Bond Graph Modeling of Centrifugal Compressor System

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Abstract

A novel method of modeling centrifugal compression systems for surge control purposes by using bond graphs is presented. By using the bond graph method, we get a simple description of compression systems based on physical phenomena and it is straight forward to get the dynamic equations. It is demonstrated that several active surge control methods can be represented by the same bond graph. It is also shown how methods for active surge control can be classified using energy flow in terms of upstream energy injection or downstream energy dissipation. A model of a compression system with recycle flow is derived in this work.

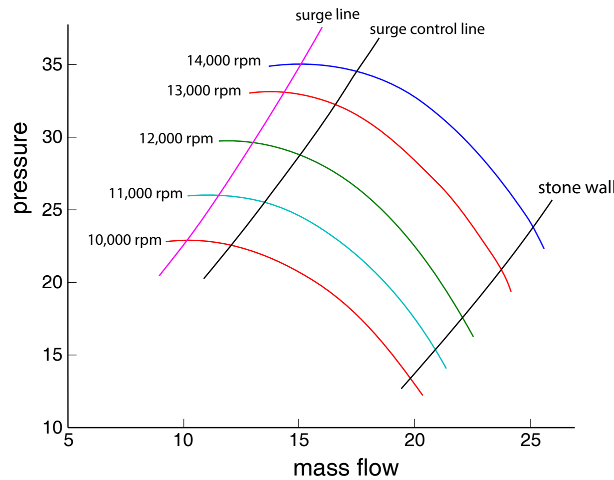


Figure 1. Compressor map for several speeds.

1. INTRODUCTION

Compressors are widely applied in industries, for examples in pipeline natural gas transportation system, extraction of metals and minerals in mining operations, natural gas reinjection plants, secondary recovery processes in oil fields, and process chemical and petrochemical plants [1].

A model of a compression system was introduced by Greitzer in [2]. The Greitzer model is able to predict transient response of a compression system subsequent to a perturbation from steady operating conditions. The model was supported by experiment results in [3].

The operating area of a compressor is commonly described by plotting compressor pressure rise against flow for varying compressor speed. This is known as a compressor map and shown in Figure 1. The stable operating area is limited for low mass flows by the so-called surge line and for high mass flows by the stone wall or choke line. Operation of a compressor at flows below the surge line would drive the compression system into an instability known as surge. This is an axisymmetric oscillation of mass flow and pressure rise and is followed by severe vibrations in the compression system. The vibrations may reduce the reliability of the system and large amplitude vibrations may lead to compressor damage, especially to compressor blades and bearings. It is of interest to study the surge phenomena because high efficiency operating points are located around the surge line. However, operating in such points may endanger the compressor by entering the surge area due to disturbances. There is a trade off between high efficiency operation and compressor stability.

Surge avoidance system (SAS) is a traditional method to prevent a compressor entering surge. These surge avoidance systems usually work by recycling flow from downstream to upstream when the operating point reach a surge control line that is located to the right of the surge line. Such surge avoidance schemes successfully ensure safe operation, but the introduction of the surge control line reduces the usable size of compressor map, thereby restricting the compressor operational envelope.

Active surge control is fundamentally different to surge avoidance as unstable equilibria are sought to be stabilized instead of avoided. The motivation behind this is to overcome some of the shortcomings of surge avoidance. Active surge control of compressors was first introduced by [4], and since then a number of theoretical and experimental results have been published. Different actuators have been introduced, and examples include recycle valves, bleed valves and throttle valves, gas injection, variable guide vanes, variable speed drives, magnetic bearing and piston-actuation [5]-[7].

Bond graph modeling is based on the energy flow in a system. It is a domain-independent graphical description of dynamic behaviour of physical systems where different domains (e.g., electrical, mechanical, hydraulics, pneumatics and thermodynamics) are described in the same way. Bond graph modeling is a form of object-oriented physical system modeling. Comprehensive bond graph literature can be found in [8]-[10] .

This paper is applying the bond graph method for compression system modeling. First, we introduce compressors, surge, surge avoidance system, active surge control, and bond graph in Section 1. Compression system is explained in Section 2. Bond graph modeling of a compression system is given in Section 3. Energy flow in a basic compression system will be studied and the dynamics equations will be derived. Section 4 will explain how surge phenomenon is related to the energy flow in the system. Section 5 discusses surge avoidance system including the bond graph model, dynamics equation and the effect of recycling flow. Active surge control system is presented in Section 6. It is shown that the various active surge control method can be represented as an additional source element at upstream junction or dissipative element at down stream junction.

2. COMPRESSION SYSTEM

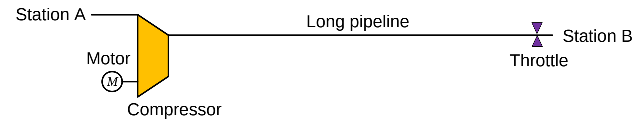
A simple diagram of pipeline gas transportation from station A to station B is shown in Figure 2(a). The system consists of a compressor, pipelines, a throttle and a motor. It is a multi-domain system and including mechanical, fluid dynamic, thermodynamic, and electric domains. This study is focused only on fluid dynamics domain and considering the system as a lump system.

During the operation, the motor drives the compressor to suck gas from station A through the inlet, increase the gas pressure, and then discharge to the long pipeline to reach station B. A throttle is placed before the outlet to change operating point by giving flow resistance. The throttling may cause the outlet mass flow to be less than the compressor discharged mass flow such that some fluid is stored in the pipeline. This is causing a capacitor effect in the pipeline.

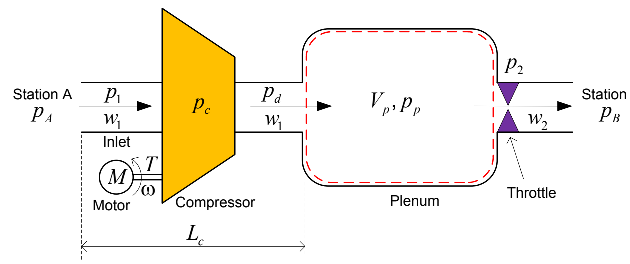
The fluid flow and pressure in the pipeline transportation system are described by a compression model in Fig 2(b). The capacitor effect in the pipeline is modelled by a plenum. The notations p_A is the pressure at station A, T is motor torque, ω is the angular motor speed, w_1 is the inlet mass flow in to compressor, p_c is the compressor pressure rise, p_d is the pressure at compressor discharge, p_p is the plenum pressure, V_p is the plenum volume, p_2 is the outlet pressure drop due to the throttle, w_2 is the outlet mass flow, and p_B is the pressure at station B.

It is assumed that the process in the system is adiabatic and pressure drop along pipeline is neglected as the throttle pressure drop is more dominant. Therefore, p_1 is equal to p_A. It is also assumed the compressor discharge mass flow is same as to the inlet mass flow. The compressor discharged pressure and the throttle pressure drop are given as follows:

$$p_d = p_A + p_c \quad \text{and} \quad p_2 = p_p - p_B. \tag{1}$$



(a) Diagram of pipeline gas transportation.



(b) Model of compression system

Figure 2. Pipeline gas transportation diagram and the compression system model.

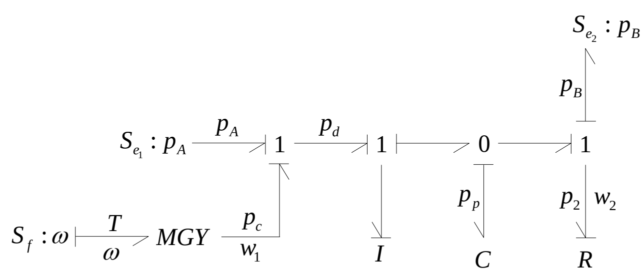


Figure 3. Bond graph model of the basic compression system.

3. BOND GRAPH MODEL OF A COMPRESSION SYSTEM

Bond graph is a domain independent modeling method based on energy flow behaviour in the system. Energy flow of each component in the system is represented by a bond which is a line with half arrow and two variables: effort (e) and flow (f). Torque and angular velocity are the effort and flow in rotational mechanical system, respectively. Pressure and mass flow are the effort and flow for the fluid system, respectively. The bonds are connected to each others by a junction. There are two kind of junctions, 0− junction and 1− junction. There is no energy storage or dissipation in the junctions. In the 1−junction, all connected bonds have equal flow and the total effort is zero. The effort of all connected bonds in a 0− junction are equal and the total flow is zero.

The bond graph model of the compression system is shown in Fig. 3. Stations A and B are modelled as the effort source elements S_{e_1} and S_{e_2} where both are providing pressure p_A

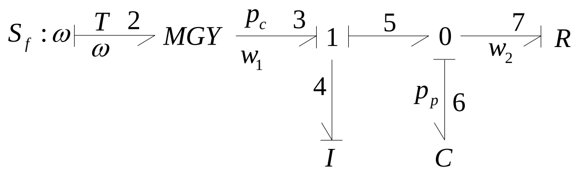


Figure 4. Simplified bond graph model of the basic compression system.

and p_B, respectively. The motor which rotates the compressor is modelled as a flow (angular velocity) source element S_f. The compressor which converts rotational speed into fluid pressure is modelled as a modulated gyrator (MGY) with the compressor pressure rise as the gyrator ratio. The throttle and plenum are modelled by $R-$element (resistor) and $C-$element (capacitor), respectively. Fluid inertia is modelled by $I-$element (inertia). However, the outlet fluid inertia is neglected.

By assuming the pressures at station A and B are the atmospheric pressure and all pressure are measured as pressure gauge, the compressor discharged pressure and the throttle pressure drop become:

$$p_d = p_c \quad \text{and} \quad p_2 = p_p. \tag{2}$$

Then, the bond graph model can be simplified as shown in Fig. 4. The numbers starting from 2 in the bond graph model are the bonds index number.

Dynamic equations of the system can be obtained by using effort-flow relation of the energy storage elements (C and I elements) as follows:

$$\dot{w}_1 = \frac{1}{I}(p_c - p_p) \tag{3}$$

$$\dot{p}_p = \frac{1}{C}(w_1 - w_2) \tag{4}$$

where I and C is the inductance and capacitance. Equations (3) and (4) have the same form as the Grietzer's model equations in [2] and are exactly the same by defining $I = \frac{L_c}{A_c}$ and $C = \frac{V_p}{a_0^2}$. The notations a_0 is the speed of sound, A_c is the inlet duct cross section area, and L_c is the length of the inlet duct. It is common in compression system analysis to present the equation in non-dimensional form. The non-dimensional form of (3) and (4) are given as follows:

$$\dot{\phi}_1 = \frac{1}{\tilde{I}}(\psi_c - \psi_p) \tag{5}$$

$$\dot{\psi}_p = \frac{1}{\tilde{C}}(\phi_1 - \phi_2) \tag{6}$$

where $\tilde{I} = B^{-1}$ and $\tilde{C} = B$. The constant B is the Greitzer constant defined by $B = \frac{U}{2\omega_H L_c}$ with $\omega_H = a_0\sqrt{\frac{A_c}{V_p L_c}}$.

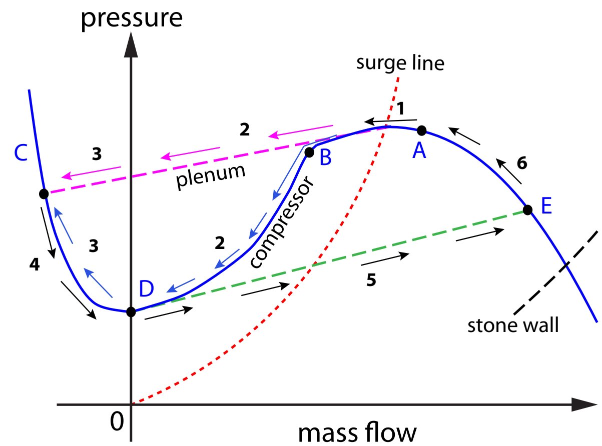


Figure 5. Compressor surge cycle.

The notation ϕ_1 is the non-dimensional compressor mass flow, ϕ_2 is the non-dimensional throttle mass flow, ψ_c is the non-dimensional compressor pressure rise, ψ_p is the non-dimensional plenum pressure, U is the mean rotor velocity, and ω_H is the Helmholtz resonator frequency.

The compressor pressure rise is given as a function of mass flow and compressor rotational speed $p_c(w_1, \omega)$ and usually presented in a compressor map provided by the compressor manufacturer. The function can also be obtained in laboratory by performing engine test to collect data and approximated by a function. [11] introduced a cubic function to approximate compressor pressure rise for constant rotational speed:

$$\psi_c(\phi_1) = \psi_{c_0} + H\left[1 + \frac{3}{2}\left(\frac{\phi_1}{W} - 1\right) - \frac{1}{2}\left(\frac{\phi_1}{W} - 1\right)^3\right] \tag{7}$$

where ψ_{c_0} is the shut-off value of the axisymmetric characteristic, W is the semi-width of the cubic axisymetric compressor characteristic and H is the semi-hight of the cubic axisymetric compressor characteristic, consult [11] for more detailed definition.

The non-dimensional throttle mass flow is defined by:

$$\phi_2 = \gamma_T \sqrt{\psi_2} \tag{8}$$

where γ_T is the throttle setting.

4. COMPRESSOR SURGE AND ENERGY

Compressor surge is a limit cycle of pressure and mass flow in compressor operation. The surge cycle is shown in Fig. 5 and is explained as follows [12]:

1) A compressor is initially operating steady at operating point A located in the stable area, but close to the surge

line. This operation results in the plenum pressure being the same as the discharge pressure. Due to a disturbance, for example by closing throttle, the compressor mass flow is reduced and the operating point should move to the new point at B located in the left side of surge line.

2) At the point B, the compressor discharge pressure is less then the plenum pressure. This condition makes the mass flow decelerate as formulated in (3) and the compressor produces less pressure discharge according to the compressor characteristic curve.

3) The mass flow continue to decelerate to zero and then to negative value (reverse flow). The compressor acts like an orifice in reverse flow. The stored energy in the plenum accelerates the reverse flow until it reaches maximum reversal flow at point C.

4) After reaching point C, the pressure is decreasing and the reverse flow is decelerated and becomes zero mass flow at point D.

5) At point D, the compressor begins operation by accelerating flow until point E.

6) The mass flow is reduced to build plenum pressure. However, the compressor operation from point E is going to point A and then to point B such that the cycle repeats.

The energy flow in the compression system is explained through the bond graph model in Fig. 4 as follows. The equation at the 1-junction is given by:

$$e_4 = e_3 - e_5 \tag{9}$$

where e is the effort and the subscript number describes the bond index number. The efforts e_3 is p_c, e_5 is p_p, and e_4 is the effort of fluid inertia. The differential equation of the inertia component is given by:

$$I\dot{f}_4 = e_3 - e_5 \tag{10}$$

where f_4 is w_1. Equation (10) is exactly same as (3).

It is known that the surge problem is due to the compressor pressure discharge being less than the plenum pressure ($e_3 < e_5$) or e_4 is negative which results the compressor mass flow deceleration. Therefore, surge can be eliminated by maintaining e_4 not negative. It can be done by increasing e_3 and/or reducing e_5 when e_4 should be negative.

The effort e_3 is the effort output of the gyrator (compressor) where the input-output relations are given by :

$$e_3 = rf_2 \tag{11}$$

$$e_2 = rf_3 \tag{12}$$

where r is the gyrator ratio. The effort e_3, which is the compressor pressure rise, increases by increasing the flow input (f_2) which is the motor speed. It is shown in Figure 1 that the compressor produces higher pressure at higher motor speed for the same mass flow.

The effort e_5 comes from the $0-$junction where the connected bonds have the same effort ($e_5 = e_6 = e_7$) and only one bond has effort inward to the junction, i.e. the bond of the $C-$element (e_6). The relation between effort and flow in the $C-$element is given by:

$$e_6 = \frac{1}{C}\int f_6 dt + e_6(0) \tag{13}$$

where f_6 is the capacitor flow which defined by:

$$f_6 = f_5 - f_7. \tag{14}$$

The effort e_6 can be reduced by making f_6 negative which can be done by increasing f_7. Reducing f_5 may also result in f_6 becoming negative. However, this is not possible physically because f_5 is the compressor mass flow (w_1) and reducing the mass flow will bring the compressor into surge. The flow f_7 is the flow of the $R-$element (throttle) where the relation between flow and effort is given by:

$$f_7 = \frac{1}{R}e_7. \tag{15}$$

The flow f_7 can be increased by reducing the R value which means more throttle opening. Therefore, the direct methods to eliminate surge are by increasing motor speed and/or opening throttle. However, those are not always applicable, for example if the surge is happen during shutting down the compressor where the motor speed and the mass flow should decrease.

5. SURGE AVOIDANCE SYSTEM

A surge avoidance system (SAS) is a common method for surge prevention in industrial compressors. The method works by recycling flow from downstream to upstream through a recycle line. A compression system equipped with SAS is shown Fig. 6. A recycle valve is the actuator to control the recycling flow. The SAS works by comparing the compressor operating point to a line called surge control line (SCL). The SCL is laid to the right of the compressor surge line (SL) and become the minimum allowed mass flow in operating the compressor. The distance between SCL and SL is called the surge margin and be defined as [13]:

$$SM = \frac{\phi_{SCL} - \phi_s}{\phi_{SCL}} \tag{16}$$

where ϕ_s is mass flow at surge line and ϕ_{SCL} is mass flow at surge control line. The surge margin is determined by the operator. The SL and SCL are shown in the compressor map in Fig 1 and in Fig 8.

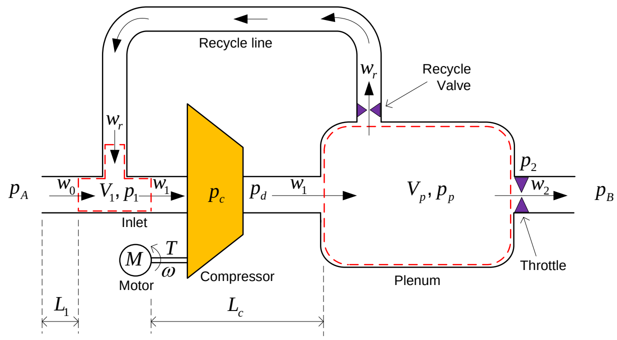


Figure 6. Compression system with recycle line.

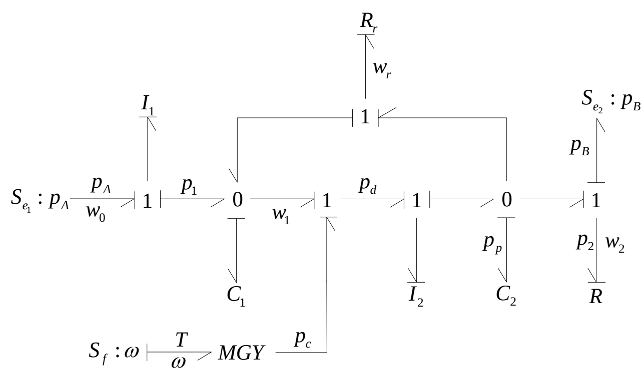


Figure 7. Bond graph model of compression system with recycle line.

When the compressor operating point is crossing the SCL, the recycle valve is opening and flowing the plenum fluid to the compressor inlet. The fluid flow causes the plenum pressure to decrease and the compressor mass flow to be accelerated such that the operating point moves to the SCL. This mechanism ensures that the operating point is not going to surge condition.

The recycling flow gives an additional fluid flow to the inlet such that the inlet states is affected. The effect is investigated by modeling the junction of inlet flow and recycling flow as a plenum and called the inlet plenum as shown in Fig. 6. The inlet plenum has two input flows (feed flow w_0 and recycled flow w_r) and one output flow (compressor inlet flow w_1).

A compressor equipped with SAS is now modelled by using bond graph. Following the procedure of bond graph modeling, the causality suggests that we also have to take account of inertia effect in the feed line due to the recycling flow. The bond graph model of the system is shown in Fig. 7. Dynamic equations of the system are then derived from the bond graph model by using the same assumptions as previously. Evaluating the effort-flow relation at the storage elements in the system (I_1, C_1, I_2, and C_2) results in:

$$\dot{w}_0 = -\frac{1}{I_1} p_1 \tag{17}$$

$$\dot{p}_1 = \frac{1}{C_1}\left[w_0 + w_r - w_1\right] \tag{18}$$

$$\dot{w}_1 = \frac{1}{I_2}\left[p_1 + p_c - p_p\right] \tag{19}$$

$$\dot{p}_p = \frac{1}{C_2}\left[w_1 - w_2 - w_r\right]. \tag{20}$$

where $I_1 = \frac{L_1}{A_1}$, $C_1 = \frac{V_1}{a_0^2}$, $I_2 = \frac{L_c}{A_c}$, and $C_2 = \frac{V_p}{a_0^2}$. The notations V_1 is the volume of the inlet plenum, L_1 is the length of the feed pipe, and A_1 cross sectional area of the feed pipe. The dynamic equations are non-dimensionalized as:

$$\dot{\phi}_0 = -\frac{1}{\tilde{I}_1}\psi_1 \tag{21}$$

$$\dot{\psi}_1 = \frac{1}{\tilde{C}_1}\left[\phi_0 + \phi_r - \phi_1\right] \tag{22}$$

$$\dot{\phi}_1 = \frac{1}{\tilde{I}_2}\left[\psi_1 + \psi_c - \psi_p\right] \tag{23}$$

$$\dot{\psi}_p = \frac{1}{\tilde{C}_2}\left[\phi_1 - \phi_2 - \phi_r\right], \tag{24}$$

where $\tilde{I}_1 = \frac{A_c L_1}{A_1 L_c} B^{-1}$, $\tilde{C}_1 = \frac{V_1}{V_p} B$, $\tilde{I}_2 = B^{-1}$, and $\tilde{C}_2 = B$. Effects of the recycling flow to the compressor inlet states are shown in (17)-(18) for the dimensional form and in (21)-(22) for the non-dimensional form.

The recycled mass flow is defined by:

$$w_r = k_{v_r} u_r \sqrt{(p_p - p_1)} \tag{25}$$

where k_{v_r} is recycle-valve constant and u_r is the valve control signal signal. The range value of u_r is from 0 to 100 %. The non-dimensional form of the recycled mass flow is given by:

$$\phi_r = k_{\gamma_r} u_r \sqrt{(\psi_p - \psi_1)}. \tag{26}$$

where $k_{\gamma_r} = \frac{k_{v_r}}{\sqrt{2\rho A_c^2}}$.

The valve control signal can be designed by using available control methods, for example PI control as given as follows:

$$u_r = K_p \phi_e + K_i \int \phi_e dt \tag{27}$$

where K_p is proportional gain, K_i is integrator gain, and ϕ_e is mass flow error defined by:

$$\phi_e = \phi_{SCL} - \phi_1 \tag{28}$$

where ϕ_{SCL} is mass flow at the surge control line. Only the non-positive mass flow error is used to generate the valve control signal, such that:

$$\phi_e = \begin{cases} \phi_e & \text{for } \phi_e \geq 0 \\ 0 & \text{for } \phi_e < 0. \end{cases} \tag{29}$$

Table 1. SIMULATION PARAMETERS

Parameter	Value	Unit	Parameter	Value	Unit
U	68	m/s	a_0	340	m/s
V_1	0.005	m^3	V_p	0.1	m^3
L_1	0.41	m	L_c	0.41	m
$A_c = A_1$	0.0038	m^2	ρ	1.2041	kg/m^3
ψ_{c0}	0.352	-	W	0.25	-
H	0.18	-	ϕ_{SCL}	0.5513	-
k_{γ_r}	0.05	-	K_p	20	-
K_i	15	-			

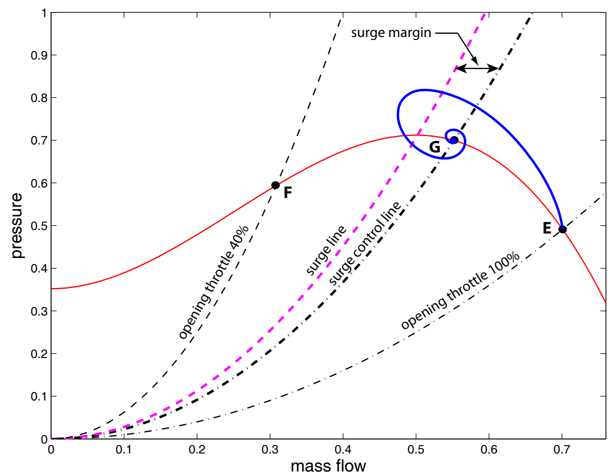


Figure 8. Trajectory of surge avoidance system.

An example simulation of a compression system equipped with SAS by using parameters in Table 1 is given as follows. A compressor is initially operating steady at point E where the throttle is fully open ($\gamma_T = 1.0$) as shown in Fig. 8. At non-dimensional time $\tau = 20$, the throttle is closed to 60% ($\gamma_T = 0.4$) such that the compressor should operate at F located to the left of SCL and also in the left of SL which is in the surge area. The operating point is moving from E to the left and crossing the SCL such that giving signal to open the recycle valve.

Figure 9 shows the recycle valve is then opening and plenum fluid is recycled to the inlet ($\phi_r > 0$). It makes the plenum pressure decrease and accelerate the compressor mass flow. Finally, the operating point stays at point G in the stable area instead of at the point F in the unstable area as shown in Fig. 8. The transient response of the inlet state affected by the recycling flow is also shown in Fig. 9 .

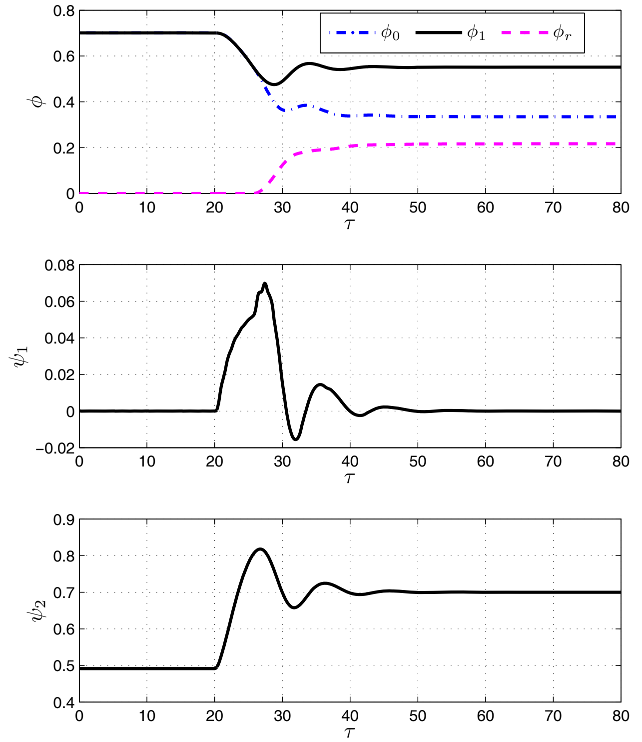


Figure 9. Surge avoidance system response and the recycling flow effect to the inlet .

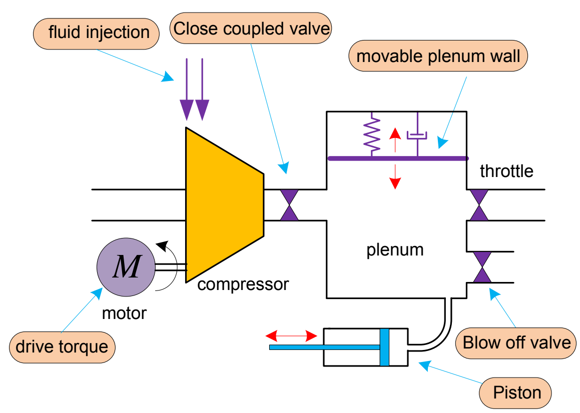


Figure 10. Several methods in active surge control

6. ACTIVE SURGE CONTROL

In previous section, it was explained from the bond graph model of the compression system that surge can be eliminated by keeping e_4 non-negative. Several of the proposed methods and actuators for active surge control are shown in Fig. 10. Basically, the actuator works to maintain e_4 positive by increasing e_3 and/or reducing e_5. Those methods can be clas-

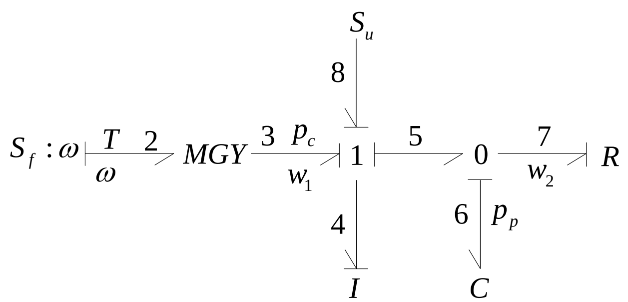


Figure 11. Bond graph model of active surge control by injecting upstream energy.

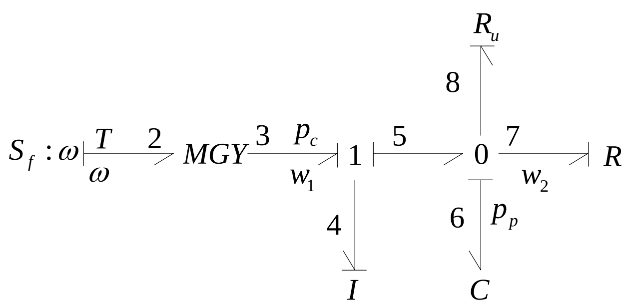


Figure 12. Bond graph model of active surge control by dissipating downstream energy.

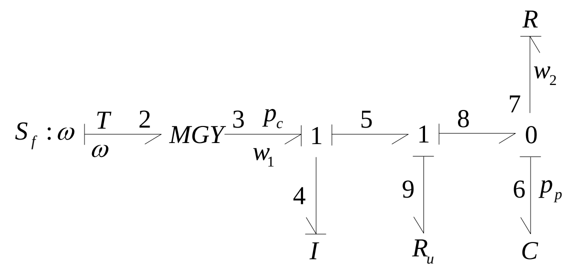


Figure 13. Bond graph model of closed couple valve active surge control

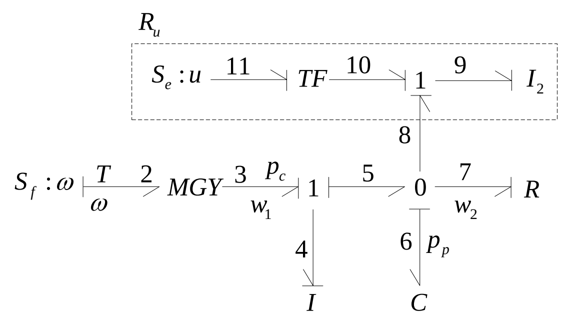


Figure 14. Bond graph model of piston-actuated active surge control

sified as upstream energy injection to increase e_3 and downstream energy dissipation to reduce e_5. A bond graph model of active surge control by upstream energy injection is shown in Fig. 11. The active actuator for injecting energy is noted by S_u. Active surge control method by using fluid injection [15] and drive torque control [6] are examples of upstream energy injection. Bond graph model of active surge control by downstream energy dissipation is shown in in Fig. 12. The active actuator for dissipating energy is noted by R_u. It is shown the effort e_5 is reduced indirectly through the flow f_8. Movable plenum wall [16]-[17], piston-actuation [7], and blow off valve [18] are examples of the R_u element in this bond graph configuration. Close coupled valve is also dissipating the downstream energy, however the valve is placed between the compressor discharge and the downstream plenum. The bond graph model of the compression system with close coupled valve is given in Fig. 13.

The active actuator S_u and R_u can be a component or a system which consists of several components. Figure 14 is showing the components inside R_u of the piston-actuated active surge control. The actuator R_u consist of an I-element, a transformer, and an S_e-element. Detail description of the system is found in [7].

Figures 13 and 14 are examples of bond graph models of active surge control using close couple valve and piston-actuation, respectively. Derivation of dynamics equation from the both bond graph models will give the same results as in [14] and [7], respectively.

7. CONCLUSIONS AND FUTURE WORKS

It was shown how modeling of compression systems based on energy flow can be done by using the bond graph method. The dynamic equations of the system were derived in a straight forward manner using the bond graph model and resulting in the same equations as the Greitzer model. The energy flow shown in the bond graph gives a simple description of surge phenomenon. The main idea of several active surge control methods was described by two kind of bond graph models, i.e.: upstream energy injection and downstream energy dissipation. The bond graph was also used to derive dynamics equation of a compressor equipped with SAS and showing the the effect of the flow recycling.

Ignoring pressure drop along the pipeline is not realistic in the real long distance pipeline system. The pressure drop is due to interaction between fluid flow and pipeline. The interaction results in heat transfer from the fluid to the pipeline. It will be a future work to take account the pressure drop along the pipeline by adding thermodynamics effect in the bond graph model and modeling the system as a distributed system.

8. ACKNOWLEDGEMENT

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REFERENCES

[1] H.P. Bloch, "*A Practical Guide to Compressor Technology* 2^{nd} *Edition*", John Wiley & Sons, New Jersey, 2006.

[2] E.M. Greitzer, "Surge and Rotating Stall in Axial Flow Compressors, Part I: Theoretical Compression System Model", J. of Eng. for Power, vol. 98, 1976.

[3] E.M. Greitzer, "Surge and Rotating Stall in Axial Flow Compressors, Part II: Experiment Results and Comparison with Theory", J. Eng. for Power J., vol. 98, 1976.

[4] A.H. Epstein, J.E. Ffowcs Williams and E.M. Greitzer, "Active Suppression of Compressor Instabilities", AIAA 10th Aeroacoustic Conf., Seattle, 1985.

[5] F. Willems and B. de Jager, "Modeling and Control of Compressor Flow Instabilities", Control Systems Magazine, IEEE, vol. 19, 1999, pp 8-18.

[6] B. Bøhagen and J.T. Gravdahl, "Active Surge Control of Compression System Using Drive Torque", Automatica, vol.44, 2008, pp 1135-1140.

[7] N. Uddin and J. T. Gravdahl, "Active Compressor Surge Control Using Piston Actuation", Proc. ASME Dynamic Systems and Control Conference, Virginia, 2011.

[8] D. C. Karnopp, D. L. Margolis, and R. C. Rosenberg, "*System Dynamics: A Unified Approach - 2nd Edition*", John Wiley, New York, 1990.

[9] W. Borutzky, "Bond Graph Modeling and Simulation of Multidisciplinary Systems - An Introduction", J. Simulation Modeling Practice and Theory, vol. 17, 2009.

[10] W. Borutzky, "*Bond Graph Methodology Development and Analysis of Multidisciplinary Dynamics System Models*", Springer-Verlag, London, 2010.

[11] F.K. Moore and E.M. Greitzer, "A Theory of Post Stall Transients in an Axial Compressors System: Part I-Development of Equation", J. Eng. for Gas Turbines and Power, vol. 108, 1986, pp 68-76.

[12] J.T. Gravdahl and O. Egeland, "*Compressor Surge and Rotating Stall: Modeling and Control* ", Springer Verlag, London, 1999

[13] R. Kurz and R.C. White, "Surge Avoidance in Gas Compression System", J. Turbomachinery, vol. 126, 2004, pp 501-506.

[14] J.T. Gravdahl and O. Egeland, "Compressor Surge Control Using a Close Coupled Valve and Backstepping", American Control Conf. Proc., New Mexico, 1997.

[15] J. E. Ffowcs Williams, M. F. L. Harper, and D. J. Alwright, "Active Stabilization of Compressor Instability and Surge in a Working Engine", J. Turbomachinery, vol.115, no. 1, 1993, pp 68-75.

[16] J. E. Ffowcs Williams and X. Y. Huang, "Active Stabilization of Compressor Surge", J. Fluid Mech., vol. 204, 1989, pp 245-262.

[17] D. L. Gysling, J. Dugundji, E. M. Greitzer and A. H. Epstein, "Dynamic Control of Centrifugal Compressor Surge Using Tailored Structures", J. Turbomachinery, vol. 113, 1991, pp 710-722.

[18] F. Willems and B. de Jager "Active Compressor Surge Control Using a One-Sided Controlled Bleed/Recycle valve", Proc. The 37th IEEE Conference on Decision and Control, Florida, USA 1998, pp 2546-2551.

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