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Modelling and Experimental Investigation of Unsteady Behaviour of a Screw Compressor Plant

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ABSTRACT

An experimental and analytical study of screw compressor operation under unsteady conditions has been carried. For this purpose a one dimensional model of the processes within a screw compressor based on the differential equations of conservation of mass and energy was extended to include other plant components, taking into account tanks and connecting piping. The analytical model was then further developed to demonstrate whole plant transient operation when a screw compressor is connected with other elements in a complete air compressor system, consisting of a positive displacement compressor, low and high pressure tanks and connections between them, and auxiliary equipment, such as control valves, storage tanks and oil and air coolers. Results of the model have been compared with experimental data and show good agreement. The model allows simulation of unsteady plant operation under various scenarios which may occur within engineering practice and calculates plant dynamics at any given time.

Key words: Mathematical modelling, Screw compressor process, Compressor plant

1. INTRODUCTION

The petrochemical and chemical sectors of the oil and gas industry, contribute significantly to the GDP of countries with developed economies. Compressor plant is often at the heart of the processes on which they operate and its failure or inefficient functioning can cost millions of dollars. Accordingly, some of the problems associated with both steady and unsteady operation of such process plant and its dynamic simulation have already been described in the following publications:

Startup and Shutdown of Process Plant

Ogbonda (1987) investigated dynamic simulation of chemical plants and stated that “information about the stability of the process during start-up and shutdown can help process engineers evolve better start-up and shutdown procedures”. The Honeywell Dynamic Engineering Studies Group (2012) confirmed that dynamic models of new plant designs, its review and testing can make shorter initial start-ups, thus making significant money and time saving for large process plants.

Jun and Yezheng (1988 and 1990) carried out experimental studies on the effects of working fluid migration during the start-up and shut-down processes in a refrigeration system operating with a reciprocating compressor. They developed a program to estimate energy losses and how to calculate their effect, with the aim of reducing energy consumption.

Fleming, Tang and You (1996) published a paper on the 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 to prevent reverse rotation in order to decrease the compressor backflow significantly. As a consequence, the shutdown torque was reduced. However, only a mathematical model was presented, invalidated by experimental data. 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 (2009) 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 (2010) 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 (2011), deals with the numerical methodology and experimental validation of start-up and shutdown transients in reciprocating compressors.

Effects of Parameter Changes

Mokhatab (2007) showed the necessity and relevance of plant dynamic modelling. This paper describes the dynamic simulation of offshore production plant where parameters, such as flow, might change frequently. A dynamic model which is able to predict the effects of severe slugging or unstable flow of the offshore process plant was developed. This model was verified by experiment and it is shown that the model can be used as “a useful engineering tool for the reliable simulation of separation facilities during normal transients and more serious upsetting conditions”. Another benefit of this model, which has been confirmed by other authors, is that “by using this model one can check if the production system handles unstable flows or if the proposed production control system stressed”. It is mentioned that slugging leads to unstable plant operation and even to its shutdown and restart. Also it is said that it is very important to have an accurate dynamic model which allows for the accurate design of the separator size to avoid oversizing, because every kilo counts on offshore platforms.

The same paper mentions two other unstable situations in plant behavior: “At the conceptual design stage, dynamic simulation studies are particularly valuable in evaluating process design options and carrying out controllability studies. During the detailed design phase, dynamic simulation can be used as a tool to check and develop start-up and shutdown procedures and examine case scenarios”.

Control Systems Design

The Honeywell Dynamic Engineering Studies group (2012) worked on different aspects of dynamic simulations including compressor control and process design and controllability. They stressed that “the expenses of damages to compressor systems can quickly run into the tens of millions dollars, not only due to the cost of equipment but also to the loss of profit during plant downtime”. Also, it is said that for big process plants it is essential to stop them just once in 2-3, sometimes even in 5 years. That’s why it is critically important to provide dynamic models including detailed compressor models, valves, tanks and pipes to know answers to questions of the type “what would happen if..” Some of these answers can be provided by dynamic simulation studies and will play a vital role in decision making for improvement in the design, or the testing of new designs before they are built (Ogbonda, 1987).

Bezzo *et al*. (2004), who studied steady-state and dynamic simulation of the purification stage for VCM (vinyl chloride) industrial plant, stated that dynamic modellingg is a powerful tool to assess control system performance and for hazard analysis in case of abnormal events. Dynamic simulators can used to design control systems and to verify its effectiveness. Also this paper demonstrates that steady-state and dynamic simulations can be used by plant engineers for better understanding of process behaviour, similar idea was concluded by other authors stated above.

The work now describes a model to simulate 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 start. 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 succeeding time steps. The sequence is repeated for the whole compressor plant system until the specified time is reached.

The model was verified by comparing predictions obtained from it with measurements obtained in a series of tests performed on a compressor test rig. A detailed description of the experiments is given in Chukanova *et al*. (2012). A part related to the model verification is presented in section 4 of this paper.

2. MATHEMATICAL MODEL OF THE SCREW COMPRESSOR PLANT

Screw compressor modeling contains the analysis of thermodynamic and fluid flow processes. Both are dependent on the screw compressor geometry and combining them in a mathematical model results in 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 compressor, for example gas leakage, heat transfer between gas and oil which extends the model for a wider range of applications: any gas or liquid-gas mixtures of known properties can be used as a working fluid, model takes into account heat-transfer between the gas and the compressor rotors or its casings and leakages between rotor-to-rotor and rotor-to-casings are taken into account as well.

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:

$$\begin{aligned} &\omega\left(\frac{dU}{d\theta}\right) = \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out} + \dot{Q} - \omega p\frac{dV}{d\theta} \\ &\omega\left(\frac{dm}{d\theta}\right) = \dot{m}_{in} - \dot{m}_{out} \end{aligned} \tag{1}$$

where θ is angle of rotation of the main rotor, h=h(θ) is specific enthalpy, $\dot{m} = \dot{m}(\theta)$ is mass flow rate p=p(θ), 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, Eq 2. The computer code also accounts for reverse flow.

Suction and discharge port velocities are calculated through Eq 2:

$$w = \mu\sqrt{2(h_2 - h_1)} \tag{2}$$

Leakage in a screw machine is a substantial part of the total flow rate and affects the compressor delivery, i.e. the volumetric and adiabatic efficiencies, 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 the 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(θ) and m(θ). Together with V(θ), 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:

$$U = (mu)_{gas} + (mu)_{oil} = \frac{mRT}{\gamma - 1} + (mcT)_{oil} \tag{3}$$

Hence, the pressure or temperature of the fluid in the compressor working chamber can be explicitly calculated from Eq 3 and equation of state of ideal gas for the given oil temperature T_{oil}.

In 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, the pressure can then be calculated explicitly from the temperature and the specific volume by means of the equation of state.

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

$$u = (1-x)u_f + xu_g \qquad v = (1-x)v_f + xv_g \tag{4}$$

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 *et al*. (2005).

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

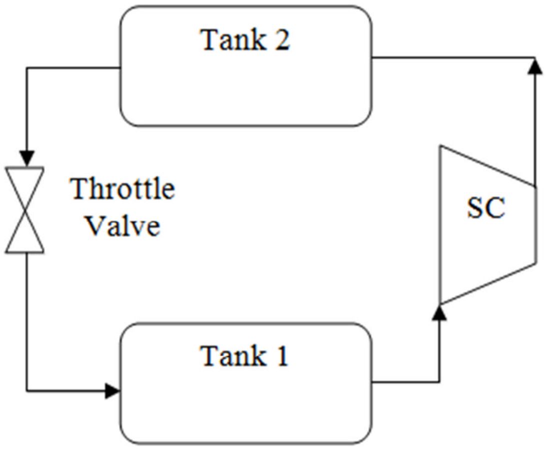

Figure 1: Scheme of Compressor Plant

All connecting pipes in the compressor plant are considered to be short enough for their volumes, 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 derived from (1) and (2) in form of finite differences are as follows.

$$\begin{aligned} m_2u_2 - m_1u_1 &= \left(\dot{m}_{in}h_{in} - \dot{m}_{out}h_{out}\right)\Delta t \\ m_2 - m_1 &= \left(\dot{m}_{in} - \dot{m}_{out}\right)\Delta t \end{aligned} \tag{4}$$

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

The ideal gas case may serve as an illustration in which the finite difference equations of thermodynamic and flow parameters can be written as:

$$p_2 = p_1 + \frac{\gamma R \Delta t}{V}(\dot{m}_{in}T_{in} - \dot{m}_{out}T_{out}) \tag{5}$$

$$\dot{m}_{out} = \mu A\sqrt{2\rho_2(p_2 - p_0)} \qquad \rho_2 = \frac{m_2}{V} \qquad T_2 = \frac{p_2}{R\rho_2} \tag{6}$$

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 for the flow through the exit throttle valve to pressure $p_{0,}$ and T_2 becomes T_{out} in the next time step. The calculation is repeated until the final time is reached.

Mass inflow and outflow are calculated as pipe flow with restrictions which comprise both line and local losses, thereby 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 the software suite which simulates the screw compressor process updated to cover the plant process and it is presented in tabular and graphical form by use of EXCEL, with mutual interchange of their input and output data. This resulted in a quick calculation allowing the bulk estimation of the unsteady behaviour of a screw compressor plant under various scenarios.

3. PRESENTATION AND ANALYSIS OF THE SIMULATED CASES

As it has been stated before, 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 give further opportunity to test closed systems, like refrigeration, air-conditioning and heat pump plants, as well as plants which operate under power cycles, like Joule, Rankine, Organic Rankine and Wet Rankine cycles.

The oil flooded compressor described in the measurements presented in the next section, was used as a basis for testing several cases which were imposed to check the plant model viability. The results are presented in a few groups, based on variation of the throttle valve area, the tank volume and the tank pressure, as well as by variation of the compressor shaft speed.

3.1 Variation of Valve Area

The input data are given in Table 1.

Table 1: Input Data for case 3.1

γ	R [J/molK]	V [m^3]	p1 [bar]	Tin [K]	Shaft Speed [rpm]	Δt [s]	A_v [mm^2]	p0 [bar]	Tout [K]
1.4	287	0.30	1	350	3000	1	See Fig 2	1	350

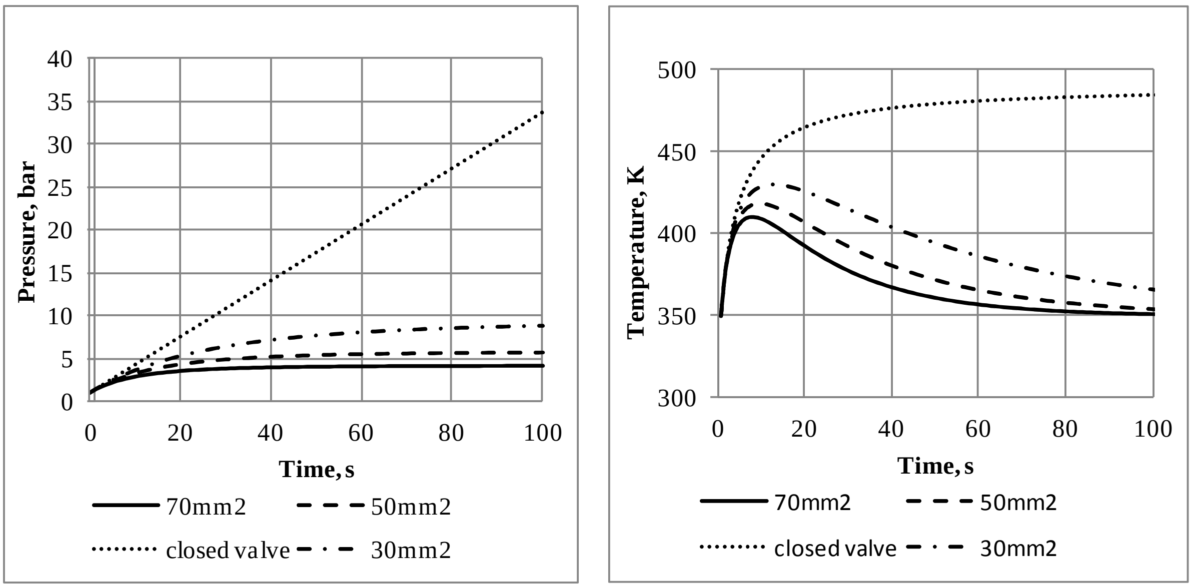

Figure 2: Pressure (left) and Temperature (right) in the tank for cases with different valve areas

From the results shown in Figure 2 it can be seen how the pressure and temperature change for different valve areas. For example, in the case of the closed valve, the pressure in the tank reached 33 bar in less than 2 minutes and the air temperature air increased from 350K (77°C) to 450K (177°C) in just 10 seconds. It might be very dangerous for an oil-injected compressor if some oil goes to the tank because of oil-flash (depending on the oil properties). If the valve is slightly open, the temperature of the air reaches its peak of 400-450K in 10 seconds and then stabilizes together with the pressure in the tank. Actually, the valve area controls the discharge pressure in the pipe.

3.2 Variation of Tank Volume

The input data are presented in Table 2.

Table 2: Input data for Case 3.2

γ	R [J/molK]	V [m^3]	p1 [bar]	Tin [K]	Shaft Speed [rpm]	Δt [s]	A_v [mm^2]	p0 [bar]	Tout [K]
1.4	287	See Fig 3	1	350	3000	1	70	1	350

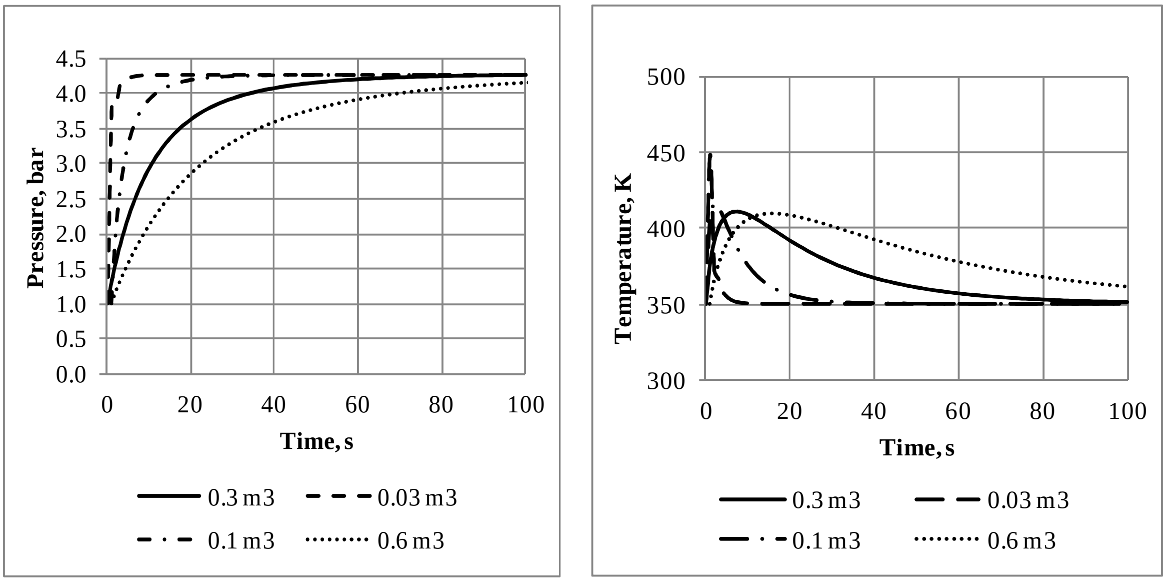

Figure 3: Pressure (left) and Temperature (right) in the tank for cases with different tank volumes

It is evident from the diagram in Figure 3 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 time when it reaches its final value. Thus for a tank of 30 litres it will be 2 seconds and for 600 litres about 2 minutes. Similarly, with the temperature: the smaller the volume, the faster it reaches its peak (400-420K) and the faster it decreases to its starting value of 350 K.

3.3 Variation of Tank Pressure

Table 3: Input data for Case 3.3

γ	R [J/molK]	V [m^3]	p1 [bar]	Tin [K]	Shaft Speed [rpm]	Δt [s]	A_v [mm^2]	p0 [bar]	Tout [K]
1.4	287	0.3	See Fig 4	350	3000	1	70	1	350

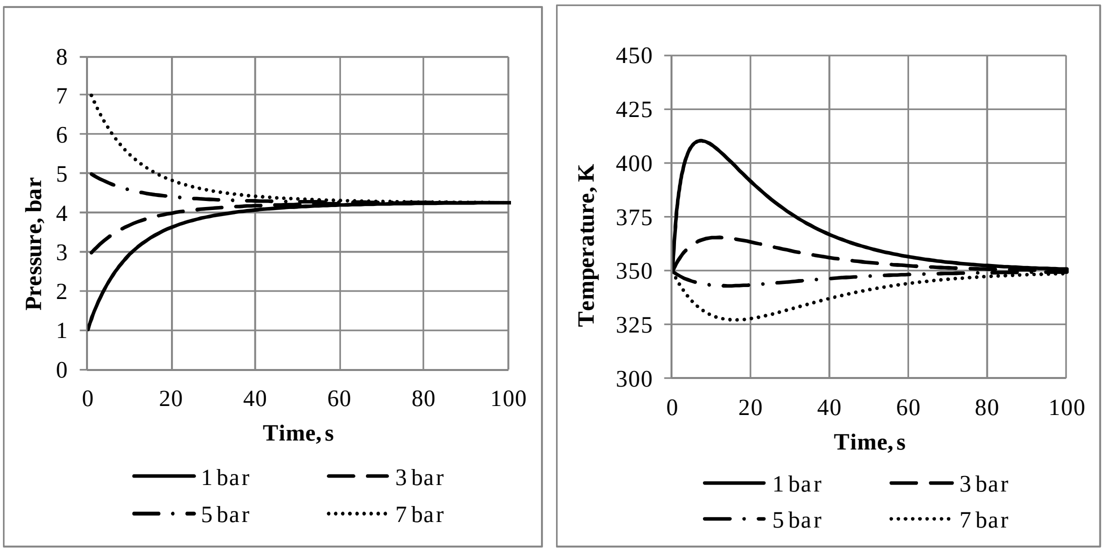

Figure 4: Pressure (left) and Temperature (right) in the tank for cases with different starting pressure in the tank

The curves in Figure 4 once again confirm that, whatever the tank starting pressure, its final value will be defined by the valve area. For all cases, the discharge pressure in the pipe came to 4.2 bar as for the previous case, shown inFigure The diagrams show that for starting pressures in the tank less than 4.2 bar the pressure will rise rapidly with the temperature and, if the starting pressure in the tank is higher than 4.2 bar the pressure and temperature drop but stabilize at 4.2 bar and 350K.

4. EXPERIMENTAL VERIFICATION OF THE RESULTS OBTAINED BY MATHEMATICAL MODEL

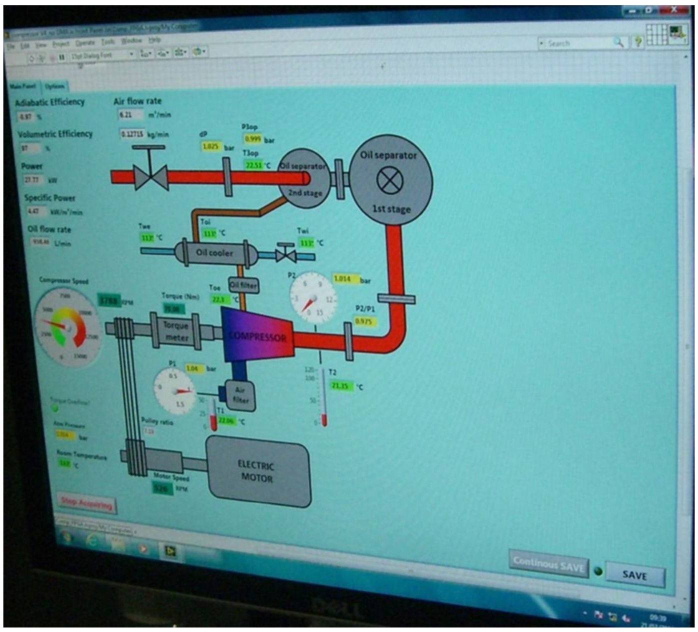

Figure 5: Test-rig Layout

An air compressor test rig presented in Figure 5 with an oil flooded twin screw air compressor was used to validate the predicted results. A 75 kW electric motor, with its speed controlled by a frequency converter, was used to drive the compressor via a six-band belt drive. The tested compressors had 4/5 and 3/5 lobe combination for the oil-flooded and dry compressors respectively, the main rotor diameter being d=128mm, while the length to diameter ratio L/d=1.55. The compressed air then passed through a two stage oil separator, consisting of two separator tanks joined together by a short pipe, for which the maximum working pressure was 15 bars. The oil cooler was a shell and tube heat exchanger.

In this system, the oil was 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 compressor plant was used to control the air pressure inside the plant.

Apart from the laboratory ambient temperature and pressure, which were manually input into the test rig computer, all measured physical quantities were obtained as electric signals and transferred to a National Instrument Labview system. Instantaneous values of pressure, temperature, speed and torque were displayed on the test data monitor. 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.

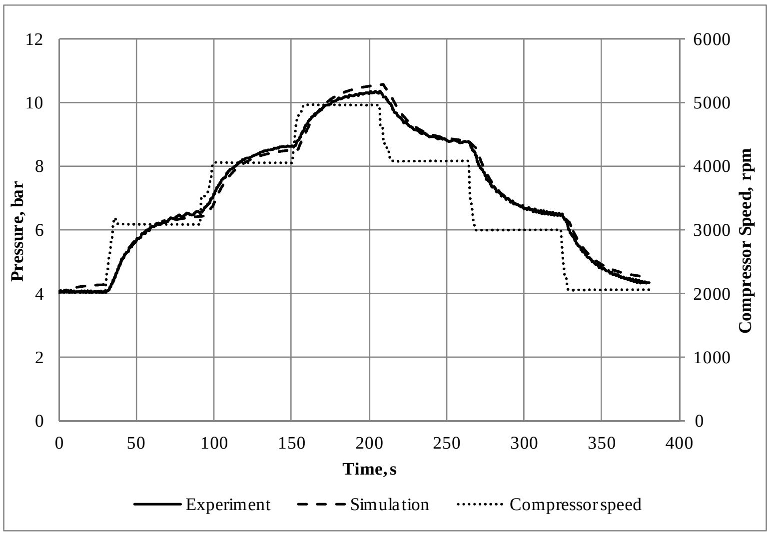

Figure 6: Comparison between experimental results and simulation data for oil-injected screw compressor plant

During experiment the compressor speed was varied up from 2000 to 5000rpm and back with step 1000rpm and time interval 60 seconds. Pressure dynamics is represented in Figure 6 and shows good agreement between experiment and simulation. The biggest difference is for the highest pressure and speed values: 10.6bar for simulation and 10.3 bar for experiment, it is around 2.8% difference.

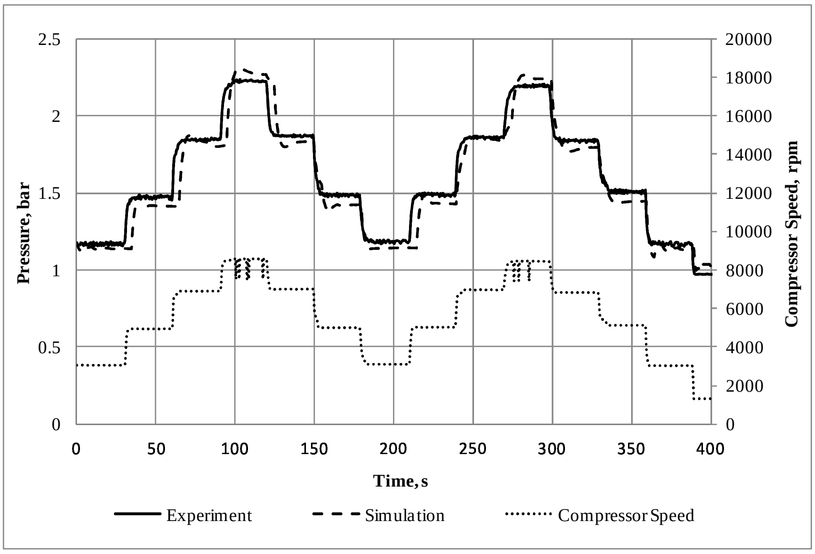

Figure 7: Comparison between experimental results and simulation data for oil-free screw compressor plant

A similar experiment with speed variation was done for oil-free compressor as well in Figure 7. Although the maximum difference between experiment and simulation around 3.9% it is still good agreement. Speed as in previous case was varied manually, from 3000rpm to 9000 rpm with step 2000rpm and time interval 30 seconds. As can be seen from diagram, pressure in oil-free compressor changes almost immediately after changing speed while pressure in oil-injected is changing gradually. It can be explained by oil thickness which creates some inertia effect.

5. CONCLUSIONS

By including the volume of the compressor plant system, within a well proven mathematical model of screw compressor performance, it was possible to calculate the interaction of compressors and their systems under unsteady conditions. The obtained prediction results agree well with measured results for air compressors. Thus the simulation procedure has been validated and may serve as a useful tool for analysis of unsteady behaviour of screw compressors and their plant. Although the results are related to the simulation of specific examples, we believe that the described approach is quite general, so that analysis and conclusions can be easily extended to the analysis of most process screw compressor plants. This model is a powerful instrument that can simulate a variety of scenarios which may occur in everyday compressor plant practice.

NOMENCLATURE

A	valve area
m	mass
m_{in}/ m_{out}	mass flow entering/ leaving the tank
p_0	atmospheric pressure
p_1	pressure in the compressor tank
p_2	pressure in the tank at the next time step
R	gas constant
T_2	temperature in the tank
T_{in}/T_{out}	temperature of the gas in/out the tank
Δt	time step
U	internal energy
U	specific internal energy
V	volume of the plant containing tank and pipes
M	flow coefficient
ρ_2	density of gas in the tank
θ	angle of rotation of the main rotor
$\dot{Q} = \dot{Q}(\theta)$	fluid heat transfer
$\dot{m} = \dot{m}(\theta)$	mass flow rate
$V=V(\theta)$	volume of the compressor working chamber
$h=h(\theta)$	specific enthalpy
$p=p(\theta)$	fluid pressure in the working chamber control volume

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