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EXPERIMENTAL INVESTIGATION OF THE INFLUENCE OF OIL INJECTION UPON THE SCREW COMPRESSOR WORKING PROCESS

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SUMARY

Presented are some results of experimental investigation - complemented by computer modelling - of the influence of oil injection upon screw compressor working process. Several parameters that characterize the oil injection were varied over ranges that were preliminary determined on the basis of the computation model. These include: oil mass flow rate, inlet temperature, droplet atomization, positions in the casing at which the oil was injected, oil jet speed and angle, time of oil retention in the working volume. The compressor performances were evaluated on the basis of measurements of all important bulk parameters: delivery rate, power consumption. power utilisation efficiency, specific power, as well as the instantaneous pressure and temperature at several positions along the working volume, from which the indicating diagram was worked out. In addition to the information on the influence of each quoted oil parameter upon the engine performances, the collected data served to verify and complement the earlier developed mathematical model of the oil influence upon the screw compressor working process, which has subsequently been employed for the computer-aided-design of two different screw compressor oil systems. The obtained experimental results and the application of the computer simulation helped in modifying the oil injection system which resulted in saving of compressor energy consumption up to 7%.

INTRODUCTION

Major advantages of screw compressors in comparison with other compressor types lie in their compactness, durabilty and wide operating range. As a rotating engine of a relatively simple design (it does not contain any oscillating elements) it can develop a high rotating speed and consequently a high power and efficiency per unit weight, as well as a wide range of operating pressure and delivery. However, the performances of a screw compressor are very sensitive to a number of design parameters which govern the thermodynamic and flow process /1, 8, 9/. A particularly strong influence is exerted by the injection of oil which, in addition to the lubrication purpose, provides better sealing and air cooling.

Classic approach to the design of new engines like screw compressors used to rest in the past upon lucid ideas of shrewd inventors and on subsequent systematic experimental testing and evaluation of effects of all relevant factors upon the work process and energy conversion efficiency. Nowadays a great deal of this lengthy and tedious work - including the verification of odd ideas - can be substituted by mathematical modelling and computer simulation of a process as a whole, or individual phenomena that constitute such a process. The solution of modelled equations can yield reliable predictions of compressor performances for the proscribed input variables, enabling thusfar an easy and fast arrival at optimum combinations of design parameters and work conditions. Mathematical modelling has become in such a way the basic ingredient and in the sdame time the major promotor of the computer-aided-design approach which serves nowadays as an unavoidable tool for optimum compressors design and optimization. A consistent mathematical model employs the exact kinematic relationship in differential form which describes the instantaneous operating volume and its changes with time or rotation angle, as well as the equations of conservation of mass and energy for the adopted control volume /11,12/. Major - still unresolved - differences among various models appear in the approach to the modelling of various accompaning phenomena, which can not easily be represented by general analytical formulations, such as fluid leakage, oil injection, heat transfer between gas on one side and screws and compressor housing on the other, and others, which influence to a large extent the final performances of a real engine. Because of these uncertainties, mathematical modelling and computer simulation have their limitations and the final answers have be obtained eventually on a test bench in a laboratory. Of course, the task is now easier since the computer optimisation has reduced to a great extent the uncertainties and tha span of possible variation of most parameters in question. This approach has been adopted in our work on development of new family of screw compressors. The mathematical model, developed and verified througn a detailed laboratory test of a selected screw compressor, served as a basis for deciding on most major parameters. The final optimization was achieved by careful and systematic experimental investigations of all factors that could not be decided upon with reliable certainty by computer modelling.

One of the important problem, which requires a careful investigation, is the influence of oil injected into the the working volume of most types of srew compressors for the purpose of sealing, cooling and lubrication. As it is well known, injected oil affects considerably the thermodynamic process in the compressor. Better sealing means less leakage of the gas and the improvement of the compression and delivery rates. In addition, an increase in the mass rate of the working fluid due to the oil injection lowers the compression temperatures, allowing in such a way higher compression rates /2, 3/. Several oil parameters that play important role in this porcess have to be optimized by the designer in order to achieve best effects and the analysis of the influences of these parameters have been subject of our investigation. These are: the size of oil droplet, oil inlet temperature, position and size of oil inlet port, velocity of the oil jet and oil viscosity. This paper focuses on the experimental investigation of the influence of oil injection upon the process in a screw compressor. A detailed study of the same problem was carried out by computer modelling, as reported in ref. /12/, and the the present work represents both, the experimental verification of the model, as well as the complementary research aimed at collecting experimentally those data that could not be obtained by the simulation model.

The experiments were carried out on two engine prototypes of major Yugoslav compressor manufacturer "Energoinvest-Trudbenik". A detailed account is presented of the experience gained in the selection of measuring methods, preparation of instruments, acquisition and processing of the measured data, with a particular emphasis on the mounting and connection of the high resolution pressure gauges within the tested engines /6/. The investigation was focussed on the analysis and optimization of three factors which contribute to a grat deal to the optimum effect of oil injection upon the screw compressor process:

- Conditions for appropriate atomization of the oil,
- Increase of the oil droplets retention time in a working space
- Determination of the optimum oil-to-air ratio which produces a minimum compressor specific power.

Each of these aspects will be elaborated in details in the flollowing chapters.

LABORATORY TEST RIG AND MEASURING EQUIPMENT

The cyclic gas compression in a screw compressor imposes a nonstationary pulsating fluid motion, and measurement of the instantaneous values of individual characteristics requires instruments of high dynamic response and a large frequency range. In the same time, due to the restricted space for their mounting, the transducers must have small overall dimensions, so that their presence does not spoil the physical and geometrical conditions of the compressor operation /5/. The following bulk operating characteristics were measured: delivery rate, power consumption, power utilisation efficiency, specific power, air- and oil temperatures. In addition, the instantaneous values of all relevant pulsating quantities were also recorded for the purpose of studying the details of the thermodynamic cycle, in particular the pressure in the compressor swept volume as the basis for devising the p-V diagram. To this purpose a new method was developed for constructing the compressor indicating diagram, which employs the recordings of the pressure changes at several discrete points along the fluid path over a period of time (range of rotating angle) /7/.

A sketch of the measuring installation is presented in Fig. 1 illustrating the main components of the compressor set, positions of the transducers and their coupling with the data acquisition and processing system. As could be seen, the compressed air, contaminated by the oil, enters first the oil separator and, after being decontaminated, it is ducted into the air reservoir. The oil separated from the air in the oil separator is taken into the oil cooler and afterwards injected back into the compressor. The investigated compressor was driven by a 75 kW 2490 rpm electrical motor coupled with a compressor via a torque transmission gear which ensured the compressor rotation speed to be increased by a ratio of 1,723. At the end of the shaft, a torque transducer was mounted together with an optical revolution indicator, connected to an amplifier and electronic counter.

Suction and discharge parameters were measured by placing the pressure and temperature senzors at the compressor inlet after the suction air filter and right at the engine exit port, respectively. A vortex flow meter for measuring the air flow rate was installed at the outlet of the oil separator just in front of the air reservoir. Oil temperature at the compressor inlet as well as at the oil cooler inlet and outlet, and the oil flow rate were also measured. A general view of a compressor test rig is given in the photograph in Fig.2.

Table 1. Measuring equipment used in experimental investigation of the screw compressor working process

Measuring value	Measuring equipment	Characteristics
Static pressure	Pressure transducer	range: 120 mb-20 b
	AVC200	output: 0-20 mA
		accuracy: 0,3 %
Dynamic pressure	Piezoelectric pressure	range:0 - 17 bar
	transducer XTME-190M	3.54 mV/bar
	Amplifier D81	output: 0 - 10 V
		accuracy: 0.1 %
Temperature	Thermocouple NiCr-Ni	range: -10 - 250 C
	Reference thermostat VST6	
	Measuring DC differential	output: 0 - 10 V
	amplifier	accuracy: 0.1 %
Shaft torque and	Torque shaft T30FN3/2	range: 0 - 2 kNm
speed of revolution		0 - 7000 rpm
		accuracy: 0.2 %
	Torque transducer MD3555	range: 0 - 5 kNm
		0 - 7000 rpm
		output: 0 - 10 v/DC
		accuracy: 2 %
	RPM transducer N3556C	range: 0 - 5000 rpm
		output: 0 - 10 V/DC
		accuracy: 0.1 %
Compressor delivery	Vortex flowmeter	range: 0 - 4 m3/min
	SWZ50Ag71.1/F2	2.5 - 30 bar
		accuracy 1%
	Signal transducer	range: 0 - 20 mA/DC
	SAW1.1UR10-S20Z	acuracy: 0.1 %
Oil flow	Rotameter OI50Ag19E/G1	range: 0 - 250 l/min
		0 - 25 bar
		accuracy: 2%
	Impuls amplifier with	range: 0 - 20 mA/DC
	current output ZIVVS20W	

The working volume of a screw compressor has a complex shape meandering through the engine along the axial and tangential direction. For this reason several pressure transducers had to be installed into the screw compressor casing and used simultaneously for recording the experimental data. To this purpose four holes were drilled in the casing along the direction parallel to the longitudinal screw axis (while the main and the gate screws were removed). A specially designed adaptor was used for installing the pressure gauge at the selected measuring place. The electric signal from the transducer was brought through the coaxial cable to the corresponding convertor and amplifier and then to the cathode osciloscope with memory on which the pressure variations in time were monitored. On the basis of known overall geometry of the compressor, the working volume and its change in time/rotating angle, the arrangement of the measuring points on the compressor housing and recordings of the instantaneous pressure at several ponts in the working space, it was possible to obtain the p-α and p-V diagrams. Typical recordings of four pressure transducers are presented in Fig.3, while the plots of indicating diagrams are presented in Fig. 6 and 7.

As could be seen from pressure recordning in Fig. 3 each of the three pressure gauges covers an angle range of about 80 degrees. By adopting the zero value of the rotating angle of the main rotor at the beginning of the compression, the ranges of the angles of rotation, covered by individual pressure gauges were respectively:

first gauge from 133.2 to 213.4 degrees, second gaguge from 211.4 to 291.6 degrees, - both covering the major portion of the compression process - and third gauge from 289.5 to 360.7 degrees, covering the lastr stage of the compression and the exhaust period. It should be noted that the exhaust process starts at the angle of 292 degrees. The oil injection into the compression chambre begins at the angle of 168 degrees and lasts through the next 79,5 degres of rotation of the main rotor.

Recording and acquisition of the measured data were obtained by the AVL TEST COMMANDER Compressor Tester. The equipment consists of Motorola mikroprocessor MC 6800 with ACIA interface modul, VDU control modul and flopy control modul, analog-to-digital converters, four six-channel multiplexers MUX (0-10 V/DC and 0-20 mA/DC), and separate sets for measuring temperature, pressure, torque and electric power consuption. Table 1. outlines basic characteristics of the data measuring and acquisition system.

OIL ATOMIZATION

In order to enhance the oil cooling efficiency (air-to-oil heat transfer) the surface area of oil droplet should be as large as possible for the same quantity of oil defined by lubricating requirements. To this purpose, the oil was atomized in small droplets by generating an oil spray in which the distribution of the droplets size (represented by droplet diameter D) is defined by the following differential equation:

$$\frac{dn}{dr} = a\, r^{p} e^{-br^{q}}$$

where a, p, b and q are distribution coefficients depending on oil to air flow and velocity ratios. The atomization of oil is achieved by a simple high pressure injection of an oil jet in the spacing between the main and gate rotors. After the During the injectionof oil, a period of oil jet destruction and a period of oil spray formation can be recognized. These time periods descrese with an encrease of air density, oil-to-air relative velocity and nozzle diameter. The mean Sauter diameter - for a general distribution of droplet sizes - is defined by

$$r = 0{,}0183\left[\frac{\rho_o \sigma_o}{\rho^{0,5} w}\right]\left(1+\frac{m}{m_o}\right)^{0,7}$$

Oil droplets are supposed to be spherical with known mean Sauter diameter. Heat transfer between the gas and oil droplets is defined by the simple cooling law $Q=\alpha A(T-T_o)\Delta\varphi/\omega$, where "$\alpha$" stands for the heat transfer coefficient, "A" is the average droplet surface area and "T" and "To" are gas and oil temperatures. The transfered heat must also satisfy another equation $Q= m\, c(T_o-T_{ob})$ where "m " is the oil mass rate, "c " is specific heat of oil, and Tob is the oil temperature in the previous time step.

The heat transfer coefficient " " at the droplet surface is obtained from the expression:

$$Nu=2+0{,}6\, Re^{0,6}\, Pr^{0,33}$$

where the oil velocity in Reynolds number is determined from the oil mass rate injected through the orifice of diameter D. In order to define the Re and Nu numbers the mean droplet diameter d was used as suggested by Sauter. The oil temperature is directly obtained from heat transfered from gas to oil by means of the equation

$$T_o=\frac{T+kT_{ob}}{1+k}$$

where $k=mc\,\omega/(\alpha A\Delta\varphi)=d\rho c/6\alpha\Delta\varphi$ is the time constant and o is the oil density. It is obvious, that for k=0 (high heat transfer coeficients or small oil droplet size) the oil temperature approaches the gas temperature.

OIL DROPLET DYNAMICS

The air-to-oil heat transfer occurs during the droplet free movement until the droplet reattaches a wall of the compressor screws or housing. Following /10/ the dominant external force governing the droplet is Stokes drag force defined by the coefficient:

$$C_k = \frac{4}{3}\frac{\mu Re C}{d^2 \rho}; \; C = \frac{24}{Re}, Re < 0{,}3; \; C = \frac{18{,}5}{Re}, Re = 0{,}3 - 1000; \; C = 0{,}44, Re \geq 1000$$

The resulting set of differential equations describing the movement of a droplet are (Fig.4):

$$\ddot{r} = r\dot{\theta}^2 - C_k \dot{r}; \; r\ddot{\theta} = -2\dot{r}\dot{\theta} - C_k(r\dot{\theta} - \omega r); \; \ddot{z} = -C_k(\dot{z} - r\, tg\alpha\, \omega)$$

The Z-axis is coaxial with the axes of the compressor screws. Oil is injected through the nozzle in compressor housing with velocity w. The initial conditions are:

$$\dot{r}(0) = w_r \qquad r(t_k) < R_{ez} \qquad r(t_k) < r_v$$
$$\dot{\theta}(0) = w_\theta / R_{ez} \qquad z(t_k) < -L$$
$$\dot{z}(0) = w_z$$

All differential equations are solved by means of Runge-Kutta fourth-order method. The calculation was performed for a range of data by varying the droplet size, nozzle location and oil velocity, and finding their optimum values which produce the most efficient air-to- oil heat transfer. Fig. 4 presents the calculated oil droplet trajectory for the case of highest air-to-oil heat transfer efficiency, expressed in terms of axial and radial positions over the time period of droplet relative movement within the air. The calculated values serve as input parameters for the oil nozzle design, expected to produce better oil atomization due to the considerable pressure drop in the nozzle and thusfar assuring an efficient oil injection.

DETERMINATION OF OPTIMUM OIL-TO-AIR RATIO

Because the temparature variation in a screw compressor working chamber is strongly dependent on oil inlet temperature and oil-to-air mass ratio, an analysis was carried out of oil temperature at the exit of the oil cooler. If the cooling capacity of the cooler is limited, the oil cooler outlet temperature is dependent on oil mass flow through the cooler. Although the processes of air compression and oil cooling are both nonlinear, it is expected that the compression process is more dependent on oil inlet temperature, which means that the minimum oil temperature at the compressor outlet (oil cooler inlet) could not be found. Hence for the purpose of mathematical modelling of screw compressor plant the oil mass variation was emplyed as an optimization parameter. The results are presented in Table 2 and in a diagram in Fig. 5.

Table 2. Determination of oil injection mass flow rate

Mo kg/s	t_i deg C	t_o deg C	Ns Wmin/m	Mo kg/s	t_i deg C	t_o deg C	Ns Wmin/m
0.069	32.1	91.6	7698	0.595	75.2	86.8	7221
0.140	49.2	87.4	7275	0.665	76.4	86.9	7226
0.210	58.5	86.7	7209	0.735	77.4	86.9	7231
0.280	64.2	86.6	7199	0.805	78.2	87.0	7234
0.350	68.1	86.6	7210	0.875	78.9	87.0	7237
0.420	70.8	86.7	7207	0.945	79.4	87.1	7240
0.490	72.9	86.8	7214	1.014	80.0	87.1	7242

It is visible from the Table 2 that there is an optimum value of oil mass flow rate resulting the minimum compressor outlet oil temperature which corresponds closely to the minimum compressor specific power. The optimum value of compressor oil flow was also confirmed experimentally yielding a basis for the compressor plant design and a plant components construction.

DISCUSSION OF THE RESULTS

The results obtained by experimental investigations and mathematical modelling of oil injection influence upon the screw compressor working process will be discussed in two different contexts: (i)- comparison of the experimental- and mathematically modelled and computed p-α and p-V diagrams (Fig 6 to 9) and (ii)- analysis and discussion overal parameters (in Table 3).

In Fig. 6 and 7 diagrams of the pressure in the compression volume are plotted over the rotation angle of the main rotor and over the compressor swept volume presenting the p-α and p-V diagrams for compressor outlet pressure of 9 b. Figs 8, 9 are plotted to show another set of p-α and p-V diagrams of the same variables for outlet pressure of 11 b . respectively. Two different oil injection systems were considered: the standard system which has been already in use in "Trudbenik" screw compressors, and a modified system with a newly designed oil nozzle . In the latter case the nozzle was placed at the position- and oriented in the direction which - in accord with both, the experimental findings and computer simulation - were supposed to yield the best air-to-oil heat transfer performances. All figures contain four curves, two for each considered oil injection system. Curves denoted by "1" and "2" refer to the standard- and "3" and "4" denote the modified system. The experimental data are denoted by dots, while the results obtained by the model computations are presented by lines. All of the results indicate that the pressure records obtained by modified oil injection system gave a lower pressure gradient during the compression, as compared with results recorded with the standard oil injection system. The phenomenon can be explained by a better air-to-oil heat transfer durig the first stages of oil injection, what reduces considerably the temperature rise of air during the compression, retaining a lower index of the polytrope. Lower values of the compressor indicating- and effective power should be expected in this case, ensuring a better power consumption efficiency and even a better compressor delivery efficiency. This phenomenon is specially evident for lower delivery pressures (9 and 11 b), and its influence becomes less important at higher outlet pressures.

A set of overall results is presented in Table 3 showing a comparison of model calculations with experimental data for the outlet pressure of 9 b. It should be emphasized that the highest air temperature (this occurs at the end of compression before the air-to-oil heat transfer is completed) obtained directly by modelling and calculated indirectly from experimental data by the heat ballance of the compressor, gave up to 30 deg C lower values with the modified oil injection system in comparison with the standard one. This temperature difference is considered to come as a consequence of two effects. Firstly the more efficient oil injection in the modified system results in better air cooling and consequently a more moderate air temperature rise as compared with the standard injection system. Secondly, the pressure slope obtained by the modified system fits better the built-in compression ratio, avoiding in such a way the external expansion (noticed in the standard system), which is more evident for the lower delivery pressures.

Table 3. A comparison of experimental and modelling results

		Oil system 1.		Oil system 2.	
		Model	Exp.	Model	Exp.
Outlet pressure	bar	9.00	9.00	9.00	9.00
Compressor delivery	m3/min	3.63	3.68	3.66	3.56
Shaft revolution speed	rpm	5151.82	5161.00	5151.82	5163.00
Teeth tip velocity	m/s	28.90	28.90	28.90	28.90
Specific power	kW/m3/min	7.30	7.12	6.76	6.84
Compressor power	kW	26.53	27.20	24.76	25.31
Oil flow	l/min	39.40	43.28	39.40	38.64
Oil inlet temperature	K	323.80	329.60	323.80	323.80
Air/oil outlet temperature	K	340.60	346.80	336.80	343.60
Air temp. after compression	K	388.00	-	357.40	-
Delivery efficiency	-	0.78	0.785	0.78	0.759
Power efficiency	-	0.50	0.523	0.54	0.572

The results of experimental investigation are compared with the results of mathematical modelling computed for experimental conditions (Figs. 6 to 9 and Table 3). The comparison indicates that the developed mathematical model of oil influence upon the screw compressor working conditions describes adequately the modelled process. All results proved the positive effects of the reconstruction of oil injection system:

- The oil mas ratio was reduced by 2-3 % as compared with previous oil systems.
- The same compresor delivery was achieved with a power consumption reduced by 2,76 - 7,42 %.
- However, since the pressure increase itself promotes better oil atomization, this improvement fades out with an increase of the delivery pressure, reducing thusfar the benefits of the reconstruction.

CONCLUSION

After a comprehensive analysis of the influence of various parameters of an oil injection system upon a screw compressor working cycle, carried out experimentally and by means of a computer simulation, a new, modified system with a more efficient nozzle, was designed for "Energoinvest-Trudbenik" screw compressors. Experimental investigations of two prototypes of screw compressor (one of them equipped with the standard-, and another with the new, modified oil injection system), enabled a detailed insight into the effects of all relevant parameters upon the screw compressor performances. In comparison with previous solutions, the new oil system proved to be more efficient and resulted in a smaller oil consumption, better air cooling and an improvement of overall compressor performances. Both, the model and experiments, lead to a conclusion that a considerable descrease of compressor specific power consumption and oil injection mass flow ratio could be achieved by a change of a nozzle diameter and its position. The achieved savings are higher for smaller compressors and for lower and moderate pressure ratios. The experiments also confirmed the validity of the basic concept of the mathematical model and its applicability to the Computer-Aided-Design of compressor systems.

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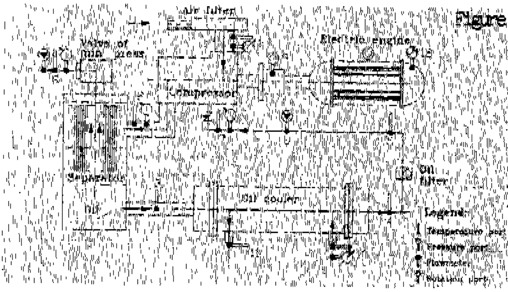
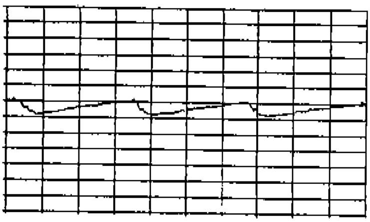
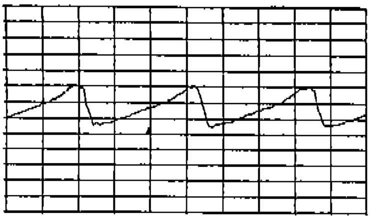


Figure 1. Shematic of the test installation
1. Compressor inlet, air
2. Compressor outlet, air
3. Oil coller inlet, oil
4. Oil cooler outlet, oil
5,6,7. Compressor inlet, oil
8. Oil separator outlet, air
9. Torque shaft and speed of revolution
10. Oil cooler inlet, cooling water
11. Oil cooler outlet, cooling water

Figure 2. A general view on the compressor testing rig

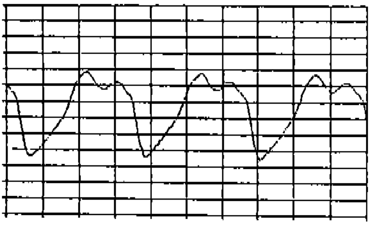


Transducer 1- Compression: 133 to 213 deg

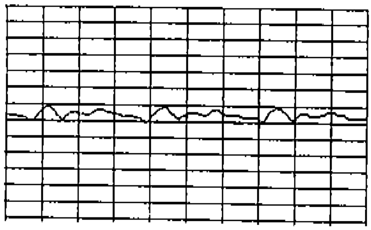


Transducer 2- Compression: 211 to 292 deg

Rate: 0,2 V/div - 1 b/div, 25 deg/div



Transducer 3- Compression and outflow: 290 to 361 deg



Transducer 4- Compressor outlet chamber - Outflow

Figure 3. Pressure transducer records in the screw compressor chamber

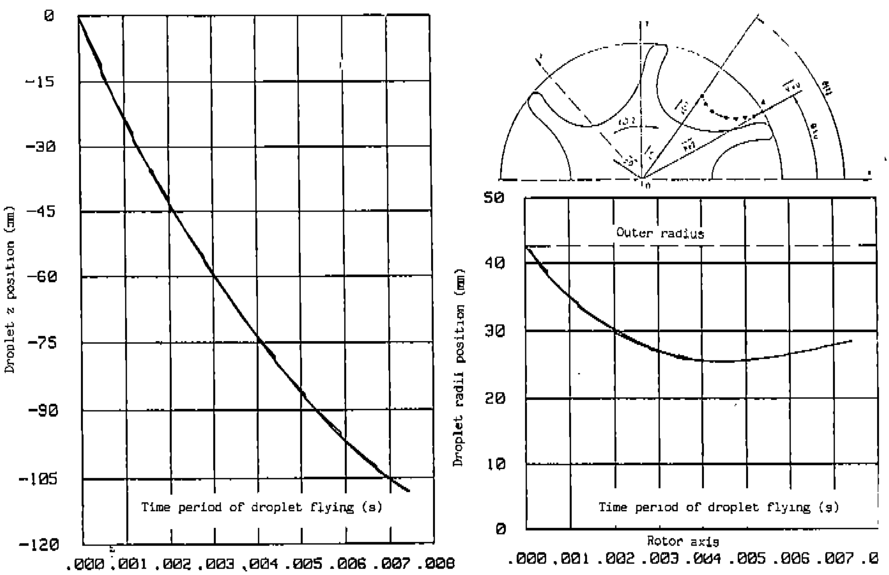


Figure 4. Oil droplet dynamics in the gate screw domain

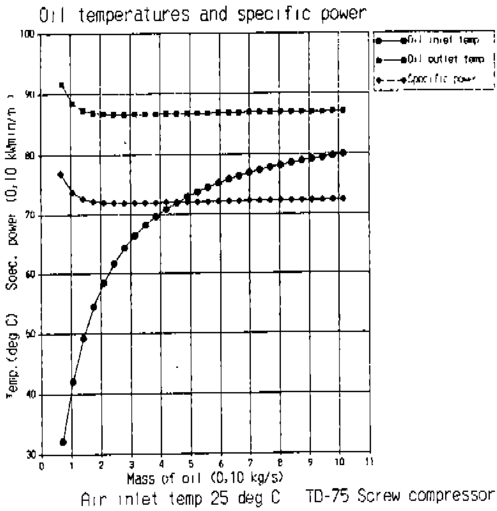


Figure 5. Oil mass ratio influence on the compressor plant overal performances

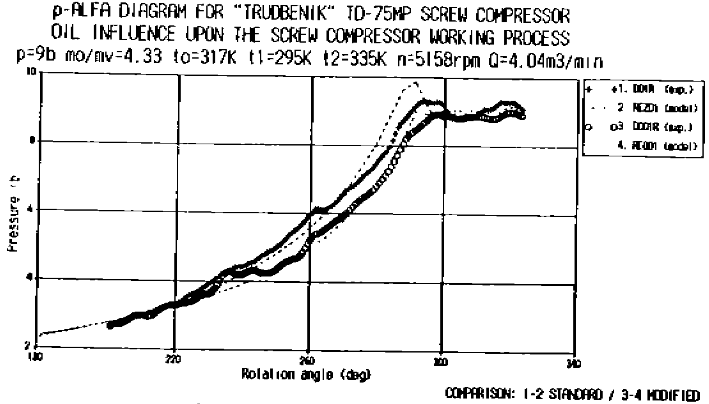


Figure 6. p-α Diagram. Oultet pressure 9 b

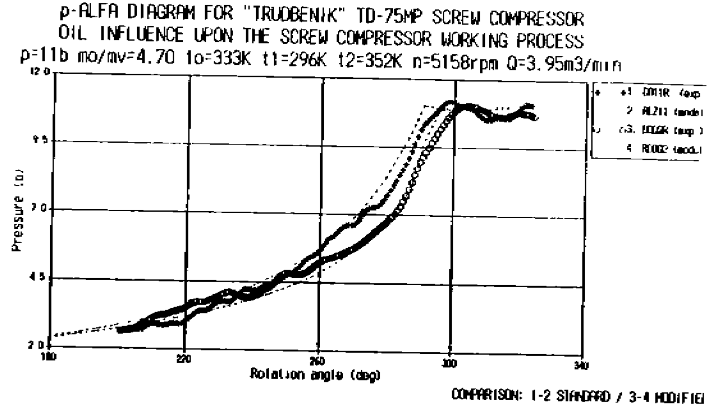


Figure 8. p-α Diagram. Outlet pressure 11 b

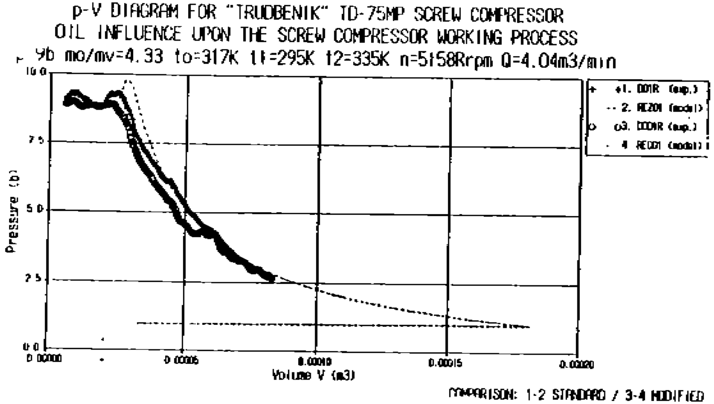


Figure 7. p-V Diagram, Outlet pressure 9 b

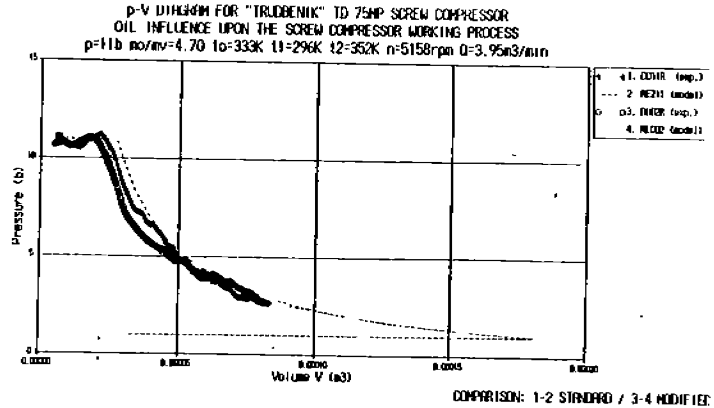


Figure 9. p-V Diagram. Outlet pressure 11 b