Warsaw University of Technology

Faculty of Automotive and Construction Machinery Engineering

INSTITUTE OF VEHEECLES

Laboratory of Combustion Engines Theory

<u>Lab work №3</u>

STUDY ON PISTON COMPRESSOR

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THE PURPOSE OF THE LABORATORY WORK

Purpose of the lab covers indicating the compressor and processing obtained indicator diagram as well as calculation of the thermodynamic parameters and energy quantities of the models of theoretical compressors; calculations of the thermodynamic parameters and energy quantities in the cycle of real compressor and analysis of the characteristics in the real compressor as compared to theoretical one.

1. EXPERIMENTAL UNITE

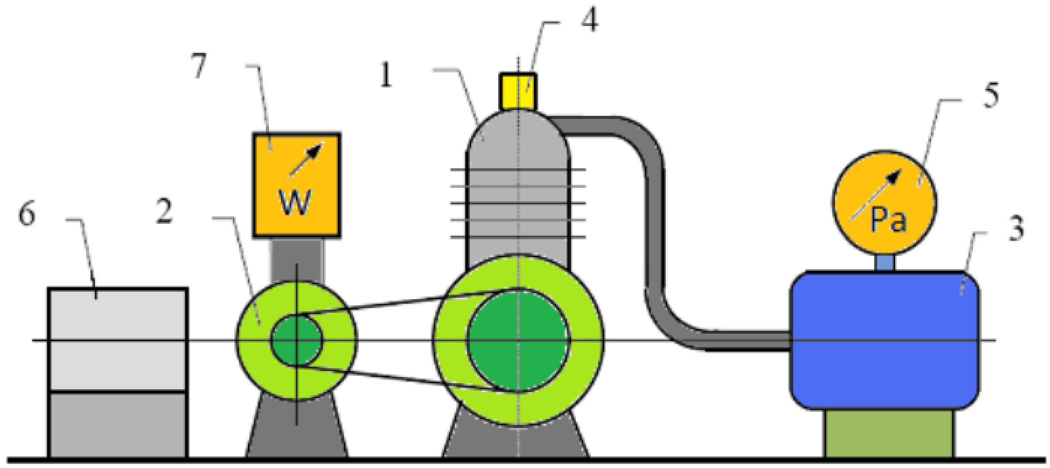


Fig. 1.1. Installation diagram: 1 – piston compressor, 2 - three-phase electric motor, 3 – collecting tank for compressed air, 4 – mechanical indicator, 5 - manometer, 6 – digital tachometer (speed of rotating shaft), 7 - wattmeter

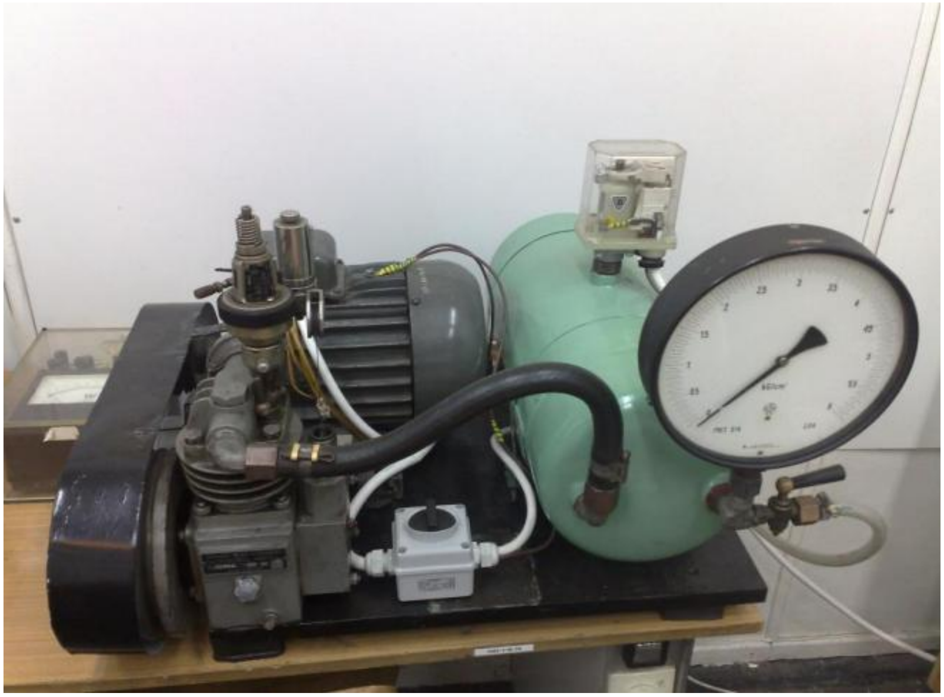


Fig. 1.2. View of the one-cylinder piston compressor used in laboratory study

Technical characteristics of experimental unite:

- ***characteristics of one-cylinder piston compressor:***
  - diameter of the cylinder d = 62 mm,
  - stroke of the cylinder s = 36 mm,
  - harmful volume of the compressor $V_{sz} = 0{,}05 \cdot Vs$
  - vacuum pressure of suction $\Delta p_1 \approx 1 \cdot 10^4$ *Pa*
- ***additional characteristics:***
  - electric motor efficiency $\eta_{el} = 0{,}6$,
  - efficiency of the belt drive $\eta_p = 0{,}95$,
  - power loss in electric motor bearings $N_{op} = 15$ *W*,
  - wattmeter constant k_w = *2W/graduation*,
- ***indicator characteristics:***
  - pressure constant, pressure graduation k_P = 11528 Pa/mm.

NECESSARY DATA FOR PREPARING LAB REPORT

Next measurements should be carried out:

- pressure in the collecting tank - p_{zb}
- ambient pressure - p_b
- ambient temperature - t_{ot}
- compressor's shaft rotation speed - n
- number of graduations on the wattmeter's scale – W

On the basis of indicator diagram the area F_{wyk} should be calculated that corresponds to compressor's indicated work L_i. Measuring results should be written in the table according to the example (table 1):

TABLE 1: Measuring results:

	p_{zb}		p_b		t_{ot}	n	Number of graduations on the wattmeter's scale W	F_{wyk}
Lp.	[kg/cm^2]	[Pa]	[mmHg]	[Pa]	[^{o}C]	[rev/min]	-	[mm^2]
1								
2								
3								

Legend: p_{zb} [kg/mm^2] - gauge pressure in the collecting tank, p_b [mm Hg]- atmospheric pressure, t_{ot} [^{o}C] – atmospheric temperature, **n** [rev/min] – speed of rotating shaft measured by tachometer, N_{el} [W] – electric power consumed by electric motor from grid, F_{wyk} [mm^2] – surface area of indicator diagram.

2. INTRODUCTION

Compressor is working machine whose purpose is to deliver gases or steam obtained from liquids under increased pressure. Compressed gases are needed in different fields of technologies that is why compressors are often been the parts of complex equipment for example refrigeration units, gas turbines and others.

Main indexes that define and characterize the compression process in real compressors are: ratio between the pressure in compressor's outlet to the pressure in the inlet also called as compression ration; final values of pressure and temperature of the compressed body, amount of the body (gas or steam) that was compressed in the time unit or so called flow rate, energy-requirement per unit weight (kg) or volume (m^3) needs the body to be compressed or ratio between amount of the body that was really compressed per one revolution of the compressor's shaft to the amount of the body that corresponds to theoretical volume of the compressor - so called compression factor or coefficient of the real flow λ.

2.1. COMPRESSORS TYPES

Compressors wherein compression process occurs in a periodical way is called displacement compressors (positive displacement compressors). Positive displacement compressors can be divided into two general categories:

- **piston compressors** – process of compression occurs inside the cylinder, where piston moves in reciprocating motion way,
- **rotary compressors -** in such compressors the process of compression occurs by means of rotating elements and gas delivery process takes place in a steady way. Rotary compressors can be divided on **vane type** where just one element carries out rotary motion - Rotasco or on compressors where two elements have rotation motion - Roots or **screw compressors**.

Compressors where full compression (up to the final pressure) occurs in one process are called **one stage compressors.** On the other hand, compressors where full compression occurs by turn in a several compression processes accompanied by intermediate cooling are called **multi-stage compressors**.

2.2. DESIGN SOLUTIONS

Piston compressors are divided on:

- one-stage and multi-stage - depending on the amount of stages of compression,
- one-sided or two-sided action - depending on if the piston compresses the gas using just one own side or two sides are employed,
- air or water cooling,
- assisted with guiding device or without guiding device - depends on if crank mechanism has guider or it doesn't.

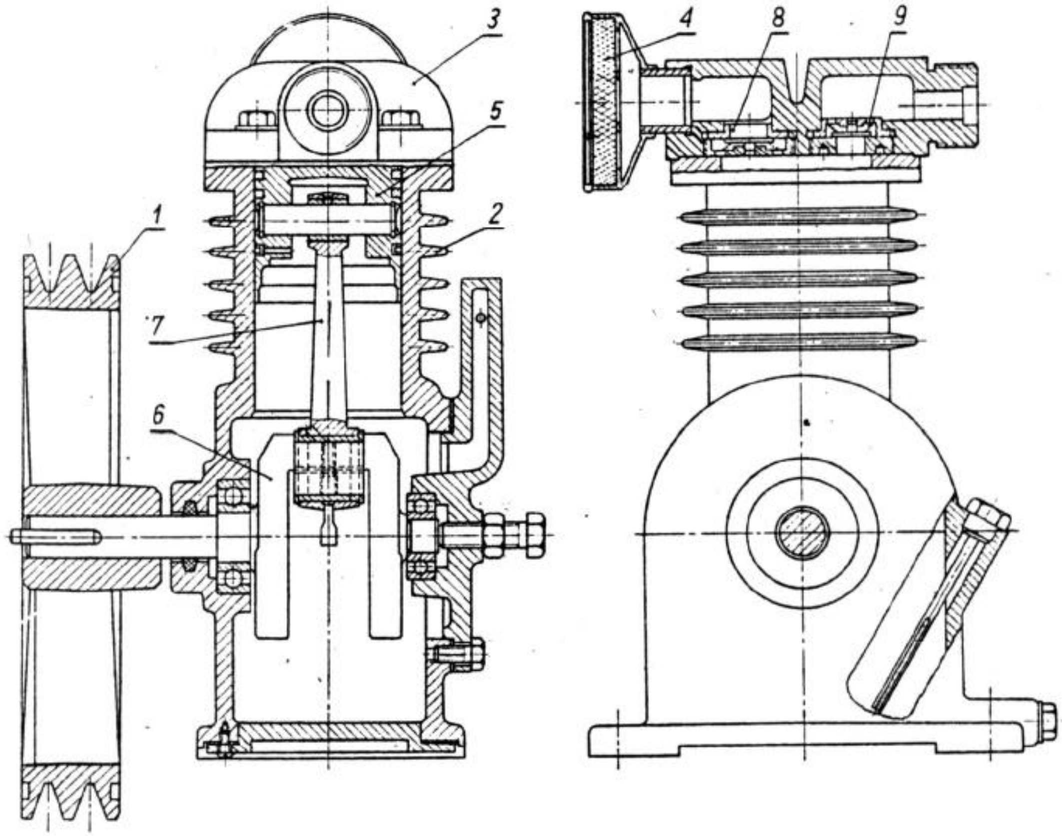


Fig. 2.1. One-cylinder piston compressor (our object of study): 1 - belt pulley with teeth, 2- cylinder, 3 - head, 4 – air filter, 5 - piston, 6 - crankshaft, 7 – connecting rod, 8 – inlet valve, 9 – outlet valve

In small and middle-size compressors the vertical and V-type arrangements of the cylinders are used. In the middle-size and big -size low-speed compressors an angle-type of arrangement (in the form of letter L) as well as horizontal arrangement is used.

If cylinders have air-cooling system, their bodies are provided with cooling fins while one end of the crankshaft has cooling fan responsible for forced air delivery to the passages between two neighbor fins.

Cylinders of the piston compressors provided with water-cooling system have special water jackets where forced circulation of cooling water occurs. Mentioned jackets are the parts of the main water cooling systems mounted in the compressors.

An advantage of air-cooled compressors over water-cooled is absence of cooled jacket that makes possible them to work in the conditions where ambient temperature is below 0 ^{0}C. Disadvantage of such compressors is less cooling rate (higher value of polytropic index in the compression process) in comparison with water-cooled compressors.

Cross-section of one-stage compressor with one-sided action, air-cooled, with vertical cylinder arrangement is presented on Fig. 2.1. As it can be seen from Fig. 2.1 the compressor has no guiding device at crank mechanism and is driven by electric motor through the belt drive. The belt pulley with teeth acts as fan.

2.3. OPERATING PRINCIPLE OF PISTON TYPE COMPRESSOR

In piston type compressor (fig. 2.1) gas compression occurs as a result of reciprocating motion of the piston 5 located inside the cylinder 2. Reciprocating motion is realized using crank mechanism that is driven by electric motor or combustion engine through gear. One operating (working) cycle of the compressor is executed during one revolution of the crankshaft 6.

Gas induction occurs when piston that located in the top return position will start movement toward bottom position after been driven from revolved crankshaft. In such conditions vacuum occurs inside the cylinder. Because of pressure difference between the cylinder and inlet (induction) channel, the inlet valve 8 opens automatically. Induction process will last until piston reaches its bottom return position. Theoretically, in this position of the piston the inlet valve 8 should be closed. After passing the bottom return position, piston starts to move upwards and compressing the gas. Compression process will finish at the moment when gas pressure inside the cylinder will be equal to the pressure outside the outlet valve 9 but this is true only in the theory. In real process, the level of in-cylinder pressure must be a little bit greater to compensate the resistance to flow in the outlet of the compressor. Anyway, finishing of the compression process usually occurs when piston is located in the mean position between two return states. At this time the outlet valve is opened and gas is pushing out from the cylinder. Pushing out process will be finished as piston will reach the top return position.

Usually self-acting valves used in piston type compressors. For such valve, opening and closing processes are managed by pressure differences on the opposite sides and plate valves usually used for such purposes.

2.4. THEORETICAL BACKGROUNDS

Theoretical compression process is usually considered regarding to ideal compressor under next conditions:

- there is no harmful volume in the compressor,
- there is no losses related to flow resistance during induction and pushing out processes which occurs when the cylinder is filled with fresh gas and when compressed gas leaves the working space,
- there is no heat exchange between cylinder walls and compressed body,
- polytropic index for compression process is constant,
- there are no friction losses, leakages and so on.

2.4.1. THEORETICAL COMPRESSOR WITHOUT HARMFUL VOLUME

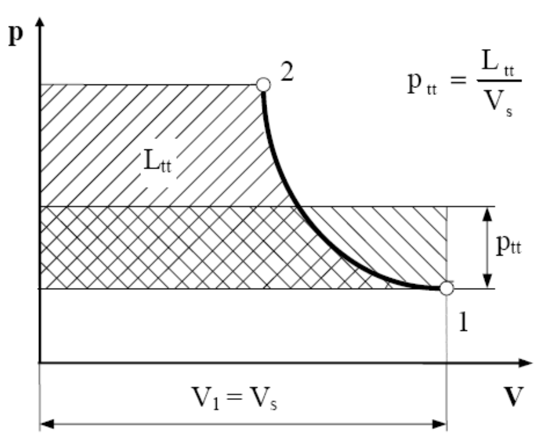


Fig. 2.2. Diagram of theoretical compressor without harmful volume

If compression process occurs with polytropic index $1 < m < k$ the next dependences can be used to describe it:

- compression work:

$$L_{tt} = \frac{m}{m-1} p_1 V_s \left[\left(\frac{p_2}{p_1} \right)^{\frac{m-1}{m}} - 1 \right] \text{ [J]}$$

- cylinder charge:

$$G_{ttcyl} = \frac{p_1 V_1}{RT_1} = \frac{p_1 V_s}{RT_1} \text{ [kg]}$$

- flow rate is equal to cylinder charge:

$$G_{tt} = G_{ttcyl} = \frac{p_1 V_s}{RT_1} \text{ [kg/cycle]}$$

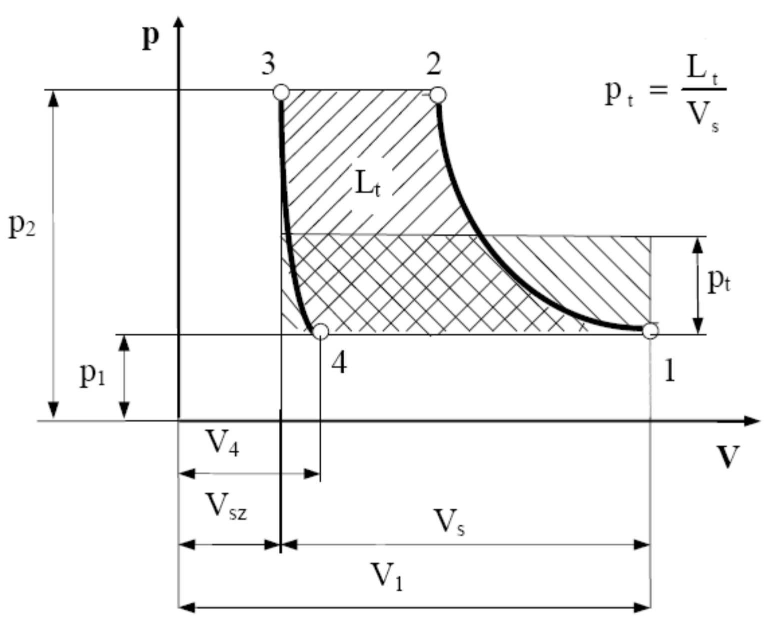
- mean excess pressure:

$$p_{tt} = \frac{L_{tt}}{V_s} = \frac{m}{m-1} p_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{m-1}{m}} \right] \text{ [Pa]}$$

- compression work for 1 kg of gas:

$$l_{tt} = \frac{L_{tt}}{G_{tt}} = \frac{m}{m-1} RT_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{m-1}{m}} - 1 \right] \text{ [J/1kg gas]}$$

2.4.2. THEORETICAL COMPRESSOR WITH HARMFUL VOLUME



Rys. 2.3. Diagram of theoretical compressor with harmful volume

Theoretical compressor with harmful volume can be considered using next assumption - compression and expansion occurs under constant polytropic index.

As a result of harmful volume existence the compressor will not suck the whole volume of the gas that corresponds to cylinder displacement V_s. The real fresh charge volume will be smaller ($V_{sz}+V_s-V_4$).

Total charge in the cylinder:

$$G_{tcyl} = \frac{p_1 V_1}{RT_1} = \frac{p_1\left(V_{sz} + V_s\right)}{RT_1} \quad \text{[kg]}$$

Flow rate of the compressor that corresponds to pushed out charge:

$$G_t = \frac{p_1\left(V_{sz} + V_s - V_4\right)}{RT_1} \quad \text{[kg/cycle]}$$

Idle compressed charge:

$$G_{sz} = \frac{p_1 V_4}{RT_1} \quad \text{[kg]}$$

If compression and expansion processes occur with polytropic index $1 < m < k$ the next dependences can be used:

- theoretical work:

$$L_t = \frac{m}{m-1} p_1 V_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{m-1}{m}} - 1\right] - \frac{m}{m-1} p_4 V_4 \left[\left(\frac{p_3}{p_4}\right)^{\frac{m-1}{m}} - 1\right]$$

since $p_4 = p_1$ and $p_3 = p_2$, then:

$$L_t = \frac{m}{m-1} p_1\left(V_1 - V_4\right)\left[\left(\frac{p_2}{p_1}\right)^{\frac{m-1}{m}} - 1\right] = \frac{m}{m-1} p_1\left(V_{sz} + V_3 - V_4\right)\left[\left(\frac{p_2}{p_1}\right)^{\frac{m-1}{m}} - 1\right] \quad \text{[J]}$$

- work consumed for compression 1 kg of gas:

$$l_t = \frac{L_t}{G_t} = \frac{\dfrac{m}{m-1} p_1\left(V_{sz} + V_3 - V_4\right)\left[\left(\dfrac{p_2}{p_1}\right)^{\frac{m-1}{m}} - 1\right]}{\dfrac{p_1\left(V_{sz} + V_s - V_4\right)}{RT_1}} = \frac{m}{m-1} RT_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{m-1}{m}} - 1\right] \quad \text{[J/1 kg gas]}$$

- mean theoretical excess pressure:

$$p_t = \frac{L_t}{V_s} = \frac{m}{m-1} p_1 \frac{V_{sz} + V_s - V_4}{V_s}\left[\left(\frac{p_2}{p_1}\right)^{\frac{m-1}{m}} - 1\right] \quad \text{[Pa]}$$

2.4.3. COMPARISON FOR THEORETICAL COMPRESSOR WITHOUT HARMFUL VOLUME AND THEORETICAL COMPRESSOR WITH HARMFUL VOLUME

- coefficient of the volumetric intaking:

$$\eta_V = \frac{G_t}{G_{tt}} = \frac{V_{sz} + V_s - V_4}{V_s}$$

- work consumed for compression 1 kg of gas:
in theoretical compressor:

$$l_{tt} = \frac{m}{m-1} RT_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{m-1}{m}} - 1 \right] \text{ [J/1 kg gas]}$$

in theoretical compressor with harmful volume:

$$l_t = \frac{m}{m-1} RT_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{m-1}{m}} - 1 \right] \text{ [J/1 kg gas]}$$

or: $l_{tt} = l_t$

2.4.4. BOUNDARY COMPRESSION RATIO

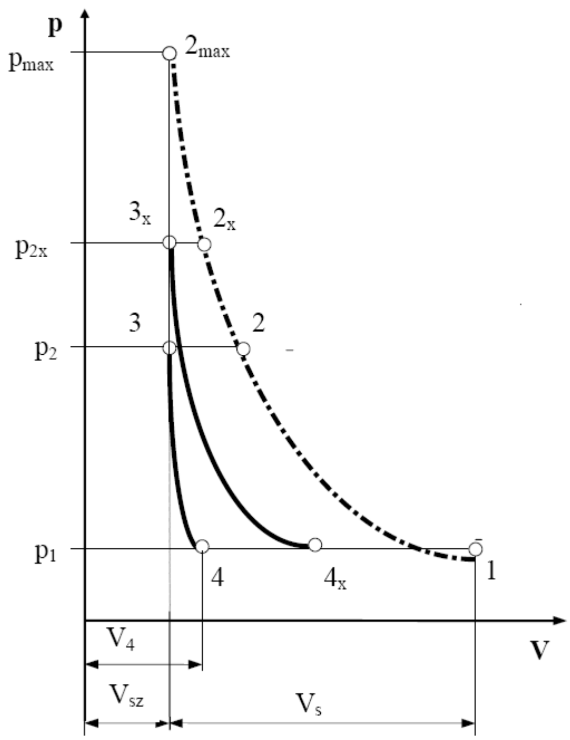


Fig. 2.4. Diagram for explanation of boundary compression ratio

As far as compression pressure increasing the flow rate of the compressor decreasing (length 4-1 becomes smaller; fig. 2.4). As compression process will reach pressure p_{2max} that corresponds to boundary compression ratio, the flow rate of the compressor will be equal to 0.

Dependences that characterize compressor are:

– relative volume of the harmful space (volume):

$$a = \frac{V_{sz}}{V_s}$$

– compression ratio:

$$\nu = \frac{p_2}{p_1}$$

– coefficient of the volumetric intaking :

$$\eta_\upsilon = \frac{V_s + V_{sz} - V_4}{V_s} = 1 + \frac{V_{sz}}{V_s} - \frac{V_4}{V_s} = 1 + a - \frac{V_4}{V_s}$$

since : $p_3 V_3^m = p_4 V_4^m$ and $V_3 = V_{sz}$,

then: $V_4 = \left(\frac{p_3}{p_4}\right)^{\frac{1}{m}} * V_{sz} = \nu^{\frac{1}{m}} * V_{sz}$,

wherefrom: $\eta_\upsilon = 1 + a - \nu^{\frac{1}{m}} * \frac{V_{sz}}{V_s} = 1 + a - \nu^{\frac{1}{m}} * a = 1 - a\left(\nu^{\frac{1}{m}} - 1\right)$

– boundary compression ratio:

$$\nu_{max} = \frac{p_{2max}}{p_1} = \left(\frac{V_1}{V_{sz}}\right)^m = \left(\frac{V_{sz} + V_s}{V_{sz}}\right)^m = \left(1 + \frac{1}{a}\right)^m$$

3.1. COMPARISON OF WORKING CYCLES OF COMPRESSORS

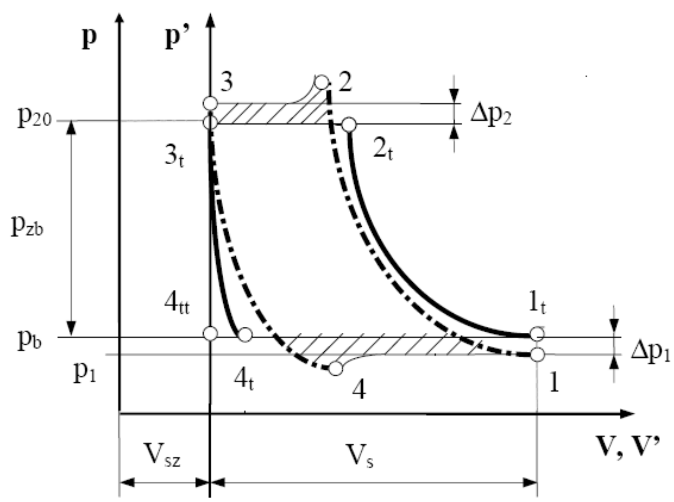


Fig. 3.1. Comparison of the real indicator cycle and theoretical cycle of the compressor with harmful volume as well as theoretical cycle of the compressor without harmful volume.

On fig. 3.1 there are diagrams of the working cycles of real compressor (indicator diagram), theoretical one with harmful volume as well as theoretical compressor without harmful volume.
Typical points on these diagrams are next:
- in coordinates p - V:
  real compressor: 1- 2 -3 - 4
  theoretical compressor with harmful volume: 1_t -2_t - 3_t - 4_t
- in coordinates p' - V':
  theoretical compressor without harmful volume: 1_t - 2_{t}- 3_t - 4_{tt}

Working cycle of the real compressor in comparison with indicator diagram of theoretical compressor with harmful volume shows that there are additional areas of work (dashed area) expended on overcoming underpressure Δp_1 and overpressure Δp_2 (pushing out pressure).

3.2. HANDLING COMRESSOR'S INDICATOR DIAGRAM

An example of compressor's indicator diagram is shown on fig.3.2. There are characteristic points and their definitions that could be used for further calculations.

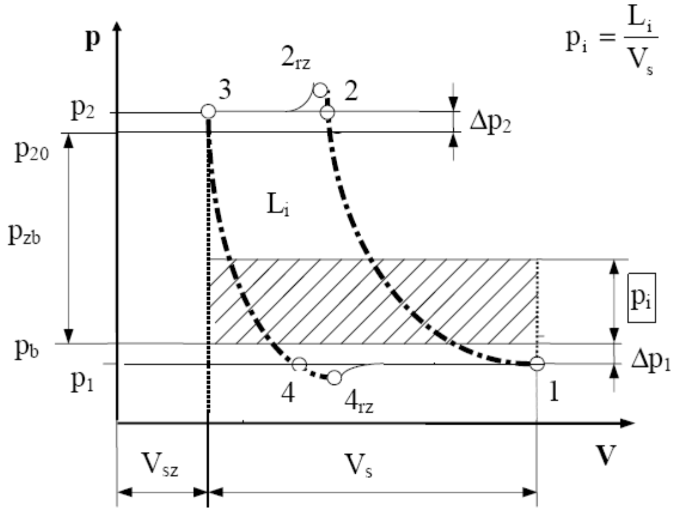


Fig. 3.2. Compressor's indicator diagram

Graduations on the indicator diagram:

- graduation of pressure: k_p = ………… [Pa/mm]
- graduation of volume: k_v = ... [m^3/mm]
- graduation of surface area: $k_F = k_{p*}k_v$ [J/mm^2]

Pressure calculations in points 1 and 2:

$p_{20} = p_b + p_{zb}$ [Pa]

$p_2 = p_b + p_{zb} + \Delta p_2$ [Pa]

$p_1 = p_b - \Delta p_1$ [Pa]

where: p_{zb} – pressure inside the collecting tank (is taken from measurements)

p_b – ambient pressure (is taken from measurements)

$\Delta p_1 = 10^4$ Pa

Δp_2 – value obtained from indicator diagram

1 mm Hg = 133,322 Pa

1 bar = 10^5 Pa

1 kG/cm^2 = 98066,5 Pa

Calculation of volumes in points 1, 2, 3, 4:

$$V_s = \frac{\pi d^2}{4} s \quad [m^3]$$

$V_{sz} = 0{,}05 * V_s$ [m^3]

$V_1 = V_{sz} + V_s$ [m^3]
where: V_2, V_4 – values obtained from indicator diagrams

Calculation of polytropic index of compression:

$$p_1 V_1^m = p_2 V_2^m$$

$$\frac{p_1}{p_2} = \left(\frac{V_2}{V_1}\right)^m$$

$$m = \frac{\ln \frac{p_1}{p_2}}{\ln \frac{V_2}{V_1}}$$ where obtained values must be in the range of $1<m<1,4$

Temperature calculations in points 1, 2:
Temperature T_1:
$T_1 = T_{ot}$ - value obtained from measurements
Temperature T_2:
From equation of polytropic compression: $p_1 V_1^m = p_2 V_2^m$

follows: $V_2 = \left(\frac{p_1}{p_2}\right)^{\frac{1}{m}} V_1 = \left(\frac{p_2}{p_1}\right)^{-\frac{1}{m}} V_1$

From state equation the volume V_1 can be calculated in the point 1:

$$V_1 = \frac{GRT_1}{p_1}$$

Volume V_2 in the point 2:

$$V_2 = \frac{GRT_1}{p_1}\left(\frac{p_2}{p_1}\right)^{-\frac{1}{m}}$$

From state equation the temperature T_2 in the point 2 can be calculated as:

$$T_2 = \frac{p_2 V_2}{GR}$$

after substitution the known value V_2, the temperature can be obtained from the next equation:

$$T_2 = \frac{p_2}{GR}\frac{GRT_1}{p_1}\left(\frac{p_2}{p_1}\right)^{-\frac{1}{m}} = T_1\left(\frac{p_2}{p_1}\right)^{1-\frac{1}{m}} = T_1\left(\frac{p_2}{p_1}\right)^{\frac{m-1}{m}}$$

Compression ratio:

$$\nu = \frac{p_2}{p_1}$$

Boundary compression ratio:

$$\nu_{gr} = \left(1 + \frac{V_s}{V_{sz}}\right)^m = \left(1 + \frac{1}{a}\right)^m$$

Indicated work:
$L_i = k_f * F_{wyk}$ [J/cykl]

where: F_{wyk} [mm^2] – the area bordered by indicator diagram

Mean indiated pressure (fig. 3.2):

$$p_i = \frac{L_i}{V_s} \quad \text{[Pa]}$$

Flow rate:

$$G_i = \frac{(p_b - \Delta p_1)*(V_{sz} + V_s - V_4)}{R*T_1} \quad \text{[kg/cycle]}$$

$$\dot{G}_i = G_i * \dot{n}_{spr} \quad \text{[kg/s]}$$

Indicated power:

$$N_i = L_i * \dot{n}_{spr} \quad \text{[W]}$$

Power of electric motor used to drive the compressor:

$N_e = 3*w*k_w$ [W]

where: w – number of graduations detected on the wattmeter during measurement
k_w – wattmeter constant (2W)

Power consumed to drive the compressor's shaft:

$N_{sp} = (N_e*\eta_{el} - N_{op})*\eta_p$ [W]

In the field of displacement compressors there are next definitions for work and power exist:

L_i, N_i - indicated work and power. Indicated work is type of work that is necessary to compress the body in real compressor without taking into account mechanical losses.

L_m, N_m – work and power of mechanical losses. Mechanical work is friction work in bearings, friction work consumed for overcoming frictions in pair: rings - cylinder face and so on. Also it could be the work required to drive auxiliary mechanisms (water pump, oil pump, fan).

L_e, N_e - effective work and power (measured on the compressor's shaft). Effective power is power that needs to be delivered to the compressor's shaft.

$N_{e\text{-}s}$ - power of the driving motor. This is power that can be taken off from the driving motor.

L_o,(E) - energy delivered to the driving motor. In the case of electric motor it will be electric energy consumed by the motor, in the case of combustion engine or steam engine it will be the amount of heat delivered to the engine through fuel combustion or steam.

3.3. CALCULATION OF THEORETICAL COMPRESSOR WITHOUT HARMFUL VOLUME

Work (area 1_t - 2_t - 3_t--4_{tt} on fig. 3.1):

$$L_{tt} = \frac{m}{m-1} p_b V_s \left[\left(\frac{p_{20}}{p_b} \right)^{\frac{m-1}{m}} - 1 \right] \quad \text{[J/cycle]}$$

Flow rate:

$$G_{tt} = \frac{p_b V_s}{RT_{ot}} \quad \text{[kg/cycle]}$$

where: R_{pow} = 287 [J/kg·K]

$$\dot{G}_{tt} = G_{tt} * \dot{n}_{spr} \quad \text{[kg/s]}$$

where: $\dot{n}_{spr}$ - speed of compressor's shaft rotation per second

$$\dot{n} = \frac{n}{60}$$ [rev/s] ; n [rev/min] - speed of compressor's shaft rotation

Power:

$$N_{tt} = L_{tt} * \dot{n}_{spr} \quad \text{[W]}$$

Mean overpressure (fig. 2.2):

$$p_{tt} = \frac{L_{tt}}{V_s} \quad \text{[Pa]}$$

3.4. CALCULATION OF THEORETICAL COMPRESSOR WITH HARMFUL VOLUME

Work (area 1_t - 2_t - 3_t - 4_t on fig. 3.1):

$$L_t = \frac{m}{m-1} p_b \left(V_s + V_{sz} - V_{4t}\right)\left[\left(\frac{p_{20}}{p_b}\right)^{\frac{m-1}{m}} - 1\right] \quad \text{[J/cycle]}$$

Volume V_{4t} can be calculated from polytropic equation of expansion 3_t - 4_t:

$$p_{3t} * V_{3t}^m = p_{4t} * V_{4t}^m$$

In view of: $p_{3t} = p_{20}$, $V_{3t} = V_{sz}$, $p_{4t} = p_b$,

$$V_{4t} = V_{sz}\left(\frac{p_{20}}{p_b}\right)^{\frac{1}{m}} \quad \text{[m}^3\text{]}$$

Coefficient of the volumetric intaking:

$$\eta_v = \frac{V_s + V_{sz} - V_{4t}}{V_s} = \frac{G_t}{G_{tt}} = \frac{\dot{G}_t}{\dot{G}_{tt}}$$

Flow rate:

$$G_t = \eta_v * G_{tt} \quad \text{[kg/cycle]}$$

$$\dot{G}_t = \eta_v * \dot{G}_{tt} \quad \text{[kg/s]}$$

Power:

$$N_t = L_t * \dot{n}_{spr} \quad \text{[W]}$$

Mean overpressure (fig. 2.3):

$$p_t = \frac{L_t}{V_s} \quad \text{[Pa]}$$

3.5. COMPARISON VALUES

Flow coefficient:

$$\lambda = \frac{G_i}{G_{tt}} = \frac{\dot{G}_i}{\dot{G}_{tt}}$$

Indicated efficiency:

$$\eta_i = \frac{l_{tt}}{l_i} = \frac{l_t}{l_i}$$

In view of: $l_i = \dfrac{L_i}{G_i}$ and $l_{tt} = \dfrac{L_{tt}}{G_{tt}}$,

$$\eta_i = \frac{L_{tt}}{G_{tt}} * \frac{G_i}{L_i}$$

In view of: $\frac{G_i}{G_{tt}} = \lambda$,

$$\eta_i = \lambda * \frac{L_{tt}}{L_i}$$

Mechanical efficiency:

$$\eta_m = \frac{N_i}{N_{sp}}$$

Compressor efficiency:

$$\eta_s = \eta_m * \eta_i$$

4. SUMMARIZING THE RESULTS OF CALCULATIONS IN THE TABLE FORM

Table 2. Selected calculation results

№	Quantity	Dimension	Value
1	m		
2	ν		
3	L_{tt}	J/cycle	
4	L_t	J/ cycle	
5	L_i	J/ cycle	
6	l_{tt}	J/1 kg of gas	
7	l_t	J/1 kg of gas	
8	l_i	J/1 kg of gas	
9	p_{tt}	Pa	
10	p_t	Pa	
11	p_i	Pa	
12	η_i		
13	η_m		
14	η_s		

5. LAB REPORT

Final lab report should include:

1. Consequence of the work.
2. Table with results of measurements.
3. Essential calculations.
4. Table with results of calculations.
5. Conclusions.

Instruction was developed on the basis of technical publications presented below as well as instructions developed in polish language by dr hab. inż. Piotr Orliński and devoted to the same topic of laboratory work.

6. LITERATURA:

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[3]. Kotlewski F. i in.: Pomiary w technice cieplnej. WNT, Warszawa 1974.