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Investigation of Start Up Process in Oil Flooded Twin Screw Compressors

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ABSTRACT

Oil injected screw compressors used for air applications and refrigeration are usually installed in simple packages which often do not include oil pump for supply of oil to the bearings and the working chamber. The period of an un-lubricated operation depends on the size of the oil system and the length of the discharge piping. When the back pressure reaches the chamber pressure, a normal mode of lubrication will start. Due to the lack of lubrication, rotors will be in direct contact with insufficient or no oil film between them, while the pressure in the compression chamber will increase causing the temperature to rise. The leakage flow will be higher than normal which will increase the overall temperature in the compression chamber. It is expected that some surface damage may occur on the rotors. In a case of a frequent start-stop mode, such operation may cause quick wear and rapid decrease in the compressor performance. This paper is expected to address such issues and to define a scope of work required to understand the oil flooded compressor process in a transient operation mode. This will allow prediction of wear in such compressors and will give some insight to the required modifications to prevent these issues and increase the compressor reliability.

1. INTRODUCTION

Investigation of transient modes of screw compressors became more and more demanding due to the fact that most of compressor plants operate permanently at unsteady conditions. In spite of importance of this problem, there is still a shortage in public knowledge and lack of verified experimental data which describe how transient operation affects the compressor performance. Currently a majority of the existing papers in unsteady operation of positive displacement compressors are related to reciprocating compressors or to the complete compressor systems, for example in refrigeration, air-conditioning, heat-pumps, as well in process industry and just a few of them describe a transient behavior of screw compressors. Mathematical models and tools need be developed to predict the screw compressor performance at unsteady conditions. Also, experimental work needs be performed to verify these models. As a result, appropriate modifications in compressor design may be done in order to improve the compressor reliability during its transient operation.

A start-up mode is one of the transient processes which affects the performance and reliability of a screw compressor and the whole compressor plant system. There are several papers in open literature which describe that process, generally they address the control problems and energy losses during the transient operation. Jun and Yezheng (1988 and 1990) experimentally studied the effects of refrigerant migration during the start-up and shut-down cycles of a refrigeration system with reciprocating compressor. They developed a program for estimation of energy losses and calculation due to this migration. Then Fleming, Tang and You (1996) published a paper on simulation of shutdown processes in refrigeration plant with a screw compressor. Their idea was to use a reverse rotation brake instead of a suction non-return valve. This prevented the reverse rotation, which led to significant decrease of the compressor backflow, which went through the clearances only and, as a consequence, reduces the shutdown torque. But this was a modelling only which still waits for experiment data for validation. A disadvantage of the reverse rotation brake might cause danger and could cause a failure if a significant rotor backlash exists. Li and Alleyne (2009) investigated the start-up and shutdown transients 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 the on-off cycle operation and validated it as a part of the 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 the start-up and shutdown transients for reciprocating compressors.

As it can be seen from the above paper review, nobody elaborated a start-stop transients for screw machines. This paper presents experimental results and their analysis in a start-up mode of the oil flooded screw compressors. These results can be further utilized in dynamic modelling of screw compressor. Some issues of the screw compressor dynamic modelling have already been considered, because the simulation potential was increased due to the much more powerful computers. Papers of Sauls, Weathers and Powell (2006) presented a transient thermal analysis of screw compressors. A control volume model based on the principles of conservation of mass and internal energy was applied in the first instance and then the derived values of pressure and temperature were used as boundary conditions for the 3-D Finite Element Method. Detailed description of such methods are presented in books by Stosic, Smith and Kovacevic (2005) for the chamber model and Kovacevic, Stosic and Smith (2007) for the screw compressor CFD. The integrated model was presented on IMechE Conference by Kovacevic, Mujic, Stosic and Smith (2007). The Integration combines benefits of the both methods and allows faster calculation and more accurate results. Krichel and Sawodny (2011) presented a model for dynamic simulation of an oil-flooded screw compressor. They split the machine into four subsystems: throttle-valve, motor, screw compressor block, and oil/air separator and presented them as a separate mathematical models. It was emphasized in their paper that the warm-up and shut-down phases require a lot of energy and that this is often ignored when studying a compressor in the quasi-stationary state. So, this again confirms that the screw compressor transients are worth of researching and that the existing and advanced mathematical models should be adopted and improved in order to predict the compressor performance during their unsteady operation.

2. TEST RIG AND EXPERIMENT DESCRIPTION

The experimental work described in this paper was performed by utilization of the existing compressor test rig equipment, which is introduced in Figure 1. The oil-flooded twin screw compressor is driven by a six-band belt drive coupled with a 75 kW electric motor which speed is controlled by a frequency converter, the two stage oil separator consists of two separator tanks joined together by a short pipe which maximum working pressure is 15 bars. The oil cooler is a shell and tube heat exchanger. This system does not have pump, the oil is injected to the compressor by means of the pressure difference between the oil separator and compressor working chamber. A motor driven throttle valve after the oil separator allows control of the air pressure inside the oil separator.

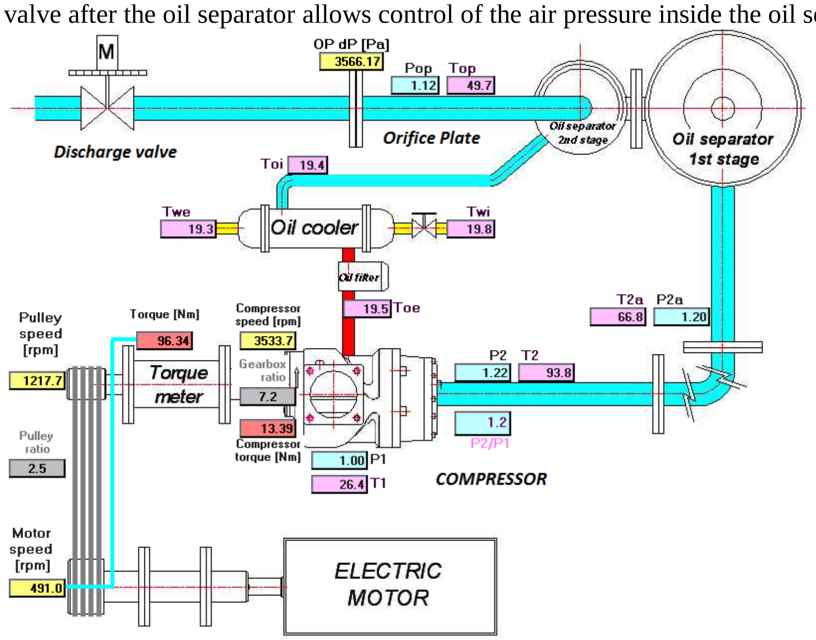


Figure 1: Test Rig Scheme

The tested compressor, presented in Figure 2 has a lobe configuration 4/5. The main rotor diameter is d=128mm, while the length to diameter ratio is L/d=1.2. Speed of the male rotor during the experiment was kept constant and equal to 3000 rpm.

Table 1: Description of the Test Rig Instruments

MEASURED PARAMETER	INSTRUMENT	SPECIFICATIONS
Compressor Speed, **N**	Tachometer (LED RPM transducer)	60 TTL pulses per revolution, Volt = 0-12v dc Accuracy= 0.1%
Compressor Torque, **M**	TRP-500 torque meter (strain gauge transducer)	max capacity: 500Nm, Calibration level: 335Nm Range = 0 - 6000 rpm, Supply volt=10v dc, Accuracy= 0.25 % of max capacity
inlet pressure, $\mathbf{P_1}$	PDCR 110/w -pressure transducer (piezoresistive type)	Operating range = 3.5bar(abs) Excite voltage=10V dc, Accuracy =0.6%,
inlet temperature , $\mathbf{T_1}$	K- type thermocouple (based on Ni/Cr-Ni/Al alloy)	Range= -200^{0}C to 1300^{0}C, Accuracy= ±2.2^{0}C sensitivity = 41 μV/°C
outlet pressure, $\mathbf{P_2}$	PDCR 922-pressure transducer (piezoresistive type)	Operating range =15 bar (abs) Excite voltage=10V dc, Output voltage= 100 mV Accuracy =0.6%
outlet temperature, $\mathbf{T_2}$	K- type thermocouple (based on Ni/Cr-Ni/Al alloy)	Range= -200^{0}C to 1300^{0}C, Accuracy=± 2.2^{0}C sensitivity = 41 μV/°C

Apart of the laboratory ambient temperature and pressure, which are manually put into the test rig computer, all measured physical quantities of the test rig are obtained as electric signals and transferred to an InstruNet data logger. Instantaneous values of pressure, temperature, speed and torque are displayed on the monitor, as presented in Figure 1.

Existing acquisition system makes measurement records twice every second and saves the data in a separate file which is used for further analysis. Before measurements, the compressor was warmed-up for 30 minutes to uniform all temperatures along the casings and to bring the oil temperature to the working level.

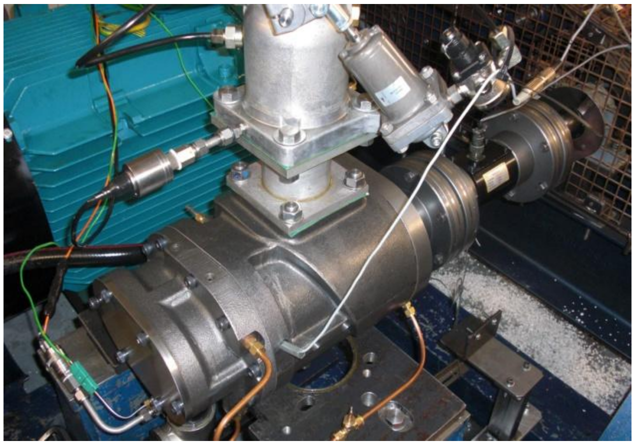


Figure 2: Tested machine

There were several types of compressor starts investigated: a start from the atmospheric pressure at the compressor discharge and a start from the higher discharge pressures, then a start with the closed discharge and a start with the open discharge, as well as a start with the closed and open suction valves. All the recorded data are then analyzed with aim to find out trends and patterns in the screw compressor working parameters.

3. RESULTS AND DISCUSSION

As it has been said above, the idea was to see the behaviour of the compressor before the oil injection started. It is expected that during the start-up the temperature rises quickly before the oil goes in, which may cause some minor damage on the rotor surface. In case of a frequent start-stop mode, such operation may cause quick wear and a rapid decrease in the compressor performance.

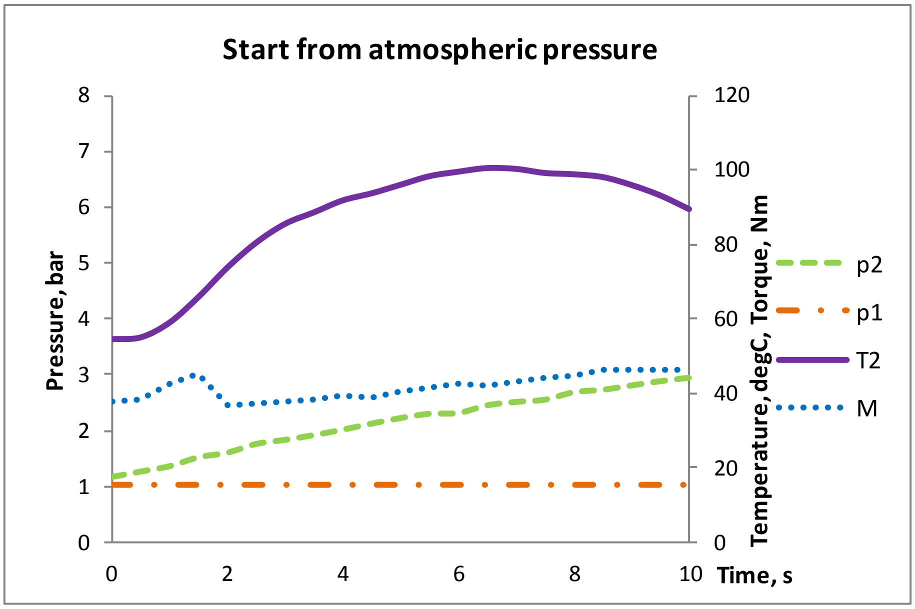


Figure 3: Start from atmospheric pressure at discharge

It can be seen from Figure 3 that, when compressor starts from atmospheric pressure, the temperature increases from 55 up to 100 °C and after 8 seconds it decreases due to the oil-injection (p1, p2, T2 and M described in Table 1). It is needed some time while the pressure builds-up in oil reservoir and oil will go in the compressor due to this pressure difference.

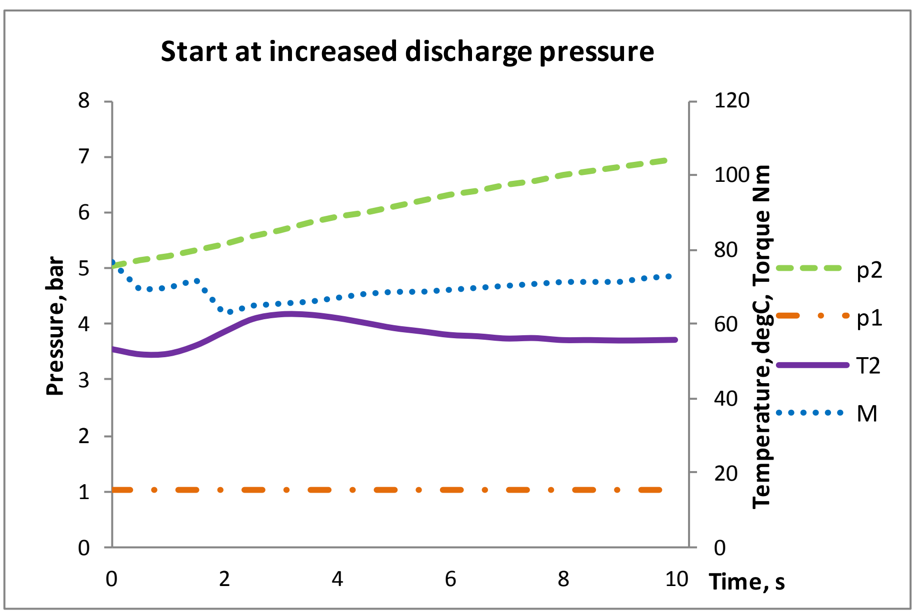


Figure 4: Start at increased discharge pressure

The situation is different when compressor starts with the pressure higher than atmospheric pressure at its discharge, as presented in Figure 4. In this case, when compressor stops, the oil flows into the compressor due to the pressure difference, but as the rotors do not rotate, the compressor will be full of oil. So, when it starts rotating, the oil will flow to the compressor discharge and the temperature will immediately drop down and it will be stabilized soon after that. In the first case, Figure 3 when compressor stops, the discharge is open and all the oil flows out of the compressor. As a consequence, when the compressor starts, the compressor is empty, there is no oil inside and there is no pressure difference for the oil flow.

This will be a period of dry contact between the rotors, which can be decreased by closing both, the compressor suction and discharge during the start. When the compressor starts, the pressure difference increases immediately due to the suction pressure drop (Figure 5). As a result there will be a significant temperature rise from 50 up to 90°C in the first 2 seconds due to the high compressor pressure ratio. Then the pressure difference reaches the required level for oil injection, the oil flows in and the temperature drops down. The un-lubricated period is decreased from 8 seconds for an ordinary start, Figure 3 down to only 2 seconds for the start with the closed suction. After some time, 32 seconds in Figure 5, the suction will be open manually and the temperature will rise again but it does not mean that there is no oil inside. The reason for that is increased inflow on suction and there is not enough oil to keep the temperature at the same level. The temperature will increase in the next 2 seconds until the pressure difference is built-up and more oil is injected. But the second increase is less significant than first, because the temperature increased from 55 up to 70 °C only during this phase.

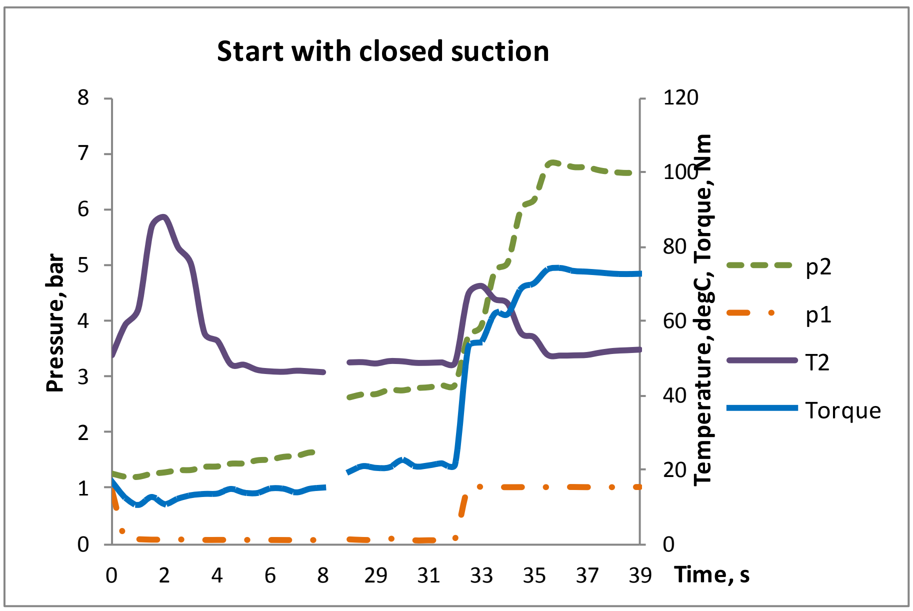
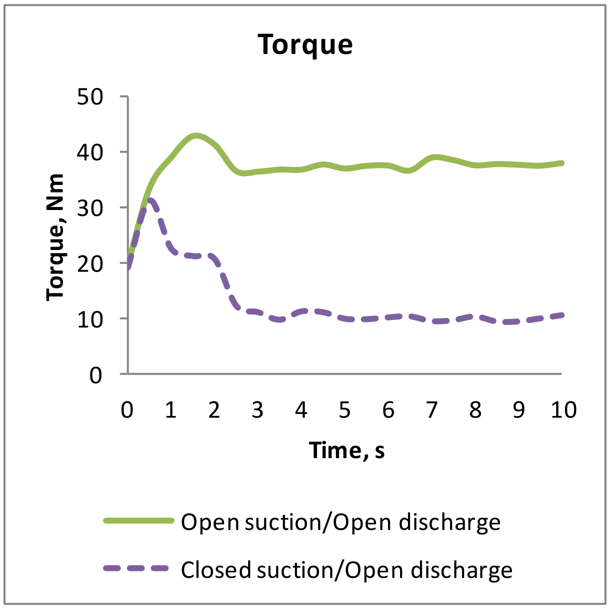


Figure 5: Start with closed suction

This is not the only benefit of starting the compressor with the closed suction. There is a comparison in Figure **6** of the ordinary start, which is similar to the process presented in Figure 3, which starts with the closed suction. As soon as the suction is closed, the compressor is started, the suction pressure gets close to the vacuum and air flow is almost zero. At the same time, the discharge pressure will be equal to the atmospheric pressure as in the case when the discharge is open. Due to the small pressure difference and low air flow, the peak of torque is approximately one third less than during the start with the open suction, and the starting torque is about four times less. As a consequence, this will result with a safer start and lower power consumption, as well as, lower noise.



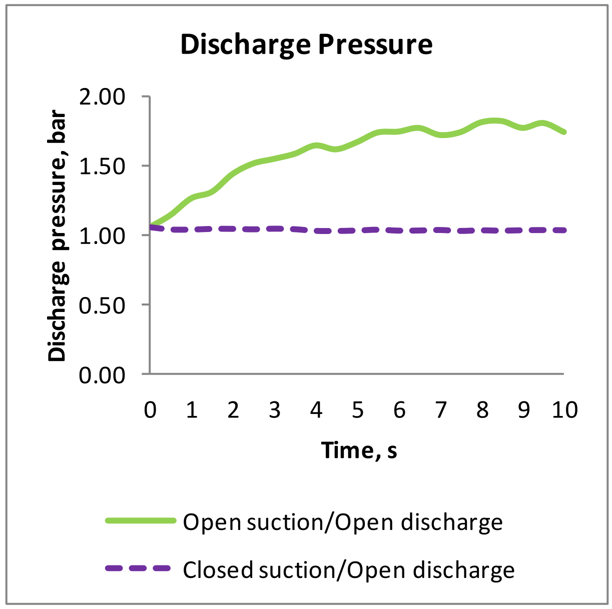
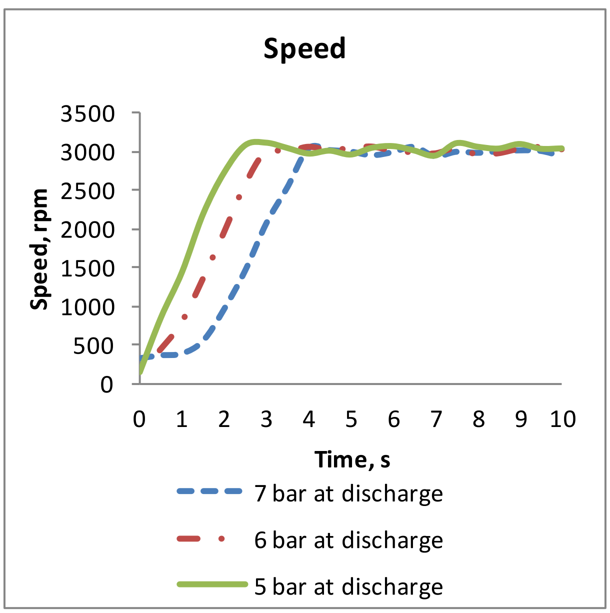


Figure 6: Comparison of start with open and closed suction

After starting the compressor under different conditions, it was found out that the higher discharge pressure, the more time is required for the compressor to achieve 3000 rpm and its acceleration is lower, as presented in Figure 7 It is clearly visible that the slowest start and the lowest acceleration is in the case of 7 bar at the compressor discharge, then at 6 bar and is the highest in the case of 5 bar at the compressor discharge. Some attempts were done to run the compressor with 8 bar at its discharge, but, since the motor current was limited, the compressor failed to start for all discharge pressure higher than 7 bar.



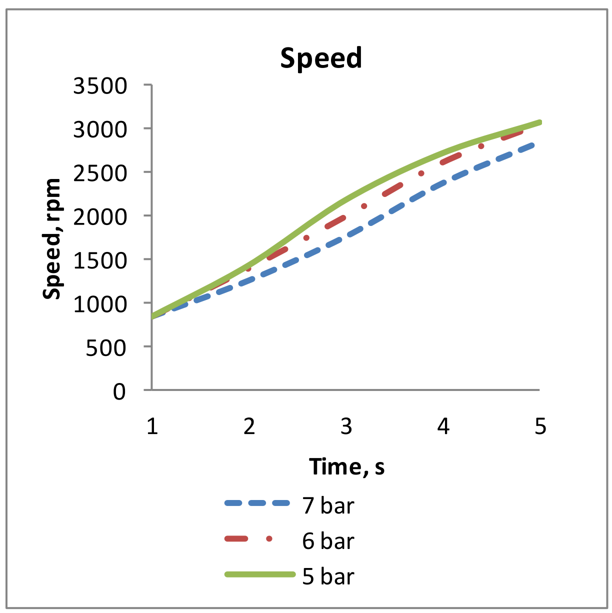
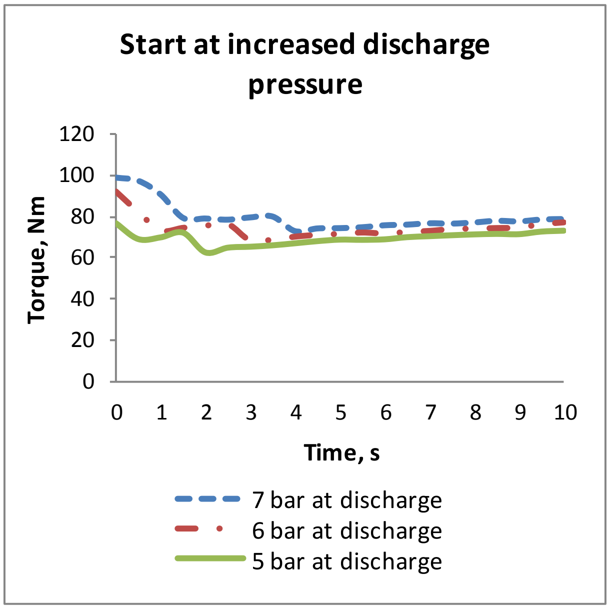


Figure 7: Speed curves for start modes with different discharge pressures

The torque diagrams shown in Figure 8 indicate that the highest torque is required to run the compressor at 7 bar, the lowest is to run it at 5 bar. When the compressor starts at 5 bar, the discharge pressure starts increasing immediately, the start at 6 bar shows decrease of pressure during the first second and then it start rising. The start at 7 bar at the compressor discharge is the slowest, the pressure decreases during the first 2 seconds and then starts rising. So, the higher the pressure during the start, the higher torque is required and as a result, the slower start. It would be useful to calculate inertia moments for these cases in the future.



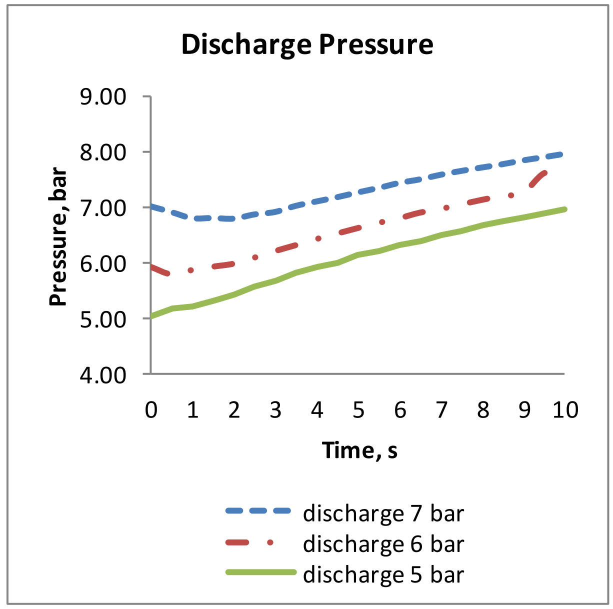


Figure 8: Torque and discharge pressure curves for starts with different discharge pressures

CONCLUSIONS

This paper gives useful insight into the screw compressor behaviour during its start-up. The experimental results presented in diagrams show such transients as speed, torque, discharge temperature and pressures for different types of the compressor start modes. This data will be used as a base for further research, like for developing a mathematical model simulationg the screw compressor transient processes and conditions.

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