Numerical analysis of unsteady behaviour of a screw compressor plant system

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ABSTRACT

In order to analyse the performance of screw compressors operating under varying load conditions, an unsteady one dimensional model of the compressor process was modified to include all plant components, including tanks and connecting piping. This was based on the differential equations of conservation of mass and energy. The results derived from this model have been verified by experiment in order to obtain a reliable tool that can simulate a variety of scenarios which may occur in everyday compressor plant practice.

Keywords: Mathematical modelling, Screw compressor process, Compressor plant

NOMENCLATURE

A	– cross section area of discharge valve	T_{in}	– temperature of the gas entering the tank
M	– mass	T_{out}	– temperature of the gas leaving the tank, equal to T_2
m_{in}	– mass flow entering the tank	Δt	– time step
m_{out}	– mass flow leaving the tank	U	– internal energy
p_0	– atmospheric pressure	u	– specific internal energy
p_1	– pressure in the compressor tank	V	– volume of the plant containing tank and pipes
p_2	– pressure in the tank at the next time step	μ	– flow coefficient
R	– gas constant	ρ_2	– density of gas in the tank
T_2	– Temperature in the tank		

1. INTRODUCTION

Design tools for the design of screw compressors and the prediction of their performance under steady operating conditions, are now widely available. However, despite the fact that such machines frequently and, in some cases even continuously operate under unsteady conditions, there are few published studies of how this affects their performance, while it is quite possible that the need to operate under such conditions may influence the compressor design. The following summarises known studies.

Jun and Yezheng (1),(2) carried out experimental studies on the effects of working fluid migration during the start-up and shut-down cycles of a refrigeration system with a reciprocating compressor. They developed a program to estimate energy losses and how to calculate how much they are due to this effect, with the aim of reducing energy consumption.

Fleming, Tang and You (3) published a paper on simulation of shutdown processes in screw compressor driven refrigeration plant. Their idea was to use a reverse rotation brake instead of a suction non-return valve, which prevented reverse rotation, leading to a significant decrease of the compressor backflow. As a consequence, a reduction of the shutdown torque occurred. However, only the mathematical model was presented and no experiment data has followed to validate it. A disadvantage of the reverse rotation brake is that it might trigger failure if there is significant rotor backlash in the compressor.

Li and Alleyne (4) investigated transient processes in the start-up and shutdown of vapour compression cycle systems with semi-hermetic reciprocating compressors. They established a model of a moving boundary heat exchanger and validated it experimentally. Ndiaye and Bernier (5) developed a dynamic model for a reciprocating compressor in on-off cycle operation and validated it as part of an experiment to justify water-to air heat pump models. A recent paper, by Link and Deschamps (6), deals with the numerical methodology and experimental validation of start-up and shutdown transients in reciprocating compressors.

This paper is the third of a series intended to give an insight into screw compressor transient behaviour. Previous papers described compressor start-up as a transient process and were presented in two papers by Chukanova, Stosic and Kovacevic, (7),(8). Experimental results and their analysis were presented and simulations included inertia effects during start up.

The work now described, covers the numerical simulation of unsteady behaviour of a screw compressor within a compressor plant, including the filling and emptying of the plant tank and associated connecting pipes during different types of compressor starts. This model has been integrated with SCORPATH (Screw Compressor Optimal Rotor Profiling and Thermodynamics), an existing compressor design program, developed in house. The model is written in FORTRAN and is based on the differential equations of mass and energy conservation, developed and tested in earlier work. It is sufficiently general to take into account dry and oil flooded compressors and various plant tanks connected by gas pipes in different combinations providing that they are characterized by one volume and one exit valve.

An interface was written to couple the compressor and plant model elements for this purpose and has been used to show how the tank pressure is affected by the gas mass flow rate, the compressor discharge gas temperature, and the volume of the tank and communicating pipes. The tank pressure is then used to calculate the compressor performance in the next time step. The sequence is repeated for the whole compressor plant system until the specified time is reached.

The model was verified by comparison of predictions obtained from it with measurement results obtained in a series of tests performed on a compressor test rig. A detailed description of the experiments is given in Chukanova, Stosic and Kovacevic (7),(8). A part related to the model verification is presented in section 3 of this paper.

2. MATHEMATICAL MODEL OF THE SCREW COMPRESSOR PLANT

Screw compressor modelling combines the analysis of thermodynamic and fluid flow processes. Both are dependent on the screw compressor geometry and combining them in a mathematical model as a complex process.

The algorithm of the thermodynamics and flow processes in a screw compressor, described here, is based on a mathematical model, defined by a set of equations which describe the physics of the complete process in a compressor. The equation set consists of the equations for the conservation of energy and mass continuity together with a number of algebraic equations defining the flow phenomena in the fluid suction, compression and discharge processes together with the differential kinematic relationship which describes the instantaneous operating volume and its change with rotation angle or time. In addition, the model accounts for a number of 'real-life' effects which may influence the final performance of a real compressor and make the model valid for a wider range of applications.

In the past, these equations have often been simplified in order to achieve a more efficient and economical numerical solution of the set. In this case, where all the terms are included, the effect of such simplifications on the solution accuracy can be assessed.

2.1 Equations governing screw compressor process

The working chamber of a screw machine together with the suction and discharge plenums can be described as an open thermodynamic system in which the mass flow varies with time and for which the differential equations of conservation laws for energy and mass are derived using Reynolds Transport Theorem.

A feature of the model is the use of the unsteady flow energy equation to compute the effect of profile modifications on the thermodynamic and flow processes in a screw machine in terms of rotational angle, or time.

The following conservation equations have been employed in the model.

The conservation of internal energy:

$$\omega\left(\frac{dU}{d\theta}\right) = \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out} + \dot{Q} - \omega p\frac{dV}{d\theta}$$

where θ is angle of rotation of the main rotor, $h=h(\theta)$ is specific enthalpy, $\dot{m} = \dot{m}(\theta)$ is mass flow rate $p=p(\theta)$, fluid pressure in the working chamber control volume, $\dot{Q} = \dot{Q}(\theta)$, heat transfer between the fluid and the compressor surrounding, $V = V(\theta)$ local volume of the compressor working chamber.

Flow through the suction and discharge port is calculated from the continuity equation. The suction and discharge port fluid velocities are obtained through the isentropic flow equation. The computer code also accounts for reverse flow.

Leakage in a screw machine is a substantial part of the total flow rate and affects the compressor delivery, i.e. volumetric efficiency and the adiabatic efficiency, the gain and loss leakages are considered separately. The gain leakages come from the discharge plenum and from the neighbouring working chamber with a higher pressure. The loss leakages leave the chamber towards the discharge plenum and to the neighbouring chamber with a lower pressure.

The leakage velocity through the clearances is considered to be adiabatic Fanno-flow through an idealized clearance gap of rectangular shape and the mass flow of leaking fluid is derived from the continuity equation. The effect of fluid-wall friction is accounted for by the momentum equation with friction and drag coefficients expressed in terms of Reynolds and Mach numbers for each type of clearance.

The injection of oil or other liquids, for lubrication, cooling or sealing purposes, modifies the thermodynamic process substantially. The same procedure can be used to estimate the effects of injecting any liquid but the effects of gas or its condensate mixing and dissolving in the injected fluid or vice versa should be accounted for separately.

In addition to lubrication, the main purpose for injecting oil into a compressor is to cool the gas. The solution of droplet energy equation in parallel with the momentum equation yields the amount of heat exchange with the surrounding gas.

The equations of energy and continuity are solved to obtain $U(\theta)$ and $m(\theta)$. Together with $V(\theta)$, the specific internal energy and specific volume $u=U/m$ and $v=V/m$ are now known. T and p, or x can then be calculated. All the remaining thermodynamic and fluid properties within the machine cycle are derived from the pressure, temperature and volume directly. Computation is repeated until the solution converges.

For an ideal gas, the internal thermal energy of the gas-oil mixture is given by:

$$U=(mu)_{gas}+(mu)_{oil}=\frac{mRT}{\gamma-1}+(mcT)_{oil} \qquad T=(\gamma-1)\frac{U-(mcT)_{oil}}{mR}$$

Hence, the pressure or temperature of the fluid in the compressor working chamber can be explicitly calculated by the equation for the oil temperature T_{oil}.

For the case of a real gas the situation is more complex, because the temperature and pressure cannot be calculated explicitly. However, since the equation of state $p=f_1(T,V)$ and the equation for specific internal energy $u=f_2(T,V)$ are decoupled, the temperature can be calculated numerically from the known specific internal energy and the specific volume obtained from the solution of differential equations, whereas the pressure can be calculated explicitly from the temperature and the specific volume by means of the equation of state.

In the case of a phase change for a wet vapour during the compression process, the specific internal energy and volume of the liquid-gas mixture are:

$$u=(1-x)u_f+xu_g \qquad v=(1-x)v_f+xv_g$$

where u_f, u_g, v_f and v_g are the specific internal energy and volume of liquid and gas and are functions of saturation temperature only. The equations require an implicit numerical procedure which is usually incorporated in property packages. As a result, temperature T and dryness fraction x are obtained. These equations are in the same form for any kind of fluid, and they are essentially simpler than any others in derived form. In addition, the inclusion of any additional phenomena into the differential equations of internal energy and continuity is straightforward. A full account of the compressor model used in this work can be found in Stosic, Smith and Kovacevic (9).

2.2 The unsteady process in a lumped volume of the plant reservoirs and connecting pipes

A two tank plant model was investigated which enables closed systems to be simulated, such as refrigeration, air-conditioning and heat pump plants, as well as plants which operate under power cycles, like Joule, Rankine and Organic Rankine cycles to be investigated. In fact, since the one tank model is a special case of this and if volume of the compressor inlet tank is left very large or infinity, it may be used to simulate atmosphere. Thus, the developed two tank model can be used to obtain the one tank model results.

A two tank plant is presented in Figure 1. Gas from Tank 1 goes to suction of screw compressor, then it discharged to Tank 2 and through throttle valve goes back to Tank 1.

All connecting pipes in the compressor plant are considered to be short enough that their volume, together with the reservoir volumes to be summed up into one lump tank volume. This assumes that all the thermodynamic properties are uniform within such a control volume. Thus the conservation equations of continuity and energy already used in the compressor model may be utilized for the tank calculations. The tank filling/ emptying equations for that analysis are as follows.

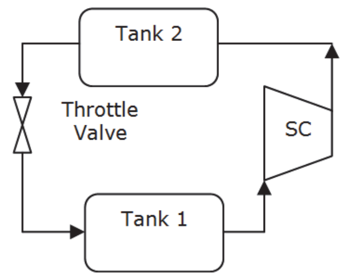


Figure 1: Two Tanks schematics

$$m_2u_2 - m_1u_1 = \left(\dot{m}_{in}h_{in} - \dot{m}_{out}h_{out}\right)\Delta t$$
$$m_2 - m_1 = \left(\dot{m}_{in} - \dot{m}_{out}\right)\Delta t$$

where indices 1 and 2 denote start and end time of filling/emptying respectively and Δt is time difference between these.

In the ideal gas case, the finite difference equations can be written as:

$$p_2 = p_1 + \frac{\gamma R \Delta t}{V}(\dot{m}_{in}T_{in} - \dot{m}_{out}T_{out})$$

$$\dot{m}_{out} = \mu A\sqrt{2\rho_2(p_2 - p_0)} \qquad \rho_2 = \frac{m_2}{V} \qquad T_2 = \frac{p_2}{R\rho_2}$$

To estimate the unsteady behaviour of a compressor plant system, the tank equations are coupled with the compressor model equations and solved in sequence to obtain a series of results for each time step. When the pressure p_2 in the tank at each time step is known, the flow and temperature m_{in} and T_{in} at the compressor discharge can be calculated. These derived values are then taken as the input parameters for the next time step. When the tank pressure p_2 is calculated. m_{out} is either known, or calculated as the flow through the exit throttle valve to pressure p_0 and T_2 becomes T_{out} in the next time step. The calculation was repeated until the final time was reached.

Mass inflow and outflow was calculated as a pipe flow with restrictions which comprised line and local losses therefore defining pressure drops within the plant communications. Since the tanks are of far higher volume than the communications, which results in far lower gas velocities, the losses in the tanks are far lower than the pipe losses.

Two levels of programming were applied, firstly the compressor and plant processes were solved separately. The compressor process was calculated through SCORPATH software and the tank model is processed, with mutual interchange of output and input data. Since this combination appeared to be slow in data transfer, the compressor and tank procedures were programmed together to get instant data exchange. This resulted in a very quick calculation allowing the bulk estimation of the unsteady behaviour of a screw compressor plant under various scenarios.

3. EXPERIMENTAL VERIFICATION OF THE RESULTS OBTAINED FROM THE MATHEMATICAL MODEL

The air compressor test rig with the oil flooded air compressor was used to validate the predicted results. An oil-flooded twin screw compressor was coupled to a 75 kW electric motor and driven by a six-band belt drive which speed is controlled by a frequency converter. A two stage oil separator consists of two separator tanks joined together by a short pipe for which the maximum working pressure is 15 bars. The oil cooler is a shell and tube heat exchanger. In this system, the oil is injected into the compressor by means of the pressure difference between the oil separator and the compressor working chamber. A motor driven throttle valve after the oil separator is used to control the air pressure inside the oil separator.

Apart of the laboratory ambient temperature and pressure, which were manually put into the test rig computer, all measured physical quantities are obtained as electric signals and transferred to an InstruNet data logger. Instantaneous values of pressure, temperature, speed and torque are displayed on the test data monitor. More details about the particular measurements can be found in Chukanova et. al. (7).

Measurement records were collected twice a second and saved in a separate file which was used for further analysis. Before measurements, the compressor and its plant were run for 30 minutes to obtain steady temperature in the compressor casing and to bring the oil temperature to its working level.

The experimental and predicted results of pressure variation during the start up, presented in Figure 2 for the starting receiver pressures 8 and 10 bars, show good agreement.

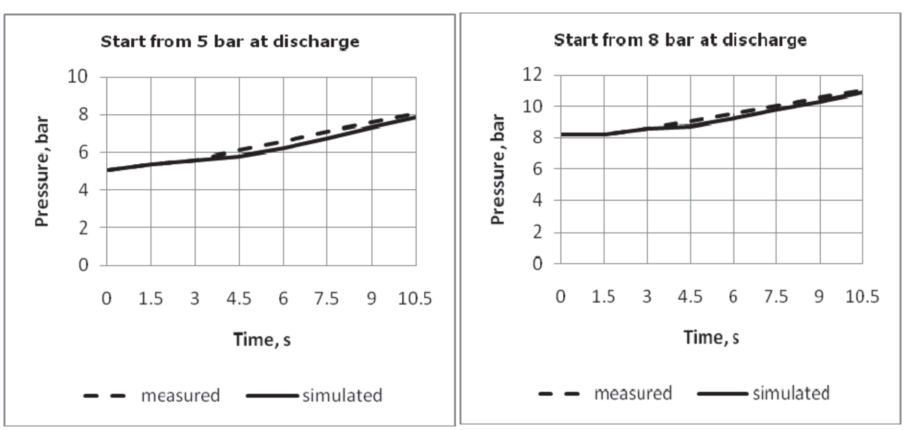


Figure 2: Diagrams of pressure rising in the tank during the compressor start

The tested compressor has a lobe configuration 4/5. The main rotor diameter is d=128mm, while the length to diameter ratio is L/d=1.55. Final speed of the main rotor during the experiment was retained constant and equal to 3000 rpm.

4. PRESENTATION AND ANALYSIS OF THE SIMULATED CASES

As previously stated, the developed model which combines the compressor and plant together gives a good opportunity to simulate various kinds of instabilities which might happen during real compressor plant operation. Several cases were presented and analysed for variety of starting tank pressures, tank volumes and valve areas, all of them for an infinite volume inlet tank, atmosphere. Then a two tank model results were presented which enables to closed cycle systems.

The same compressor was used as the basis for experimental testing, as presented in the previous section, where several cases were considered to check the plant model viability. The results are presented in groups, showing the effects of varying the throttle valve area, the tank volume and the tank pressure, as well as by varying the compressor shaft speed.

4.1 Variation of Valve Area

γ	R J/molK	V m^3	p_1 bar	Tin K	Shaft Speed rpm	Δt s	A_v m^2	p_0 bar	T_{out} K
1.4	287	0.30	1	350	3000	1	See below	1	350

Case 1 – A_v=70mm^2, Case 2 – 50 mm^2, Case 3 – closed valve, Case 4 – 30 mm^2

From the diagrams in Figure 3 it can be seen how the pressure and temperature change for different valve areas. For example for the case of the closed valve pressure in the tank reaching 33 bar in less than 2 minutes and the temperature of the air increasing from 350K (77°C) to 450K (177°C) in just 10 seconds. In fact, this confirms how the valve area can be used to control the discharge pressure.

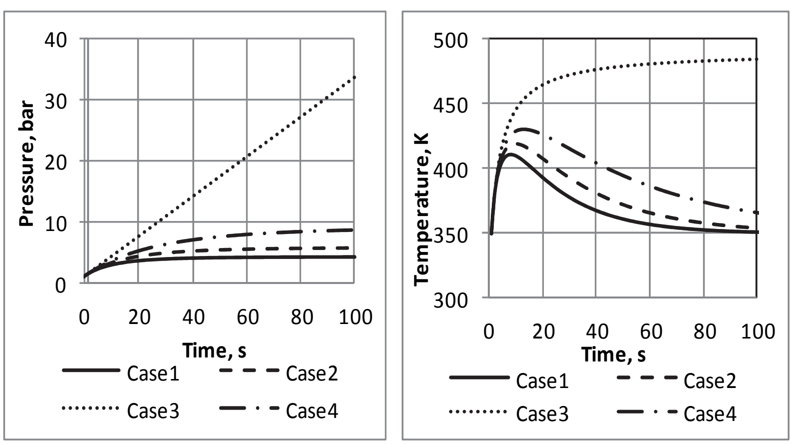
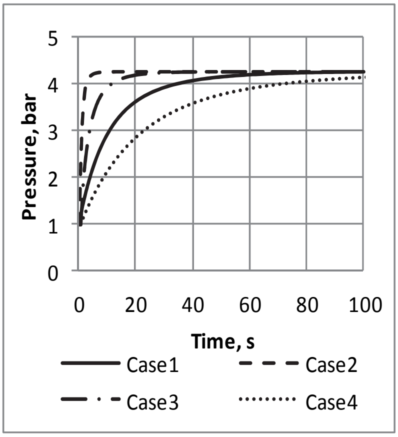


Figure 3: Pressure (left) and Temperature (right) in the tank for cases with different valve areas

4.2 Changing of Tank Volume

The tank volume is varied as follows: Case 1 – V=0.3m^3, Case 2 – 0.03m^3, Case 3 – 0.1m^3, Case 4 – 0.6m^3.

It can be seen from the diagrams in Figure 4 that for a given throttle valve area, the final discharge pressure will be the same for different tank volumes. It is only a question of the time for it to reach its final value: for a tank of 30 litres it will be 2 seconds, for 600 litres about 2 minutes. Similar characteristics apply to the temperature: the less volume the faster temperature reaches its peak (400-420K) and the faster that it returns to its initial value of 350K.



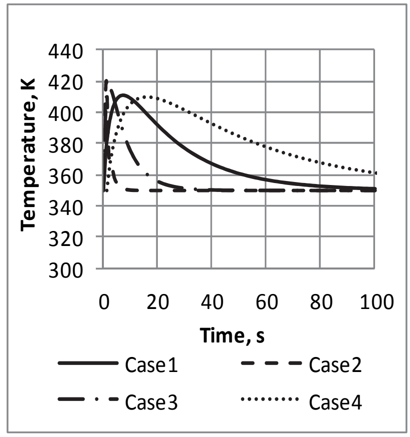
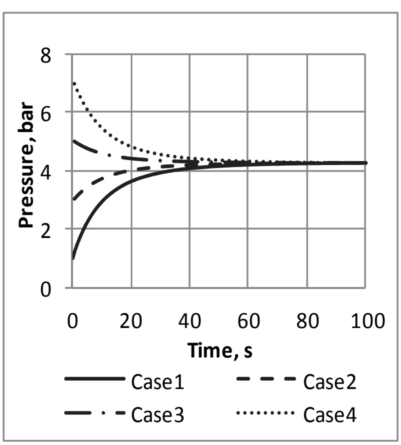


Figure 4: Pressure (left) and Temperature (right) in the tank for cases with different tank volumes

4.3 Changing of Tank Pressure

The pressure in the tank was varied as follows: Case 1 – p_2=1bar, Case 2 – 3bar, Case 3 – 5bar, Case 4 – 7bar.

The diagram in Figure 4 confirms that whatever the starting pressure, it will reach a final value which is determined by the valve area. This is 4.2 bar in all cases, as presented in Figure 5. It is shown that for starting pressure lower than 4.2 bar the pressure will quickly rise together with temperature. Conversely, if pressure if the starting tank pressure is above 4.2 bars, the pressure and temperature will rapidly fall to their final values.



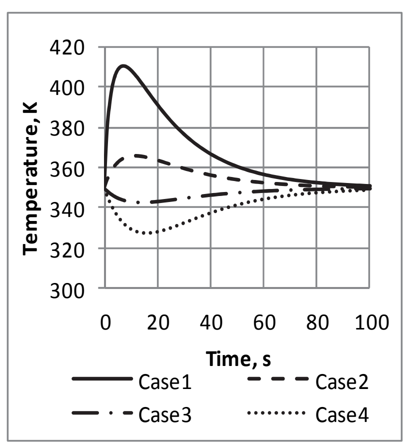


Figure 5: Pressure (left) and Temperature (right) in the tank for cases with different starting pressure in the tank

4.4 Two Tank Case

A two tank plant model was then developed to enable closed systems to be simulated. In fact, the one tank model is a special case of this and if volume of the compressor inlet tank is left very large, or infinity to simulate atmosphere, the one tank model results will be obtained.

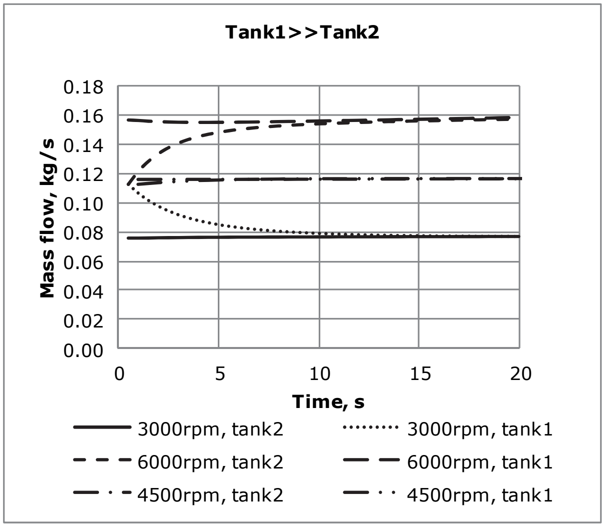


Figure 6: Pressure in the Tanks 1 and 2 for different shaft speeds

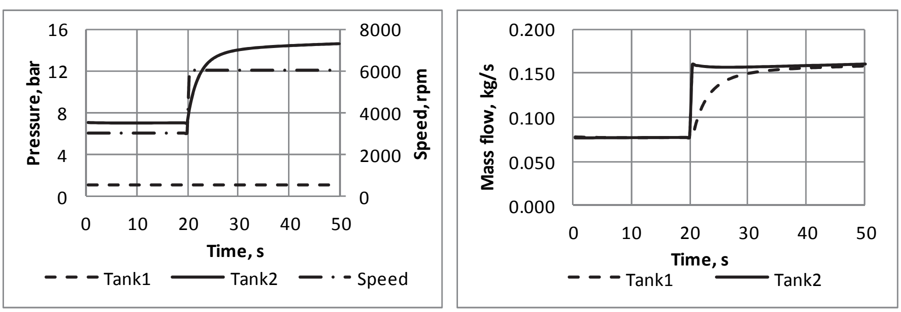


Figure 7: Pressure in the tank, left and mass flow in and out, right for speed variation from 3000 to 6000rpm

Figure 7 left shows the effect of a sudden change in the compressor shaft speed during the plant operation. The pressure in the discharge tank is doubled when the shaft speed is doubled. In this case, the volume of the inlet tank is kept much larger than that of the discharge tank.

As soon as the shaft speed is increased from 3000 to 6000rpm the pressure in Tank 2 starts to rise, but the pressure in Tank 1 remains almost constant because of its larger volume. As a result, the mass flow rate to Tank 2 doubles immediately, but the flow rate from Tank 2 into Tank 1 needs some time to reach this value, as a result of the increase of pressure in Tank 2, as shown in Figure 7 right.

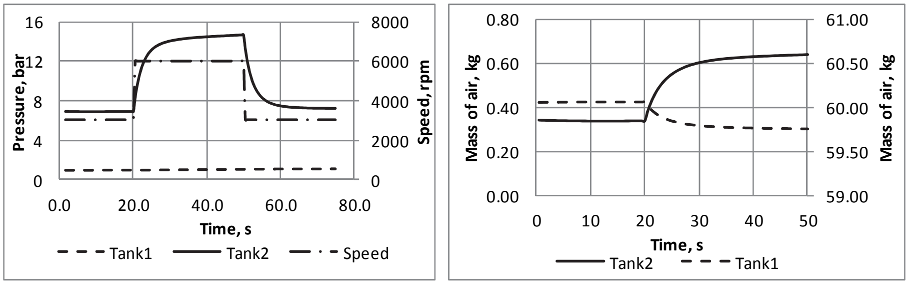


Figure 8: Pressure, left and mass, right in both tanks while changing speed from 3000 to 6000rpm and back to 3000

The resulting variation of mass contained in the tanks with time is presented in Figure 8 right. If the speed increases from 3000rpm to 6000rpm and then is brought back to its original value, the pressure history, as shown in Figure 8 left confirms that the pressure reaches its starting value.

5. CONCLUSION

By including the tank volume and other elements of a compressor plant system, into a well proven mathematical model for estimating screw compressor performance, it was possible to calculate the interaction between compressors and their systems under unsteady conditions. The predicted results, thus obtained agree well with measured results. Thus the simulation procedure has been validated and can be used as a useful and convenient tool for the analysis of unsteady behaviour of screw compressors in their plant. The modelling techniques were developed in a step by step iterative process starting with a simple analytical model and systematically taking more into account factors. This model is a powerful instrument that simulated a variety of scenarios which may occur in everyday compressor plant practice.

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