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NUMERICAL STUDY OF UNSTEADY FLOW REGIME IN A NATURAL GAS PIPE

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In this study, isothermal gas flow characteristics in a long pipeline are monitored and controlled using two mathematical models and the finite difference approach. These models, which take into account continuity, motion, and the equation of state, are based on one-dimensional nonlinear partial differential equations. Results are consistent with the body of literature, supporting their applicability in the present. Higher inclinations have a noticeable impact on output flow rates when examining the effects of pipe inclination on gas flow under similar pressure. A significant contribution of this work is the analysis of stabilization time across inclinations, which reveals pressure shifts and flow steadying after 5000 s. These findings are useful for the energy sector because they help pipeline operators and engineers maintain a safe, effective gas flow.

Keywords: Gas flow pipelines, Non-stationary flow, Real-time monitoring, Stability, Inclination.

NOMENCLATURE					
A	cross-sectional area	[m^2]	Q	fluid flow rate	[kg/s]
a	sound velocity	[m/s]	t	time	[s]
D	diameter of the pipe	[m]	u	velocity	[m/s]
f	friction coefficient		Δt	time step	
g	acceleration due to gravity	[m/s^2]	Δx	spatial step	
L	pipe length	[m]	θ	constant parameter	
N	number of nodes.		ρ	fluid density	[kg/m^3]
P	fluid pressure	[N/m^2]	φ	inclination angle	[rad]

1. Introduction

The non-stationary gas flow in pipes is a common phenomenon in various industrial installations designed for transporting or utilizing gaseous fluids. These installations can have diverse natures and multiple applications. Examples illustrating this phenomenon include pipes in compressed air supply systems for pneumatic servo circuits, supply skids for gas turbines or internal combustion engines, as well as gas transport and distribution networks.

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In such systems, variations in operating conditions can cause non-uniform gas flow leading to disturbances that may affect gas parameters at any point in the circuit. Therefore, real-time monitoring and control of gas flows are crucial to ensure the safe and efficient operation of these installations.

Non-stationary flow occurs when one or more parameters at the system inlet or outlet vary. This phenomenon is characterized by the propagation of a wave, the amplitude and speed of which depend on the physical properties of the fluid and the geometry of the installation. The duration of this phenomenon, sometimes referred to as transient, can vary widely depending on the length of the gas pipeline and the magnitude of parameter variations

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In most cases, the parameters observed are pressure or density, as well as flow velocity or flow rate in an isothermal flow. In the case of a gas pipeline with small parameter variations, the transient can be long-lasting. Conversely, in the case of short circuits with large parameter variations, the transient can be rapid. Therefore, understanding and characterizing non-stationary flow are essential for maintaining efficient and safe gas flow in industrial installations.

In the field of studying unsteady flow in gas pipelines, several researchers have made significant contributions. Stoner [1] employed the characteristic method to solve the equations governing this type of flow. The approach's validity was confirmed by comparing the results with experimental transients.

Kessal [2] conducted a numerical study on isothermal compressible fluid flow utilizing two different methods. The first method involved a predictor-corrected scheme suitable for analyzing fast fluid flow in short gas pipelines, while the second method employed a simple finite difference scheme designed for slow fluid flow. The obtained results were found to be consistent with those obtained using other methods.

Emara-shabaik et al. [3] evaluated the effectiveness of numerical schemes presented in their paper for developing real-time pipeline simulators. These schemes demonstrated stable simulations with fast response times. While the CTCS (centred-time centred-space) method was free of oscillating errors, it still had residual errors. On the other hand, the Mac Cormack (MC) method yielded the most accurate results among the three methods investigated. Additionally, as the mesh size was reduced, the CTCS method's accuracy approached that of the MC method.

Gato and Henriques [4] presented a numerical model based on the discontinuous Galerkin method of Runge Kutta to study the dynamic behavior of gas flow in pipes. They specifically focused on examining the compression wave that occurs following the abrupt closure of a valve downstream. Furthermore, they analyzed the effect of partial reflection of pressure waves during the transition between pipes with different sections.

Nouri-Borujerdi [5] performed simulations of transient adiabatic compressible gas flows in a long pipe after a catastrophic pipe rupture. They employed the high-order finite difference method and divided the pipe into high and low-pressure sections. As a result, a significant pressure drop was observed at the rupture point.

Bermúdeza et al. [6] investigated non-isothermal compressible gas flow in a pipe using the finite volume method. The study considered various factors such as wall friction, height, and heat exchange with the external environment.

Their findings contribute to the understanding of the thermodynamic behavior of gas flow in pipelines. Akhmetzyanov and Salnikov [7] focused on modeling unsteady gas flow using multilevel methods based on a generalized solution to the initial boundary value problem for nonlinear partial differential equations. Their models described the unsteady pressure and flow distribution in gas transport systems with discontinuous coefficients.

Dayev [8] employed variable differential pressure flow meters to measure the quantity of natural gas. They applied the similarity theory and obtained analytical correlations, ultimately deriving an equation for the gas expansion factor.

Fan et al. [9] investigated the transient composition of gas pipeline networks by combining the control volume method for hydraulic simulation and an unsteady heat transfer model for thermal simulation. To validate their findings they compared the obtained results with experimental data.

Koo [10] explored transient flow in natural gas pipelines using the pressure-dependent line-corrected characteristic method. Through verification experiments focused on a slow transient real-world problem, they confirmed the feasibility of this method for practical pipeline analysis applications.

2. Governing Equations

The non-stationary gas flow in pipelines is typically analyzed using a one-dimensional approach, which is sufficient for describing the phenomenon. The governing equations are derived from the equations of continuity, momentum, energy, and the thermodynamic equation of state. In the literature, studies are often focused on either adiabatic or isothermal cases. When gas flow transitions are rapid, it is assumed that pressure changes occur instantaneously, and there is no time for heat transfer between the gas in the pipeline and the surrounding environment. Conversely, in the case of slow transient flows caused by fluctuations, it is assumed that the gas in the pipe has enough time to reach thermal equilibrium with the surroundings at a constant temperature. For this particular study, an isothermal flow is considered to approximate real gas dynamics application. The pressure and mass flow equations used in the model are obtained from (Emara-shabaik et al., [3]).

$$\frac{\partial P}{\partial t}+\frac{a^2}{A}\frac{\partial Q}{\partial x}=0 \tag{1}$$

$$\frac{\partial Q}{\partial t}+A\frac{\partial P}{\partial x}+\frac{fa^2}{2AD}\frac{Q|Q|}{P}+\frac{Ag\sin\varphi}{a^2}=0 \tag{2}$$

In the model B, which is presented below, have used the following equations for density and velocity obtained by (Nouri-Borujerdi [5])

$$\frac{\partial \rho}{\partial t}+\frac{\partial \rho u}{\partial x}=0 \tag{3}$$

$$\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+\frac{1}{\rho}\frac{\partial P}{\partial x}=-\frac{2f}{D}u^2 \tag{4}$$

3. Numerical Methods

Despite the risk of error in some cases, numerical methods are commonly used to solve complex problems, offering a versatile alternative to conventional analytical solutions but with varying degrees of resolution and performance.

The choice of the appropriate numerical method depends on the complexity of the problem to be solved and can range from simple to very sophisticated approaches.

In this work, the central finite difference scheme is used to discretize the governing equations in space, while an advanced form is employed for the time domain. More specifically, the transient pressure and flow terms are treated as follows:

3.1 Model A

$$\left(\frac{\partial P}{\partial t}\right)_i^{t+\theta}=\frac{P_i^{t+1}-P_i^t}{\Delta t}+0(\eta) \tag{5}$$

$$\left(\frac{\partial Q}{\partial t}\right)_i^{t+\theta}=\frac{Q_i^{t+1}-Q_i^t}{\Delta t}+0(\eta) \tag{6}$$

The differential terms in space for P and Q are discretized like this

$$\left(\frac{\partial P}{\partial x}\right)_i^{t+\theta} = \theta\frac{P_{i+1}^{t+1} - P_i^{t+1}}{\Delta x} + (1-\theta)\frac{P_{i+1}^{t} - P_i^{t}}{\Delta x} + 0(\Delta x) \tag{7}$$

$$\left(\frac{\partial Q}{\partial x}\right)_i^{t+\theta} = \frac{Q_{i}^{t+1} - Q_{i-1}^{t+1}}{\Delta x} + (1-\theta)\frac{Q_{i}^{t} - Q_{i-1}^{t}}{\Delta x} + 0(\Delta x) \tag{8}$$

Therefore, the governing partial differential equations can be approximated by the following difference equations

$$P_i^{t+1} + \frac{\alpha_1}{2}\theta Q_{i+1}^{t+1} - \frac{\alpha_1}{2}\theta Q_{i-1}^{t+1} = P_i^t - \frac{\alpha_1}{2}(1-\theta)(Q_{i+1}^t - Q_{i-1}^t) \tag{9}$$

$$Q_i^{t+1} + \frac{\alpha_2}{2}\theta P_{i+1}^{t+1} - \frac{\alpha_2}{2}\theta P_{i-1}^{t+1} = Q_i^t - \frac{\alpha_2}{2}(1-\theta)P_{i+1}^t + \frac{\alpha_2}{2}(1-\theta)P_{i-1}^t - \alpha_3\frac{Q_i^t\left|Q_i^t\right|}{P_i^t} + \alpha_4 P_i^t \tag{10}$$

Where: $\alpha_1 = \dfrac{a^2\Delta t}{A\Delta z}$; $\alpha_2 = \dfrac{A\Delta t}{\Delta z}$; $\alpha_3 = \dfrac{fa^2\Delta t}{2AD}$; $\alpha_4 = \dfrac{Ag\sin\varphi\Delta t}{a^2}$

3.2 Model B

In order to compute the flow that crosses the pipe's area in time dt, equations are created along the control volume dv with distance dx in figure 1. The pipe is considered to be inclined by an angle.

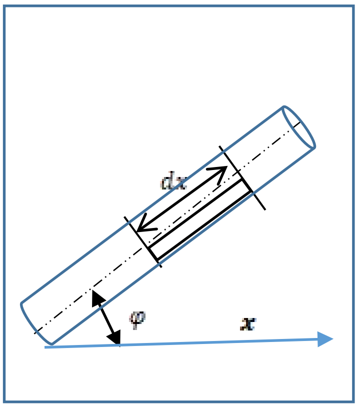

Fig.1 Flow through a control volume

The finite difference representation of Eq. (3) for density at a given grid point is written as follows.

$$-\frac{u_i^n \Delta t}{2\Delta x}\rho_{i-1}^{n+1} + \rho_i^{n+1} + \frac{u_i^n \Delta t}{2\Delta x}\rho_{i+1}^{n+1} = \rho_i^n - \frac{\Delta t \rho_i^n}{2\Delta x}\left(u_{i+1}^n - u_{i-1}^n\right) \tag{11}$$

Equation (4) can be discretized for parameters *u* and *t* respectively as

$$-\frac{u_i^n \Delta t}{2\rho_i^n \Delta x}u_{i-1}^{n+1} + u_i^{n+1} + \frac{u_i^n \Delta t}{2\rho_i^n \Delta x}u_{i+1}^{n+1} = u_i^n - \frac{\Delta t}{2\rho_i^n \Delta x}\left(P_{i+1}^n - P_{i-1}^n\right) - \frac{f_D}{2D}\left(u_i^n\right)^2 \tag{12}$$

3.3 Boundary conditions

The initial and boundary conditions of the above problem are given as follows:

$P\ (x,0) = p(x)$: initial pressure distribution

$Q\ (x,0) = q(x)$: initial flow distribution

$P\ (0,t) = p_0$: inlet pressure

$P\ (L,t) = p_e$: exit pressure

For the other variables u and ρ a gradient of 0 is imposed at the pipe inlet and outlet.

4. Results and discussion

Two isothermal models based on equations (p,q) model A and equations (ρ,u) model B are validated by collecting pressure data upstream and downstream of Emara-shabaik et al. [3]. The inlet pressure is 3.5 bar and the outlet pressure is fixed at 1 bar.

4.1 Mesh sensitivity

To ensure the accuracy of our calculations, the variation of our results has been tested by choosing different numbers of points along the pipeline. Specifically, 100, 110, and 120 points were tested for a distance of 90 km. In fig. 2, 3, 4, and 5, the distances are being scaled to the total length of the pipe, denoted by x.

The variations of P and Q as a function of dimensionless distance x/L for different numbers of nodes, denoted by N, are illustrated in the following figures. It is observed that the curves for all cases are completely superimposed, indicating that our results are not sensitive to variations in the number of nodes.

4.1.1 Mesh sensitivity of model A

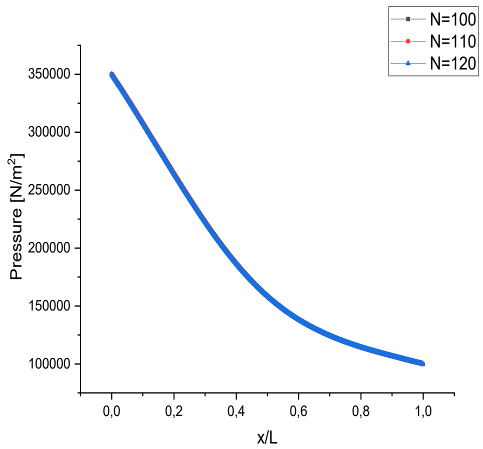

Fig.2. Pressure profile (model A) for t =500 s

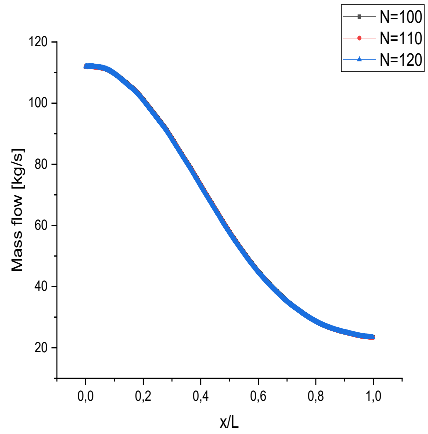

Fig.3. Flow profile (model A) for t =500 s

4.1.2 Mesh sensitivity of model B

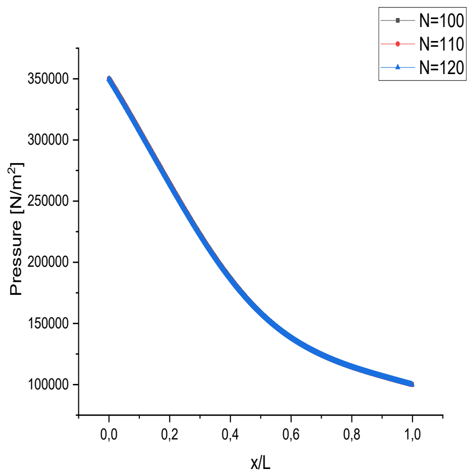

Fig.4. Pressure profile (model B) for t =500 s

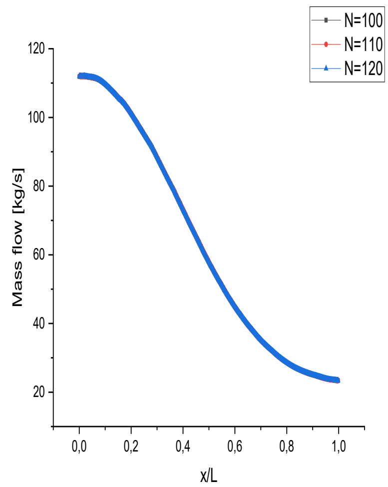

Fig.5. Flow profile (model B) for t =500 s

4.2 Validation of the two models (A and B)

The results of Emara-shabaik et al. [3] are being used to validate our numerical computing code, which is built upon two models of equations. The pressure variation at the pipe's intake and outlet is being compared with that calculated using models A and B in fig 6, 7, and 8 for durations of 500 s, 1250 s, and 5000 s, respectively, under the identical boundary conditions and time as those taken into consideration by Emara-shabaik et al. [3]. A perfect concordance is observed in all three cases. Due to the linear pressure drops at constant temperature, a decrease in P is observed in all of the scenarios that are being looked at as the exit is being approached. As time goes on, the pressure drop across the pipe is decreasing, revealing a concave curve. With the same authors, the flow rate is being compared as a function of x/L for two distinct time periods, 500 s and 1250 s. A discrepancy from the reference curve is seen for the curve in Fig. 9 at time 500 s, which could be due to different methodologies. However, as shown in Fig. 10, there is a good agreement between the two curves for a longer time period of 1250 s.

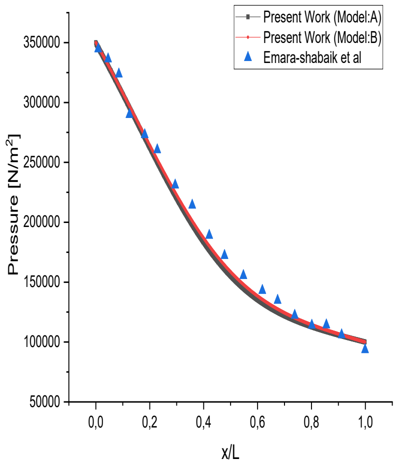

Fig.6. Pressure profile for t=500 **s**

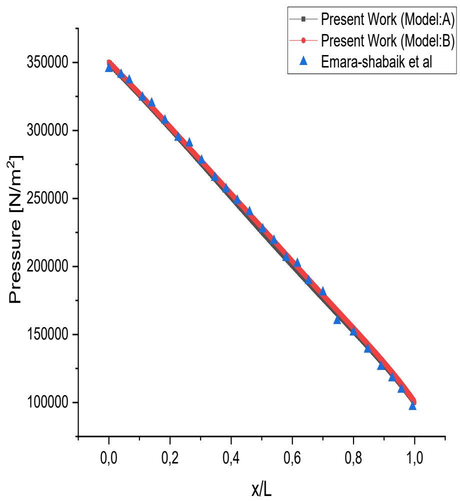

Fig.7. Pressure profile for t=1250 s

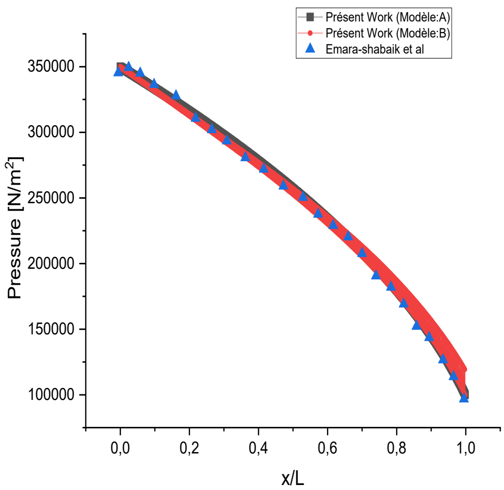

Fig.8. Pressure profile for t=5000 s

With the same authors, the flow rate is compared as a function of x/L for two distinct time periods, 500 s and 1250 s. A discrepancy from the reference curve is seen for the curve in Fig. 9 at time 500 s, which could be due to the different methodologies. However, as shown in Fig. 10, there is a good agreement between the two curves for a longer time period of 1250 s.

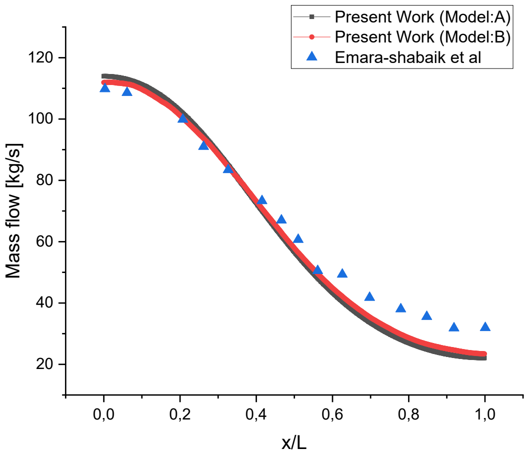

Fig.9. Flow profile for t=500 s

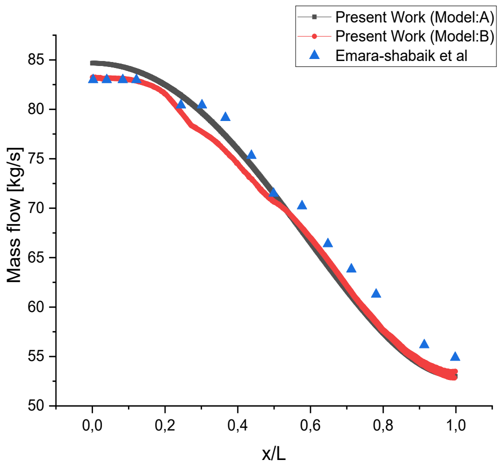

Fig.10. Flow profile for t=1250 s

4.3 Effect of Pipeline Inclination on Control Parameters

In this section, the influence of the tilt angle on the pressure profile along the pipe for t=1000 s is presented, as depicted in Fig.11. Simulation calculations are being performed for each angle to obtain the pressure and flow profile along the pipe. At the inlet, a maximum pressure value of 3.5 bar is observed, and it decreases almost linearly when the pipe is perfectly horizontal (i.e., the angle is zero). As the angle increases, the pressure drop at the inlet becomes more gradual, followed by a linear and parallel drop, and concludes with a sharp decrease at the outlet while maintaining higher pressures at larger angles.

By imposing inlet and outlet pressures, the flow rate varies with the angle. Specifically, the flow rate increases as the angle increases; the highest flow rate is achieved by tilting the pipe between 0 and π /8. Between π /6 and π /4, the curves are nearly parallel with almost constant values at the inlet and a linear decrease until the gas leaves.

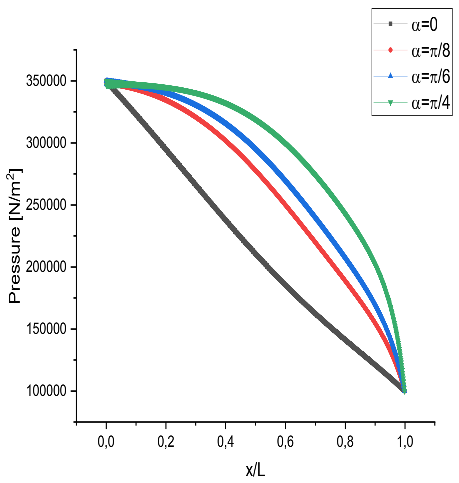

Fig.11. Pressure variation as a function of

φ (Tilt angle) model A at t=1000 s

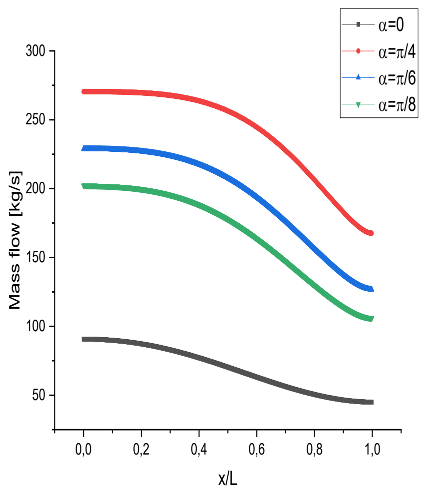

Fig.12. Variation of flow rate as a function of

φ (Tilt angle) model A at t=1000 s

4.4 Pressure and Flow Stability as a Function of Inclination Angle

4.4.1 Pressure stability

The stability time of the pressure and flow profiles is determined by testing the solutions at different time intervals (t=1000, 2000, 5000, and 6000 s) for each angle of inclination.

The pressure variation for different time intervals ranging from 500 s to 5000 s is being illustrated in Fig.10. The pressure values are increasing between 1 bar and 3.5 bar along the pipe and are reaching constant values beyond 5000 s.

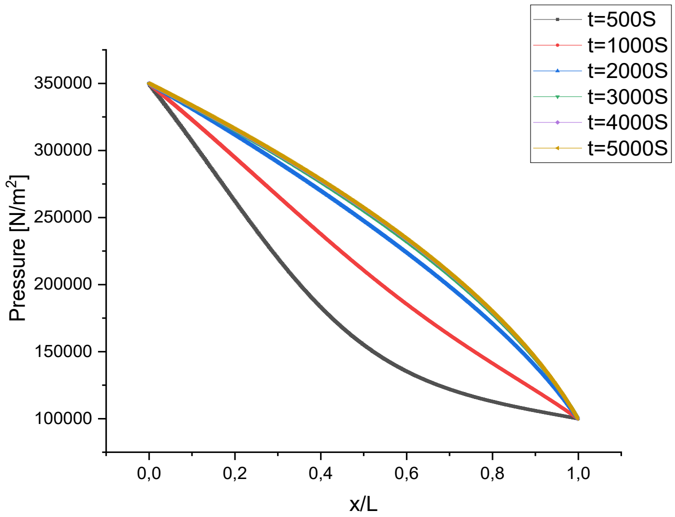

Fig.13. Pressure stability for angle 0

Angles of π /4, π /6, and π /8 are being considered, and variable time intervals ranging from 1000s to 6000 s has been tested. It is being observed that beyond 5000 s, the pressure values (p) are not changing, indicating the attainment of a steady state. With the increase in angle, the outlet is being approached by the maximum pressure value, resulting in a constant flow rate.

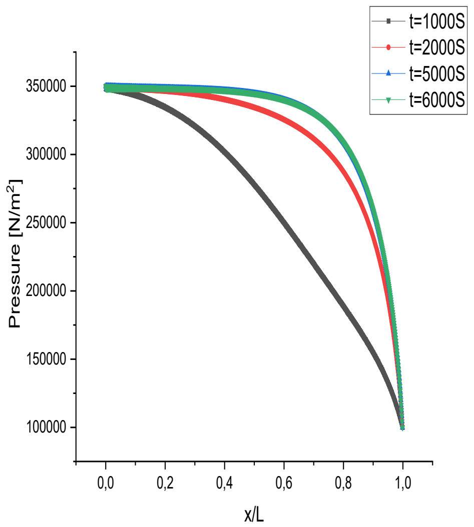

Fig.14. Pressure stability for angle π /8

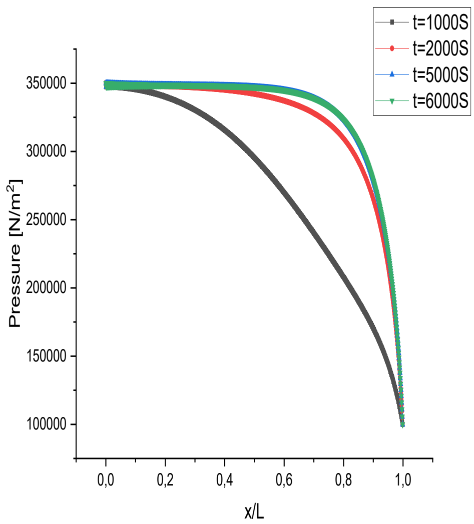

Fig.15. Pressure stability for angle π /6

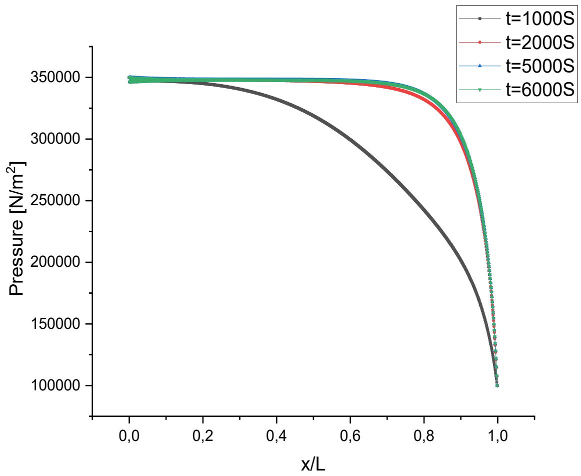

Fig.16. Pressure stability for angle π /4

4.4.2 Stability of flow rate and velocity

The stability of model A for the flow rate is being examined in this section as a function of time. The curves for several time intervals are presented until stability is achieved. The flow rate is starting at a maximum value at the inlet, which is decreasing towards the outlet for all the chosen time intervals shown in Fig. 17. As time is increasing, the flow rate is starting to even out, resulting in a constant flow rate throughout the pipe, particularly at 5000 s. It is worth noting that the flow rate is behaving differently when the pipe is horizontal.

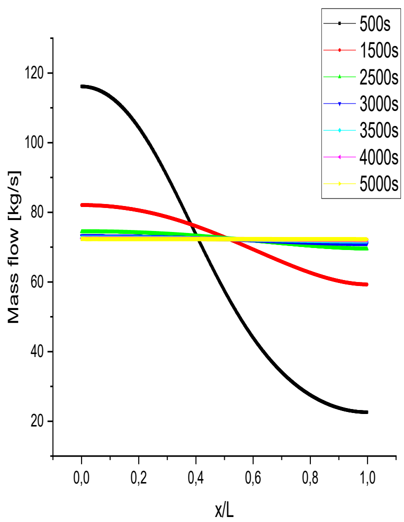

Fig.17. Flow rate stability for angle 0

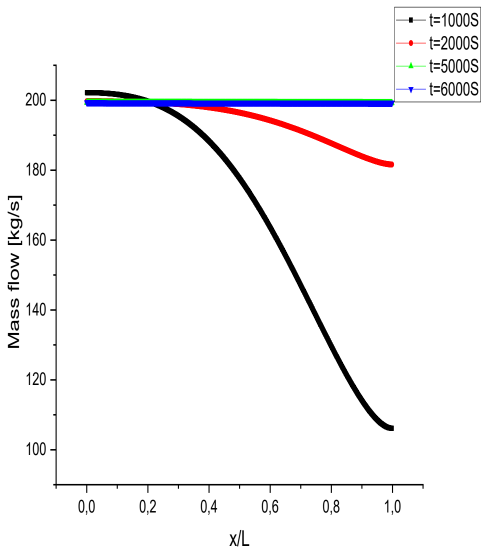

Fig.18. Flow rate stability for angle π /8

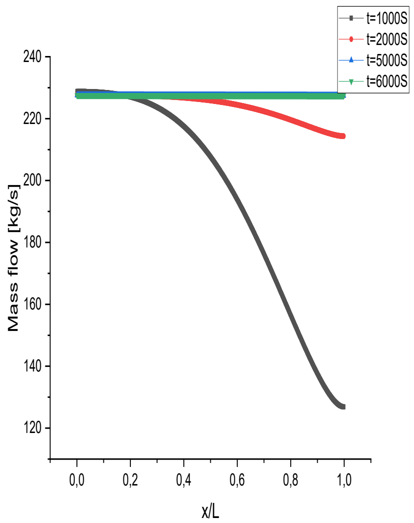

Fig.19. Flow rate stability for angle π/6

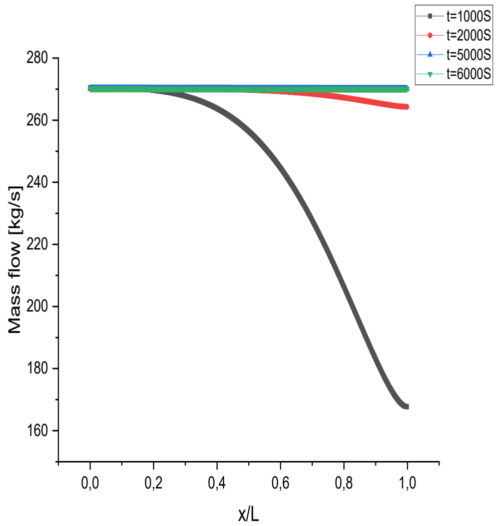

Fig.20. Flow rate stability for angle π /4

It is worth noting that the flow rate is behaving differently when the pipe is horizontal. It is decreasing at the inlet and increasing at the outlet by approximately the same amount of flow. However, when the angle of inclination changes, a weak decrease at the inlet is being observed, and a strong growth of the flow rate is being observed at the outlet of the pipe with time until the flow rate becomes constant in any cross-section of the pipe.

5. Conclusions

In the context of a gas flow pipe, this study presents useful and simple computational techniques that do not require sophisticated programming. The central finite difference method is being used for the numerical simulation of one-dimensional transient flow. Two model equations are being employed, and our results for pressure and flow rate are being compared to those of Emara-shabaik et al. [3]. The literature is showing a good agreement with our results in quantifying the evolution of pressure and flow rate along the pipe.

The effect of inclination angle on these parameters is also being investigated. An increase in flow rate at the inlet is being achieved when inlet and outlet pressures are imposed and the inclination is increased. Conversely, a significant increase in flow rate is being observed at the outlet to establish a uniform flow rate in the pipe and attain a steady state.

In future work, other numerical methods will be tested, and the presence of leakage in the pipe will be explored, which will hold high relevance for industrial applications.

Acknowledgements

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