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Control of Flow Rate in Pipeline Using PID Controller

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Abstract— In this paper a PID controller is utilized in order to control the flow rate of the heavy-oil in pipelines by controlling the vibration in motor-pump. A torsional actuator is placed on the motor-pump in order to control the vibration on motor and consequently controlling the flow rates in pipelines. The necessary conditions for asymptotic stability of the proposed controller is validated by implementing the Lyapunov stability theorem. The theoretical concepts are validated utilizing numerical simulations and analysis, which proves the effectiveness of the PID controller in the control of flow rates in pipelines.

Keywords—component, formatting, style, styling, insert

I. Introduction

Classic PID approaches as well as controllers are updated and expanded during the years, from the primary controllers on the basis of the relays as well as synchronous electric motors or pneumatic or hydraulic systems to current microprocessors. Currently, many techniques for the tuning as well as design of PI and PID controllers are proposed [1]. The method proposed in [2] is the most widely utilized PID parameter tuning methodology in chemical industry and is considered as a conventional technique. Basilio and Matos [3] suggested a new method with less complexity in order to tune the parameters of PI controllers of the plant with monotonic step response. The methodology of internal mode principle is utilized in [4] and [5] in order to extract the gains of PID and PI controllers. Exhaustive investigation [6-8] revealed that the outcomes of P control are very sensitive to the sensing location as well as the quantity of phase shift. By suitable selections of these variables, the P control can be completely efficient in annihilating the vortex shedding or minimizing its strength. Furthermore, it is demonstrated that the increment in the proportional gain can results in the decrement of the velocity fluctuations in the wake and the strength of vortex shedding. Nevertheless, a large gain causes instability in the system [9-10].

In order to implement the control law, the primary step is to determine a desired output response of a particular system to an arbitrary input over a time interval, that can be carried out by system identification [11]. Generally, it is feasible to generate a model on the basis of a complete physical illustration of the system. Nevertheless, this model contains complexity, also has high calculation costs [12]. The secondary step is to define the parameters of the PID controller. There exist various literatures associated with the methodologies for tuning of PID controllers applied in various controller structures [13].

Flow control is a major rapidly evolving field of fluid mechanics. There have been various concepts of flow control in drag reduction, lift enhancement, mixing enhancement, etc. [14-16]. Fadlun et al. [17] implemented the concept of [18] to a finite-difference methodology where a staggered grid is used. In [19] a digital pulse feedback flow control system utilizing microcontroller as well as feedback sensing element is developed. Surprisingly, even though the flow control methods are widely spread, investigating the stability of the control system is very rare. In [20-21] the P, PI and PID controls are proposed for flow over a cylinder with a Reynolds number blow the 200. The aim of control in these studies is the attenuation or annihilation of vortex shedding behind a bluff body. The only investigation on the implementation of PI and PID controls to the flow over a bluff body is carried out in [21]. In the recent years artificial neural networks (ANNs) have become popular and many interesting ANN applications have been reported in engineering field[22-30]. In [31] the authors have develop flow modeling and optimization methods using neural networks. In [32] a neural network controller approach is proposed in order to control the pump flow rate. Surprisingly, even though the flow control methods are widely spread, investigating the stability of the control system is very rare.

This paper deals with the modeling and control of flow rate in heavy-oil pipelines. For this aim, the PID control algorithm is utilized to control the flow mechanism in pipelines. A torsional actuator is placed on the motor-pump in order to control the vibration on motor. The stability of the PID controller is verified using Lyapunov stability analysis. The stability analysis of the controller results in a theorem, which validate that the system states are bounded. The theoretical concepts are validated using numerical simulations and analysis, which proves the effectiveness of the PID controller in the control of flow rates in pipelines. Further, it is the first attempt to place a torsional actuator on the motor-pump in order to control the vibration on motor and hence control the flow rates in pipelines.

This paper is structured as follows. Firstly, in Section II the pump model system is established. The PID control method is described in Section III. In this section, sufficient conditions for the controller under the Lyapunov stability theorem are designed. A numerical example is presented in Section IV to illustrate the results. Finally, the conclusions are provided in Section V.

II. Materials and Methods for Modelling of the System

Flow control loop system is basically a feedback control system. The structure of the pump model system is shown in Fig.1. which is an open loop system. If there is unwanted vibration in the motor, the stability of the flow rate will hamper. Therefore, it is important to change the open loop system to a closed loop system by implementing a controller so as to control the stability of flow rate by controlling the vibration in the motor.

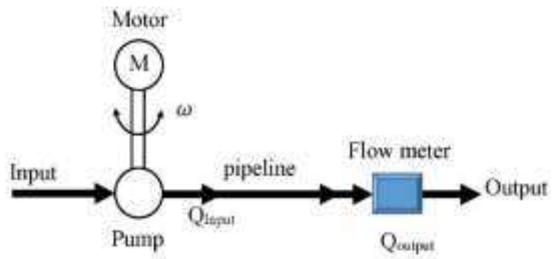

Fig.1. Scheme of open loop model

A. Modelling of the pipeline

The proposed model consists of induction motor, which causes a rotation in pump and consequently can lead to flow of heavy-oil in pipelines as shown in (Fig.1). This flow model can be illustrated in the form of partial differential equation (PDE) [33].

The linear global theory associated with flow stability is rooted in eigendecompositions of the linearized flow operators. The direct as well as adjoint eigendecompositions associated with these kinds of operators generate information related to the stability of the operator, the acceptance of initial conditions as well as external forcing, also the sensitivity to spatially localized disturbances.

From the viewpoint of incompressible, constant-density, constant-viscosity flows associated with Newtonian fluids, the nonlinear Navier–Stokes equation is defined for a nondimensional velocity field $(x,t): \mathbb{R}^n \times \mathbb{R} \to \mathbb{R}^n$, pressure field $p(x,t): \mathbb{R}^n \times \mathbb{R} \to \mathbb{R}^n$ and Reynolds number $Re > 0$ as below equation,

$$\frac{\partial u}{\partial t} = -\frac{\nabla p}{\rho} - u.\nabla u + F_f \tag{1}$$

Where,

ρ is the density in $\frac{kg}{m^3}$

u is the flow velocity in $\frac{m}{s}$,

∇ is the divergence,

p is the pressure in $\frac{kg}{m.s^2}$,

t is time in s,

F_f is termed as the summation of external force and body forces

By implementing the mass balance into the Equation (1) the following is concluded [34],

$$\nabla.u = 0 \tag{2}$$

Equation (1) can be rewritten as,

$$\frac{\partial u}{\partial t} = -\frac{1}{\rho}\frac{\partial p}{\partial x} + F_f \tag{3}$$

Let $\frac{\partial p}{\partial x}$ be the change of pressure in two different points, moreover for achieving a numerical stability of computation, it is essential to partition pipeline into various segment (generally identical), hence the flow in pipeline can be stated as,

$$\frac{\partial u_i}{\partial t} = -\frac{1}{\rho L}(p_i - p_{i-1}) + F_f, \quad i = 1, \dots, n \tag{4}$$

where L is taken to be the distance between two sections. Now let,

$$\alpha p_i = p_{i-1}, \quad i = 1, \dots, n \tag{5}$$

where α is termed as the coefficient of pressure changes in sections

$$\frac{\partial u_i}{\partial t} = -\frac{1}{\rho L}(1-\alpha)p_i + F_f, \quad i = 1, \dots, n \tag{6}$$

The loss of the friction under the conditions of laminar flow conforms with the Hagen–Poiseuille equation [35-36]. For a circular pipe having a fluid of density (ρ) and Kinematic viscosity υ, the hydraulic slope F_f can be described as,

$$F_f = \frac{64}{Re}\frac{u^2}{2gD} + F_b = \frac{64\upsilon}{2g}\frac{u}{D^2} + F_b \tag{7}$$

where g is the gravity, D is the diameter of the pipes and F_b is the shape force vector in pipes. By substitution of (7) in (6), the following equation can be extracted,

$$\frac{\partial u_i}{\partial t} = -\frac{1}{\rho L}(1-\alpha)p_i + \frac{64\upsilon}{2g}\frac{u}{D^2} + F_b, \quad i = 1, \dots, n \tag{8}$$

The pressure $p(x,t)$ in the pipeline can be described as,

$$p = \frac{F}{A} \tag{9}$$

where F is the force inside the pipeline, also A is the cross section in the pipe.

By taking into consideration $F = ma = m\frac{d^2x}{d^2t}$, and $u = \frac{dx}{dt}$, also by substitution of (9) in (8) the following equation is extracted,

$$\frac{\partial}{\partial t}\frac{\partial x_i}{\partial t} = -\frac{(1-\alpha)}{\rho AL}\frac{\partial^2 x_i}{\partial t^2} + \frac{64\upsilon}{2gD^2}\frac{\partial x_i}{\partial t} + F_b \tag{10}$$

Therefore,

$$-\frac{(1-\alpha)+\rho AL}{\rho AL}\frac{\partial^2 x_i}{\partial t^2} + \frac{64\upsilon}{2gD^2}\frac{\partial x_i}{\partial t} + F_b = 0 \tag{11}$$

If an external force, f_p generates by pump, (11) can be rewritten as follows,

$$\Gamma\ddot{x} + \Phi\dot{x} + f_b = f_p \tag{12}$$

where $x \in \mathbb{R}^2$, $\Gamma \in \mathbb{R}^{2\times2}$, $\Phi \in \mathbb{R}^{2\times2}$, $f_b = [f_{b1}\ f_{b2}]^T \in \mathbb{R}^{2\times1}$, $f_p = [f_{p1}\ f_{p2}]^T \in \mathbb{R}^{2\times1}$.

Since in this work two pipelines are used, (12) can be rewritten as,

$$\begin{aligned} \gamma_1\ddot{x}_1 + \varphi_1\dot{x} + f_{b1} = f_{p1} \\ \gamma_2\ddot{x}_1 + \varphi_2\dot{x} + f_{b2} = f_{p2} \end{aligned} \tag{13}$$

Since the pump is supplying pressure to the pipes for the maintaining the flow rates, so the pipes will have the same external force, $f_{p1} = f_{p2}$.

B. Modelling of the actuator

In order to reduce the vibrations of the motor caused by the external forces (f_p) a torsional actuator is placed on the motor, see Figure. 2.

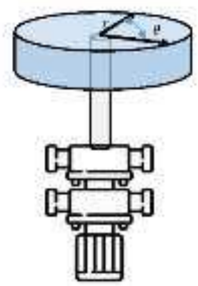

Fig. 2. Torsional actuator with motor-pump arrangement

The motor and the pump are interconnected with the help of a shaft. The main purpose of the motor is to drive the pump. The pump with the help of the motor initiate a flow of fluid in the pipe. Any unwanted vibration in the motor will result in the vibration in the pump, which will result in improper flows in the pipe. Therefore, it is important to control the vibration in the motor, so as to control the vibration in the pump for making a stable flow of fluid in the pipelines. For this purpose, a torsional actuator having a motor and disk arrangement as shown in figure 3 is placed on the top base of the pump The main intention of the torsional actuator is to control the vibration on motor and consequently controlling the flow rates in pipelines.

The inertia moment of the torsional actuator is defined as,

$$J_t = m_t r_t^2 \tag{14}$$

where m_t is considered to be the mass of the disc, and r_t is the radius of the disc. The torque produced by means of the disc is defined as

$$u_\theta = J_t(\ddot{\theta}_t + \ddot{\theta}) \tag{15}$$

where $\ddot{\theta}$ is taken to be the angular acceleration of the motor and $\ddot{\theta}_t$ is taken to be the angular acceleration of the torsional actuator.

In order to reduce the torsional response, the directions of $\ddot{\theta}_t$ as well as $\ddot{\theta}$ are taken to be different. The friction of the torsional actuator is defined as [37],

$$f_d = c\dot{\theta}_t + F_c \tanh(\beta\dot{\theta}_t) \tag{16}$$

where c is taken to be the torsional viscous friction coefficient, β is the motor constant, F_c is taken to be the coulomb friction torque, also tanh is considered to be the hyperbolic tangent which depends on β and motor speed. The final torsion control is expressed as,

$$u_\theta = J_t(\ddot{\theta}_t + \ddot{\theta}) - f_d \tag{17}$$

C. Modelling of the pump

The general equation of the pump supplying pressure to the pipe for flow control can be demonstrated as [38],

$$T\dot{\omega} = \tau - (\tau_p - n\omega) \tag{18}$$

where,

ω is angular velocity,

τ is motor torque,

τ_p is frictional torque of the motor,

n is load constant,

T is rotations inertia time constant.

The equation (18) can be modified as follows,

$$\ddot{x}_p = \frac{\tau - (\tau_p - n x_p)}{T} \tag{19}$$

Where $\ddot{x}_p$ is the flow acceleration of the pump. Since $\tau, T, \tau_p,$ and n are known quantities of pump so $\ddot{x}_p$ can be estimated.

The external force generated by the pump is

$$f_p = m_p \ddot{x}_p \tag{20}$$

where $\ddot{x}_p$ is the acceleration of the motor and m_p is volumetric mass of the pump.

The shape force vector f_b can be modeled as a linear or a nonlinear model.

From (12), by considering shape force vector f_b as non-linear model, the following analysis is illustrated:

In simple non-linear case, (12) becomes

$$\Gamma\ddot{X} + \Phi\dot{X} + f_b = f_p \tag{21}$$

where f_b is taken to be non-linear.

III. THE TUNING METHOD BASED ON PID CONTROLLE

Since 1700's the control of continuous process has been carried out by utilizing feedback loop. System with feedback control contains drawback which is related to the instability of the system. In order to resolve this problem an appropriate controller should be chosen and also it must be ideal for the monitoring system. The proportional feedback control is uncomplicated and relatively easy to implement. Nevertheless, its outcome is completely sensitive to the sensing location as well as feedback gain. It is concluded from the control theory that these drawbacks of the P control should be overcome by adopting I as well as D controls. Nevertheless, there exist very few studies investigating the application of the PID control for fluid-mechanics problems. In addition, there are a very limited number of studies dealing with the P control and PI controller for pipeline. Due to this lack of investigations, this paper aims to develop a PID control for flow rate in the pipeline.

The control mechanism is demonstrated in Fig. 3 which shows the entire control process of the flow rate of the heavy-oil in pipelines.

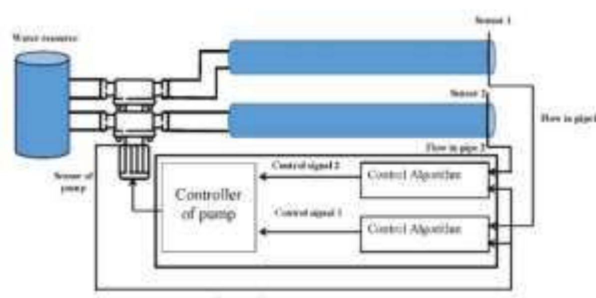

Fig. 3. Structure of system

The PID control is considered as a control law in which the existence of output for feedback is essential. This practical control method is widely utilized in the control society. In the PID control, the controller is made of a simple gain (P control), an integrator (I control), a differentiator (D control) or some weighted composition of these possibilities [39], see Figure 4.

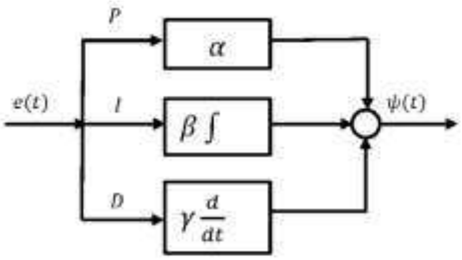

Fig. 4. PID controller

The PID control is expressed as,

$$\Psi(t) = -\kappa_p e(t) - \kappa_i \int_0^t e(t) d\tau - \kappa_d \dot{e}(t) \tag{22}$$

where k_p, k_i, as well as k_d are positive definite and k_i is the integration gain. For the flow control, X^d is desired reference and also $X^d = \dot{X}^d = 0$. Hence, equation (22) is rewritten as below,

$$\Psi(t) = -\kappa_p X - \kappa_i \int_0^t X d\tau - \kappa_d \dot{X} \tag{23}$$

For analyzing the PID controller, equation (22) can be stated as below,

$$\begin{gathered} \Psi(t) = -\kappa_p X - \kappa_d \dot{X} - \vartheta \\ \vartheta = \kappa_i \int_0^t X d\tau \ , \quad \vartheta(0) = 0 \end{gathered} \tag{24}$$

The closed-loop system of eq. (12) along with the PID control (eq. (23)) is demonstrated as below,

$$\begin{gathered} \Gamma \ddot{x} + \Phi \dot{x} + F_g = -\kappa_p X - \kappa_d \dot{X} - \vartheta \\ \dot{\vartheta} = \kappa_i X \end{gathered} \tag{25}$$

In matrix form, the closed-loop system is defined as,

$$\frac{d}{dt}\begin{bmatrix} \vartheta \\ X \\ \dot{X} \end{bmatrix} = \begin{bmatrix} \kappa_i X \\ \dot{X} \\ -\Gamma^{-1}(\Phi \dot{X} + F_g + k_p X + k_d \dot{X} + \vartheta) \end{bmatrix} \tag{26}$$

Here the stability of the PID control demonstrated by eq. (23) is analyzed. The equilibrium of eq. (26) is presented by $[\vartheta \quad X \quad \dot{X}] = [\hat{\vartheta} \quad 0 \quad 0]$. As at equilibrium point $X = 0$ as well as $\dot{X} = 0$, the equilibrium is $[f(0), \quad 0, \quad 0]$. For moving the equilibrium to the origin, the following is defined,

$$\hat{\vartheta} = \vartheta - f(0) \tag{27}$$

Therefor, the final closed-loop equation is defined as,

$$\begin{gathered} \Gamma \ddot{X} + \Phi \dot{X} + F_g = -\kappa_p X - \kappa_d \dot{X} - \vartheta + f(0) \\ \vartheta = \kappa_i x \end{gathered} \tag{28}$$

For analyzing the stability of equations (28), the following properties are required,

Property 1. The positive definite matrix Γ should satisfy in the below condition

$$0 < \lambda_{min}(\Gamma) \le \|\Gamma\| \le \lambda_{Max}(\Gamma) \le \bar{\gamma} \tag{29}$$

such that $\lambda_{min}(\Gamma)$ as well as $\lambda_{Max}(\Gamma)$ are considered as the minimum and maximum eigenvalues of the matrix Γ , respectively also $\bar{\gamma} > 0$ is taken to be the upper bound.

Property 2. f is taken to be Lipschitz over $\tilde{x}$ and $\tilde{y}$ if

$$\|f(\tilde{x}) - f(\tilde{y})\| \le \Omega \|\tilde{x} - \tilde{y}\| \tag{30}$$

As F_g is first-order continuous functions also satisfies in Lipschitz condition, Property 2 is hereby established.

The lower bound of F_g can be calculated as below,

$$\int_0^t f dx = \int_0^t F_g dx + \int_0^t d_u dx \tag{31}$$

The lower bound of $\int_0^t F_g dx$ is stated as $-\hat{F}_g$ also $\int_0^t d_u dx$ as -$\hat{D}_u$. Therefore, the lower bound of Ω is defined as,

$$\Omega = -\hat{F}_g - \hat{D}_u \tag{32}$$

The stability analysis of PID control approach is given by the below mentioned theorem.

Theorem. By taking into consideration the structural system of equation (12) controlled by the PID control approach of equation (22), the closed-loop system of equations (28) is taken to be asymptotically stable at the equilibriums $\left[\vartheta - f(0), X, \dot{X}\right]^T = 0$, if the following gains are satisfied,

$$\begin{gathered} \lambda_{min}(\kappa_d) \ge \frac{1}{4}\left(\frac{1}{3}\lambda_{min}(\Gamma)\lambda_{min}(\kappa_p)\right)^{1/2}\left[1 + \frac{k_e}{\lambda_{Max}(\Gamma)}\right] - \lambda_{min}(\Phi) \\ \lambda_{Max}(\kappa_i) \le \frac{1}{6}\left(\frac{1}{3}\lambda_{min}(\Gamma)\lambda_{min}(\kappa_p)\right)^{1/2}\left[\frac{\lambda_{min}(\kappa_p)}{\lambda_{Max}(\Gamma)}\right] \\ \lambda_{min}(\kappa_p) \ge \frac{3}{2}[\Omega + \Xi] \end{gathered} \tag{33}$$

IV. Numerical results

For the numerical analysis purpose and for the validation of the novel control strategy, the various parameters associated with the flow control are described in Table1:

TABLE 1. PARAMETERS ASSOCIATED WITH THE FLOW CONTROL

ρ	1240	kg/m^3
υ	1.604×10^{-3}	m^2/s
g	9.81	m/s^2
L	100	m
α	0.95	-
D	0.05	m
A	1.96×10^{-3}	m^2
W	30	kg
r_t	0.3	m
m_t	1.5	kg

The implemented software in this paper is Matlab/ Simulink. Simulations are presented to show that the motor vibration can be attenuated to a significant level by using the torsional actuator with the developed controllers thus validating the effectiveness of the proposed control approach using PID controllers. A simulation period of 20s is considered for evaluation. For the simulation purposes, the weight of the torsional actuator is considered to be 5% of the motor and pump weight in combination.

The Theorem proposed in this paper generates sufficient conditions for the minimum amounts of the proportional as well as the derivative gains. This Theorem validates that both proportional and derivative gain must be positive as negative gains can make the systems unstable. The PID gains are selected within the stable range by the stability analysis in order to ensure the efficiency.

Since the maximum flow rate of the pipeline is $13\,m^3/s$ so the other non linear force associated with Ω has to be less than $13\,m^3/s$. Hence, we select $\Omega = 13\,m^3/s$, and

$$\lambda_{min}(\Upsilon_1) = 1.0002\,, \lambda_{min}(\Upsilon_2) = 1.000206, \lambda_{min}(\Xi_1) = 2.09\,, \lambda_{min}(\Xi_2) = 2.09$$

From Theorem 2, we use the following PID gains

$$\lambda_{min}(\kappa_{p1}) \geq 453, \lambda_{min}(\kappa_{d1}) \geq 8, \lambda_{Max}(\kappa_{i1}) \leq 928 \quad (34)$$

Also for pipe 2 we have,

$$\lambda_{min}(\kappa_{p2}) \geq 453, \lambda_{min}(\kappa_{d2}) \geq 8, \lambda_{Max}(\kappa_{i2}) \leq 928 \quad (35)$$

From ranges mentioned by eq. (34) and (35), the best value of gains are

$$\kappa_{p1} = 458, \kappa_{d1} = 110, \kappa_{i1} = 650, \kappa_{p2} = 480, \kappa_{d2} = 115, \kappa_{i2} = 645$$

Two subsystem blocks of milling model, one in the absence of control mechanism and another with control mechanism are generated for comparing the outcomes. The flow rate from the pump is the input to the flow model. Numerical integrators are utilized in order to calculate the velocity as well as the position from the acceleration signal. The control signal from the controller subsystem block is given to the Torsional actuator simulation block in order to produce the essential control forces.

Figure 5, 6 represents the vibration attenuate in motor. From these Figures, it can be concluded that PID controller is performing good in minimizing the vibration. Figure 7, 8 ero and are stable, which proves the effectiveness of PID controller.

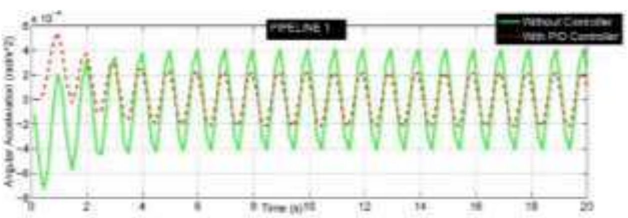

Fig. 5. Comparison of motor vibration attenuation using PID controller for pipeline 1

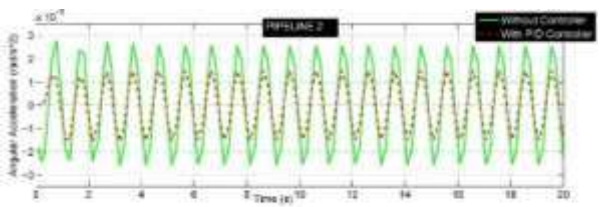

Fig. 6. Comparison of motor vibration attenuation using PID controller for pipeline 2

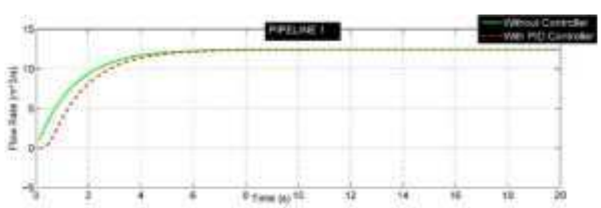

Fig. 7. Stability of flow rate with using of PID controller in pipeline 1

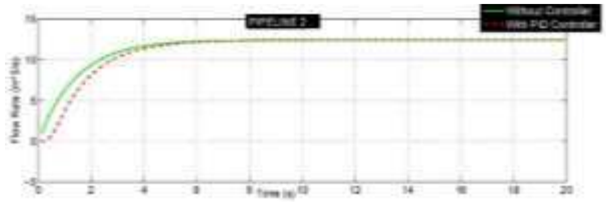

Fig. 8. Stability of flow rate with using of PID controller in pipeline 2

V. CONCLUSIONS

In this paper, a novel active control strategy for the attenuation of motor vibration is proposed which consequently controls the flow rate in heavy-oil pipelines. The important theoretical contribution associated with the stability analysis for the PID controller is developed. The required stability conditions are obtained for the purpose of tuning the PID gains. By utilizing Lyapunov stability analysis, the sufficient conditions for the minimum amounts of the proportional, integrator as well as the derivative gains are obtained. The numerical simulation and analysis validates the effectiveness of PID controllers in the minimization of motor vibration to control the flow rate in pipelines. The main contributions of this paper are:

1) In this work, the stability of PID controller is validated which has not been given importance in earlier researches considering the flow rate control.

2) The technique of using torsional actuator on the motor-pump arrangement is entirely a new concept.

Future work is intended towards the development of the experimental setup for further investigation and the improvement of the controller by fuzzy methods.

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