Proceedings of IPC2006
6th International Pipeline Conference
September 25-29, 2006, Calgary, Alberta, Canada

IPC2006-10260

OPTIMIZING THE ACTIVE SPEED CONTROL UNIT FOR IN-LINE INSPECTION TOOLS IN GAS

Dr. Florian Rahe
ROSEN Technology & Research Center
Physics Department
Am Seitenkanal 8
49811 Lingen (Ems)
Germany
Email: FRahe@RosenInspection.net

ABSTRACT

The simulation and optimization of an active speed control unit for an intelligent in-line inspection tool is presented. A simple model was developed describing the dynamics of such a tool in a gas pipeline. The dynamics are represented by a set of coupled differential equations incorporating tool velocity, pressure, differential pressure over the tool and bypass area. These equations were solved numerically by implementation in a simulation environment. Control rules for an active speed control unit were integrated in this model to simulate the effect of regulation activities on the reactions of the tool. The rules were chosen to represent the capabilities of a real unit (electronics and mechanics). The results of the simulation are compared with the behavior of a tool in a 40" gas pipeline and the regulation algorithm is optimized with respect to the capabilities of the existing control unit.

INTRODUCTION

Pipelines are used all around the world for transporting different kinds of media like natural gas, crude oil and refined products. For effective operation the pipelines have to be cleaned from time to time with special devices to remove debris or residual products. These devices have been developed further to intelligent tools providing nondestructive testing (NDT) of the pipeline. These techniques include geometric inspections, like gauging the internal bore or determining the position of the pipeline with the help of an inertial measurement unit, and further NDT techniques. These techniques include Magnetic Flux Leakage (MFL), Electro-Magnetic Acoustic Transducer (EMAT) and Ultrasonic Testing (UT). Despite all the differences, these techniques have one thing in common: To get accurate results the inspection tool should move at a moderate constant speed. The optimal speed range for most tools is in the range of $0.5-4\ \mathrm{m/s}$ ($1.1-8.9\ \mathrm{mi/h}$). The flow velocity in a pipeline can vary over a wide range from nearly $0\ \mathrm{m/s}$ ($0\ \mathrm{mi/h}$) to more than $10\ \mathrm{m/s}$ ($22.4\ \mathrm{mi/h}$) especially in gas pipelines. Because of the pressure loss very high flow velocities are typical for gas pipelines. Even higher speeds can be observed and therefore active speed control units (SCU) are developed, which allow bypassing some amount of media through the body thus reducing the velocity of the tool [1]. Additionally in gas pipelines velocity spikes can occur and one possible solution to overcome these excursions is by a fixed or a variable bypass [2]. ROSEN uses active SCU for their larger tools and optimizes them with finite element methods (see figure 1) so they can bypass up to $7\ \mathrm{m/s}$ ($15.7\ \mathrm{mi/h}$) of gas flow through a 40" CDP (corrosion detection tool, MFL technique). This allows a wide range for regulation of the tool speed.

As a first approach for a speed control algorithm a simple threshold discrimination was used. If the tool is running 10% faster than the desired velocity the valve of the SCU shall be opened. If the tool is running 10% slower than the desired speed the valve shall be closed. Within this area of $\pm 10\%$ of the desired speed no regulation shall be done. The velocity was aver-

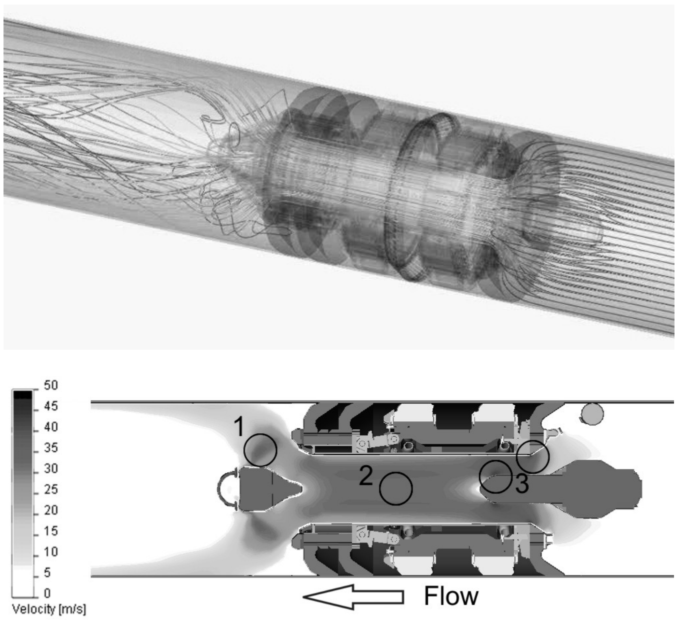

Figure 1. ON TOP: FLOW PATHS OF GAS THROUGH THE OPTIMIZED GEOMETRY OF A 40"CDP WITH SPEED CONTROL. AT THE BOTTOM: DIFFERENTIAL VELOCITIES OF GAS BYPASSING THE TOOL. AT THE OUTLET (1) THE VELOCITY IS ABOUT $35\ \mathrm{m/s}$ (78.3 mi/h), IN THE CENTER (2) $30\ \mathrm{m/s}$ (67.1 mi/h) AND AT THE EDGES OF THE INLET (3) $45\ \mathrm{m/s}$ (100.7 mi/h).

aged over 2.5 s so as not to regulate against short disturbances like welds. Frequently the threshold condition is checked. This procedure worked with the first speed control tools, since the regulation range was lower and it was in most cases regulated to maximum bypass. With the new optimized mechanical design the regulation range of the valve is higher. In a gas pipeline the SCU started to let the tool speed oscillate around the desired velocity. A part of the velocity profile can be seen in figure 3. Even if these fluctuations are not critical, this behavior is unwanted and an optimized speed algorithm had to be developed.

Attempts have already been made in the literature to propose an optimized speed regulation algorithm [3]. They require position, velocity, the velocity of bypass flow through the tool, differential pressure. The regulation parameter in that setup, which can be identified as a measure of the bypass area, can be changed from one state (i.e. fully closed) to the other state (i.e. fully opened) within a second. The proposed procedure based on these assumptions shows a remarkable performance, but the assumptions do not reflect the actual physical conditions. With the tool design, which is actual realized, the speed and the differential pressure will be the only measurement values, which are known by the speed control algorithm. Also, the valve needs in the given example about 13 s for a full operation (changing from closed to fully opened) which is not adresses by these algorithms. Additionally the calculations, which are needed for such an algorithm, require too much calculation time for the processors. These restrictions prevent a useful implementation of the algorithms presented in the literature sofar.

To develop a speed control algorithm fitting the above mentioned restrictions a model had to be developed to simulate the tool dynamics. All of the existing models are quite complex and have to perform numerical integrations and finite difference methods at each time step [3–7]. This means a high amount of implementation and computation time. Since a fast solution was needed, a much simpler model was developed to reproduce the observed speed oscillations and to propose and generate an optimized speed control algorithm.

MODELING

In the Following a simple model containing the most important parameters will be set up. The scheme can be seen in figure 2. This model contains the amount of medium (number of gas molecules) pumped in and out the pipeline per time ($\dot{N}_{P,up}$, $\dot{N}_{P,down}$), the pipeline parameters (volume, pressure and contained media) up- and downstream (V_{up}, p_{up} and N_{up} for upstream parameters), the tool velocity (v_T) and the amount of bypassed media per time ($\dot{N}_{BP}$).

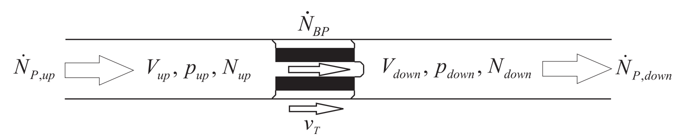

Figure 2. MODEL FOR A TOOL WITH BYPASS IN GAS PIPELINE.

For modeling it is assumed that the tool is only affected by a certain control volume around it and the remainder can be neglected. Additionally all pressure changes shall be instantaneous. This will make it possible to implement the model without the need of finite difference methods for the calculation of the media dynamics up- and downstream of the tool. For the modeling the following assumptions were made:

1. the gas is ideal
2. the flow is one phase
3. the pipeline diameter is constant
4. the friction is based on constant static and constant sliding friction
5. the tool moves horizontally
6. the pressures in a control volume up- and downstream of the tool shall be instantaneous

For a simple model it can be assumed that the temperature will be constant, since earlier investigations [5] have shown that results from isothermal models differ only very little from results obtained by non isothermal models. Therefore the ideal gas law can be used:

$$p \cdot V = N k_B T =: M \tag{1}$$

Where M can be seen as a measure of how much media is contained in the volume V with pressure p. With a constant temperature T the expression $N k_B T$ can be directly replaced by M with no constraints.

Gas flow

For modeling the gas flow it is assumed that the medium flow v_F outside a control volume around the tool is constant. This simplification can be justified for short sections of the pipeline relevant for the tool dynamics. The constant medium flow with the constant overall pipeline volume will result in an overall average pressure of p_0. If A is the cross section of the pipeline the amount of pumped medium is:

$$\dot{M}_{P,up} = -\dot{M}_{P,down} = p_0 \cdot v_F \cdot A = const. \tag{2}$$

Since we are interested in the tool behavior in a straight and horizontal section of a pipeline, the pipeline before and after the tool can be treated as homogeneous. For the simplicity of the model only a fixed control volume traveling with the tool will be considered and it is assumed that the remainder can be neglected. The pressure changes in the control volume upstream (V_{up}) and downstream (V_{down}) of the tool was assumed to be instantaneous.

The result is the following set of mass conservation equations in these two volumes:

$$\dot{M}_{up} = \dot{M}_{P,up} - p_0 \cdot v_T \cdot A - \dot{M}_{BP}(p_0, \Delta p) \tag{3}$$

$$\dot{M}_{down} = \dot{M}_{P,down} + p_0 \cdot v_T \cdot A + \dot{M}_{BP}(p_0, \Delta p) \tag{4}$$

Here $\Delta p(t) = p_{up} - p_{down}$ is the pressure difference over the tool, which has an influence on the amount of media passing the bypass ($\dot{M}_{BP}(p_0, \Delta p)$). v_T is the velocity of the tool and determines via the pipeline cross section and the overall pressure the amount of medium transported due to the tool movement.

The mass conservation equations are coupled via the ideal gas law (1) with the fixed control volume around the inspection device. The upstream and downstream part of the volume are coupled via the media pasing by ($\dot{M}_{BP}(p_0, \Delta p)$). It is assumed that the bypass is an opening in the tool with a certain area A_{BP}. Formula (5) describes the dependency of the velocity v_{BP} in such a bypass on the differential pressure Δp over such a bypass:

$$\Delta p = \frac{\xi}{2} \rho {v_{BP}}^2 \tag{5}$$

Here ξ is the pressure drop coefficient of the valve and ρ is the medium density. This equation is mainly used to calculate the differential pressure over the tool from the velocity difference of the medium and the tool [3, 5–7]. This gives the impression that the differential pressure is caused by the velocity difference between tool and medium. This may lead to a wrong interpretation, since there can still be a differential pressure even if there is no velocity difference. Here the formula is used to calculate the velocity of the medium through the bypass hole:

$$v_{BP} = \sqrt{\frac{2|\Delta p|}{\rho \xi}} \tag{6}$$

In this case formula (5) is interpreted as representing the differential pressure over the tool, which causes the medium to flow through the bypass. The amount of bypassing gas per time $\dot{M}_{BP}$ can then be calculated by:

$$\begin{aligned} \dot{M}_{BP}(p_0, \Delta p) &= p_0 \left\{ v_T + v_{BP}(\Delta p) \right\} A_{BP} && (7) \\ &= p_0 A_{BP} \left\{ v_T + \operatorname{sign}(\Delta p) \sqrt{\frac{2|\Delta p|}{\rho \xi}} \right\} && (8) \end{aligned}$$

The speed control is designed to control the amount of bypassed medium by changing the cross sectional area of the bypass in the tool. Therefore A_{BP} will be time dependent and the area will be controlled by the designed control algorithm.

Tool dynamics

The dynamic of the tool is described by the position x. With the mass m of the tool it follows:

$$m \cdot \ddot{x} = \begin{cases} 0 & \text{if } v_T = 0 \wedge F_D(t) < F_{FR,st} \\ F_D(t) - F_{FR,sl} & \text{if } v_T > 0 \end{cases} \tag{9}$$

The tool dynamics are determined by the 'driving force' F_D and the static ($F_{FR,st}$) and sliding ($F_{FR,sl}$) frictional force. It is assumed that the tool can hardly move backward, which yields for a wide range of MFL tools. The main driving force of the tool is caused by the differential pressure acting on the cross sectional area of the tool ($F_D = \Delta p \cdot A$).

If the medium velocity caused by tool speed and bypassed medium is smaller than the flow velocity, the medium upstream the tool has to be slowed down. The momentum of the medium will be transferred to the tool. To overcome some additional numerical calculations with finite difference methods a linear velocity distribution ($v_G(x)$) is assumed from the flow velocity at a certain distance $v_G(L) = v_F$ to the medium velocity directly at the tool $v_G(0) = v_M$:

$$v_G(x) = v_F \cdot \frac{x}{L} + v_M(1 - \frac{x}{L}) \tag{10}$$

From this fixed velocity distribution with a fixed medium density it is possible to calculate the momentum change of the media on one side of the tool:

$$\begin{aligned} F_M &= \int_0^L \rho A \dot{v}_G dx && (11) \\ &= \int_0^L \rho A \frac{\partial v_G}{\partial x} \dot{x} dx = \int_0^L \rho A \frac{\partial v_G}{\partial x} v_G dx && (12) \\ &= \rho A (v_M - v_F)(v_M + v_F/2) && (13) \end{aligned}$$

If the medium velocity at the tool is higher than the pump velocity the tool is accelerating the medium in the downstream part of the pipeline. Assuming again a linear velocity distribution delivers the same result as in formula (13). This is the momentum change of the medium in the control volume around the tool. Due to momentum conservation the tool will have the opposite momentum change ($F_{MT} = -F_M$). So all forces on the tool can be combined to:

$$m \cdot \ddot{x} = \begin{cases} 0 & \text{if} \quad \begin{array}{c} v_T = 0 \ \wedge \\ (F_D + F_{MT}) < F_{FR,st} \end{array} \\ F_D + F_{MT} - F_{FR,sl} & \text{if} \quad \begin{array}{c} v_T > 0 \ \vee \\ (F_D + F_{MT}) > F_{FR,st} \end{array} \end{cases} \tag{14}$$

With the mass conservation equations of the control volume (3, 4), the equation for the bypassed media (8) and the balance of forces on the tool (14) all governing equations for the model are set up.

Speed control

The designed speed control works by changing the bypass area via a motor driven valve. During the run only the tool speed should be used by the control algorithm. The valve can be controlled by the states stop, opening and closing. Here the valve velocities are fixed, in our case about 13 s for one full operation from closed to open. In the simulation this is integrated by an algorithm which decides depending on the tool velocity in which state the valve should be set (opening, closing, and stop). The bypass area A_{BP} will be calculated by integration, which will then have an influence through formula (8) on the tool dynamics.

IMPLEMENTATION OF THE MODEL

To solve the model the above described differential equations have to be integrated. The solving of the differential equations is done numerically by an implementation in a modeling environment. This environment is designed for simulation and includes functions for numerical integration of variables. It deals by itself with time steps to ensure high accuracies and offers different methods for numerical integration. The implementation was sectioned in five blocks:

1. Control volume upstream the tool
2. Control volume downstream the tool
3. Tool dynamics
4. Bypassed medium through the tool
5. Control algorithm

This sectioning makes it easy to exchange a control algorithm by an optimized one and the results can be compared directly. The first three blocks contain an integration of the differential equations. The two other blocks describe the SCU divided into the hardware, which is responsible for the bypassed medium and the control of the hardware.

RESULTS

In the first step the model was tested by comparing the results of the simulation with the observed tool behavior. Since a few parameters from the run (i.e. static fricion, sliding friction, pressure drop coefficient of the valve and medium density) are not known exactly, comparison was used to determine these parameters by fitting the model to the reality. Some of them can be estimated from other parameters like the sliding friction from the differential pressure. But there is still some space for optimization.

Comparing the model

The used parameters in the end are mass of tool $m = 2600$ kg (5732 lb), pipeline square section $A = \pi \cdot (40"/2)^2$, sliding friction $F_{FR,sl} = 0.45\,\text{bar} \cdot A = 36.5\,\text{kN}$, static friction $F_{FT,st} = 1.15 \cdot FR_{FR,sl}$, pump flow $v_F = 7.5$ m/s (16.8 mi/h), overall pressure $p_0 = 35$ bar (507.6 PSI), medium density $\rho = 20\,\text{kg/m}^3$ ($1.25\,\text{lb/ft}^3$) and the pressure drop coefficient of the valve $\xi = 3$. The maximal bypass area was set to $A_{BP,max} = 0.17A$. These values are all in the range of the real values. The optimal speed was chosen to be 1 m/s (2.2 mi/h) as in the run.

To get realistic velocity data for the input of the control algorithm some bandwidth limited white noise in the size of the usually observed noise is added to the calculated velocity. Using the above parameters in the model with the old control algorithm resulted in a similar behavior to the observed speed profile. A comparison of the speed profile and the opening position of the valve for simulated and run data is shown in figure 3. On top, the results of the simulation are shown. The upper curve is the velocity profile and below is the opening position of the speed control valve. Both curves fit well to the real run data which is shown below. These results show that the model can be used to

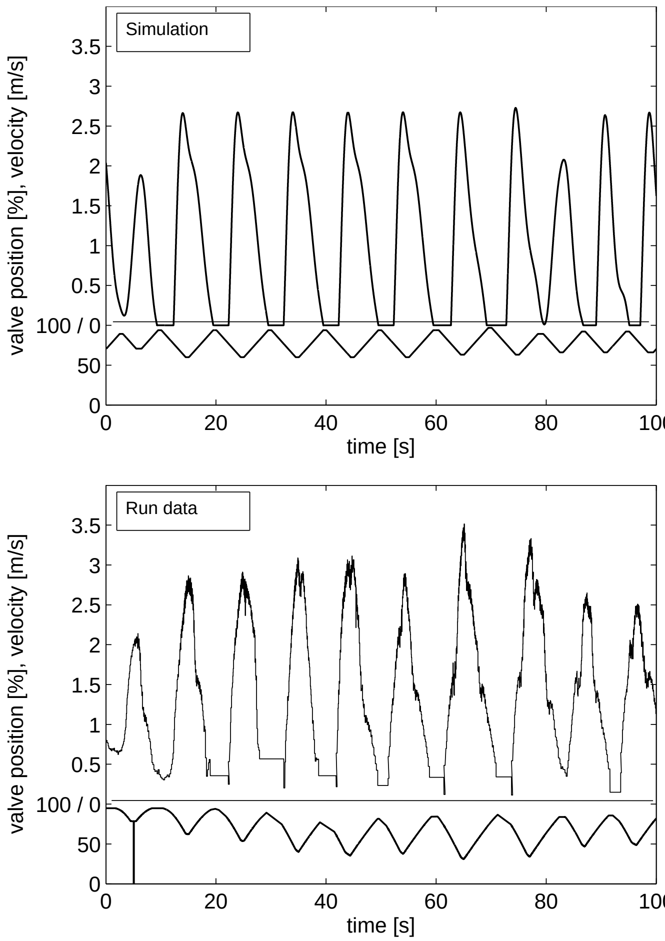

Figure 3. COMPARISON OF SIMULATED TOOL BEHAVIOR (ABOVE) AND OBSERVED TOOL BEHAVIOR (BELOW). THE COMPARISION SHOWS A GOOD AGREEMENT WHICH SUPPORTS THAT THE MODEL IS SUITED FOR SIMULATIONS OF A TOOL SPEED CONTROL.

simulate the tool behavior and the regulation by the speed control in a straight pipeline.

Source of oscillation

Analyzing the simulation and the raw data it is possible to identify the source of the oscillations. There are two main sources: The gas dynamics around the tool and the used regulation algorithm.

Due to the compressibility gas an inspection device tends to oscillate around the flow velocity if the device has left this velocity. If the tool was slowed down the differentail pressure will increase and accelerate the tool. The pressure will increase till the tool speed reaches the flow velocity. But then the differential pressure is higher than needed and the tool will accelerate further till the differential pressure decreases. At that stage the tool is faster than the flow. Now the differential pressure decreases and the tool is being slowed down. When the tool reaches the flow velocity the differential pressure is now too low and the tool slowes down further. This oscillation is damped and is caused by the compressibility of the gas up and downstream the tool. This setup can be compared by two springs which will hold the tool. Any deviation from the steady state will result in an oscillation. Additionally the reactions of the device on changes will be delayed and will result in an overshooting. An example of such behavior is shown in figure 4 where the start of the tool with a fixed bypass of $A_{BP} = 0.1 \cdot A$ is simulated. A similar behavior was also observed in other simulations [3].

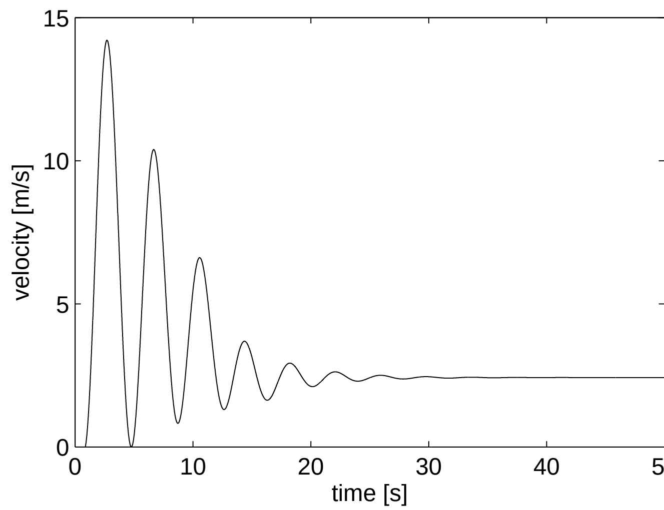

Figure 4. OSCILLATIONS OF A TOOL AFTER A STOP WITH A FIXED BYPASS AREA.

The regulation algorithm itself also causes a delay. The averaging over 2.5 s results in a delay of 1.25 s for the algorithm. Additionally the processing can cause a delay. In summary a delay of approximately 2 s may occur. This is approximately a quarter of the oscillation time. A quarter delay means a phase shift of $\pi/2$, which shows that the observed oscillation is caused by the delay of the algorithm and the delay of the tool reactions.

Optimizing the algorithm

As already mentioned an optimized speed control algorithm was already proposed in the literature [3]. It is clear that very effective algorithms can be designed if more parameters than only the tool velocity are known. An attempt similar to this was tried by calculating the first (and second) derivative of the velocity. With these the velocity was extrapolated by integration methods. This procedure shows nearly an optimal performance with the capabilities of the mechanics. But introducing some noise in the velocity data, like it is always observed, made the proposed algorithm unstable. The reason is clear, since differentiating noisy data will result in a lower signal to noise level. Integration on this basis for data extrapolation can result in strong deviations. Therefore another concept had to be developed. The general framework for the algorithm is defined by being simple, stable and the only input shall be the tool velocity.

The identified sources for the oscillation give hints for the solution of the problem. First of all the algorithm tried to regulate against short term fluctuations (shorter than 10 s). Then the tool need some time to reach the steady state and the regulation activities have a quite strong influence on the tool behavior. Additionally the window size for no regulation activities was too narrow. To overcome the regulations against short term fluctuations the velocity was chosen to be averaged over 10 s. Then, the window, in which no regulation is done, was chosen to be $\pm 10\%$ of the target speed or at least ± 0.4 m/s (0.89 mi/h). To overcome the strong reactions of the tool the speed of the valve had to be reduced. Since the motor speed could not be slowed down, a kind of pulse width modulation was used, which means a fixed full pulse of 0.5 s followed by a variable waiting time is being used to steer the valve. The waiting time was varied depending on the difference between tool velocity and optimal velocity. The waiting times were chosen to be between 0 and 10 s.

This new algorithm was tested in the same boundary conditions as the old algorithm was tested. A comparison of the performance when starting the tool is shown in figure 5. On top, the speed profile and at the bottom the position of the valve is shown. The dotted line represents the new and the solid line represents the old algorithm. Due to the long averaging time the response time of the new algorithm is in the beginning higher. After a certain time both algorithms show an additional stop. This stop is caused by the high average velocity of the tool after starting to run and the bypass which is opened. After the first stop the old algorithm runs into the observed speed oscillations. The new algorithm needs only to make two additional short regulations to reach the optimal speed without any oscillations.

This comparison shows that the new algorithm works well. To overcome the speed peaks when the tool is starting more parameters of the tool dynamics have to be known by the steering algorithm. The most interesting parameter would be the differential pressure acting on the tool. But since the speed fluctuations are relatively fast (below 5 s) the response time of the valve with 13 s will be too lsow to regulate against these peaks. If the response time could be reduced, a more effective speed control regulation could be realized. Faster response times of such

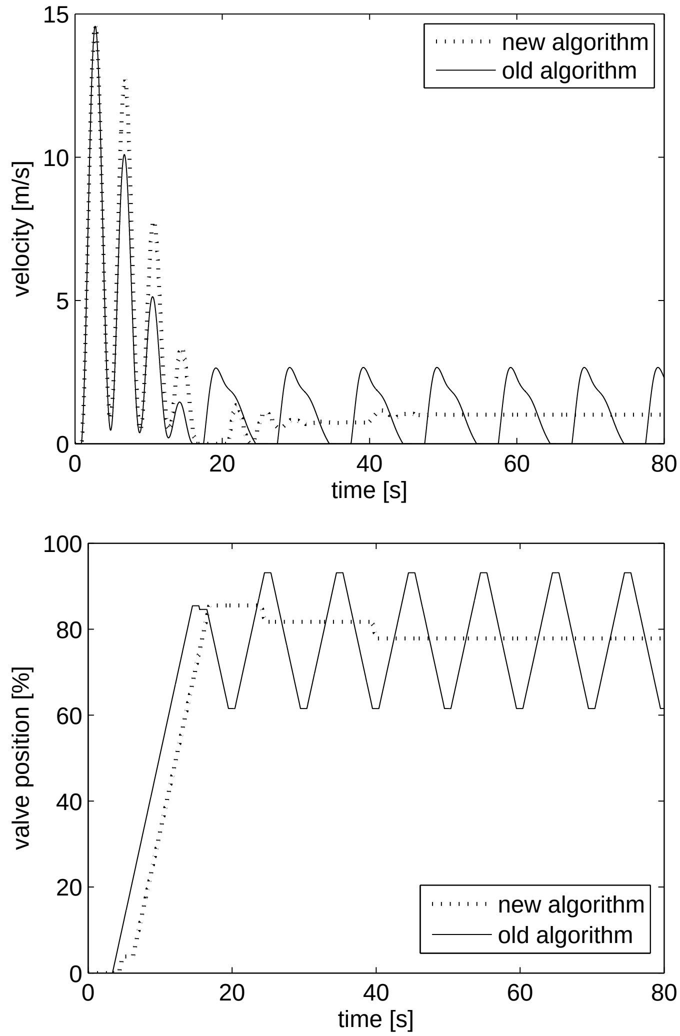

Figure 5. COMPARISION OF THE PERFORMANCE OF THE NEW (DOTTED LINE) AND THE OLD (SOLID LINE) SPEED CONTROL ALGORITHMS. ON TOP THE SPEED PROFILE AND AT THE BOTTOM THE POSITION OF THE VALVE IS SHOWN. IT CAN CLEARLY BE SEEN THAT THE NEW ALGORITHM WORKS WELL.

valves result in higer power consumtion and higher mechanical stress. The power consumption can be especially critical when considering operation times of more than one day. Therefore a highly optimized and laborious speed regulation algorithm regulating against short velocity peaks will not be proposed at this time.

CONCLUSION

In this paper a simple model is developed which describes the dynamic of an intelligent pipeline inspection tool with an active speed control in a gas pipeline. This model is based on mass and linear momentum conservation equations. With certain assumptions it is possible to reduce the model to a set of differential equations, which can be integrated numerically. The model was implemented in a simulation environment and the results were compared with the run of a real tool and they showed a good agreement. Based on the analysis of the tool behavior a new speed control algorithm was proposed and the algorithm was tested with the simulation. The new algorithm performed very well and it was realized in the new tool software.

REFERENCES

[1] U. Thuenemann, J. W.-K., 2003. "Control your speed". *World Pipelines*, April, pp. 31–33.

[2] L. Matthews, M. Kennard, A. O. D., 2003. "Velocity control of pigs in gas pipelines". In *Pipeline Pigging & integrity Technology*, J. Tiratsoo, ed., 3rd ed. Scientific Surveys Ltd, PO Box 21, Baconsfield HP9 1NS, UK, chapter 1, pp. 35 – 47.

[3] T. T. Nguyen, H. R. Yoo, Y. W. R. S. B. K., 2001. "Speed control of pig using bypass flow in natural gas pipeline". In Proceedings of the IEEE Internationael Symposium on Industrial Electronics, Vol. **2**, IEEE, pp. 863 – 868.

[4] P. C. R. Lima, H. Y., 1999. "Modeling of pigging operations". In Proceedings of the SPE Annual Technical Conference and Exhibition, Vol. **2** of *Production Operations and Engineering*, Society of Petroleum Engineers, pp. 863 – 868.

[5] A. O. Nieckele, A. M. B. Braga, L. F. A. A., 2000. "Transient pig motion through non-isothermal gas and liquid pipelines". In Proceedings of the 2000 Internationael Pipeline Conference, J. R. Ellwood, ed., Vol. **1**, American Society of Mechanical Engineers, pp. 611–618.

[6] A. O. Nieckele, A. M. B. Braga, L. F. A. A., 2001. "Transient pig motion through gas and liquid pipelines". *Journal of Energy Resources Technology,* **123**, December, pp. 260–269.

[7] L. F. A. Azevedo, A. M. B. Braga, A. O. N. P. R. S. M., 1999. "Simulating pipeline pigging operations". In Proceedings of the Pipeline Pigging Conference & Exhibition, pp. 1 – 21.