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Pressure regulator stability analysis in electropneumatic

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ABSTRACT

This paper aims at giving a analytical approach of the stability analysis of control pressure components in pneumatic domain. Some authors have given some results in linear case especially in the hydraulic context [Meritt 1960, Mac Cloy et al 1980, Margolis 1997, Alirand et al. 2002]. The structure and the technologies of commonly used pneumatic pressure control component are firstly introduced, and then a non-linear model is proposed as the basis for the stability analysis. The classical linear method, based on the tangent linearized model is applied to determine the analytical stability conditions using Routh-Hurwitz criteria. The root loci are finally studied according to a set of equilibrium points which corresponds to various conditions of use. In this last part, an existing pressure regulator is used as an example and allows the development of an analysis concerning the dynamic performances on realistic basis. To conclude, the proposed approach gives a set of design rules which enables the system parameters to be sized according to the required dynamic performances and stability.

KEYWORDS

Pressure control components, modeling, stability analysis, root locus, design.

1 INTRODUCTION

Pressure control components such as reducing valves, boosters and pressure regulator play an essential role in many pneumatic circuits. In spite of their simple structure, such components are often the source of oscillating behaviors that will be the cause of noise and disturbance of the normal operation. Even today the development and optimization of pressure components is a rather complex problem that requires a relatively large amount of experimental testing. The reasons are mainly related to the various flow conditions of the variable and fixed pneumatic orifices, the wide range of applications

that differ from each other in regard with the controlled volumes, the pressure levels, and the connected circuit (length of pipes, fittings). The compressibility and the potentially high velocity of the fluid cause the various non-linearities of the system resulting in many modeling difficulties such as jets, flow forces, thermal exchanges, displacement of sonic throttle and more. New design requires, in the first hand a good understanding of the influence of certain parameters on the dynamic behavior and the stability of the valve, and in the other hand a good knowledge of the effects of the geometry on the flow behavior.

When it is possible, analytical approaches are essential to reduce the design time and to avoid the direct developments using old designs and only validated by simulation. Non-linear and linear analyses are relevant in many cases as a first level in the design stages in order to reach the desired specifications of the component. This paper proposes analytical results concerning the static and dynamic behaviors of pneumatic poppet valves [] which may be easily extended to a wide range of valves, including electro-pneumatic valves.

2 PRESSURE COMPONENT STRUCTURE

The studied pneumatic poppet valve is a pneumatic pressure regulator commonly used in truck brake systems. The structure presented figure 1 is similar to many other valve designs used in automotive or railway applications or in pneumatic circuits. The function of these valves is to control the upstream pressure of a circuit or the pressure in a volume in spite of mass flow rate consumption. Poppet valves are usually preferred to spool valves when leakages have to be avoided but to the detriment of pressure accuracy around zero opening. The outlet pressure is obtained with the help of a pilot part which can be actuated pneumatically, mechanically or electrically. In spite of the existing technology differences for the pilot parts [Faisan], the power stages of these poppet valves present many structural similarities as shown in figure 2.

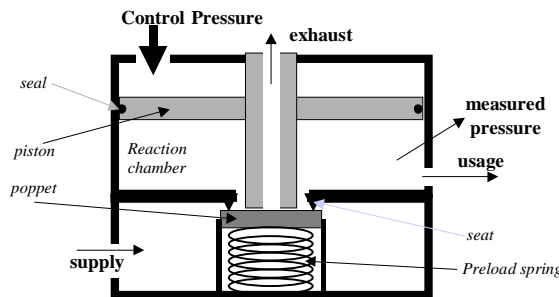


Figure 1: Pressure regulator

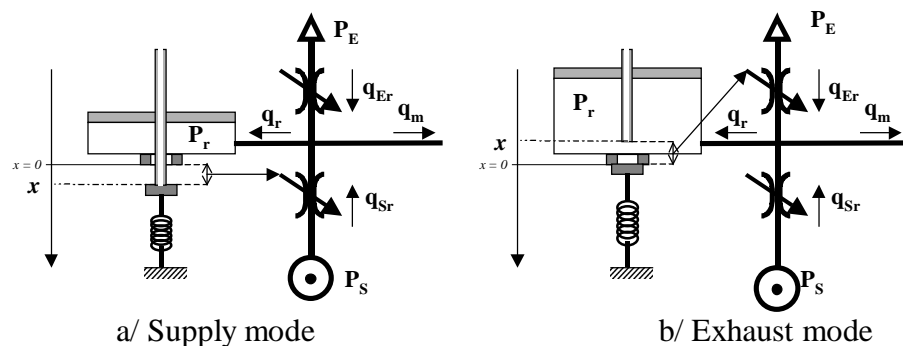


Figure 2: Structure of pressure control components

The flow performances of the pressure control components are related to the characteristic of each orifice constituting the pneumatic Wheastone half bridge and

therefore they have to be accurately characterized for accurate simulation purposes. The supply and exhaust orifices are connected to a volume, called reaction chamber, that distributes the flow to the downstream circuits. At this step of the design, the choice of the outlet connections or fittings may influence the behavior of the component. If they are undersized, the flow is no more limited by the control orifices and a strong coupling with the control orifices may introduce a displacement of the sonic throttle. In this case, the stability of the valve is also affected. As it could be limiting to base this work on a priori standard load circuit, it is assumed here that there isn't any coupling in between the pressure control component and the load circuit.

The pressure in the reaction chamber is used as the feedback in the control loop. In the case of electro-pneumatic components, a pressure transducer may be used and the closed loop may integrate an electronic controller. As the form of the controller can modify the global behavior of the valve, such components are not studied in this work.

The displacement of the mobile part is driven by a pilot force controlled pneumatically, mechanically or electrically. It is generally constituted of a piston, a poppet and preload springs. The role of the piston is, in one hand to open the supply orifice by pushing the poppet and in the other hand to release the poppet on its seat, closing the supply orifice, and to open the exhaust orifice. This last orifice is placed either inside the piston, either through the poppet. The mobile part presents generally a short range of displacement and introduces mainly two problems at the design stage. Firstly, for energy consumption reasons, no leakage is allowed in between the reaction chamber and the pilot part. Membranes may be used but the displacements are shorter than with pistons. The friction effect (dry and viscous friction) introduced by the solution consisting in piston and seal presents advantages in regard with the stability but to the detriment of pressure accuracy. Secondly, the contact phenomenon in between the piston and the poppet is analytically difficult to handle because it introduces a change of model structure.

The energy exchanges and the structure of this system may be represented by the 2 bond graphs of figure 3; each representation is corresponding to a functioning mode (supply or exhaust) and the model switches according to the piston and poppet relative position. The causal loops displayed on the figures show clearly the coupling in between the different energy storages (dot lines) but also their connection to dissipations (plain lines) which will have a damping effect.

Figure 3: Structure in supply mode (a) / structure in exhaust mode

3 MODELING ELEMENTS

Because of the change of structure and the discontinuities it introduces, the proposed analysis will consider separately the two cases without trying to solve the problem when the change occurs. The model is here slightly simplified in order to obtain a non-linear model usable for design and analysis purposes. Air is considered as a perfect gas and a polytropic law will express the thermal exchanges. The mass flow rate characteristic of each orifice is supposed to be known using the ISO6358 standard; that is to say for any opening position of the orifices, the sonic conductance and the critical pressure ratio are given. This approach guaranties realistic results concerning the power stage. The used friction model based on Karnopp [] assumptions considers viscous, stiction and striction terms.

In supply conditions ($x > 0$), the piston and the poppet are supposed in contact, constituting a single inertia and the exhaust orifice stays closed.

Mechanical equation

$$\begin{cases} \frac{dv}{dt} = \frac{1}{(m_p + m_{pop})} (F_d - A_r P_r - K(x + x_p) - F_f - bv) \\ \frac{dx}{dt} = v \end{cases} \quad (1)$$

Pressure in the reaction chamber

$$\frac{dP_r}{dt} = \frac{krT_s}{V_r(x)} \left(q_{Sr}(x, P_r) - q_m - \frac{P_r}{rT_s} \frac{dV_r}{dt} \right) \text{ with } \frac{dV_r}{dt} = -A_r v \text{ and } V_r(x=0) = V_{r0} \quad (2)$$

In exhaust conditions ($x < 0$), the supply orifice is closed and the poppet stays on its seat. Only the piston displacement is considered.

Mechanical equation

$$\begin{cases} \frac{dv}{dt} = \frac{1}{m_p} (F_d - A_r P_r - F_f - bv) \\ \frac{dx}{dt} = v \end{cases} \quad (3)$$

Pressure in the reaction chamber

$$\frac{dP_r}{dt} = \frac{krT_s}{V_r(x)} \left(q_{Er}(x, P_r) - q_m - \frac{P_r}{rT_s} \frac{dV_r}{dt} \right) \text{ with } \frac{dV_r}{dt} = -A_r v \text{ and } V_r(x=0) = V_{r0} \quad (4)$$

Orifice model based on ISO6358 Standard

For each opening position:

$$\text{the flow is sonic, if } \frac{P_{down}}{P_{up}} \leq b(x) \text{ and } q(x, P_{up}, T_{up}) = C(x) \sqrt{T_{up}/T_0} P_{up} \quad (5)$$

the flow is subsonic,

$$\text{if } \frac{P_{down}}{P_{up}} > b(x) \text{ and } q(x, P_{up}, P_{down}, T_{up}) = C(x) \sqrt{T_{up}/T_0} P_{up} \left(\sqrt{1 - \frac{\left(\frac{P_{down}}{P_{up}} - b(x) \right)^2}{(1 - b(x))^2}} \right) \quad (6)$$

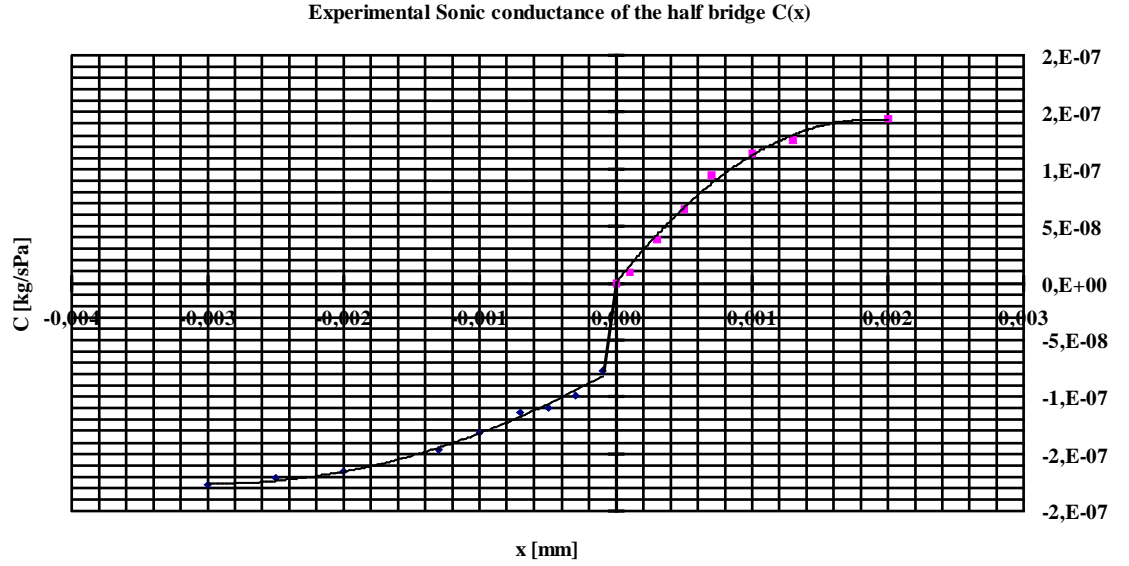


Figure 4: Sonic conductance of the studied valve

Figure 4 shows the conductance sonic obtained for the valve used as an example in this work. In order to present exploitable analytical results, it is assumed that the flows are in sonic conditions and that the temperature in the reaction chamber is equal to the external temperature T_S . In one hand, the sonic condition hypothesis is usually respected in supply conditions as the supply pressure is generally very high compared to the controlled pressure in order to reach correct performances. In the other hand, the study of the stability in exhaust conditions is only interesting if the flow is significant, this means that the control pressure has to be higher than the exhaust pressure. Therefore the pressure in the reaction chamber is supposed to respect the following conditions (7) and only equation (8) is used in the analytical developments.

$$\frac{P_E}{P_r} \leq b(x) \text{ and } \frac{P_r}{P_S} \leq b(x) \quad (7)$$

$$q_{Er}(x, P_r) = C(x)P_r \text{ and } q_{Sr}(x) = C(x)P_S \quad (8)$$

4 STEADY STATE CONDITIONS AND LINEARISATION

On the basis of the proposed model and hypotheses, the equilibrium set is determined for given conditions of desired pressure, expressed here as an equivalent force on the piston F_d^e , and a fixed mass flow rate at outlet q_m^e . In the supply mode (10), the position of the poppet is directly obtained from the knowledge of the sonic conductance whereas it is necessary in the exhaust mode (9) to first determine the pressure in the chamber from the mechanical equation before solving the flow balance equation. It shows also that the stiction force has an important effect on the reached steady state, but its action may be limited by increasing the section of the piston A_r . In supply mode, the precision of the pressure control is also increased when the poppet spring force is low compared to the control force. As precision is required in both modes, the conclusion is that the stiffness of the poppet spring must be sized only to avoid leakages.

As an example the static characteristics of the studied valve are shown figure 6 according to the desired pressure at outlet. The influence of the spring stiffness and dry friction appears clearly at low control pressures.

Exhaust mode

$$\begin{cases} v^e = 0 \\ \left(F_d^e - |F_{fst}| \right) / A_r \leq P_r^e \leq \left(F_d^e + |F_{fst}| \right) / A_r \\ C(x^e) = q_m^e / P_r^e \end{cases} \quad (9)$$

Supply mode

$$\begin{cases} v^e = 0 \\ C(x^e) = q_m^e / P_s \\ \left(F_d^e - K(x^e + x_p) - |F_{fst}| \right) / A_r \leq P_r^e \leq \left(F_d^e - K(x^e + x_p) + |F_{fst}| \right) / A_r \end{cases} \quad (10)$$

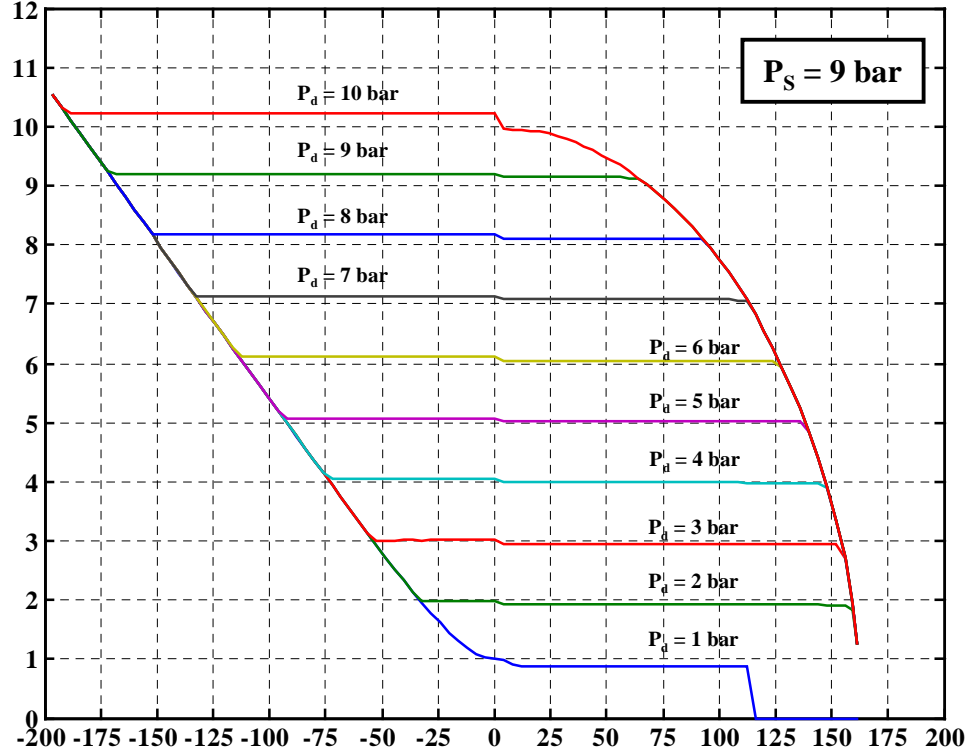


Figure 6: Static characteristics of studied valve

Around the previous steady state points, the linearised model is given by equations 12 (exhaust mode) and 13 (supply mode). It shows clearly that the dynamic behavior is slightly different according to the usage conditions.

Assuming that:

$$\begin{cases} dP_r = P_r - P_r^e \\ dv = v - v^e \\ dx = x - x^e \end{cases} \quad \begin{cases} dF_d = F_d - F_d^e \\ dq_m = q_m - q_m^e \end{cases} \quad (11)$$

Exhaust mode

$$\frac{d}{dt} \begin{bmatrix} dP_r \\ dv \\ dx \end{bmatrix} = \begin{bmatrix} \frac{krT_S C(x^e)}{(V_{r0} - A_r x^e)} & \frac{kP_r^e A_r}{(V_{r0} - A_r x^e)} & \frac{krT_S P_r^e}{(V_{r0} - A_r x^e)} & \frac{dC(x^e)}{dx} \\ -\frac{A_r}{m_p} & -\frac{b}{m_p} & 0 & 0 \\ 0 & 1 & 0 & 0 \end{bmatrix} \begin{bmatrix} dP_r \\ dv \\ dx \end{bmatrix} + \begin{bmatrix} 0 \\ \frac{1}{m_p} \\ 0 \end{bmatrix} \begin{bmatrix} dF_d \\ dq_m \end{bmatrix} \quad (12)$$

Supply mode

$$\frac{d}{dt} \begin{bmatrix} dP_r \\ dv \\ dx \end{bmatrix} = \begin{bmatrix} 0 & \frac{kP_r^e A_r}{(V_{r0} - A_r x^e)} & \frac{krT_s P_s}{(V_{r0} - A_r x^e)} \frac{dC(x^e)}{dx} \\ -A_r & -b & -K \\ m_p + m_{POP} & m_p + m_{POP} & m_p + m_{POP} \\ 0 & 1 & 0 \end{bmatrix} \begin{bmatrix} dP_r \\ dv \\ dx \end{bmatrix} + \begin{bmatrix} 0 & -krT_s \\ 1 & 0 \\ m_p + m_{POP} & 0 \\ 0 & 0 \end{bmatrix} \begin{bmatrix} dF_d \\ dq_m \end{bmatrix} \quad (13)$$

5 ANALYTICAL ANALYSIS OF THE STABILITY

Although the linear analysis is, in this context, limited due to the highly non-linear system, it provides analytical results which are crucial at the design stage. The study of the system stability is based here on the Routh-Hurwitz criterion []. It provides a set of conditions on the characteristic polynomial coefficients, which insure that the real parts of the polynomial roots are negative. These conditions, obtained without computing the roots of the characteristic polynomial, may be expressed analytically and design criteria can be deduced.

From the linear models 12 and 13, the form of the characteristic polynomials is given by equation 14. As the coefficient of s^3 is positive (and equal to 1), the Routh-Hurwitz criterion imposes the same conditions (15).

$$s^3 + a_1 s^2 + a_2 s + a_3 = 0 \quad (14)$$

$$\begin{cases} a_1 > 0 \\ a_3 > 0 \\ a_1 a_2 - a_3 > 0 \end{cases} \quad (15)$$

5.1 Supply mode

In supply mode the coefficients of the characteristic polynomial are given by equation 16. The three first conditions are always verified and therefore the stability condition is deduced from equation 17.

$$\begin{cases} a_1 = \frac{b}{m_p + m_{pop}} \\ a_2 = \frac{1}{m_p + m_{pop}} \left(K + \frac{kP_r^e A_r^2}{(V_{r0} - A_r x^e)} \right) \\ a_3 = \frac{A_r}{m_p + m_{pop}} \frac{krT_s P_s}{(V_{r0} - A_r x^e)} \frac{dC(x^e)}{dx} \end{cases} \quad (16)$$

$$a_1 a_2 - a_3 > 0 \Leftrightarrow \frac{b}{m_p + m_{pop}} \left(K + \frac{kP_r^e A_r^2}{(V_{r0} - A_r x^e)} \right) - \frac{krT_s A_r P_s}{(V_{r0} - A_r x^e)} \frac{dC(x^e)}{dx} > 0 \quad (17)$$

Pneumatic Stiffness

As the spring stiffness is generally negligible compared to pneumatic stiffness (§ 4), the condition becomes:

$$\frac{b}{m_p + m_{pop}} > \frac{rT_s P_s}{A_r P_r^e} \frac{dC(x^e)}{dx} \quad (18)$$

The relation (18) shows that a larger section of the piston A_r and evidently lower masses ($m_p + m_{pop}$) **will improve the stability**. But for a given design, increasing the pressure in the reaction chamber P_r^e (that is consequently the desired pressure) has a positive effect on the stability. Finally, it appears that the flow performances of the power stage are antagonist to a good stability, as high supply pressures P_s but also steep evolutions of the sonic conductance according to the poppet position $dC(x^e)/dx$ are unfavorable.

5.2 Exhaust mode

When the component is in exhaust mode, the coefficients of the characteristic polynomial are now given by equations (19). From modeling hypotheses the sonic conductance $C(x^e)$ is negative when $x^e < 0$ (exhaust mode), therefore the coefficients a_1, a_2, a_3 are positive, and the Routh-Hurwitz criterion imposes the condition (20).

$$\begin{cases} a_1 = \frac{b}{m_p} - \frac{krT_s}{(V_{r0} - A_r x^e)} C(x^e) \\ a_2 = \frac{1}{m_p} \left(\frac{kP_r^e A_r^2}{(V_{r0} - A_r x^e)} - \frac{krT_s}{(V_{r0} - A_r x^e)} b C(x^e) \right) \\ a_3 = \frac{A_r}{m_p} \frac{krT_s P_r^e}{(V_{r0} - A_r x^e)} \frac{dC(x^e)}{dx} \end{cases} \quad (19)$$

$$a_1 a_2 - a_3 > 0 \Leftrightarrow$$

$$\left[\frac{b}{m_p} - \frac{krT_s C(x^e)}{(V_{r0} - A_r x^e)} \right] \left[kP_r^e A_r^2 - krT_s b C(x^e) \right] - krT_s P_r^e A_r \frac{dC(x^e)}{dx} > 0 \quad (20)$$

This relation may be expressed as a 2nd degree polynomial (21) in $C(x^e)$ and a new set of conditions (23) have to be verified corresponding to the sign of the polynomial form (22).

$$\frac{krT_s b}{(V_{r0} - A_r x^e)} C(x^e)^2 - \left[\frac{b^2}{m_p} + \frac{kP_r^e A_r^2}{(V_{r0} - A_r x^e)} \right] C(x^e) + P_r^e A_r \left[\frac{A_r}{rT_s} \frac{b}{m_p} - \frac{dC(x^e)}{dx} \right] > 0 \quad (21)$$

$$x^2 + bx + g > 0 \text{ if } d^2 = b^2 - 4g < 0 \text{ or } \begin{cases} d^2 = b^2 - 4g \geq 0 \\ x < \frac{1}{2}(-b - d) \text{ or } x > \frac{1}{2}(-b + d) \end{cases} \quad (22)$$

As $b < 0$ and because in exhaust mode $C(x^e) < 0$, the stability conditions becomes :

$$d^2 = b^2 - 4g < 0 \text{ or } \begin{cases} d^2 = b^2 - 4g \geq 0 \\ C(x^e) < -\frac{1}{2}(b + d) \end{cases} \quad (23)$$

$$\text{with } \begin{cases} b = -\frac{(V_{r0} - A_r x^e)}{krT_s b} \left[\frac{b^2}{m_p} + \frac{kP_r^e A_r^2}{(V_{r0} - A_r x^e)} \right] \\ g = \frac{(V_{r0} - A_r x^e) P_r^e A_r}{krT_s b} \left[\frac{A_r}{rT_s} \frac{b}{m_p} - \frac{dC(x^e)}{dx} \right] \end{cases}$$

These final relations are still difficult to handle but it can be deduced that decreasing the sonic conductance evolution according to the orifice opening $dC(x^e)/dx \approx 0$ (true for large opening) is favorable to the stability conditions. This is corresponding to the stability conditions obtained in the previous case but now the controlled pressure has to be low.

In order to obtain an analytical approach in the unfavorable cases, that is when the viscous friction is low, it may be remarked that the conditions (22) have to be verified for any x^e , that is even if $dC(x^e)/dx$ is larger than $\frac{A_r}{rT_s} \frac{b}{m_p}$ (at low opening). A last reduction is to assume that the pneumatic stiffness is very large compared to b^2/m_p (generally verified in any cases).

These assumptions lead to the following expressions:

$$\begin{cases} b = -\frac{P_r^e A_r^2}{brT_s} \\ g = -\frac{(V_{r0} - A_r x^e) P_r^e A_r}{bkrT_s} \frac{dC(x^e)}{dx} \end{cases}, \text{ that implies } d^2 > 0 \quad (23)$$

Therefore, the final stability condition can be reduced to (24):

$$C(x^e) < \frac{1}{2} \frac{P_r^e A_r^2}{brT_s} \left[\sqrt{1 + \frac{4brT_s}{A_r} \frac{dC(x^e)}{dx}} + 1 \right] \approx \frac{(V_{r0} - A_r x^e) dC(x^e)}{A_r k dx} + \frac{P_r^e A_r^2}{brT_s} \quad (24)$$

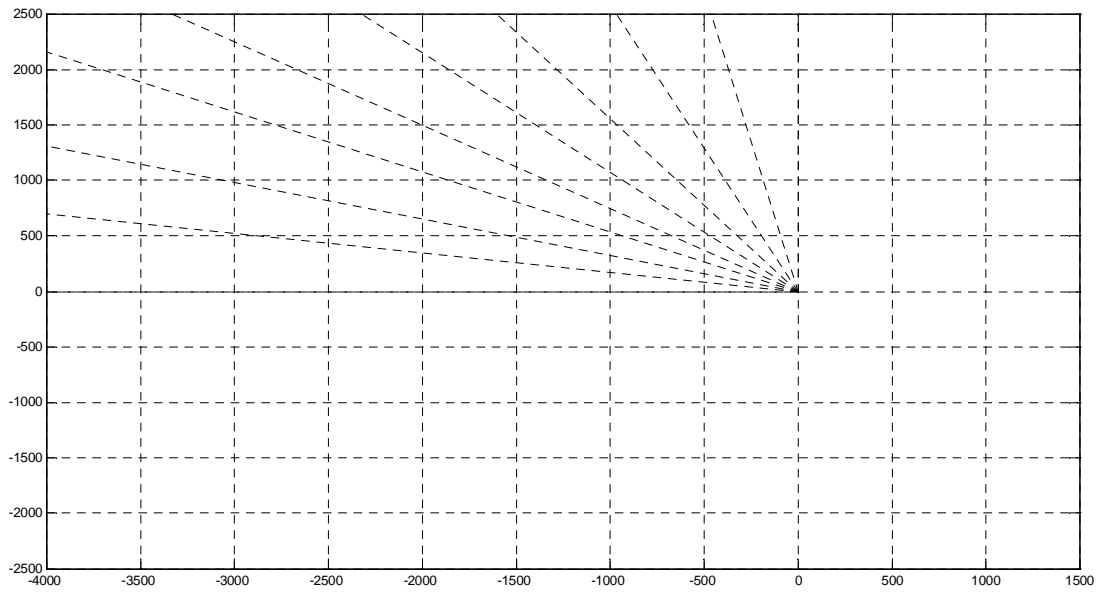
In exhaust mode the stability is then mainly related to the evolution of the sonic conduction according to the orifice opening. In the contrary of the supply mode, the stability is now facilitated when the desired pressure is low. And similarly, steep evolutions of the sonic conductance according to the poppet position $dC(x^e)/dx$ are unfavorable. The relation (24) can be useful at the design stage of the exhaust orifice and it could be interesting to extend the work in this direction.

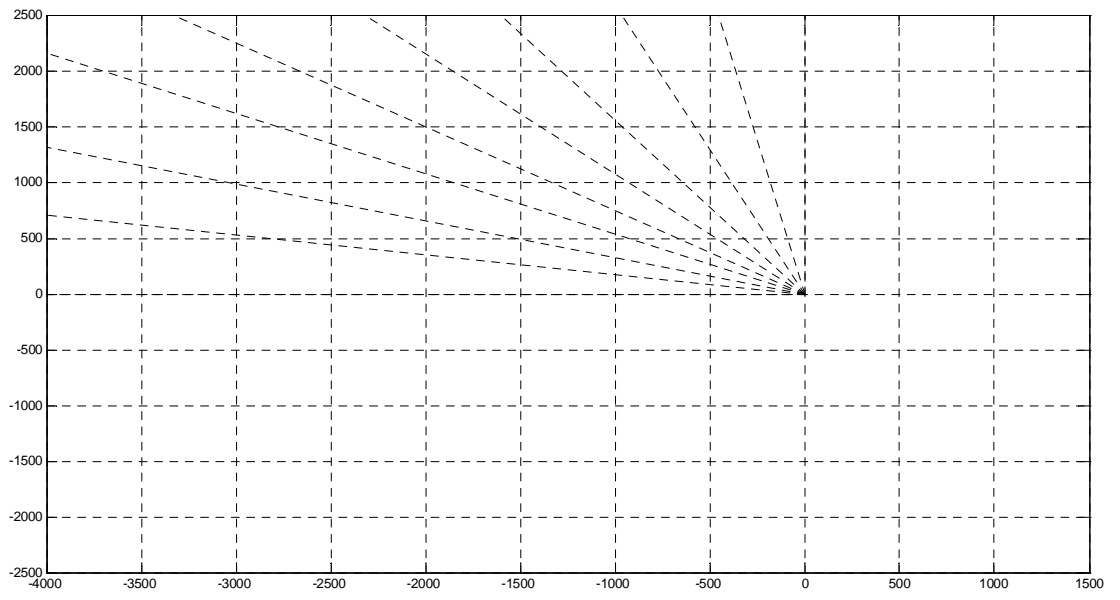
6 ROOT LOCUS

The previous part showed that an analytical approach is possible to determine the stability conditions and give an interesting help at the design stage. But the stability study is more complete if the eigenvalues of the system may be identified. Even in the linear approach, it is difficult to obtain a right analytical approach concerning the roots of the characteristic polynomial. Only a numerical approach is here proposed with two main objectives. The first one is to confirmed the previous study on the stability of the pressure component according to the functioning mode. The second goal is to give a idea of dynamic performances in terms of frequency associated to each mode and its damping ratio.

The studied pressure regulator is used here as an example and the root loci explore the viscous friction required to reach dynamic performances according to various values of desired pressures and outlet mass flow rates.

Figure 8 shows the root loci for a set of desired pressures values in supply mode (8a) and exhaust mode (8b). Figure 9 shows the root loci for a set of outlet mass flow rate values in supply mode (9a) and exhaust mode (9b).





half-bridge. Nevertheless, design precautions have been determined in the exhaust mode of the valve. Thus to avoid unstable behaviors, viscous friction has to be increased drastically. It can be noticed that flow forces and turbulences may have substantial damping effects if they are rightly placed and may be used to limit the required viscous friction. Therefore, it seems interesting to look at fluid dynamic approaches because they may provide pertinent information concerning the flow profiles in valves and naturally concerning fluid-born damping effects.

From the analytical approach of the conditions, it is not surprising to observe similar behaviors according to the values of the controlled pressures on the root loci. It appears also that the time responses are corresponding to the response of a 2nd order system in all cases. In spite of a large bandwidth, the low damping of the corresponding modes have a negative effect on the response in the time domain. The viscous friction can be adjusted in order to improve the damping of the modes but to the detriment of the dynamic. It can be concluded that a right compromise has to be done concerning the viscous friction.

Finally, it is important to notice that the load circuit of the pressure control components may play a crucial role in damping or feeding the natural oscillating behavior of the system. It is obvious that pipes could introduce a disastrous coupling with the system, but also that the direct connection with a volume through a low section fitting will naturally damped the whole behavior. An interesting solution consists in adding a second chamber, called utilization chamber, in between the reaction chamber and the wheastone half-bridge. Using a calibrated orifice, this solution has the same damping effect as a volume at outlet.

Future work focuses on the modeling of flow forces and reduction of the non-linear model. The first objective is related to the use of local approach like Computational Fluid dynamics to derive macroscopic models taking into account flow forces, effects of turbulence. Secondly, as it has been shown in this work, the response of the system in the time domain is relatively simple and the reduction of the system to a global dynamic completed by a set of non-linear static characteristic will improve the time delay required for the design of pneumatic circuit such as brake circuit, and gas supply networks.

NOMENCLATURE

m_p	piston mass (kg)
m_{pop}	poppet mass (kg)
A_r	Piston area (m ²)
P	Pressure (N)
K	stiffness spring (N/m)
x	piston position (m)
x_p	preload spring position (m)
F	Force (N)
b	viscous friction coefficient (N/(m/s))
v	piston velocity (m/s)
k	polytropic constant
r	perfect gas constant related to unit mass (J/kg/K)
T	temperature (K)
q_m	mass flow rate (kg/s)
V	Volume (m ³)
$C(x)$	Sonic conductance (kg/s/Pa)
$b(x)$	critical pressure ratio
t	time (s)
V	volume (m ³)
δ	variation around equilibrium set

Subscript

f	dry friction
st	stiction friction
r	reaction chamber
E	exhaust
S	supply
d	desired

Superscript

e	equilibrium
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