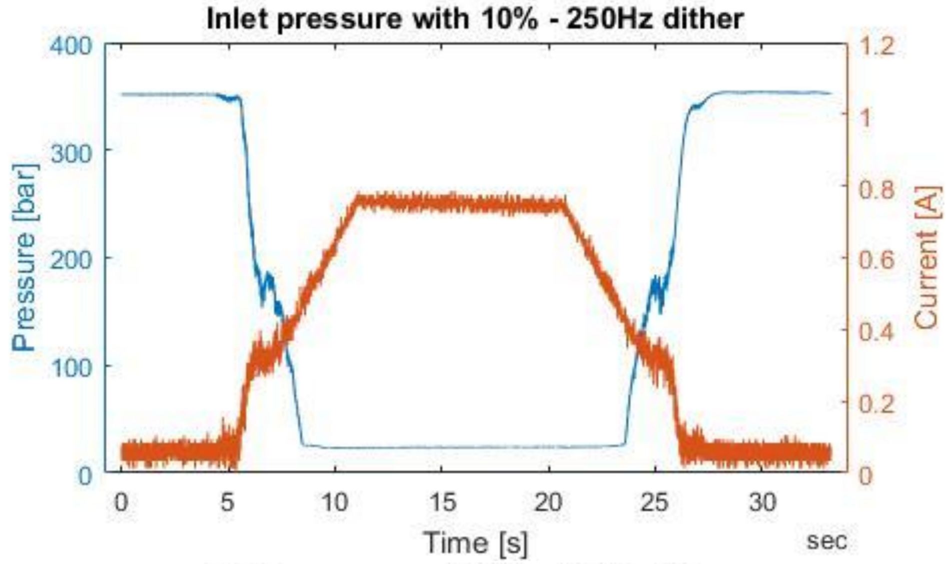

Inlet pressure with 10% - 200Hz dither
Pressure [bar]
Current [A]
Time [s]
sec

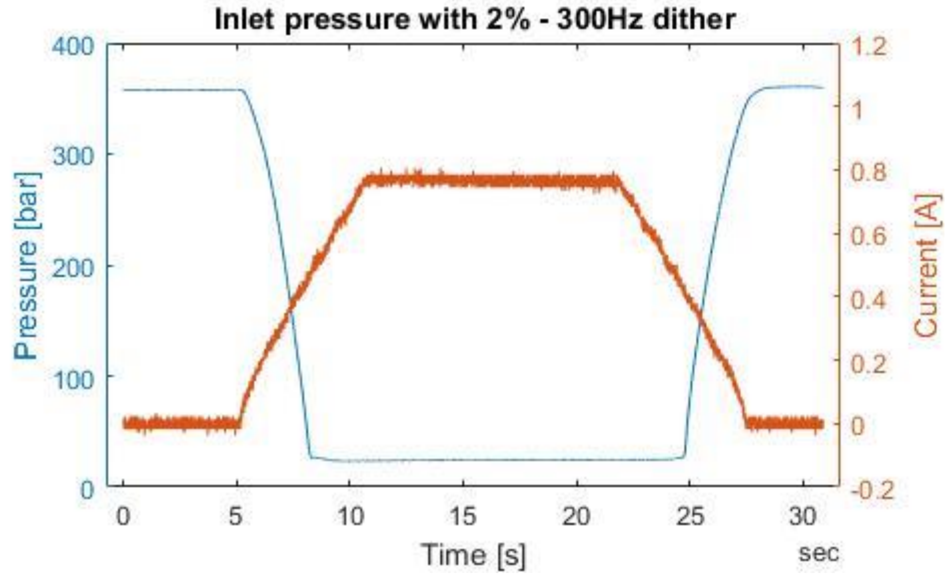

Inlet pressure with 2% - 250Hz dither
Pressure [bar]
Current [A]
Time [s]
sec
0
5
10
15
20
25
30
35
100
200
300
400
-0.2
0.2
0.4
0.6
0.8
1
1.2

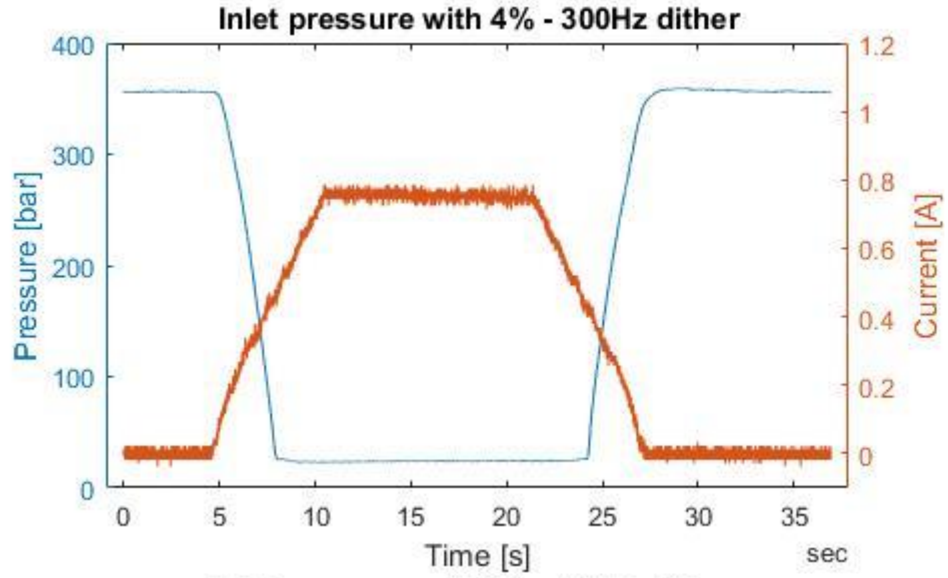

Inlet pressure with 4% - 250Hz dither
Pressure [bar]
Current [A]
Time [s]
sec
0
5
10
15
20
25
30
35
100
200
300
400
-0.2
0.2
0.4
0.6
0.8
1
1.2

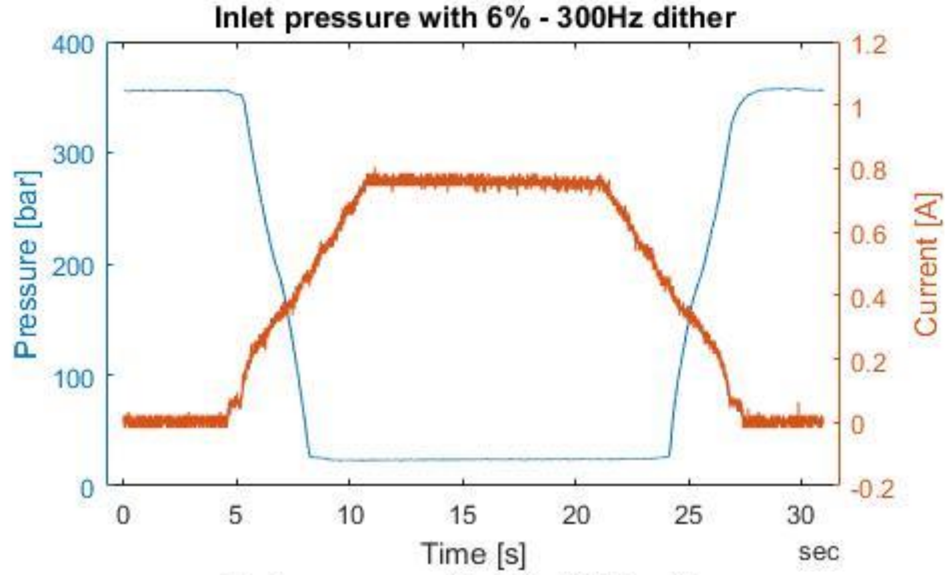

Inlet pressure with 6% - 250Hz dither
Pressure [bar]
Current [A]
Time [s]
sec
0
5
10
15
20
25
30
100
200
300
400
-0.2
0.2
0.4
0.6
0.8
1
1.2

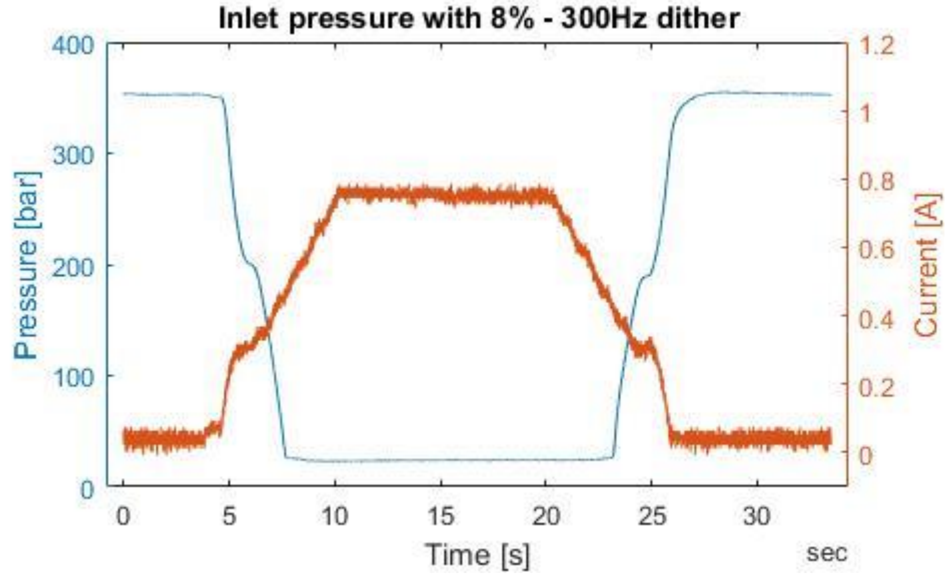

Inlet pressure with 8% - 250Hz dither
Pressure [bar]
Current [A]
Time [s]
sec
400
300
200
100
0
1.2
1
0.8
0.6
0.4
0.2
0
0
5
10
15
20
25
30

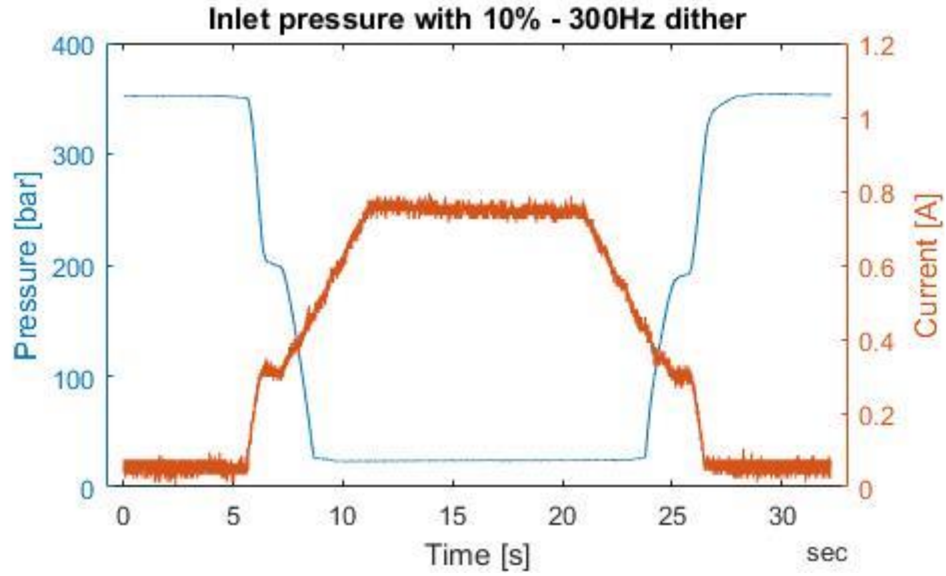

Inlet pressure with 10% - 250Hz dither
Pressure [bar]
Current [A]
Time [s]
sec
400
300
200
100
0
1.2
1
0.8
0.6
0.4
0.2
0
0
5
10
15
20
25
30


Inlet pressure with 2% - 300Hz dither
Pressure [bar]
Current [A]
Time [s]
sec
400
300
200
100
0
1.2
1
0.8
0.6
0.4
0.2
0
-0.2
0
5
10
15
20
25
30


Inlet pressure with 4% - 300Hz dither
Pressure [bar]
Current [A]
Time [s]
sec
0 5 10 15 20 25 30 35
0 100 200 300 400
0 0.2 0.4 0.6 0.8 1 1.2


Inlet pressure with 6% - 300Hz dither
Pressure [bar]
Current [A]
Time [s]
sec
0 5 10 15 20 25 30
0 100 200 300 400
-0.2 0 0.2 0.4 0.6 0.8 1 1.2


Inlet pressure with 8% - 300Hz dither
Pressure [bar]
Current [A]
Time [s]
sec
0 5 10 15 20 25 30
0 100 200 300 400
0 0.2 0.4 0.6 0.8 1 1.2


Inlet pressure with 10% - 300Hz dither
Pressure [bar]
Current [A]
Time [s]
sec
400
300
200
100
0
1.2
1
0.8
0.6
0.4
0.2
0
0
5
10
15
20
25
30