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Compressor Surge Control Using a Variable Area Throttle and Fuzzy Logic Control

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Abstract

This paper presents a new fuzzy control approach to compressor surge control. A variable area throttle is used in this strategy. The controller uses measurements of the mass flow rate and the change of mass flow rate to determine the control action. By using the change of mass flow rate, the controller does not need to know where the operating point is. One advantage of the proposed controller is that it does not need a mathematical model of the controlled compressor and it works on any centrifugal compressors. The fuzzy rules are transferable and only the scaling factor and the throttle capacity need to be changed when the controller is applied to another compressor of the same type. Simulation results show that the proposed controller gives good performance in various situations.

Keywords: Centrifugal compressors, surge, fuzzy control, active surge control.

1. Introduction

Centrifugal compressors were first invented in the 19th century by a French professor named Auguste Rateau. These compressors are relatively dependable, trouble-free, come in different sizes and almost any gas could be compressed by them (Bloch, 1995). However, the performance of centrifugal compressors is generally limited by an instability known as surge. Surge is a phenomenon that occurs at low compressor mass flow rates causing the compression system to operate on the positively sloped region of the compressor characteristic (Fink et al., 1992, Greitzer, 1981). It is characterised by periodic pressure and mass flow rate oscillations throughout the compression system. The mass flow rate fluctuates in magnitude and sometimes it even reverses its direction (Lüdtke, 2004) which may lead to a lot of process

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disruptions. This instability limits the range of operation of the compressor and jeopardises its stability.

Over the years, many measures have been introduced to overcome the problem of surge in compressors. Traditionally, the problem has been tackled by using surge avoidance techniques (Staroselsky and Ladin, 1979). However, this well established method limits the operational range of the compressor and reduces its efficiency (Willems and de Jager, 1999, Gravidahl, 1998). As a result, active surge control was introduced as an alternative approach to deal directly with the surge instability rather than avoiding it.

Active surge control has been a subject of active research for around two decades. The method was first introduced by Epstein *et al.* (1989). The approach has the advantage of allowing stable compressor operation in previously unstable, high-performance areas and also has the advantage of relatively low-energy consumption, since it operates on small amplitude disturbances when it tries to stabilise surge in its earliest stages (Epstein *et al.*, 1989). Many actuators for active surge control have been reported in the literature and many examples are presented in Willems and de Jager (1999). In many of the cases, proportional feedback is used as the control law and has been shown to be successful (Pinsley *et al.*, 1991, Simon *et al.*, 1993).

Although linear approaches have shown to be successful in actively suppressing the instability caused by surge, nonlinear approaches seem to be more promising since the problem itself is a nonlinear one. Badmus *et al.* (1996) used a nonlinear approach in controlling an axial compressor based on input-output feedback linearising control. The method gave greater performance in comparison to a linear controller (Badmus *et al.*, 1996). Adaptive control has also been reported in the literature by Blanchini and Giannattasio (2002). The problem with nonlinear techniques, however, is that they involve a lot of tedious mathematical manipulation and do not offer a simple solution.

In this study, a new nonlinear approach using fuzzy logic control is presented for compressor surge control. The advantages of this method lie in its simple design procedure and its good performance. Although fuzzy control has been successfully used in many other applications, its application to active surge control has not been

reported in the literature to the authors' knowledge. In this paper, the authors propose the use of a variable area throttle to stabilise the surge instability. This is usually already present in conventional anti-surge schemes. The use of a variable area throttle for active surge control is also reported in the literature by Willems and de Jager (1998) and Pinsley et al. (1991). Willems and de Jager (1998) used the mass flow rate signal to stabilise surge at a point to the left of the surge line. On the other hand, Pinsley et al. (1991) used the pressure signal as a feedback signal. However, Simon et al. (1993) showed that the maximum range of stabilisation that could be obtained with these combinations is not big compared to using a combination of Closed Coupled Valve (CCV) and the mass flow rate signal. The use of CCV, however, achieves this stabilisation at a cost of reducing the pressure output of the machine as demonstrated in Gravdahl (1998). In addition, Gravdahl (1998) developed a very complicated nonlinear controller for active surge control. This paper presents a simple fuzzy logic control strategy that depends on intuition in its rules and gives very satisfactory results at the end. These results are compared to those obtained by Gravdahl (1998) in terms of performance.

The paper is organised as follows. Section 2 describes compressor surge and its causes. Section 3 outlines the proposed methodology for active surge control. A compressor model used for simulation studies in this paper is given in Section 4. Section 5 outlines the fuzzy controller developed in this paper. Simulation results under various operating conditions are presented in Section 6. Some concluding remarks are given in Section 7.

2. Background information

2.1 Compressor dynamics and surge fundamentals

The dynamics of centrifugal compressors are usually described using a compressor performance map. A compressor performance map (also known as the compressor characteristic) contains curves that show the expected pressure rise across a compressor for given RPM (Revolution Per Minute) and mass flow rate values assuming gas composition and temperature are kept constant. This compressor characteristic is uniquely defined by the compressor's geometry, operating conditions, gas properties and other variables (Bloch, 1995). Figure 1 shows an example of how a typical compressor characteristic map looks like. A surge point is a point of minimum

mass flow rate and maximum pressure (for a given RPM curve) that the compressor can tolerate before entering into a surge cycle (Bloch, 1995). The surge line in Figure 1 connects all the surge points for each RPM curve. Operating the compressor to the left of the surge line will cause the compressor to undergo a surge cycle that may damage it. The branches of the curves to the left of the surge line are normally not shown because it is forbidden to operate the compressor in that region (Lüdtke, 2004). Hence, for a stable compressor operation the operating point must be kept to the right of the surge line at all times.

Like the compressor characteristics, a throttle characteristic shows the expected pressure drop across a throttle given a mass flow rate and the throttle area values. The general shape of a typical throttle characteristic looks like the ones shown in Figure 2, where curves (2) and (3) correspond to decreasing throttle areas from curve (1), and curve (4) represents increasing throttle area from curve (1). The throttle characteristic line also represents the pressure requirements or the resistance of the compression system it represents (Willems and de Jager, 1999).

Figure 3 shows a configuration of an idealised pumping system (a compressor in this case) as described in (Greitzer, 1981). In this system, an incompressible fluid is pumped in from a large reservoir to a closed tank that contains a compressible gas. This tank discharges into a larger reservoir through a throttle valve. In this example, the two reservoirs are assumed to be at the same pressure level (but they do not necessarily have to be equal). If the end effects are neglected, then the inlet of the compressor and the exit of the throttle can be considered to be at the same pressure (Greitzer, 1981). As a result, the essential components of the system include: the pump (compressor), the mass storage capability of the closed volume (plenum), the throttle which controls the system mass flow rate and also represents the system pressure requirements (e.g. losses due to resistance in the piping or effects of subsystems) and finally the inertance of the incompressible fluid in the compressor inlet and throttle ducts. These components are shown in Figure 3 surrounded by the dotted ellipse. In a steady-state situation, the net mechanical energy provided by the compressor should be dissipated by the throttle (Willems and de Jager, 1999). Therefore, generally, the steady-state operating point is set by two conditions, namely that the pressure rise in the compressor equals the pressure drop across the throttle and

that the mass flow rate through the compressor equals the mass flow rate through the throttle (Greitzer, 1981). These two conditions mean that the operating point is at the intersection of the compressor characteristic and the throttle as shown in Figure 4.

2.2 Static and dynamic instabilities

Stability is defined by Betchov and Criminale (1967) as “the quality of being immune to small disturbances”. Taking this definition into account, let us analyse the stability of the compression system described in the previous section. Consider the compressor characteristic shown in Figure 5 which shows the characteristic of a constant speed compressor (fixed suction conditions and gas composition are assumed). The grey region is the surge region (unstable region) and the region on the right is the operating region. The dotted portion of the curve is usually an approximation of the physical system as it is very hard to measure experimentally (Gravdahl, 1998). This portion is usually omitted because it is forbidden to operate in this region (Lüdtke, 2004). Greitzer (1981) shows that there are two types of stabilities to be considered in a centrifugal compressor, namely, static and dynamic instabilities. Static instability is related to the departure of the operating point from the original operating point under the influence of small disturbances. Static stability is guaranteed as long as the throttle line has a greater slope than that of the compressor characteristic at any given steady-state operating point (refer to Greitzer (1981) for more details). Dynamic instability, however, is of more interest when dealing with surge since this is the criterion that causes the oscillations in the mass flow rate and pressure. In addition, it is the dynamic instability that is usually violated first before static instability. A system could be statically stable but dynamically unstable. This is illustrated in Figure 5, where at both points, OP_1 and OP_2 , the slope of the throttle line is greater than that of the compressor characteristic, indicating that the system is statically stable. However, the system is dynamically stable at OP_1 but dynamically unstable at OP_2 . Therefore, static stability is a necessary but not a sufficient condition for dynamic stability. Usually dynamic instability occurs when the intersection point between the compressor characteristic and the throttle line is in the positively sloped region of the compressor characteristic (for example OP_2) (Greitzer, 1981). Therefore to treat surge, the dynamic instability needs to be stabilised (active surge control) or avoided (anti-surge control).

2.3 Causes of dynamic instability

Surge might be initiated by a rise in the discharge pressure as a result of a trip of a downstream compressor, blockage, excessive throttling or when the mass flow rate taken from a pipeline by consumers is at a much smaller rate than that put into the pipeline by the machine. This increase in discharge pressure will cause the plenum pressure to rise because of flow accumulation (D-A in Figure 5). If the pressure of the plenum rises to a value higher than what the compressor could produce (point A in Figure 5), the mass flow rate reverses (A-B in Figure 5). In that process, the plenum depressurises until its pressure becomes less than what the compressor can produce again (B-C in Figure 5), and forward mass flow rate is restored (C-D in Figure 5). At that point, if the discharge pressure is still high, the cycle (A-B-C-D in Figure 5) will repeat over and over again. In addition to mass flow rate reversal, which is highly undesirable, surge might lead to high vibrations, temperature increases, and rapid changes in thrust that can damage the seals or even drive the stationary parts against the rotating parts (Bloch, 1995) and might cause damage to the equipment which in return will result in high maintenance costs.

3. Methodology

The compressor performance map in Figure 5 can be divided into two regions, namely, a safe region (negatively sloped portion to the right of the surge point) and a surge region (positively sloped region to the left of the surge point). In the safe region, at any given mass flow rate value, the system does not surge because there is no big difference between the pressure in the plenum on one hand and the pressure in the compressor on the other hand. In other words there is no dynamic instability in this region. However, the same is not true in the surge region. It can be seen from Figure 5 that there is a difference between the pressures of the plenum and that of the compressor in this region (the dash dotted line is the pressure of plenum while the dotted characteristic is the pressure of the compressor). This imbalance causes the reversal of the mass flow rate and it is this that makes this region unstable. Therefore, what keeps the compressor dynamically stable in the safe region is the absence of the pressure imbalance between the compressor and the plenum. Thus, the solution to

avoiding surge lies in making sure that these two quantities are always the same in the surge region too. Increasing the area of the throttle valve whenever any small difference in the pressure is detected will ensure that the pressure of the plenum is brought down to match that of the compressor. This will ensure that no pressure imbalance is present between the plenum and the compressor and therefore no gas acceleration between these two elements.

The next question is how to let the controller know about the pressure difference or imbalance between the compressor and the plenum? In other words, which signal can the controller use to keep the pressure difference between the plenum and compressor close to zero? The change of mass flow rate signal (which is the acceleration of the mass of gas) could be used. The change of mass flow rate is caused by pressure imbalance. Therefore, if the change of mass flow rate is zero or close to zero there is no pressure imbalance or small pressure imbalance and consequently no flow reversal. Hence, what the controller should do is simply to monitor this signal and keep it always at zero or very close to zero. That is if change of mass flow rate becomes negative (i.e. the gas is slowing down) the plenum pressure should be decreased to avoid it from getting to the point of reversing. This could be done by increasing the throttle area whenever necessary to allow some of the gas flowing out of the plenum and decrease the pressure in the plenum. If the change of mass flow rate becomes positive (i.e. the gas is gaining speed) the pressure of the plenum should be increased. This could be done by restricting the amount of air that leaves the plenum through the throttle by reducing its area. The controller needs to act when the change of mass flow rate is still small, this will ensure that any problem is suppressed early and with minimum energy before the imbalance results in a surge oscillation. Looking at this from a different angle, what the controller is trying to achieve is to mimic the stable operation of the negatively sloped region of the compressor characteristic by modifying the way the system dissipates some of the energy put in by the compressor. One of the advantages of using the change of mass flow rate signal is that it does not require the controller to know where the operating point is. This is important since the throttle line does not represent the amount of valve opened by the operator only, it also includes all kind of resistances that are experienced by the system that might not be measurable (i.e. blockages). All the controller is concerned about is to balance the pressure between the compressor and the plenum at any given time without worrying

about where the operating point is. Operation at the steady-state point is guaranteed by the static stability criterion (Greitzer, 1981) as long as the slope of the throttle line is greater than that of the compressor characteristic.

In summary, there are two types of stabilities to look at: static and dynamic stabilities. According to Greitzer (1981), static stability is a necessary but not sufficient condition for dynamic stability and surge is mainly caused by the later. Therefore, if it can be assumed that static stability is always guaranteed in a given compressor then, the only thing to worry about is the dynamic stability. Dynamic stability can be maintained if the change of mass flow rate is kept at zero at any given time.

4. Compressor model

One of the first models reported in the literature was developed by Emmons et al. (1955). However, it is the model proposed by Greitzer (1976a) that really got wide acceptance and paved the way for further studies of surge control. Since the main focus of this paper is on centrifugal compression systems, rotating stall is not needed in the model. The Greitzer (1976a) model was first introduced for axial compressors but many researches have successfully used it on centrifugal compressors (Hansen et al., 1981, Pinsley et al., 1991). A schematic of the compression system modelled by Greitzer (1976a) is shown in Figure 6. The model consists primarily of a compressor working in a duct and discharging into a plenum, a throttle is connected to the plenum to control the mass flow rate. The plenum is assumed to be large compared to the dimensions of the connected ducts so that velocities and fluid acceleration in it could be considered negligible (Moore and Greitzer, 1986). The pressure inside the plenum is regarded to be spatially uniform, and the flow in the ducts is assumed to be incompressible. The model only applies to compressors with low inlet Mach numbers and small temperature rises compared to the ambient temperature. Additional assumptions include: the compressor behaves quasi-steady, the rotational speed variations are insignificant, and finally the overall temperature ratio of the system is considered to be near unity (Willems, 2000). The Greitzer (1976a) model has the ability to predict the changes in mass flow rate and pressure reasonably well as various experimental studies have shown (Greitzer, 1976b).

A main feature of the model is that it is nondimensional. This is advantageous because it reduces all the speed lines in a compressor map to one line. Changing the speed of the compressor or the size of the plenum corresponds to changing only one parameter known as the B-parameter.

In Moore and Greitzer (1986) the model was defined as:

$$\Phi_c = \frac{1}{f_c(\Psi_c(\Phi_c) - \Psi)} \quad (1)$$

$$\Psi = \frac{1}{(4B^2 f_c)(\Phi_c - \Phi_T(\Psi))} \quad (2)$$

where B is known as the B -parameter and is defined as:

$$B = \frac{U}{2\alpha_s} \sqrt{\frac{V_p}{A_c L_c}} \quad (3)$$

The meanings of the model parameters are listed in Table 1. The compressor characteristic is usually described by the following cubic equation:

$$\Psi_c(\Phi_c) = \psi_{c0} + H \left(1 + \frac{3}{2} \left(\frac{\Phi_c}{W} - 1 \right) - \frac{1}{2} \left(\frac{\Phi_c}{W} - 1 \right)^2 \right) \quad (4)$$

where the parameters ψ_{c0} , H and W are determined from the steady-state measurements of the compressor characteristic (Willems et al., 2002).

The throttle characteristic is given by:

$$\Phi_T(\Psi) = (Y_T + \Delta)\sqrt{\Psi} \quad (5)$$

where,

$$\Delta = \kappa_c C_c \quad (6)$$

Here, $(Y_T + \Delta)$ is the overall gain of the varying area throttle, Y_T is the gain required at steady state and Δ is the extra gain that will be added or subtracted from the steady-

state throttle gain. C_c is the maximum gain that could be added to or subtracted from \mathcal{Y}_T , and $u_c \in [-1, 1]$ is the signal that determines the fractional increase or decrease in Δ .

5. Fuzzy controller design

A schematic of the proposed control strategy is shown in Figure 7. The mass flow rate and change of mass flow rate measurements are fed into the Fuzzy Logic Controller (FLC) and the FLC in turn makes a decision on the proper action to be taken. The controller used in the simulations is a Mamdani-type (Mamdani, 1974) FLC with a typical IF-THEN rule structure. The defuzzification method used in the FLC is the centre of gravity defuzzification method.

5.1 Structure of the FLC

The controller consists of two input variables and one output variable. The input variables are the *Flow Region* (Input 1) and the *Change Of Flow* (or derivative of mass flow rate) (Input 2). The output variable is the signal that determines the fractional increase or decrease in the overall throttle gain necessary to stabilise the system and is named *Valve Opening*. The nondimensional *Flow Region* range was set to be from 0 to 1. Since it is a common practice in fuzzy control design to normalise the inputs and the outputs of the controller between -1 and 1 or 0 and 1, the range of Input 2 was set to be from -1 and 1 (part (b) of Figure 8). To ensure that the signal is mapped correctly into the normalised controller Input 2, a scaling factor (SF) is used as shown in Figure 7. The scaling factor value used for this study was set to SF=1000. By normalising the inputs and outputs, the FLC can now be used on different compressors and without changing the membership functions inside the controller. If the input signal range of the new compressor is somewhat different from the existing one, then only the value of the scaling factor needs to be changed. Moreover, by using this method, the scaling factor could be used as a tuning parameter to optimise the performance of the FLC. The FLC output range was set between -1 and 1, where -1 and 1 corresponds to the minimum and maximum openings of the throttle valve above or below the steady-state value respectively.

Figure 8 shows the membership functions used for the two inputs (a and b) and the output of the controller (c). For Input 1 the flow region was divided into three fuzzy sets: “safe region”, “surge line” and “surge region”. A triangular membership function was used for the “surge line” fuzzy set, a sigmoid function for the “safe Region” and a Z-shaped function for the “surge region”. Input 2, the change of flow, was divided into negative, zero and positive sections using triangular membership functions. Finally, the output was divided into five fuzzy sets: “Close Fast”, “Close”, “Do Nothing”, “Open” and “Open Fast”. Triangular membership functions were used for those fuzzy sets.

5.2 Fuzzy control rules

Table 2 gives the rules used in the FLC. As explained before, the controller’s main function is to keep the pressure difference between the plenum and the compressor close to zero. The change of mass flow rate signal contains information about the direction that the flow will take in the future. This is directly related to the difference in pressure between the plenum and the compressor. Therefore, if the flow is in the surge line region or the surge region itself, the controller needs to be ready to act. Also, whenever the change of mass flow rate is negative the controller should increase the opening of the throttle to release some of the pressure in the plenum until the change of mass flow rate is close to zero. If the change of mass flow rate is positive, the controller will decrease the throttle opening to add some resistance to the system. The amount of extra opening and closing of the throttle will be proportional to the change of mass flow rate value. The larger the change of mass flow rate value the more throttle opening/closing will be required.

Figure 9 shows the control surface for the fuzzy controller. As the figure shows, because the compressor is inherently stable for mass flow rate values larger than the surge line ($\Phi > 0.4955$), the controller output is zero in those regions. The controller output is also zero whenever the change of mass flow rate is zero.

6. Simulation Results

6.1 Uncontrolled situation with surge

All the simulations in this paper were carried out using a fixed-step ode3 (Bogacki-Shampine) solver in the MATLAB[®]/SIMULINK[®] environment with a fixed step size of 2. The model parameters are given in Gravdahl (1998).

To demonstrate the behaviour of the system in the unstable area of the compressor map, the throttle gain, γ_T , was set to 0.5 (corresponding to $\Phi = 0.3929$). The initial values for the nondimensional mass flow rate and pressure rise were set to 0.75 and 0.32 respectively while the fuzzy controller was switched off throughout the simulation ($u_c = 0$). Figure 10 shows the mass flow rate and pressure fluctuations against ζ (the nondimensional time). Figure 11 shows the same results in the compressor map view.

As it can be seen from Figure 11, the throttle line intersects with the compressor characteristic in the surge region (denoted by \star). The operating point shifted from the initial condition along the compressor characteristic in the negatively sloped portion of the characteristic. At the surge line, the pressure of the plenum becomes higher than that of the compressor and the flow reverses and causes surge as expected. This matches exactly with the explanation in previous sections.

6.2 The controlled situation

The fuzzy controller described earlier was implemented on the simulated compressor under investigation. The system was then simulated with the throttle gain set to 0.5 ($\Phi = 0.3929$) and controller being switched on this time. The scaling factor of the fuzzy controller was set to 1000 which was obtained by trial and error. The results are presented in Figure 12 and Figure 13. As it can be seen from Figure 12, the controller succeeds in stabilising the system at the new equilibrium point ($\Phi = 0.3929$ and $\Psi = 0.6175$) even though it is to the left of the surge line (the unstable region). Part (a) and part (b) of Figure 13 show the mass flow rate and pressure of the system against ζ respectively. Part (c) and part (d) show the change of mass flow rate signal and the control signal respectively. It can be observed from the figure that there was no control action until the mass flow rate approached the surge line (even though the change of mass flow rate signal was negative). At that point, the controller increases

the gain of the throttle to allow more flow and prevent surge. This positive control signal is a result of having negative change of flow (part (c) of Figure 13). The action of the controller was gradually moving towards zero as the system approached its equilibrium point as required. However, once the change of mass flow rate signal became positive, the control action became negative. Positive change of mass flow rate might not cause reversal of the mass flow rate, but it may undermine the performance of the control system. Therefore, the control action becomes negative to increase the network resistance and prevent gas acceleration.

Figure 14 compares using a variable area throttle and a bleeding valve. The left portion of Figure 14 shows the case where the controller is controlling a throttle valve by increasing or decreasing its area around the steady-state value. The right portion of Figure 14 shows the situation when the controller is only allowed to increase the mass flow rate of the system but not decrease it. This might occur when a bleed valve is used that remains shut and only allowed to open when the change of mass flow rate is negative (i.e. when there is a potential that the system will surge). The controller has no effect on the throttle here. It can be noted from Figure 14, that when using only a bleed valve, the controller can prevent surge. However, when the operating point changes slightly to a higher mass flow rate (say by increasing the area of the throttle) while still being in the unstable region, the pressure in the plenum will drop suddenly and a pressure imbalance between the plenum and the compressor will be created. This will result in the flow to accelerate in a forward direction (positive change of flow). Since the controller is not allowed to increase the resistance of the network, the pressure difference will initially increase due to positive slope of compressor characteristic. However, eventually the pressure of the plenum will balance with that of compressor again at a point in the stable region (negatively sloped region). Nevertheless, even though the pressure is balanced, the mass flow rate through the throttle will be less than that through the compressor because the intersection point is in the unstable area. This will lead to gas accumulation in the plenum and to negative change of mass flow. Once the flow return to the surge region (negative change of flow), the controller will be able to stabilise it again. In contrast, when a throttle controller is allowed to increase and decrease its area the resulting response is much better. Not only that it prevents surge, but it also makes the mass flow rate change

straight to the new steady-state point without the need of going back to the stable region first.

Figure 15 shows the same comparison but against ζ . Notice the transient response from the lower steady-state to the higher steady state in the top two panels of Figure 15. Obviously, the transient in the left is much smoother and much better. This is due to the decreasing of the throttle area (adding resistance) when the change of mass flow rate is positive as shown in the control signal (bottom left of the figure).

6.3 Uncertainty in the location of the throttle line

Figure 16 shows a case with 4 different throttle gains ($\gamma_T = 0.55, 0.5, 0.45$ and 0.4 respectively). As it can be observed from the figure, the controller succeeds in stabilising the compressor at different locations in the surge region without the need of re-tuning any of its parameters. The dashed vertical lines show the location where the throttle line is supposed to intersect with the compressor characteristic for all four cases. The small circle (\circ) marks the starting point of the simulation while the small star (\star) marks the finishing point. Since there was no parameter change in the controller in all four cases, the results show that this controller is robust in the face of uncertainties in the location of the throttle line. This also demonstrates that the controller does not need to know the location of operating point in order to operate.

6.4 Comparison with the controller proposed by (Gravdahl, 1998)

Gravdahl (1998) proposed the use of a CCV to control surge based on the findings of (Simon et al., 1993). The linear analysis in (Simon et al., 1993) showed that using a CCV has the potential of providing better control compared to other actuation forms such as bleed valves when coupled with mass flow rate measurements. Figure 17 shows a comparison between the results obtained in this study and those of (Gravdahl, 1998). The throttle gain used for that method was set to 0.61 while for the fuzzy method the throttle gain is set to 0.5121 . This is to ensure that the steady-state mass flow rate value for both systems is the same ($\Phi = 0.405$). It can be seen from Figure 17 that both methods achieve stability at the same mass flow rate point. However, the method employed by Gravdahl (1998) results in pressure drop across the CCV which results in the operating point to be operated at low pressure value as demonstrated on the figure. In contrast, the proposed fuzzy controller with a variable area throttle

achieved stabilisation at $\Phi = 0.405$ without the pressure drop that comes with the CCV. It can be argued that this means the strategy employed in this paper is therefore more efficient, as it achieves stabilisation without the need of losing some of the pressure that the compression system produces at the first place. It is worth noting that, the controller employed in (Gravdahl, 1998) is a nonlinear controller derived using the backstepping methodology (Karstic et al., 1995). The approach involves a lot of tedious mathematical manipulations. The simple FLC proposed in this paper is designed by intuition and operating experience. The method proposed in this paper also uses the existing throttle (or will require only changing the throttle) and does not need coupling the compressor with a CCV. The control strategy is also model free and does not require any knowledge of the location of the operating point. The controller only needs to know the location of the surge point which is readily available from the suppliers of the compressors. The biggest advantage of the FLC method, however, is the simplicity of its design. This allows the operators in industry to understand the controller which might help in convincing the decision makers to shift from conventional anti-surge control strategies to active surge control.

Figure 18 shows the same comparison but this time against ζ . However, in this case the controller was switched on at $\zeta = 1600$ in both cases. The throttle gain was set so that stabilisation is achieved at $\Phi_s = 0.405$ as before. In the figure, the CCV case is marked with the dashed line and the method proposed in this paper by the solid line. For the mass flow rate case, both controllers achieve stabilisation at the required mass flow rate point. However, for the pressure case, the figure shows a big difference in the pressure at which this stabilisation was achieved. It is apparent that the CCV method wastes a lot of the pressure that the compressor has produced at the first place to achieve this stabilisation.

However, one must realise that even though the controller can stabilise the compressor at any given operating point, this comes at a cost. Table 3 below shows the minimum required value of C_c to achieve stabilisation for different γ_T values. Columns one and two show the chosen values of γ_T and the corresponding mass flow rate at the intersection with the compressor characteristic respectively. Column three shows the minimum required C_c value to achieve stabilisation. Column four

shows the value of C_c as a percent of the magnitude of Y_T . Finally, column five shows the reduction in surge point achieved in each case.

As it can be seen from the table, the smaller the value of Φ_c that the system needs to be stabilised at the larger the value of C_c required. Figure 19 shows the same results graphically. The figure shows the throttle lines when $u_c = 0$ (solid line) i.e. when there is no control action and when, u_c is either fully open or fully closed i.e. $u_c = \pm 1$ (dotted lines). When u_c is fully open or closed, Δ becomes equal to C_c . Therefore Δ is used in the figures instead of C_c . As it can be seen from the figure, the smaller the value of Y_T the larger the value of Δ required to achieve minimum stabilisation. This could also be interpreted as follows. Recall that at full throttle opening ($u_c = 1$) the throttle's overall gain is $(Y_T + C_c)$. Therefore, at steady state (when $u_c = 0$), the overall throttle gain as a percentage of the full gain is given as $\left(\frac{Y_T}{Y_T + C_c}\right)$. Using values from Table 3, when $Y_T = 0.6$ the overall gain at steady state ($u_c = 0$) will be given as:

$$\frac{Y_T}{Y_T + C_c} = \frac{0.6}{0.6 + (0.6 \times 5\%)} \approx 95\%$$

This means that the throttle valve has to be at approximately 95% opening to allow for the extra 5% needed for stabilisation when $Y_T = 0.6$. Similarity, it could be shown that for $Y_T = 0.45$ the throttle must be at approximately 60% opening when at steady state to allow for the extra 40% needed for stabilisation.

6.5 The effect of disturbance

Disturbances occur in all physical systems and compressors are no exception. In a gas turbine, pressure disturbances may take place due to combustion induced fluctuations which will also accelerate the mass flow rate (Gravdahl, 1998). Mass flow rate disturbances may occur from processes upstream of the compressor or other compressors in series (Gravdahl, 1998). These disturbances are referred to as time varying disturbances. Another type of disturbances that effect compressors are constant disturbances (Gravdahl, 1998). Gravdahl (1998) pointed out that this type of disturbances is of a particular importance because if a constant negative mass flow disturbance pushes the equilibrium over the surge line, it will initiate surge. In

addition, the constant pressure disturbance and the constant flow disturbance could be thought of as uncertainty in the compressor and throttle characteristics respectively (Gravdahl, 1998).

The model shown below is adopted from Gravdahl (1998) and is the same model presented in Eq(1) and Eq(2). However, here the model includes both types of disturbances mentioned earlier.

$$\dot{\Phi}_c = \frac{1}{t_c(\Psi_c(\Phi_c) - \Psi + \Psi_d(\zeta) + d_\phi)} \quad (7)$$

$$\dot{\Psi} = \frac{1}{4B^2 t_c} (\Phi_c - \Phi_T(\Psi) - \Phi_d(\zeta) - d_\psi) \quad (8)$$

where $\Phi_d(\zeta)$ and $\Psi_d(\zeta)$, are the time varying mass flow rate and pressure disturbances and d_ϕ and d_ψ are the constant mass flow rate and pressure disturbances respectively. The rest of the variables are the same as before. It can be noticed from the model equations that the pressure disturbance affects the mass flow rate output and the mass flow rate disturbance affects the pressure output.

6.6 Constant disturbances

Figure 20 and Figure 21 show simulation of surge oscillations being induced by a constant disturbance. Figure 20 shows this in the compressor map view and Figure 21 shows different system signals against ζ . Initially, the controller is switched off and the compression system is operated in the stable region with a throttle setting of $\gamma_T = 0.65$ which gives a stable operation. At $\zeta = 1000$ the constant disturbances $d_\phi = -0.1$ and $d_\psi = 0.05$ are brought into the system, resulting in the state of the system being pushed over the surge line. As a result, surge takes place. At $\zeta = 2000$ the FLC is switched on and as shown by both Figure 20 and Figure 21, the surge oscillations are stabilised by the controller at a new operating point.

6.7 Time varying disturbances

In this section, the throttle is set as $\gamma_T = 0.5$ so that the throttle line intersects with the compressor characteristic in the unstable region. This will result in surge oscillations.

At $\zeta = 1750$ the controller is switched on. Both the flow and pressure disturbances here are taken to be white noise signals varying between ± 0.05 .

As shown in Figure 22, the system succeeds in stabilising the compressor even in the presence of time varying disturbances. An interesting thing to note is that, in the case where only a bleed valve (Control valve) is used, the results are not as satisfactory as those in Figure 22. The controller succeeds in preventing surge but not in allowing the system to operate to the left of the surge line at the required steady-state point as shown in Figure 23. This is because, as mentioned before, when the flow moves to the right (in the compressor map view) the controller has to wait for the system to go to the stable region and come back before it can apply control action on it again. However, since the frequency of the time varying disturbances is much higher than the natural restoration frequency of the compressor, the flow will be moved again to the stable region before it gets to the point where the control action could influence the flow.

7. Conclusions

A novel fuzzy logic control strategy to control the surge instability is presented in this paper. The controller relies solely on intuition in its design which makes its operation very easy to understand. The controller uses measurements of the mass flow rate and the change of the mass flow rate to determine the appropriate action to take. One of the advantages of using the change of mass flow rate signal is that it does not require the controller to know where the operating point is. The actuator of choice is a variable area throttle valve that is used to increase or decrease the resistance that the compressor experiences so as to stabilise surge. One of the main advantages of this method is that it does not need a model of the compressor in its design. Another advantage of the method is that the fuzzy control rules are transferable and only the scaling factor and the throttle capacities need to be changed when the controller is used on another compressor of the same type. Simulation investigations show promising results. When compared to the results of (Gravdahl, 1998) the proposed controller gave stabilisation at same mass flow rates without the pressure drop that usually accompanies the use of a CCV. It was also noted from the study that the proposed approach becomes less efficient for smaller stabilisation mass flow rates. This is because the partial throttle opening required for application of the control

strategy decreases with smaller stabilisation mass flow rates. This means more energy loses due to pressure drop across the throttle will result with smaller partial throttle openings. However, the controller also shows the ability of giving satisfactory results in the face of both constant and time varying disturbances. Finally, it is also demonstrated through simulation that the proposed FLC is robust in the face of uncertainty in the location of the throttle line.

Acknowledgement

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Table 1. Parameters used in the model

Parameter	Meaning /value
Φ_c	The annulus averaged mass flow coefficient (nondimensional)
Ψ	The nondimensional plenum pressure.
$\Phi_T(\Psi)$	Throttle characteristic
U	The constant compressor tangential speed (mean rotor velocity)
a_s	The speed of sound = 340 m/s
C_c	The maximum gain that could be added to or subtracted from Υ_T . In this paper the maximum value used is $C_c = 0.29$.
V_p	Plenum volume = 1.5 m ³
A_c	The flow area in the compressor = 0.01 m ²
L_c	Length of duct and compressor = 3 m
B	The Gretizer B -Parameter defined as: $B = \frac{U}{2a_s} \sqrt{\frac{V_p}{A_c L_c}} = 1.8$
l_c	The effective flow passage length of the compressor and its ducts (nondimensional) = 13.33
t	time
$\xi = Ut / R$	The nondimensional time variable used by Moore and Greitzer (1986)
R	The mean compressor radius.
$\dot{\Phi}_c$	The derivative of Φ_c with respect to ξ , $\dot{\Phi}_c = \frac{d\Phi_c}{d\xi}$
$\dot{\Psi}$	The derivative of Ψ with respect to ξ $\dot{\Psi} = \frac{d\Psi}{d\xi}$
Υ_T	Throttle gain.
ψ_{c0}	Shut off value of compressor characteristic.
H	Compressor characteristic semi height.
W	Compressor characteristic semi width.

Table 2. Fuzzy control rules

		Flow region		
		Surge	Surge Line	Safe
Change of flow	Negative	<i>Open Fast</i>	<i>Open</i>	<i>Do Nothing</i>
	Zero	<i>Do Nothing</i>	<i>Do Nothing</i>	<i>Do Nothing</i>
	Positive	<i>Close Fast</i>	<i>Close</i>	<i>Do Nothing</i>

Table 3. Minimum required values for C_e at different stabilisation points.

V_T	Φ_e at intersection point	Minimum required C_e	C_e as % of V_T	Reduction achieved in surge point
0.60	0.4872	0.03	5%	3.25%
0.55	0.4423	0.12	21%	12.17%
0.50	0.3929	0.21	41%	21.98%
0.45	0.3409	0.29	65%	32.30%

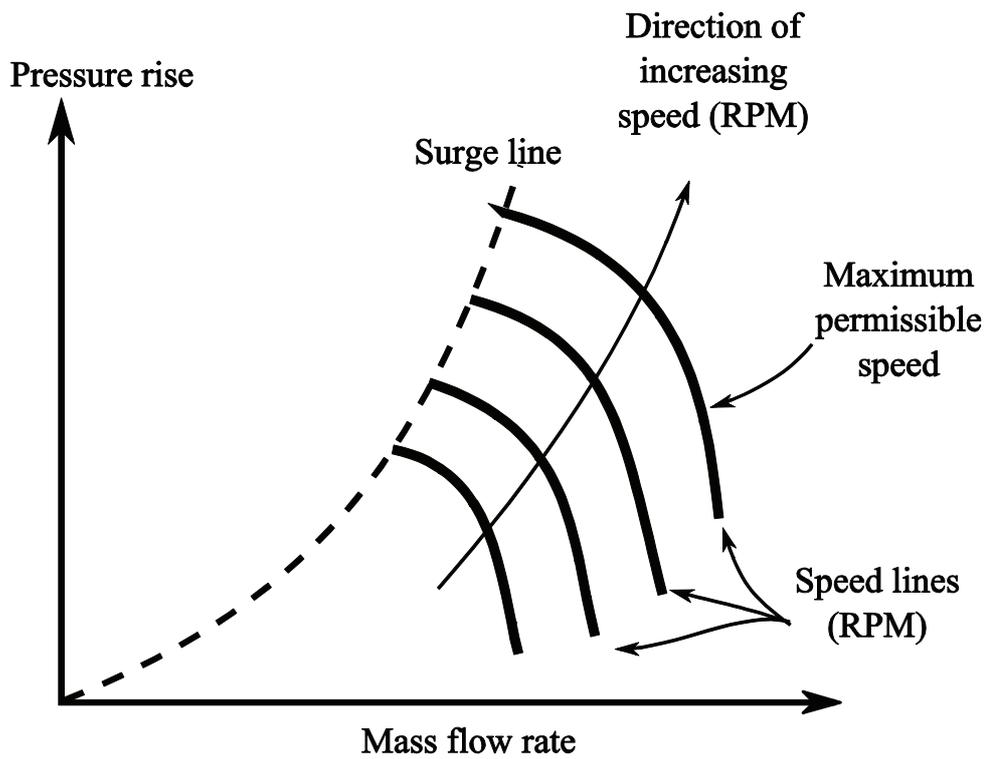


Figure 1. Compressor performance map

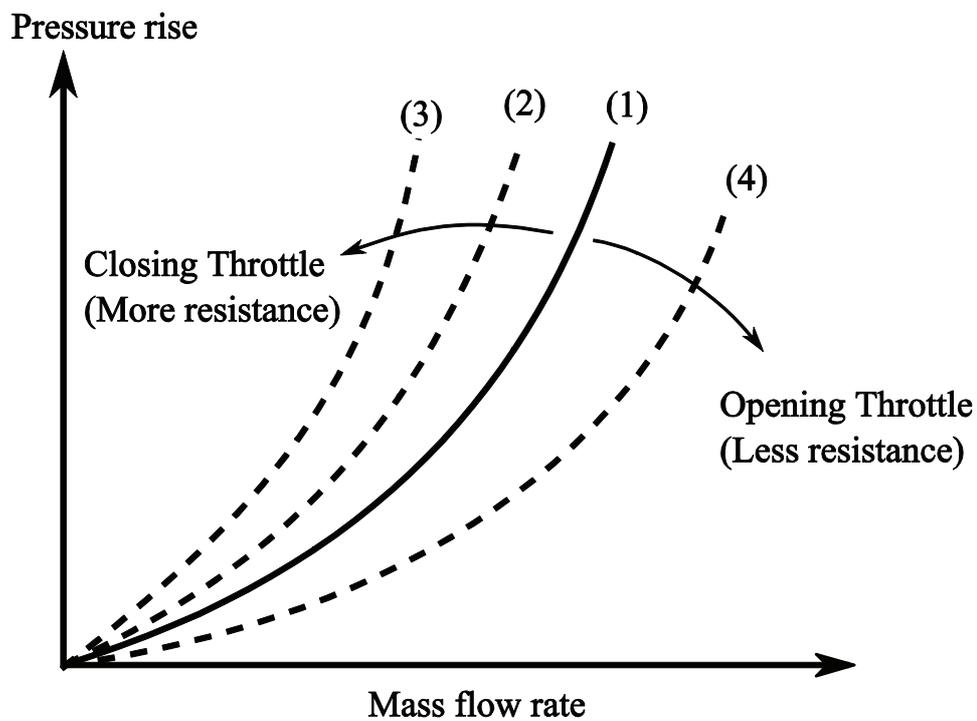


Figure 2. Throttle characteristic map

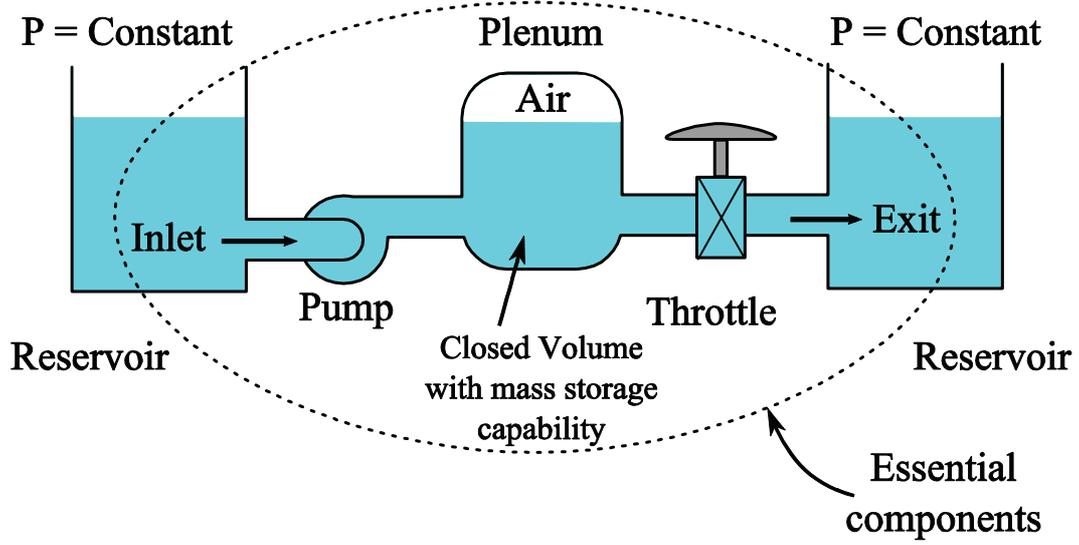


Figure 3. An idealised pumping system as explained in (Greitzer, 1981)

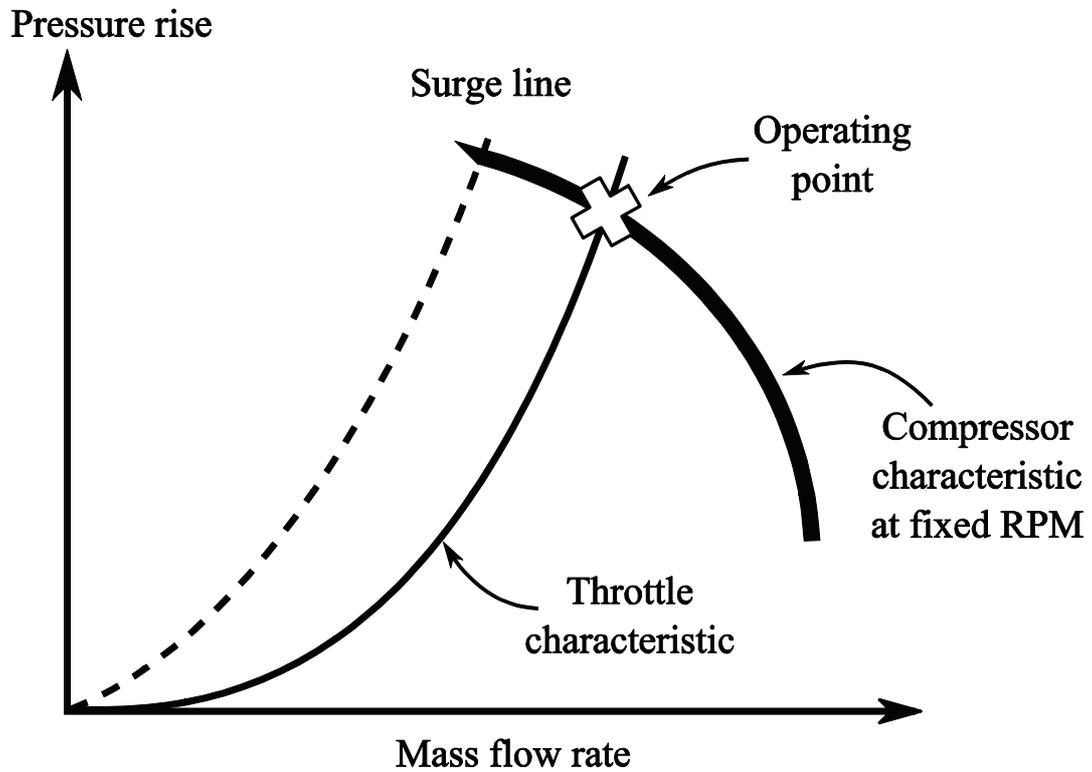


Figure 4. Operating point of a compressor

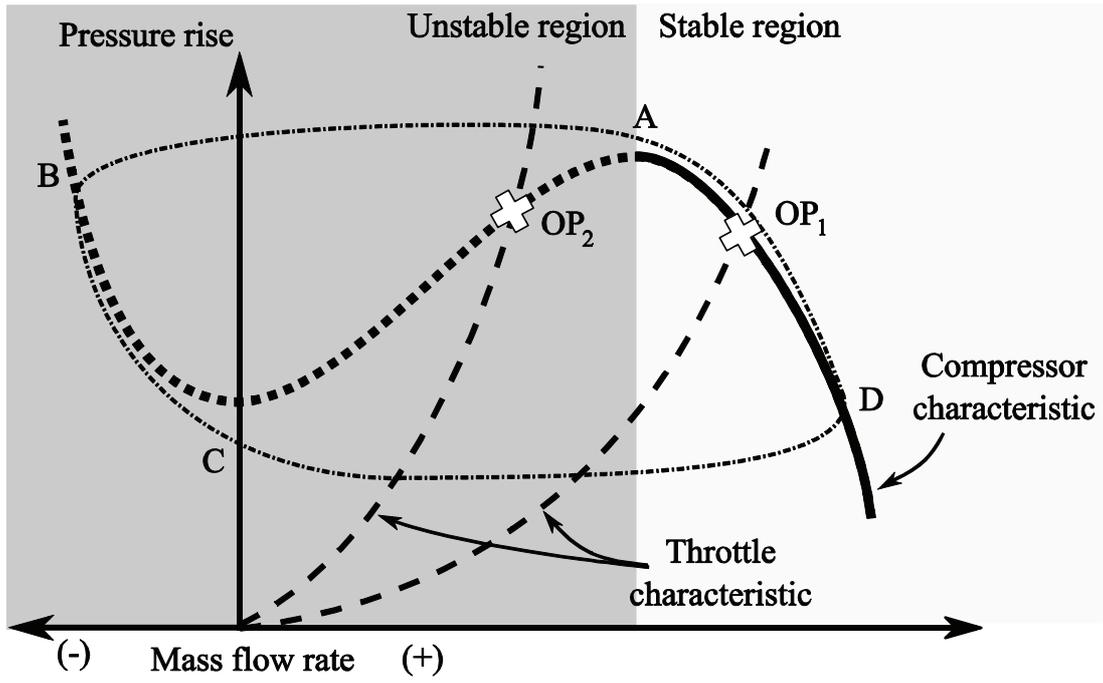


Figure 5. Stable and unstable regions of the compressor performance map

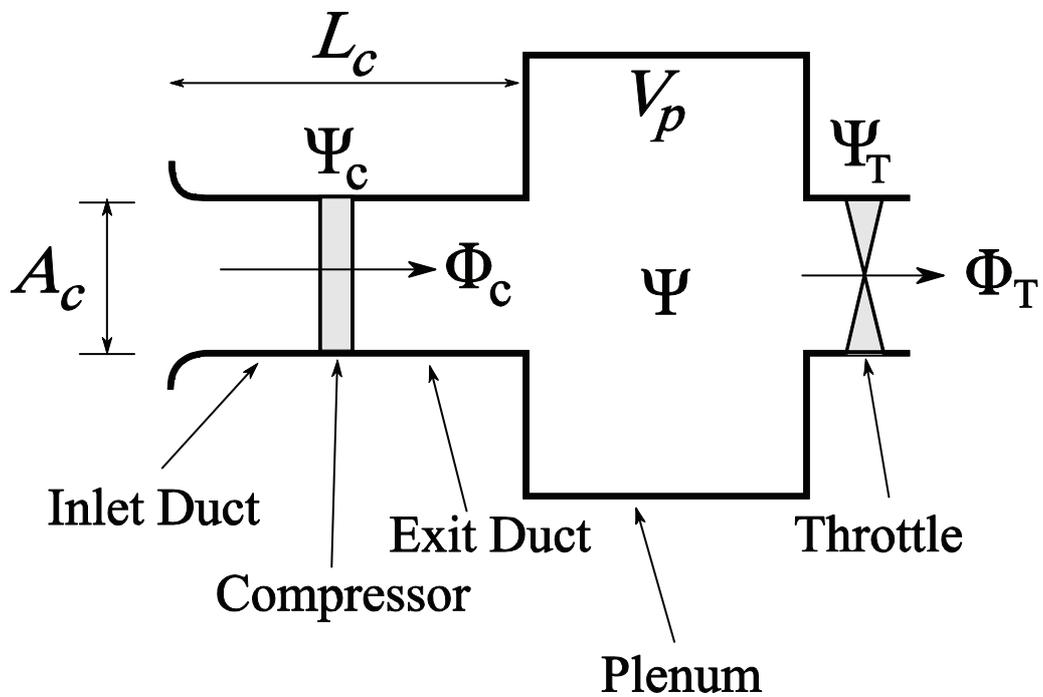


Figure 6. The compression system of Greitzer (1976a)

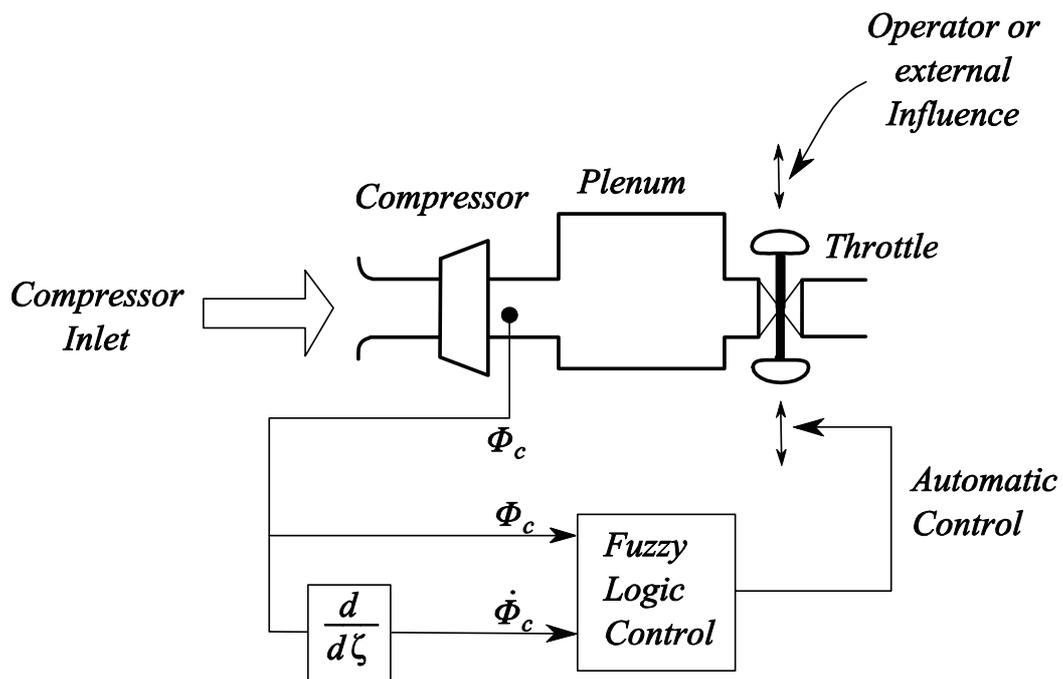


Figure 7. A schematic of the overall control strategy

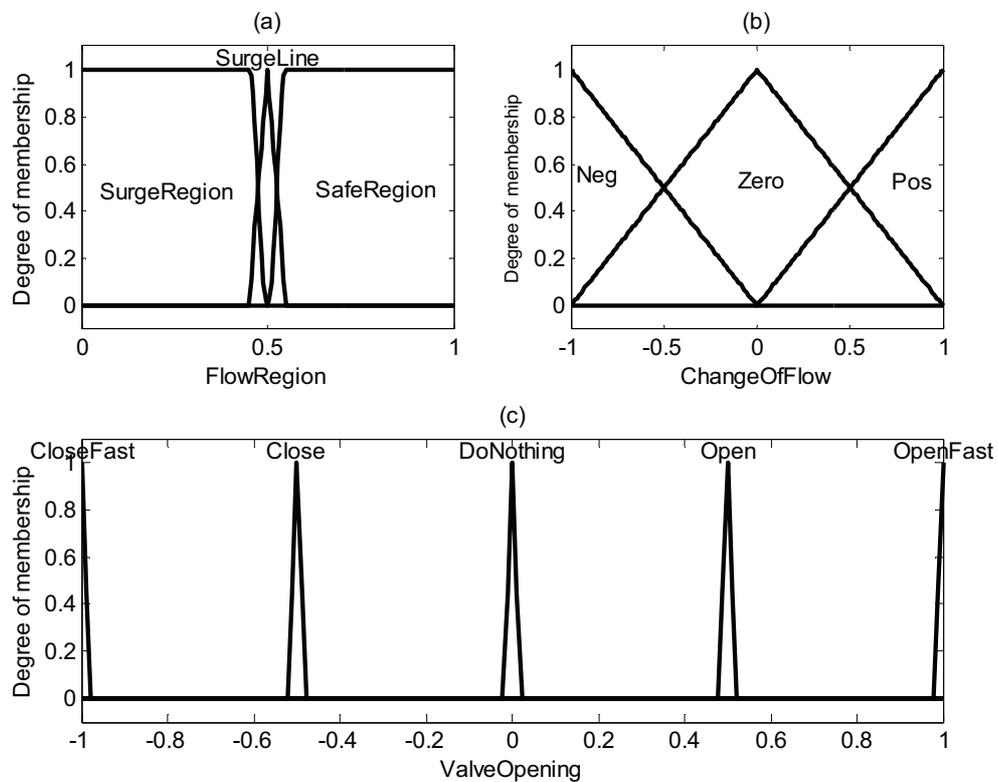


Figure 8. Input and output membership functions

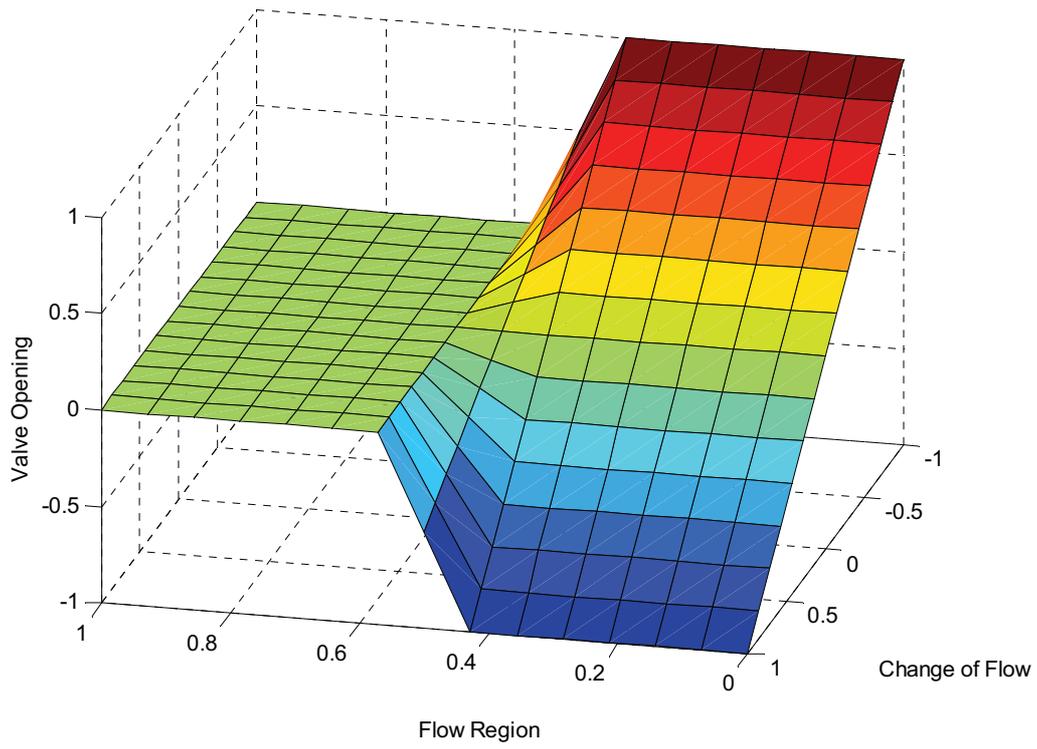


Figure 9. Control surface for the fuzzy system

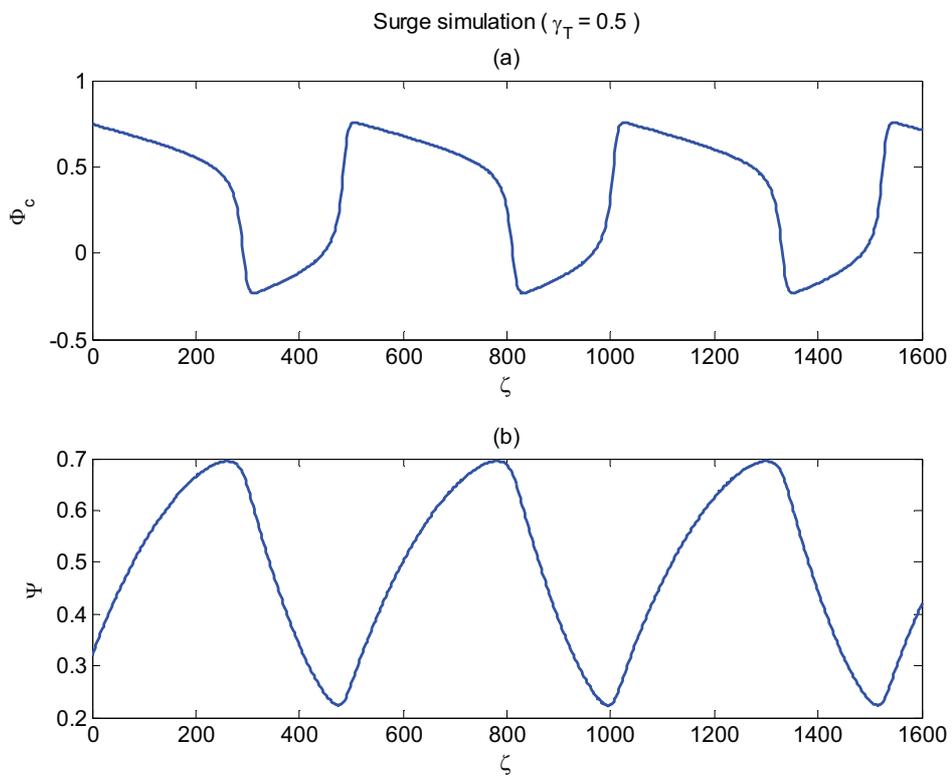


Figure 10. Mass flow and pressure rise against ζ

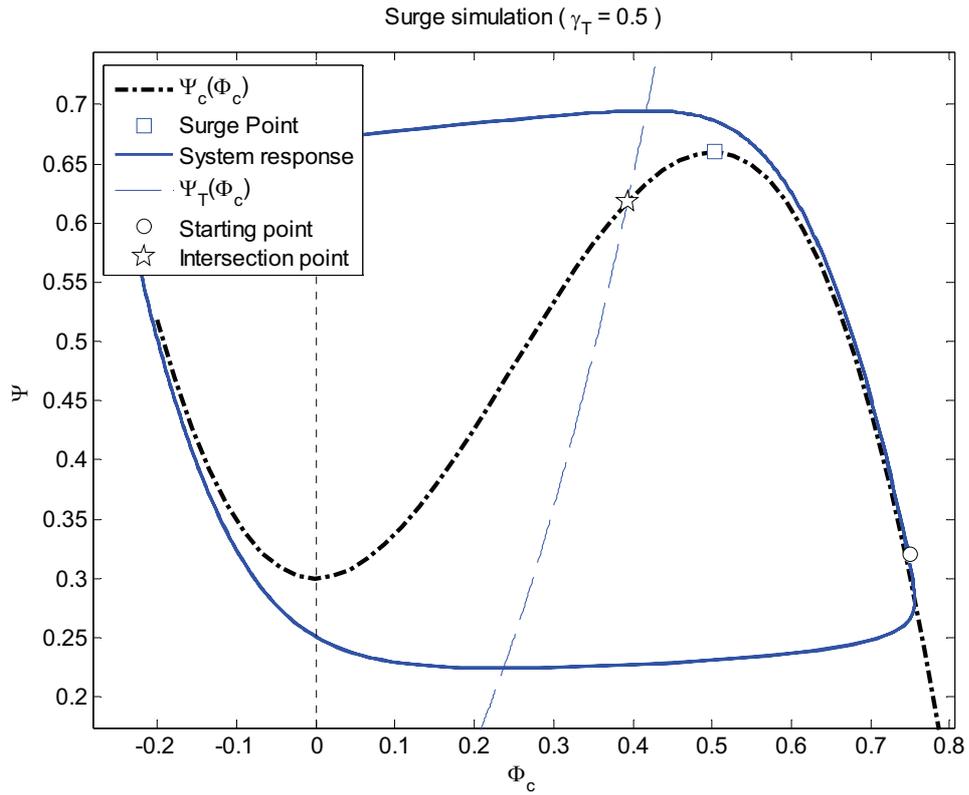


Figure 11. Surge simulation in compressor map view

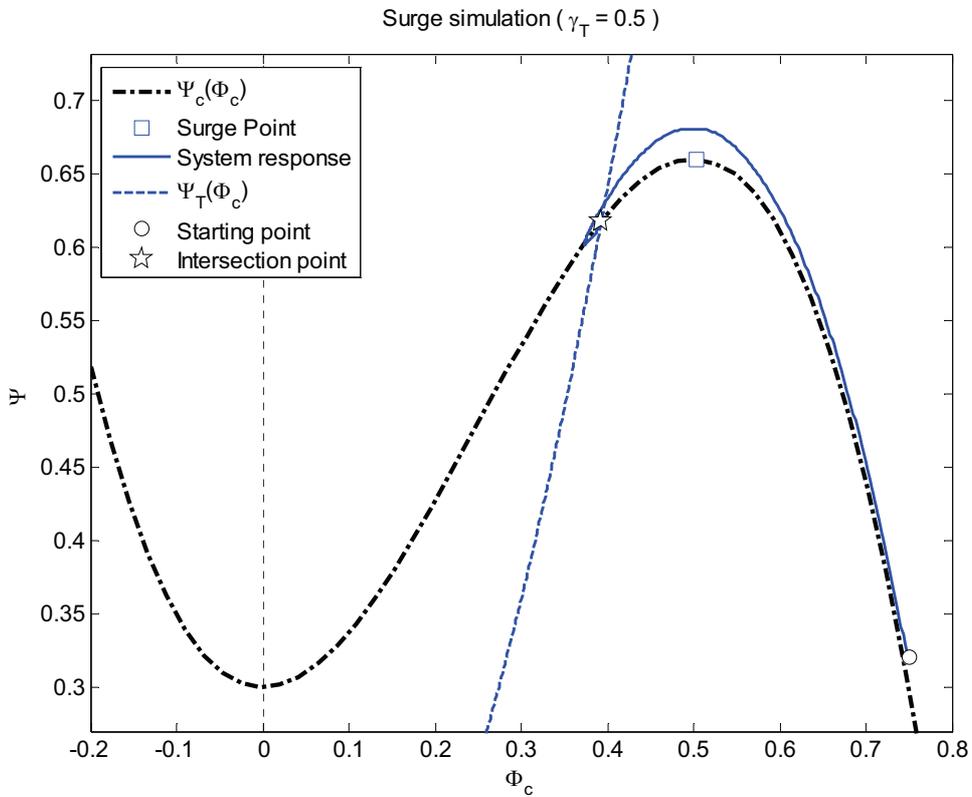


Figure 12. Stable operation in the surge region with fuzzy controller switched on ($\gamma_T=0.5$)

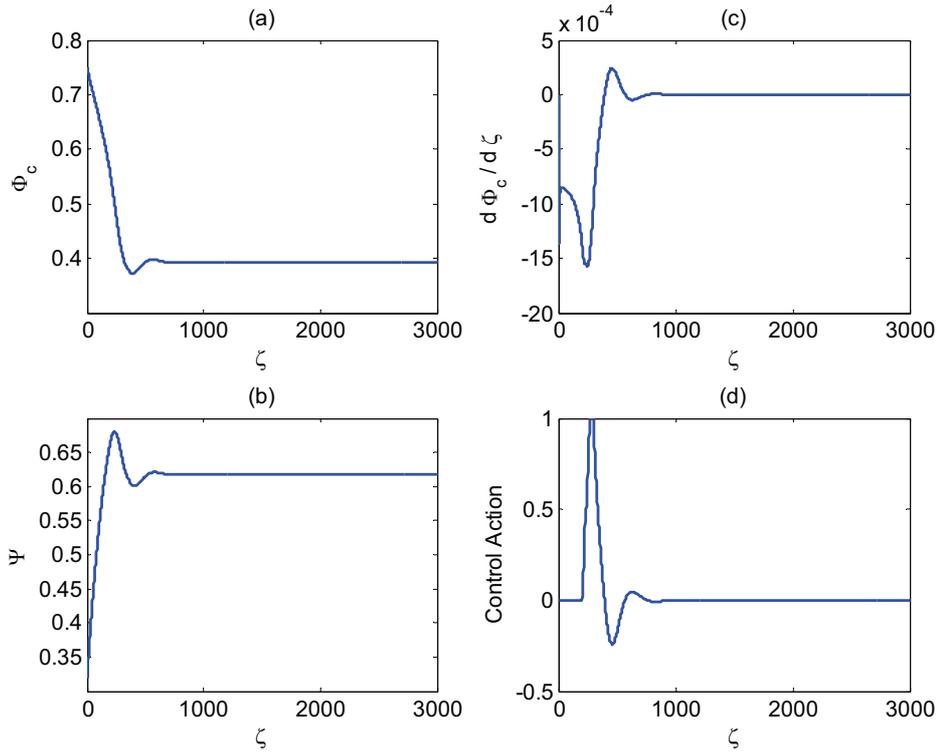


Figure 13. Stable operation with fuzzy controller (responses against ζ)

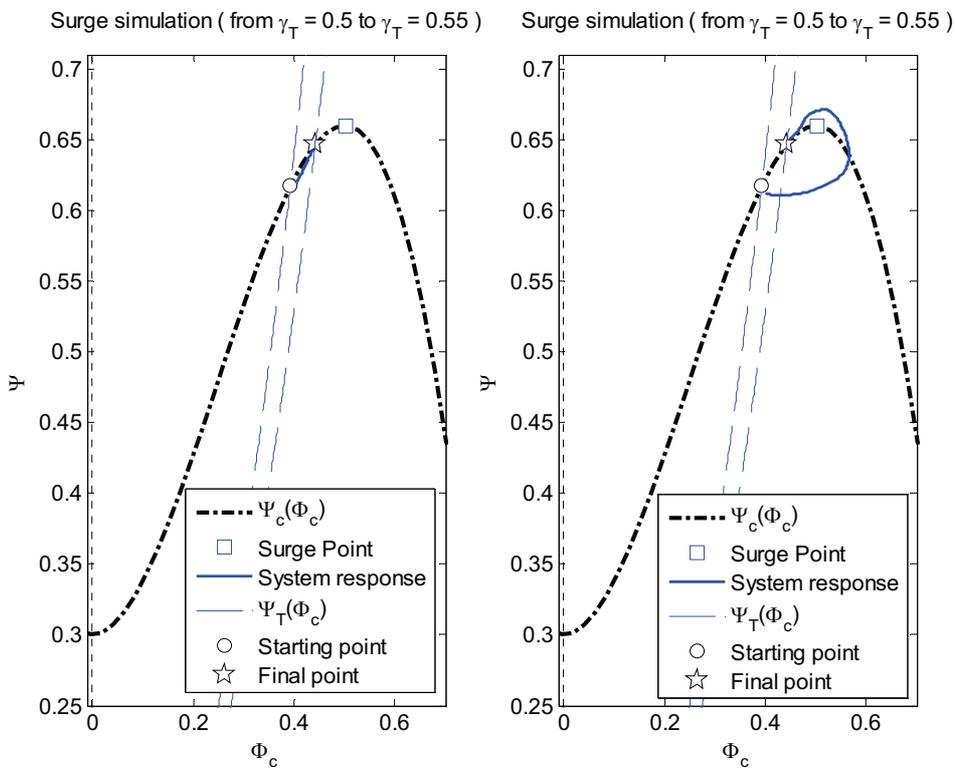


Figure 14. Comparison between using a variable area throttle and a bleeding valve

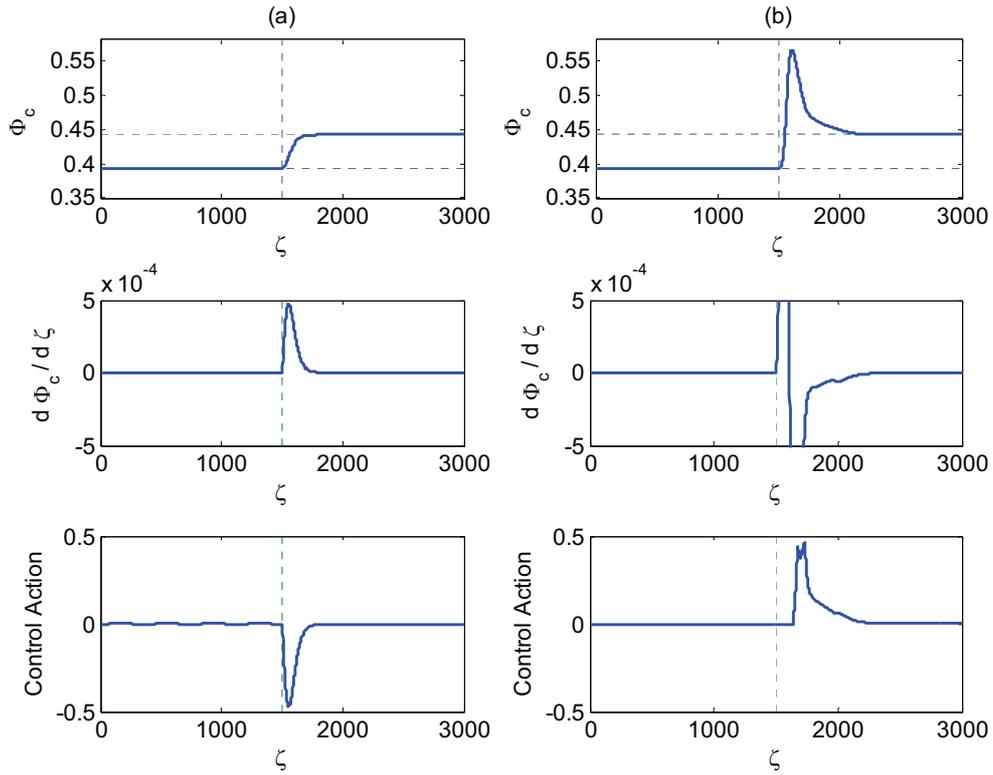


Figure 15. Comparison between using a variable area throttle and a bleeding valve (against ζ)

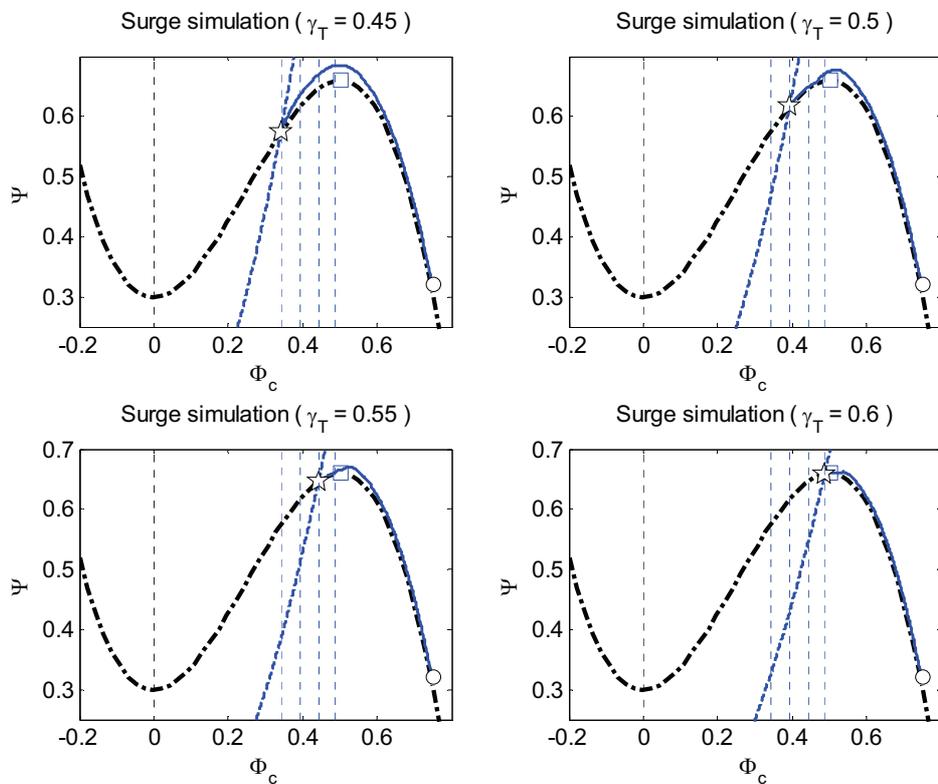


Figure 16. The operation of the FLC at 4 different operating points

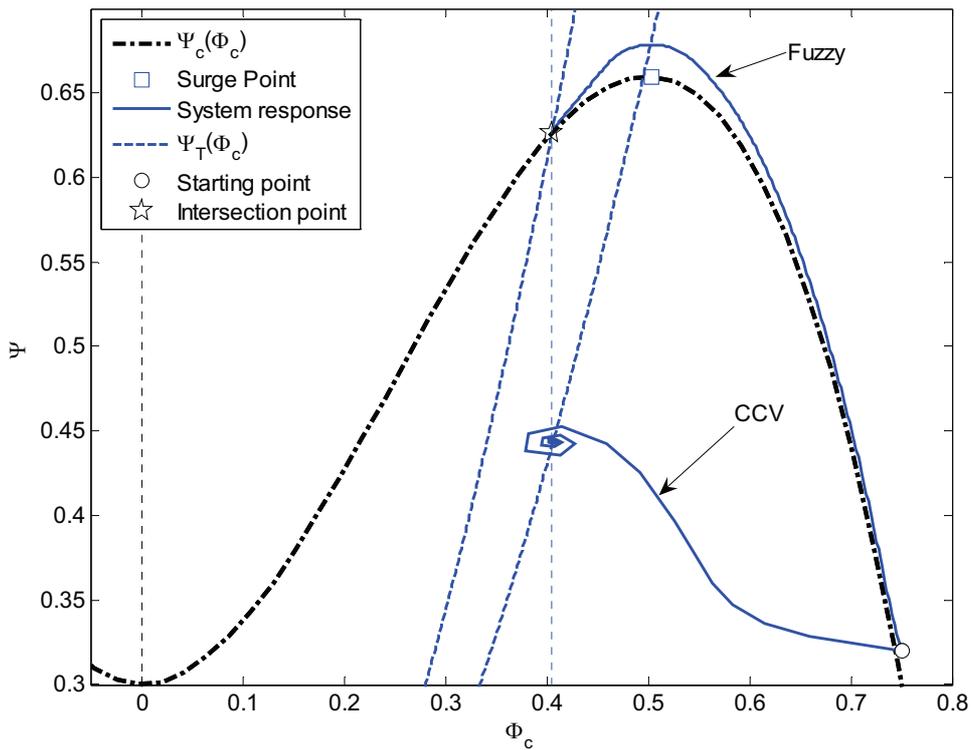


Figure 17. Comparison between the CCV approach and the varying throttle approach

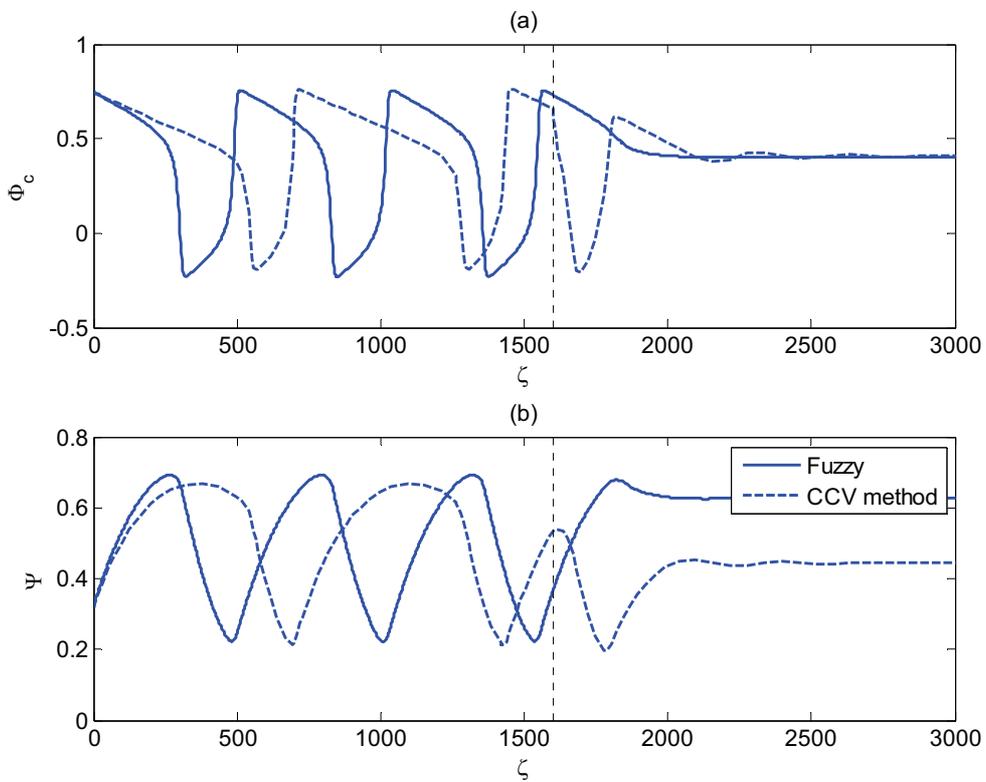


Figure 18. Comparison between CCV method and variable throttle area method (against ζ)

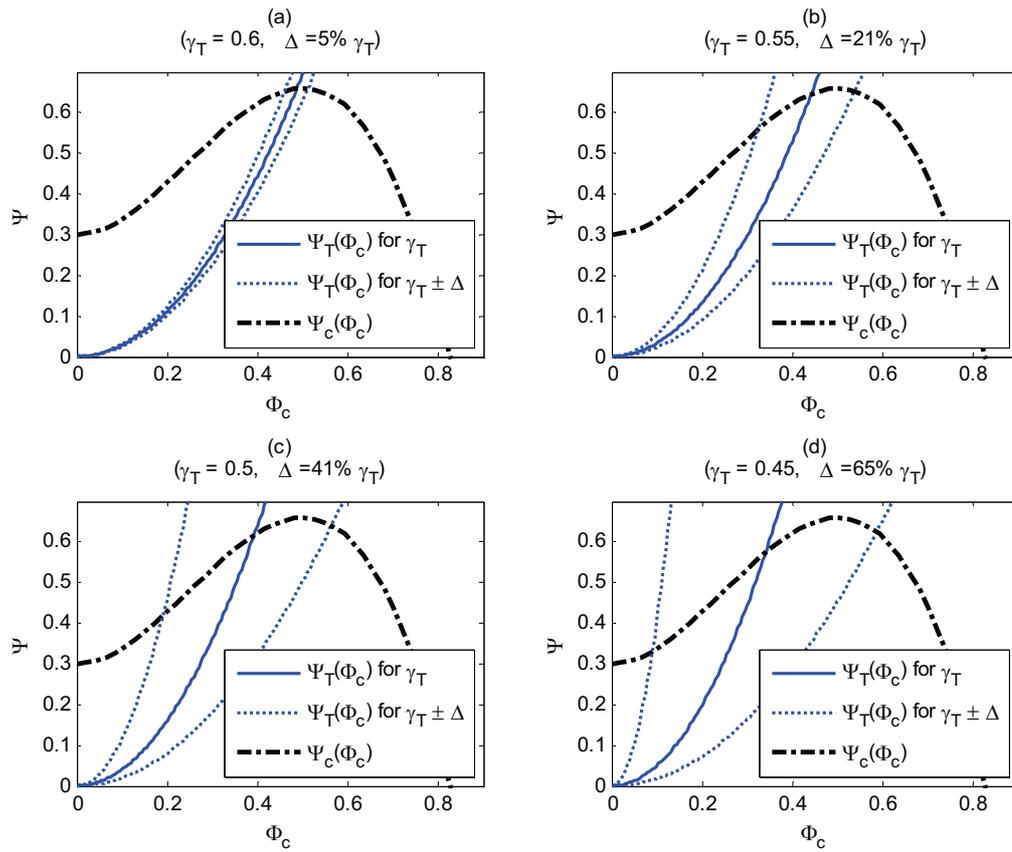


Figure 19. The minimum required Δ as a percent of γ_T .

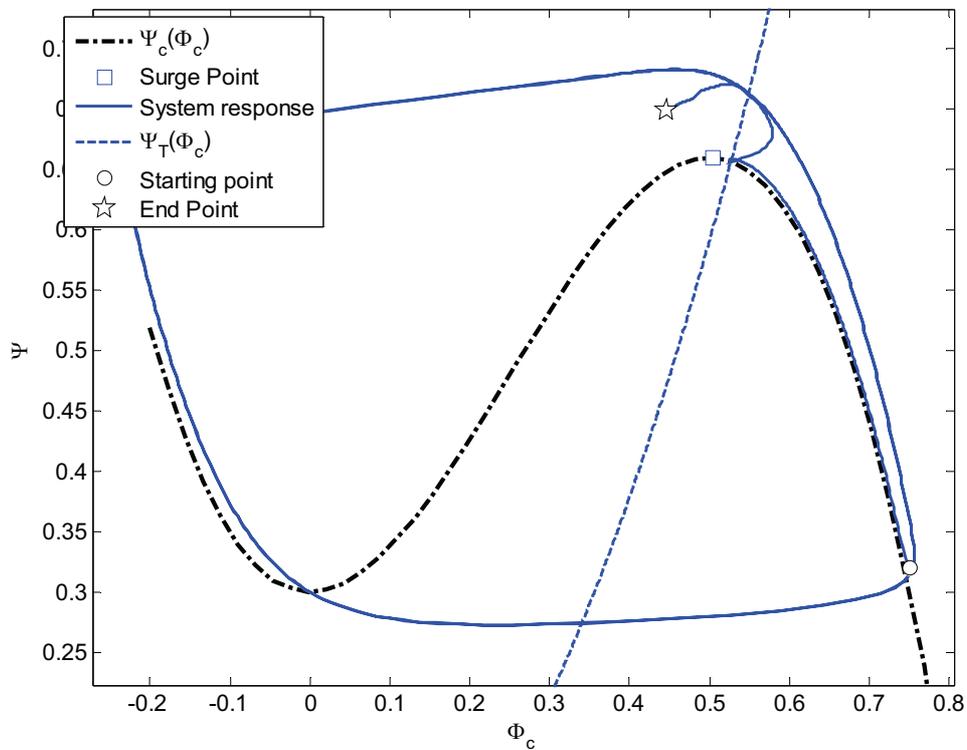


Figure 20. Effect of constant disturbances on the FLC scheme.

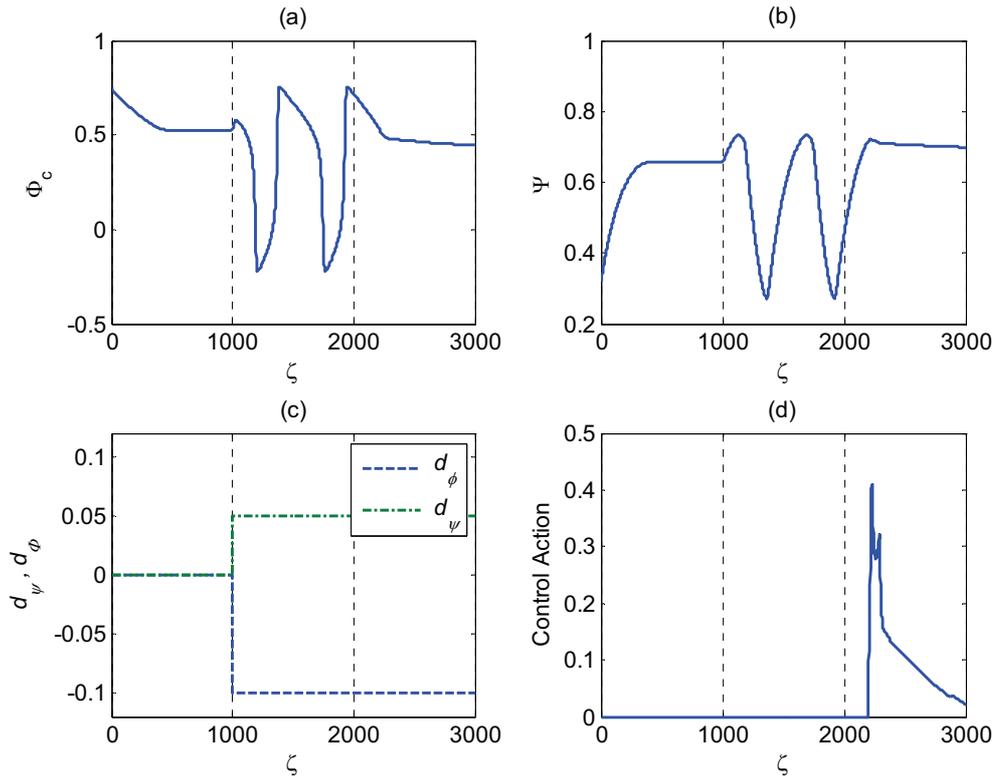


Figure 21. The effect of constant disturbances on the FLC scheme (against ζ)

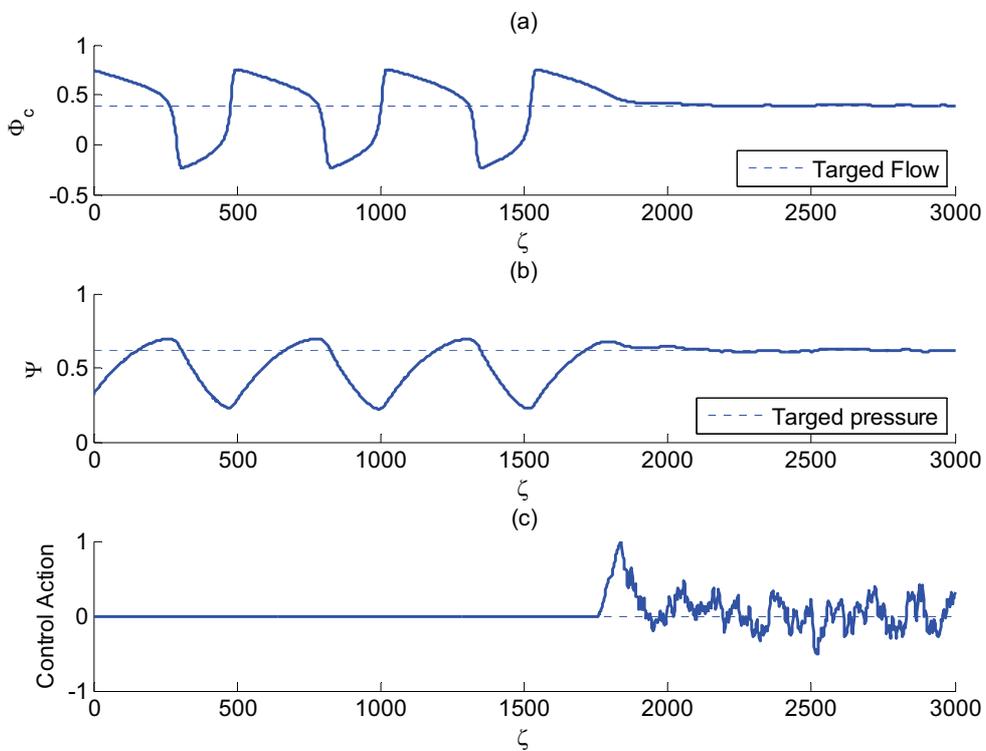


Figure 22. The effect of time varying disturbances

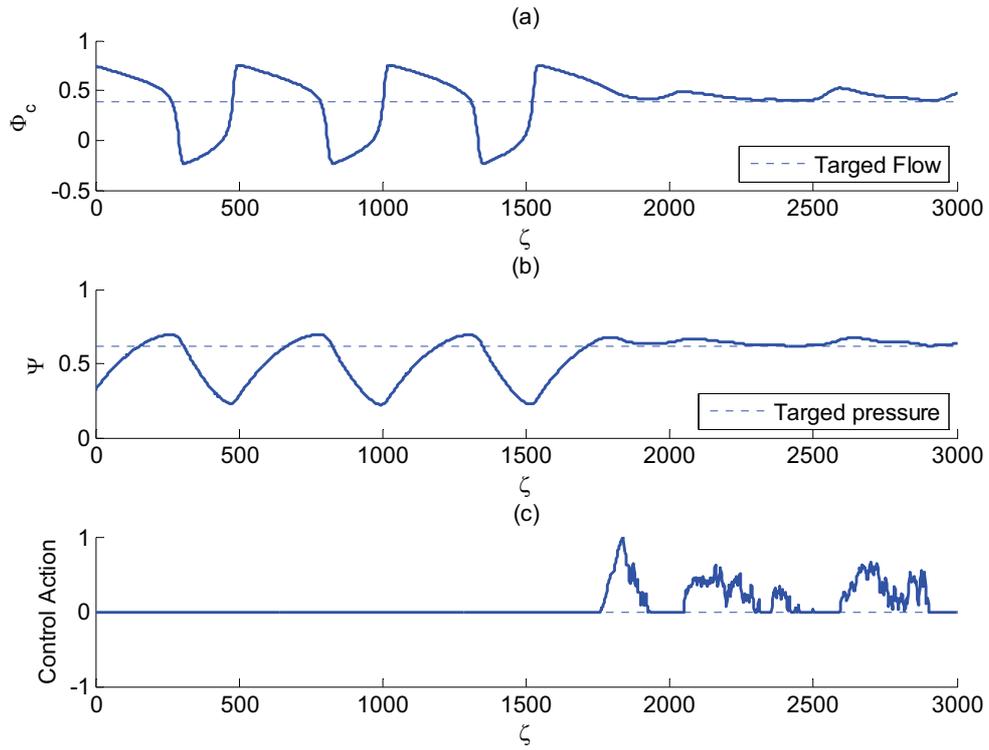


Figure 23. The effect of time varying disturbances when a bleed valve is used (one side control)