Abstract: This paper describes friction compensation based control of an industrial pneumatic actuator for the variable geometry turbocharger (VGT). Adaptive LuGre model for friction has been used in this work. Friction force that is responsible for hysteresis and hunting phenomenon in the position control is compensated with two different control strategies designed to improve low velocity position tracking. The first approach is related to the knocking the input signal to reduce position error. In the second approach, an adaptive LuGre model based friction observer is proposed for friction compensation. At the end both approaches are compared with the help of experiments on the actuator.

Keywords: Pneumatic actuators, Knocker, Hunting phenomenon, Friction observer, compensation

1. INTRODUCTION

The trend of engine performance enhancement has led to the idea of exhaust gas after treatment. This improves combustion efficiency and reduces NOx emission as well (Moulin and Chauvin (2009)). In the automotive industry, electric-pneumatic actuators are widely used to control the vanes of Variable Geometry Turbochargers (VGT). Due to high performance demands on the air path system, these actuators require very accurate control over their dynamics. Such accuracy is not achievable with simple control techniques in many cases, mainly due to nonlinearities in these systems.

The use of such an electro-pneumatic actuator for a VGT system has been discussed in this paper. In the turbocharger application pneumatic actuators are used to regulate the turbine speed by changing the angle of the vanes of VGT. The turbine speed is a function of the vane opening angle, as shown by Kolmanovsky et al. (1997). The rotation of the turbine turns the coupled compressor to produce boost pressure inside the intake manifold. A complete diagram of the VGT system is shown in the Fig.1.

A pneumatic actuator works by converting the force due to pressure difference into mechanical motion. The physical contact between mechanical parts induces friction in the system. Friction causes an important nonlinear hysteresis within pneumatic actuators. Besides friction, the variation of aerodynamic force on the vanes also contributes to the nonlinear behavior of the actuator. The aero-dynamic force is a reaction force exerted by the exhaust gases on the vane vanes, and hence on the actuator. This force is difficult to calculate analytically due to complex geometry of VGT system, and has not been discussed in detail in the available literature (to the best of our knowledge).

Friction is one of the main factors affecting the positioning accuracy of the actuator system. Specially, when controlled with common PID controllers, limit cycles can be observed in the actuator position Canudas de Wit et al. (1995). To avoid such problems, Laghrouche et al. (2007) and Brun et al. (1999) have proposed robust control laws for the control of pneumatic actuators. Among other techniques friction compensation based on friction models to eliminate the nonlinear effect of friction have also been considered in contemporary researches (See for example Canudas de Wit et al. (1995)). A survey of static and dynamic friction models, and existing techniques for its compensation has been presented by Helouvy et al. (1994). The observer based friction compensation techniques have been addressed in detail by Altizer (2004). Canudas de Wit and Lischinsky (1997) developed an adaptive LuGre model for friction to incorporate the the influences of external forces and temperature on the LuGre model.

The techniques for friction compensation discussed above are mainly model-based. Several comprehensive studies have been carried out in friction modeling. A brief review about different friction models has been presented in the article by Olsson et al. (1998). The LuGre model (extension of the Dahl model) manages to capture most of the nonlinear phenomena associated with friction. However, it does not predict the reversal point memory Altizer (2004).

In this work an adaptive LuGre friction model is presented for an electro-pneumatic VGT actuator. A complete model for actuator has already been developed and validated with LuGre model in the article Mehmod et al. (2010). The adaptation characterizes the variation in hysteresis due to aerodynamic force, as a function of diaphragm position and pressure at the inlet of turbine. It is shown in this work that linear PID controllers are not appropriate for control of such actuators and they result in limit cycles. Two solutions have been proposed to eliminate limit cycles with the help of friction compensators. In first approach, a knocker signal is introduced in the control signal. While, in the second approach, An observer based tracking with error correction is proposed for an adaptive LuGre model.

The paper is organized as follow. In section 2, a complete model of the pneumatic actuator with adaptive LuGre model is presented. In section 3, problem of the linear PID controller for position tracking are discussed and two solutions are proposed to compensate for the nonlinearities. In section 4, experiments are performed to implement friction compensation techniques.
2. ACTUATOR MODEL

This section describes the physical modeling of the actuator used for Variable Geometry Turbocarrier. A complete turbocarrier system is shown in Fig.1. Electro-pneumatic actuator consists mainly of two parts: Electro-pneumatic pressure converter (EPC) and a pneumatic actuator (air valve). The EPC regulates air pressure in the pneumatic actuator by either supplying atmospheric pressure or vacuum to the actuator chamber. Pressure difference in the pneumatic actuator produces a linear motion in its diaphragm. This motion is converted to rotary motion via rotating mechanism called unison ring that actuates all VGT vanes together. These guided vanes regulate exhaust gas flow to the turbocarrier. A detailed model for the electro-pneumatic VGT actuator has been presented in Mehmood et al. (2011). Pneumatic actuator is equivalent to a mass-spring-damper model. Pressure inside the actuator chamber causes the movement of the actuator diaphragm, which is connected to the vanes of turbocarrier. The following force balance equation for the diaphragm represents the actuator dynamics:

\[ m_d \dddot{x}_d = F_{act} - F_{act} - b_d \dot{x}_d - F_0 - k_{sm} x_d - F_f - F_{aero} \]  \( \text{(1)} \)

Forces \( F_{act} \), \( F_{act} \), and \( F_{sm} \) are the forces on the diaphragm due to actuator pressure, atmospheric pressure and equivalent spring-membrane force, respectively. \( x_d \), \( m_d \), and \( b_d \) are the position, the equivalent mass and damping coefficient of the diaphragm, respectively. \( F_0 \) is the pre-loaded spring force and \( k_{sm} \) is the spring and membrane constant that produce spring force. Forces acting on the diaphragm are defined below:

\[ F_{act} = p_{act} A_d \quad \text{and} \quad F_{act} = p_{atm} A_d \]

\( p_{act} \), \( p_{atm} \) and \( A_d \) are the actuator pressure, atmospheric pressure and diaphragm cross sectional area, respectively. \( F_f \) and \( F_{aero} \) are the friction and aerodynamic forces. These forces will be addressed in detail in the following section.

Pressure modeling in actuator chamber: Using ideal gas equation, pressure inside the actuator can be modeled as a function of the air gas constant, volume, temperature and mass inside the actuator chamber.

\[ p_{act} = \frac{RT_{act}}{V_{act}} m_{act} \]  \( \text{(2)} \)

The air mass inside the actuator chamber varies due to air flow, and hence the pressure. Taking the time derivative of the equation (2) we obtain

\[ \dot{p}_{act} = \frac{RT_{act}}{V_{act}} \dot{m}_{act} \]

The mass flow has been mapped as a function of the applied PWM signal \( U \) and the difference between the actuator pressure and the PWM dependent equilibrium pressure \( \delta p = p_{act} - p_{eq}(U) \). Fig.2 shows estimated mass flow as a function of applied signal \( U \) and pressure difference \( \delta p \). From curve fitting of this map, we obtain

\[ \dot{m}_{act} = \lambda(U, \delta p) = \phi(\delta p) + \sigma(\delta p) u \]

\( \phi \) and \( \sigma \) are fourth order polynomials. Next we will present adaptive friction modeling using the dynamic LuGre model.

Fig. 2. Cartography for mass flow as a function of applied signal and pressure difference

2.1 LuGre Friction Model

The LuGre friction model is a dynamic model that can simulate important friction phenomena, useful for control of mechanical systems. Some of the important phenomena are hysteresis, stick slip, Strubeck effect etc (See for example Alpeter (2004) and Olsson et al. (1998)). The LuGre model is based on the hypothesis that friction force arises due to microscopic irregularities between the surfaces in contact (Camadas de Wit et al. (1995)). These irregularities or asperities have been modeled as elastic bristles. Fig.3 shows model parameters and formation of bristles between the sliding surfaces. The dynamics of the LuGre model are given in equations set (5).

\[ F_f = \sigma_0 z + \sigma_1 \dot{z} + f(v) \]

\[ \dot{z} = v - \frac{\sigma_0}{g(v)} \]

\( \text{Where } F_f \) is the friction force and \( \sigma_0 \) and \( \sigma_1 \) are coefficients, \( f(v) \) is the velocity dependant damping force, that prevent the model to behave as a linear spring at small displacements. The LuGre model takes the Strubeck effect into consideration through the function \( g(v) \) Soren et al. (2007). Function \( g(v) \) is given below

\[ g(v) = \left( \frac{F_e}{F_s} + \left( F_s - F_e \right) \exp \left( -|v|/s_c \right) \right) \]

\( v_s \) is the sliding speed coefficient determines the stribeck curve and \( g(v) \) is such that \( F_s \leq g(v) < F_s \) (See Fig.3(d)). It has been observed that friction within the actuator varies with the influence of external aerodynamic force. Hence, an adaptive model is essential that incorporate these external perturbations.
2.2 Adaptive LuGre Model

Partially known LuGre model parameters do not depict the cause of increase in hysteresis due to aero dynamic force. Moreover, Aerodynamic force acts as a viscous friction force. Canudas de Wit and Lischinsky (1997) introduced adaptive mechanism to deal with structured normal force and temperature variation. An adaptive LuGre model is shown in equation (7).

\[ F_f = \hat{\theta}F_f \]  

Where \( \hat{\theta} \) represents variation in the model parameters. We obtained the adaptive friction model by considering that aero-dynamic force is the variation in the LuGre model parameters.

\[ F_f = F_f + F_{\text{aero}} = \left[ 1 + \frac{F_{\text{aero}}}{F_f} \right] F_f \]  

Where \( F_f = (\alpha_0 z + \alpha_1 \dot{z} + f(v)) \) is the friction force obtained from LuGre model. In the case under-discussion, adaptive model is proposed as function of aero dynamic force. This variation is also the function of turbine pressure \( p_{\text{trb}} \) and the VGT’s vanes angle \( C_r \).

\[ \hat{\theta} = \left[ 1 + \frac{f(p_{\text{trb}}, C_r) \text{sign}(x_d)}{F_f} \right] \]  

Fig.4 shows estimated friction for ramp signal when exhaust gas pressure at the inlet of the turbocharger is varied from 0mBar to 3000mBar. A complete state space model of the actuator with adaptive LuGre friction model is shown in the following section. Two methods for friction compensation have been discussed. Their effectiveness has been shown through experiments on the real systems.

3. FEEDBACK CONTROL WITH FRICTION COMPENSATION

It has been observed experimentally that friction gives rise to limit cycles in servo drives where the controller has integral action; for references see Canudas de Wit et al. (1995). This phenomenon is referred to as hunting. To further illustrate the properties of our friction model we will investigate its application to our actuator. First we will use it to show that it predicts limit cycle oscillations in our actuator with PID control. We will then use it to design observer based friction compensators and a knocker to achieve better tracking performance. Fig.5 shows hunting phenomenon in the experiments with the simple PID control. Whereas, integral action of the controller that cause limit cycles (hunting phenomenon), is shown in Fig.6. As we have mentioned above, the hunting effect is the result of the controllers integral action working on the stick-slip. Friction compensation is a viable option against the Hunting effect. In the following section, two methods for friction compensation have been discussed. Their effectiveness has been shown through experiments on the real systems.

3.1 Knocker

A very simple way to eliminate some effects of friction is to use a dither signal, i.e. a high frequency signal added to the control.
\[ \dot{x}_1 = x_2, \]
\[ \dot{x}_2 = \left( \left( A_d (p_{atm} - x_4) - b_d x_2 - (F_0 + k_{sm} x_1) - \dot{\theta} \left( \sigma_0 x_3 + \sigma_1 \dot{x}_3 + f(v) \right) \right) / m_d \right) \]
\[ \dot{x}_3 = x_2 - \sigma_1, \]
\[ \dot{x}_4 = \frac{RT_{act}}{V_0 - A_d x_1} \left( \lambda (U, \delta p) \right) + \frac{x_4}{V_0 - A_d x_1} A_d x_2 \]

(10)

3.2 Observer based friction compensation

The disadvantage of using dither is that controller output switches to compensate friction, which may cause to the fatigue of mechanical parts. Moreover, power dissipation increases with over-switching. It is of course more appealing to make a model-based control that uses the model to predict the friction to compensate its nonlinear behavior. Since the model is dynamic and has an unmeasurable state, an observer is necessary. Friction can be compensated using the observed state with tracking error based correction. Consider the friction model presented in eq. (5) and (7) and assume that the parameters \( \sigma_0, \sigma_1 \) and \( \sigma_2 \) and the function \( g(v) \) in the eq. (6) are known. The state \( z \) is, however, not measurable and hence has to be observed to estimate the friction force. For this we use a nonlinear friction observer given by

\[ \ddot{z} = v - \sigma_0 \frac{|v|}{g(v)} \dot{z} - ke \]

(12)

Where \( k \) is a positive constant and \( e = x_d - x_{ref} \), is the error between actual and reference position. The friction force obtained from the state observer is given by eq.(13). Where \( \dot{\theta} \) describes the adaption of external nonlinear forces.

\[ \dot{F}_f = \dot{\theta} \left( \sigma_0 \dot{z} + \sigma_1 \dot{z}^2 + f(v) \right) \]

(13)

A block diagram of the complete system with position control and correction based friction observer is shown in the Fig.8. Control input to the system is calculated using feedback linearization with following control law

\[ u = -H(s)e + \dot{F}_f + m_d x_{ref} + b_d x_{ref} + k_{sm} x_{ref} - F_{eq} \]

(14)

with

\[ F_{eq} = F_{aim} - F_{act} - F_0 \]

Where \( x_{ref} \) and \( H(s) \) represent the reference position and the linear PID controller (with parameters \( k_p, k_i \) and \( k_d \) as proportional, integral and differential gains for controller), respectively. If we chose \( H(s) \) such that the transfer function

\[ G(s) = \frac{\sigma_1 s + \sigma_0}{m_d s^2 + b_d s + k_{sm} + H(s)} \]

(15)
is strictly positive real function (SPR), then observer error \( \hat{F}_f = F_f - \hat{F}_f \), and the position error \( e = -G(s)\tilde{z} \), will asymptotically go to zero. Where \( \tilde{z} \) is the difference between the observer state and the real friction state. Using the Kalman-Yakubovitch Lemma and LaSalle’s theorem it is proven that above system is asymptotically stable, for \( \tilde{z} \to 0 \) this means that the error \( e \) will tend to zero. The proof of the theorem is detailed in Canudas de Wit et al. (1995) and Khalil (1995). The only disadvantage with observer based friction compensation is that we need a precise friction model which is not the case in knocker based compensation.

4. MODEL VALIDATION

The experiments were conducted on a the pneumatic valve and EPC used with the DV6TED4 commercial engine. Experimental results were obtained using the test bench shown in the Fig.9. The test bench is equipped with two pressure sensors (Druck PTX 610-1176). These sensors measure pressure of both actuator and reservoir chambers, with relative gauge pressure ranging from 0 to –1 Bar. Position of the diaphragm is measured using a potentiometer mounted on the pneumatic actuator. Data is acquired using Compact-RIIO modules (NI-9215) for analog input. The EPC, Pierburg device, is energized with H-Bridge DC servo drive module (NI-9505). Sampling frequency for the CompactRio is fixed to 1KHz. In Fig.10, the output results for the position tracking of the VGT pneumatic actuator are shown using the knocker as a compensation tool. It can be seen that the oscillation in the actuator have successfully been removed. The controller output is shown in the Fig.11. The error difference with the knocker can be seen in the Fig.12. The position error is reduced to 0.067mm which is ten times less than its value 0.82mm with PID. In Fig.13, the output results for the position tracking of the VGT pneumatic actuator are shown for friction observer. The controller output is shown in the Fig.14. The error difference with the friction observer can be seen in the Fig.15. The position error is also reduced in this case which shows the effectiveness of this approach. Experimental results shown above are performed on the real system with \( \dot{\theta} = 1 \). This implies that there is no external force due to exhaust gas pressure. For the time being, validation of controller with external perturbation forces is not possible because of test bench limitations. So, to validate our controller, the aerodynamic force is induced in the system, using adaptive LuGre model when exhaust gas pressure at the inlet of turbine is kept to be 1500 mBar. The estimated aerodynamic force is added as an external perturbation force. So, the performance of the controller with external force is shown in Fig.16 for both knocker and friction observer. Simulation results prove that model based friction observer is more robust and give better performance as compared to the knocker.

5. CONCLUSION

Results presented in this paper have been obtained through experimentation on a pneumatic actuator used for VGT system. Considering the effects of aerodynamic force an adaptive LuGre model is presented. The Hunting effect caused by friction stick-slip and controller integral action, have been explained and illustrated with simulation results. Friction compensation
Fig. 13. Friction compensation with feedback observer

Fig. 14. Control signal applied to the actuator

Fig. 15. Error in the position tracking with and without feedback observer

Fig. 16. Simulation results with friction compensation techniques when exhaust gas produces aerodynamic force


REFERENCES