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PISTONS FOR INTERNAL COMBUSTION ENGINES

Tutorial
to laboratory work on the course
"Aircraft piston engines"

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Описано призначення поршневої групи двигунів внутрішнього згоряння, особливості їхньої роботи, конструкції поршнів, поршневих кілець і поршневих пальців. Наведено приклади поршнів авіаційних і наземних двигунів. Подано відомості про матеріали, з яких виготовляються деталі поршневої групи, а також формули для розрахунку деталей на міцність.

Для студентів спеціальності «Авіаційні двигуни і енергетичні установки» при виконанні лабораторних робіт.

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Tutorial introduces the information of functions of piston assembly, features of their operation, features of pistons, piston pins and piston rings. There are examples of land and aircraft engines presented in the tutorial. The tutorial also involves information about materials used to manufacture parts of piston assembly. There are also given formulas for the strength analysis of parts of piston assembly.

This tutorial is profitable for students studying “Aircraft Engines and Power Plants” to prepare for laboratory activities.

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CONTENTS

1 OVERVIEW OF PISTON CONSTRUCTION AND THEIR OPERATING CONDITIONS

The piston of internal combustion engine serves to form the variable closed gas-tight volume inside **the cylinder** and to transfer forces from expanding gas in the cylinder to the connecting rod. There are two main reasons to make the gas-tight volume:

1) hot gases leaking out the cylinder heat the piston and increase the crankcase temperature that is absolutely unacceptable (Fig. 1);
2) piston must prevent oil getting from inside the crankcase into the above piston volume, because oil that have got into the cylinder cannot be combusted and is exhausted jointly with combustion byproducts; besides, big masses of oil in the combustion chamber result in intense carbonization and forming deposits that in their turn deteriorate the operation of spark plugs or fuel jets in diesels.

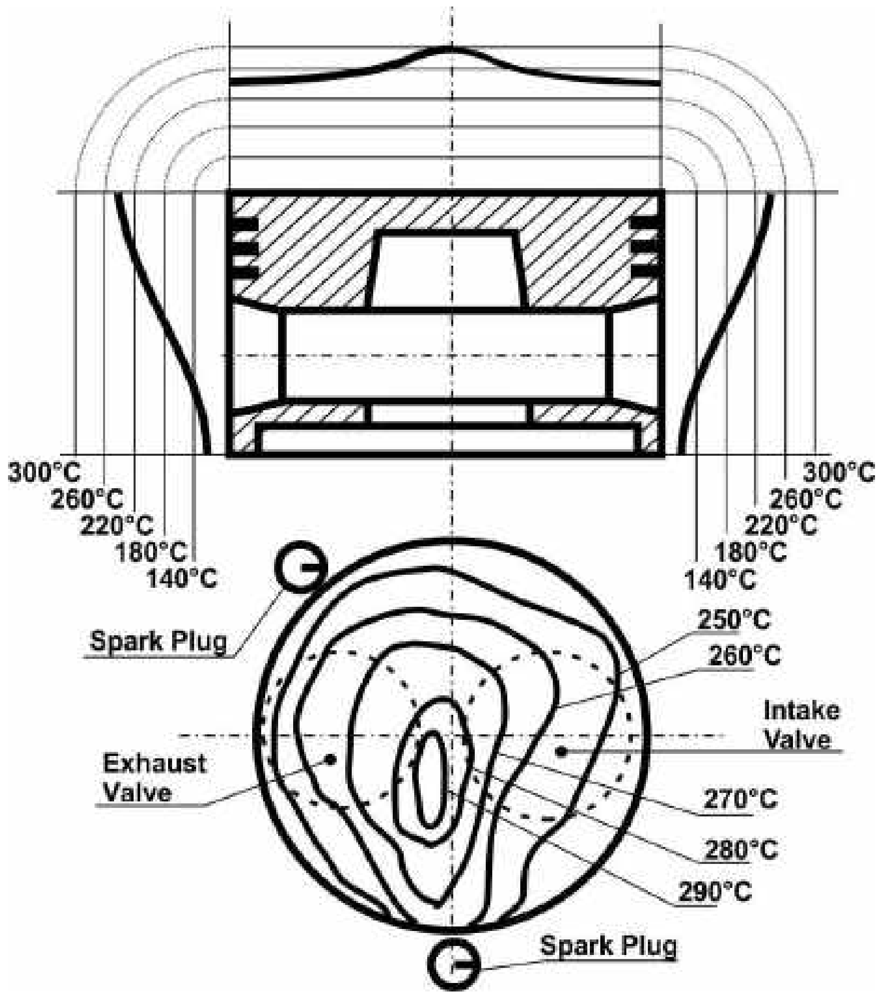


Figure 1 –Temperature distribution on piston surface

During operation the piston interacts with red-hot gases, which strongly heat it and complicate its operating conditions. Arranging piston cooling is a very complex task because oil film separates piston and cylinder wall and because of its smaller diameter, piston contacts with cylinder wall only by the part of its side surface. For these reasons the temperature of the piston is much higher (up to ***300°C***) than the temperature of the cylinder during engine operation. The surface temperature distribution is presented in Fig.1.

Besides, piston is also acted by pressure of expanding gases and inertia force. Transferring these loads from piston to the connecting rod reveals the additional side force guided normally to piston side surface. This force is a source of piston side surface wearing and extra mechanical losses due to the necessity to overcome the friction.

Thus, piston (piston assembly) is made the following demands:

- the reliable sealing of above piston volume;
- enough strength and rigidity to stand acting loads;
- efficient heat removal that allows providing suitable thermal conditions for operation;
- absence of oil leaking from crankcase to displacement volume;
- low wear of friction surfaces;
- the lowest possible friction between piston and facing it cylinder wall;
- minimum mass resulting minimum inertia force. This force depends on squared angular velocity ω^2 and reaches big values for fast engines; is also increases the loads acting the crank mechanism.

Meeting these requirements is a very complex problem because some of them contradict each other. That is why pistons mostly operate at maximum permissible heat and mechanical loading that limits its lifetime and boosting ability.

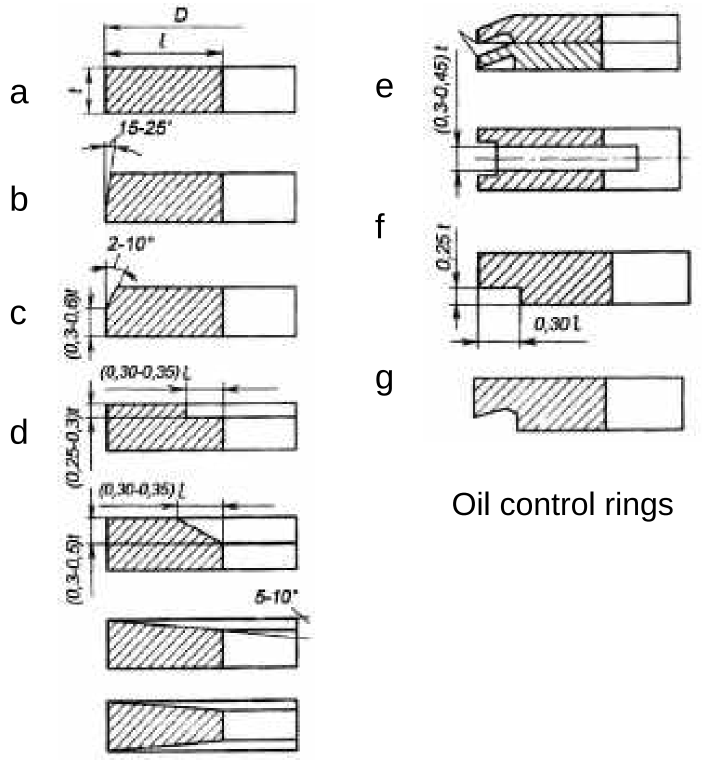
Piston assembly consists of piston, piston rings and piston pin. Piston is a cartridge-like structure with massive flat, convex or concave piston crown. Side surface can be decomposed into two parts according to their functions and operation conditions. The first, upper one, abutting piston crown has grooves for piston rings mounting. This part of side surface is heated the most because it serves to bleed the heat from piston crown to cylinder wall and also to transfer gas forces from **piston crown** to **piston pin**. Piston pin is mounted inside the **piston-pin bosses** that are in contact with side walls and directly with piston crown.

The second, the lower one, usually termed as **piston skirt** serves to transmit the force to cylinder wall wearing itself out. The temperature of the skirt is much lower than the temperature of the upper part.

The depth of the piston must be shortened to decrease the mass of the piston. But the shorter piston is, the higher specific loads act its side surface that in its turn increases the wear out of the piston. Because of this, the piston depth varies from engine to engine to meet the particular requirements, but the general trend is kept. Mass reduction is attained by manufacturing pistons from aluminum alloys. Aluminum alloys allow lower operating temperatures of the piston.

Temperature and linear thermal expansion coefficient of aluminum piston are higher than the same parameters of steel or cast iron cylinder sleeve. The clearance between piston and facing it cylinder sleeve is set to prevent piston «sticking» during operation.

The first compression ring operates in the most difficult conditions. Overcoming the friction of piston rings takes ***40…50 %*** (sometimes up to ***60 %***) of all mechanical losses in the engine. The distribution of losses in the three-ring schemes: 1^{st} ring – ***60 %***, 2^{nd} – ***30 %*** and 3^{rd} – ***10 %***.



Compression rings

Oil control rings

Figure 2 – Profile of compression and oil control rings

Compression rings (sometimes sealing rings) mounted in the upper part of the piston prevent working gases blow-by in the crankcase. These rings are split (the split is termed “locks”) and of bigger diameter than inner diameter of the cylinder in standalone state.

The profiles of piston rings are presented in Fig. 2. Have been fitted into the "grooves" of the piston inside the cylinder, the side face of the piston ring is pressed to cylinder bearing surface being acted by elasticity force and pressure. Circumferential pressure distribution (pressure profile) on cylinder wall depends on ring profile in standalone state and sizes of the cross-section. The ring in standalone state is profiled to provide uniform circumferential pressure profile.

The ring is run-in at low modes with the lowest probability of the blow-by and the minimum consequences of possible blow-by. This is done to provide the snug engagement of piston ring to cylinder wall (the problems are usually caused by deviations from form and sizes during manufacturing). Run-in usually takes some hours. For run-in time shortening side face of compression ring is manufactured conical (cone angle is ***0,5…2°***). The most probable place for blow-by is the split of piston ring. To minimize or even to eliminate the probability of the blow-by the piston usually has two piston rings with splits arranged at opposite sides of the piston. Excluding sealing purposes piston rings have another important purpose. As it was already mentioned, the piston rings are snug engaged to cylinder wall that makes them major heat conductors transferring heat from piston to cylinder. If there is no oil cooling of the piston, then compression rings take away 70 % of all heat present in the piston.

Compression rings are very good to prevent hot gases getting from above piston volume to crankcase, but they are very inefficient to prevent oil film getting to the above piston volume. The process oil gets to the displacement volume consists of two phases:

- when piston travels from top dead center to bottom dead center, the film covering cylinder bearing surface gets to the end clearance between the piston ring and "groove";
- then piston travels from bottom dead center to top one, oil in the end clearance is forced to above piston volume (Fig. 3).

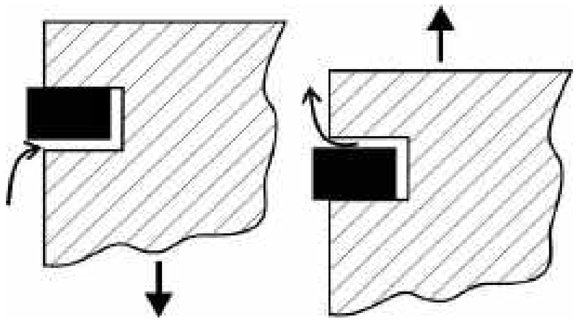


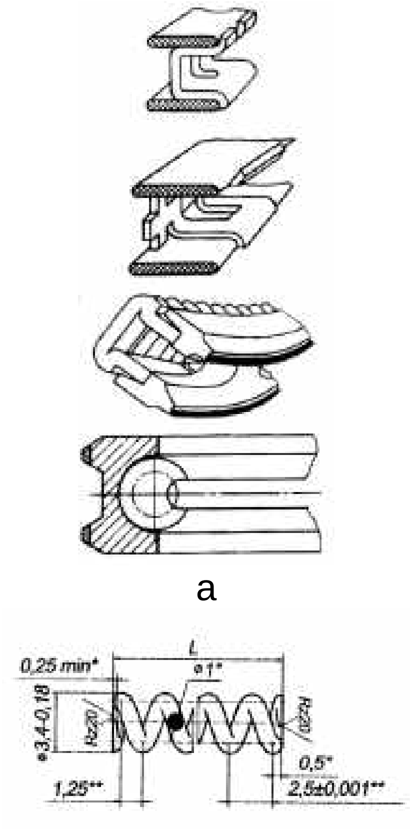
Figure 3 –The scheme of oil flows through oil control rings (pumping effect of the ring)

Hence, compression ring of the piston operates as a pump that pumps the oil to displacement volume. Pressure and elastic force acting the ring are not enough to provide snug engagement and effective oil film removal from cylinder wall. To minimize oil getting to above piston volume piston has one more ring of special construction termed oil control ring or scraper ring. **Oil control** ring are of the following features:

1. The ring is pressed against the cylinder wall not over the entire height, i.e. not over the entire surface. The reduction of contact surface is attained by circular chamfering, or by turning circular groove at the outer ring surface. Modern oil control rings (Fig. 4, a) are multiple-piece with low elasticity. The multiple-piece ring is pressed to the cylinder bearing surface by special springs (Fig. 4, b). Reduction of the ring supporting face at slight change in elasticity results in better ring pressing to cylinder wall. This provides more perfect oil removal from cylinder wall.
2. To minimize pumping effect of the oil control ring, the clearance between the ring and the groove is designed to be smaller than between compression ring and the groove. Fitting with fewer clearances becomes possible because oil control rings operate in areas with lower temperatures than compression rings.
3. Oil control ring (see Fig. 2, e) and groove for this ring have drillings. Below the groove there is one more turned groove with the set of apertures drilled along the circumference. All these apertures are aimed to remove the oil from

4. cylinder wall back to crankcase. Hence, pressure in the groove drastically drops, and leaks of oil to the above piston volume decrease.

Different coverings like chrome plating, phosphating etc., are widely applied for piston rings. Modern constructions generally comprise two or three compression rings and one (rarely more) oil control ring. Generally there are three or four rings. Old constructions have had up to six and more rings. Rings are manufactured from cast iron or from alloyed steel ***4Х5Ф1С***.



a

b

Figure 4 – The multiple-peace oil control rings: a – rings of different construction; b – spring

Piston pin connects piston with connecting rod. Edges of the piston pin are mounted inside the bosses of the piston. The central part of the piston pin is inside the rod bushing in the upper part of the connecting rod termed **rod small end**. During engine operation there is a force that acts along the piston trajectory causing bending and shearing stresses. Besides, the piston pin is worn out because it turns round inside the rod bushing and inside the bosses.

There are two methods to mount pin in the piston:

a) **the floating pin mounting**, which allows pin displacement in the bosses and in the rod bushing;
b) **the set screw pin mounting** when the piston pin is secured in the rod bushing or in the bosses.

Nowadays only floating pin mounting is applied because of more uniform wear out of piston pin and rod bushing. Hard floating pin may travel axially without any limitations wearing out or even scaring the **cylinder bearing surface**. For these reasons the axial displacement of piston pin is limited by special blank covers made of soft aluminum alloy (see Fig. 6) or by snap rings. The snap rings are fitted in the grooves, which are machined in bosses of the piston at the edges of the piston pin (see Fig. 7).

Piston pins are hollow for mass reduction purposes. Sometimes the hole in the piston pin is by-conical («hourglass») with minimum area in the center of the pin. Piston pins of such form are very similar to the beam of equal the flexural strength. The pins are usually **carbonized** or **nitrided** for wear out reduction.

2 CLEARANCES

The clearance between the piston and the cylinder barrel is much smaller during engine operation than in idle state because of the following reasons:

- temperature of working piston is much higher than the temperature of barrel;
- piston is manufactured from material with higher linear heat expansion coefficient than the barrel.

The clearance reduction can be assumed as

$$\Delta\delta = \boldsymbol{D}\left(\alpha_{\eta} \cdot \Delta \boldsymbol{t}_{\eta} - \alpha_{\eta} \cdot \boldsymbol{t}_{\eta}\right). \quad (1)$$

E.g. the clearance between the piston of diameter ***150 mm*** heated to temperature ***260°C*** and the cylinder barrel heated to temperature ***140°C*** is

$$\Delta\delta = \boldsymbol{150} \cdot \left(\boldsymbol{0{,}000011} \cdot \boldsymbol{140} - \boldsymbol{0{,}000024} \cdot \boldsymbol{260}\right) = -\boldsymbol{0{,}705\ mm}\,. \quad (2)$$

Negative value indicates that the clearance becomes smaller. For considered above reasons the idle clearance between piston and cylinder barrel is chosen to ensure “stickless” piston travel during operation. But operation clearance must not be too big, because this results in reduction of contact surface between piston and cylinder. Contact via small surface increases the specific pressure on the side surface of the piston and deteriorates lubrication. Besides, strikes, which appear at the moment when side force changes its hand, become more intense. The clearance between piston and cylinder barrel is not equal along piston height, because upper part of the piston is of higher temperature than lower one during operation. So, idle clearance in the upper part is set bigger than in the lower part of the piston.

The clearance between piston and cylinder barrel is usually specified by two values – minimum and maximum clearance. The examples of this clearance for some engines are presented in Table 1.

Table 1 – Clearance between piston and cylinder barrel, mm

Fluid cooled RE				**Air cooled RE**
Part	VK-105	MAN D0826	ROTAX-582	AI-14
Upper part	0,96–1,05	-	0,06–0,08	0,96–1,05
Lower part (skirt)	0,53–0,64	0,13–0,364	0,09–0,12	0,53–0,64

The clearance between piston and cylinder barrel increases with time because of friction parts wearing out. That is why each engine type is established limiting rates for the clearances in maintenance. The clearance between piston and cylinder barrel after repair or inspectionin maintenance is generally increased by ***0,1…0,2 mm*** referring initial rate.

Clearances of piston rings and piston pin. Height mounting fit between groove and the upper compression ring varies within limits ***0,09…0,15 mm***. The rest compression rings operate at lower temperatures and hence fitted with smaller clearance (***0,05–0,1 mm***). Oil control rings are mounted in zones with rather low temperatures. Their height mounting fit is much smaller than for other rings and on average equals ***0,04–0,08 mm***.

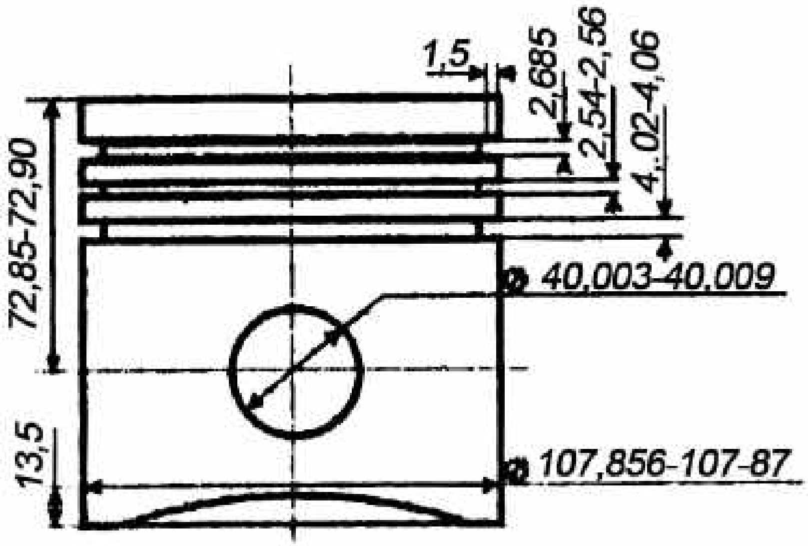


Figure 5 – Sizes of piston by ALKAN

The gap between floating pin and road bushing equals ***0,04–0,06 mm***. At the same time the clearance is allowed to increase in maintenance (the reason is the wear out of friction surfaces)up to ***0,08 mm***. Generally piston pin is loosely fitted in the piston bosses (the clearance is ***0–0,02 mm*** at indoor temperature). Fitting itself occurs at higher temperatures of the piston. When the piston is heated during the operation, clearance between piston pin and bosses becomes almost equal to the clearance between piston pin and rod bushing. The precise clearances in joints are provided by settingtight limit for all main sizes of piston-cylinder unit.

The sizes of piston by ALKAN for the engine MAN D0826 are presented in Fig. 5. An analysis of Fig.5 reveals that:

- tolerance scope of piston skirt at the broadest place is ***19 µm***,
- tolerance scope of the hole for piston pin fitting is ***6 µm***;
- tolerance scope of distance between piston crown and pin axis is ***50 µm***.

3 EXAMPLES OF EXISTING CONSTRUCTIONS

Piston of AM-38 RE (Fig. 6 and 7) is stamped from aluminum alloy. The piston crown is slightly convex. The inner side of the piston crown is ribbed for higher rigidity and better heat withdrawal. Bulky bosses of the piston are gradually turned into piston crown. The rigidity of the skirt is improved by axial and two annular ribs in the bottom part. There are five grooves for six piston rings in the upper part of the piston. Four top are compression rings, the rest two, mounted in common groove, are oil control rings (see Fig. 2). The topmost compression ring has cylindrical outer side surface (Fig. 2, a), the rest compression rings are turned to have conical outer side surface with apex angle 2° (see Fig. 2, b).

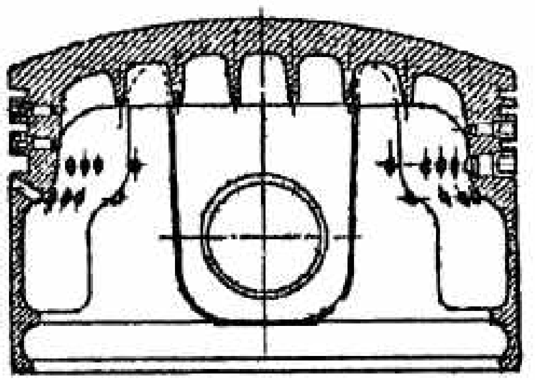


Figure 6 – Piston of V-type AM-38 reciprocating engine

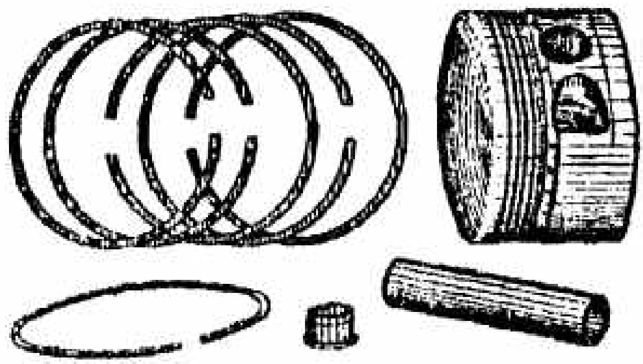


Figure 7 –Parts of AM-38 piston unit

Oil control rings have the form very similar to scraper (see Fig. 2, g) for better oil removal from cylinder housing surface. For better scraped off oil removal there are 17 holes drilled in groove for oil control rings and 9 holes are drilled in annular groove beneath the oil-control rings. There are also pits milled in the end surface of each oil control ring for the same purposes. All piston rings (except topmost one) are fixed from turning over by special stopper. Stoppers are mounted circumferentially-spaced. Floating piston pin is restricted from lengthwise displacement during operation by two aluminum blank covers. The end faces of blank covers are machined spherical. The radius of the sphere is some lower than the radius of the cylinder. There are small drillings in the blank covers aimed to eliminate possible pressure rise in the internal cavity of the piston pin. Outer surface of the piston pin is carbonized, grinded and polish.

Piston of VK-105 RE (Fig. 8) is stamped from aluminum alloy. The piston crown is flat. The upper part of the piston side surface is machined conical with cone angle facing piston crown. This is done keeping in mind thermal expansion of the piston during operation. The middle part of the piston side surface is oval. The major axis of the oval is perpendicular to piston axis and minor axis is parallel to piston axis. Such a form of side surface allows eliminating friction in the part of the piston skirt that does not take up side loads.

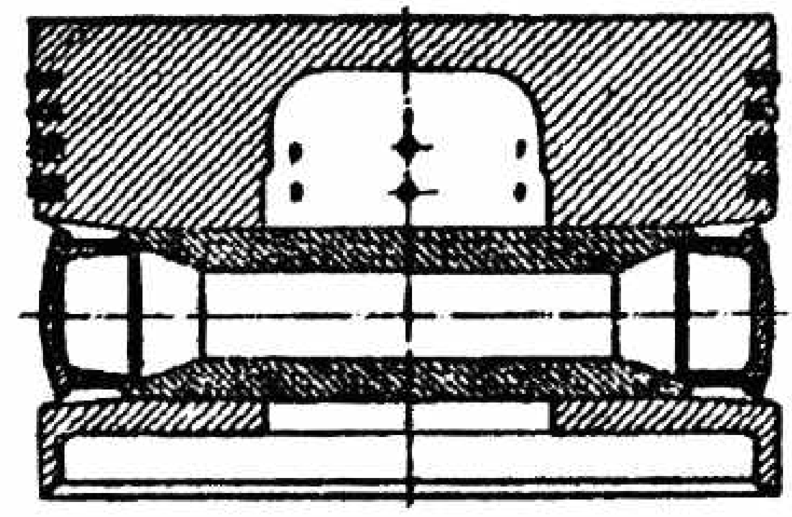


Figure 8 – Piston of VK-105 RE

There are four grooves for six piston rings in the upper part of the piston. There are two compression rings in the two top grooves and four oil control rings are in the lower two grooves (two rings in common groove).The side (operation) surface of the topmost compression ring is cylindrical, and of the lower compression ring is conical. There are also pits milled in the end surface of each oil control ring for removing oil inside the piston. The piston pin is restricted from lengthwise displacement during operation by two aluminum blank covers. The surfaces of piston bosses that are in contact with piston pin are milled special oil supplying grooves. They are to lubricate piston pin.

Piston of ASH-82 RE (Fig. 9) stamped from aluminum alloy. The piston crown is flat. The reverse surface of the crown is densely ribbed. The bosses are directly joined with the piston crown. There are four grooves in the upper part of the piston and one is in the bottom part. There is one compression ring in each of three top most grooves. There are two oil control rings in the fourth groove. There is one more groove beneath the hole drilled for piston pin mounting. This groove is for extra oil control ring.

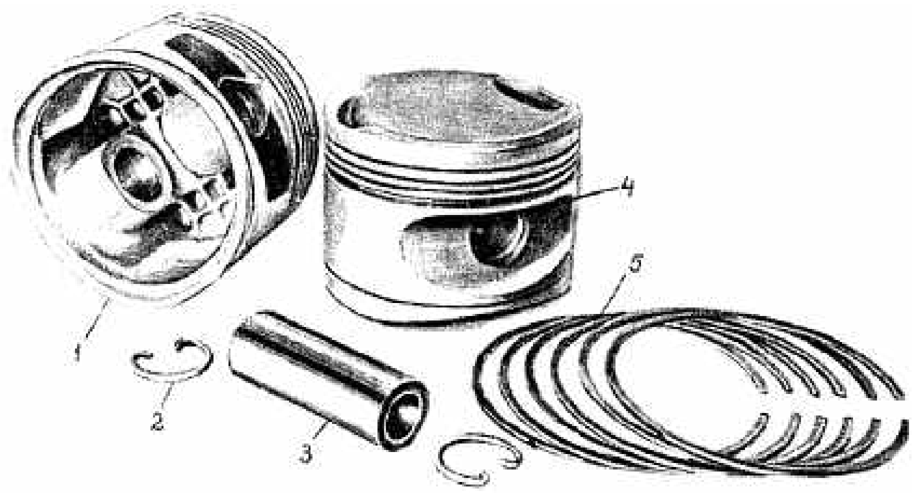


Figure 9 – Piston of ASH-82 RE

Oil control rings have the form very similar to scraper (see Fig. 2, g; Fig. 10, 4, 5, 6). The upper oil control rings are faced down by sharp edge, and the lower one is faced up by the sharp edge. Such positioning oil control rings provides permanent oil circulation between oil control rings. This reduces friction and wear out of piston and cylinder barrel.

Piston pin is prevented from lengthwise displacements by two rectangle retaining snap rings fitted in corresponding grooves from both sides of bosses. There are some millings termed **valve relief pockets** that prevent piston striking the valve head during operation.

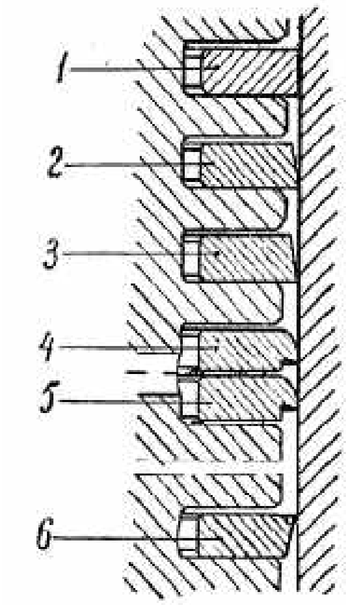


Figure 10 – Piston rings of ASH-82 RE

Piston of 6TD RE. (The piston is presented in this learner`s guide because considered engine meets most demands that are usually made to power plants of aircrafts. In the thirties there was reciprocating engine Jumo-207 by Junkers in use with the piston of the similar design).

For higher reliability purposes, the piston was designed composite. The piston 2 (Fig. 11) is formed from aluminum alloy AK4-1. Side surface of the piston is barrel-like on height; transversal cross-section of piston is oval. Being heated during operation piston becomes cylindrical. There is a microrelief on the side surfaces of the intake and exhaust pistons (Fig. 12). The side surface of the intake valve is gridded (c) for better oil absorption. The side surface of the exhaust valve as more heat stressed part is grooved for better oil absorption (d) and provision lower temperatures of piston.

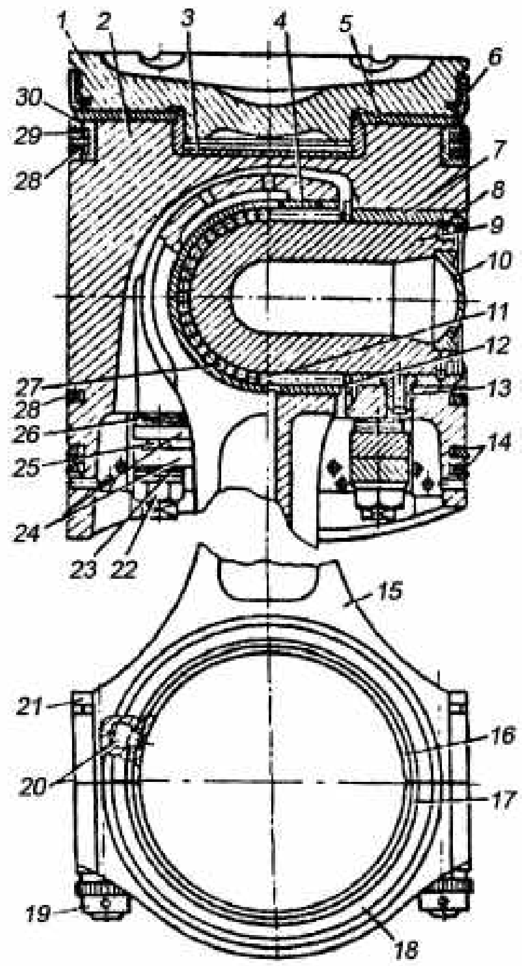


Figure 11 – Piston and connecting rod of 6TD RE

Piston pin is rigged in pressed in steel nitrated bushes 7 (see Fig. 11). The bushes are restricted from cranking by pins 13.Steel ring-holder 30 for two compression rings 28 and 29 is pressed on piston. The insert 3 is pressed in piston crown aimed to center shield 1 and spacer 5. There are three grooves machined beneath piston pin: the upper one is to fit additional compression ring 28, the rest two are to fit oil control rings 14. There are some oil returning holes to remove oil, which was scraped from cylinder housing surface. The shield serves to prevent overheating of aluminum part of the piston. The shield is manufactured from steel ***20X25H20C2*** (***ЭИ 283***) with improved properties of surface that is a part of combustion chamber. The improvement is achieved by thermo diffusion chromium-plating. Combustion diffusion chromium-plating. Combustion chamber is formed by pits of opposed moving pistons. To ensure unimpeded

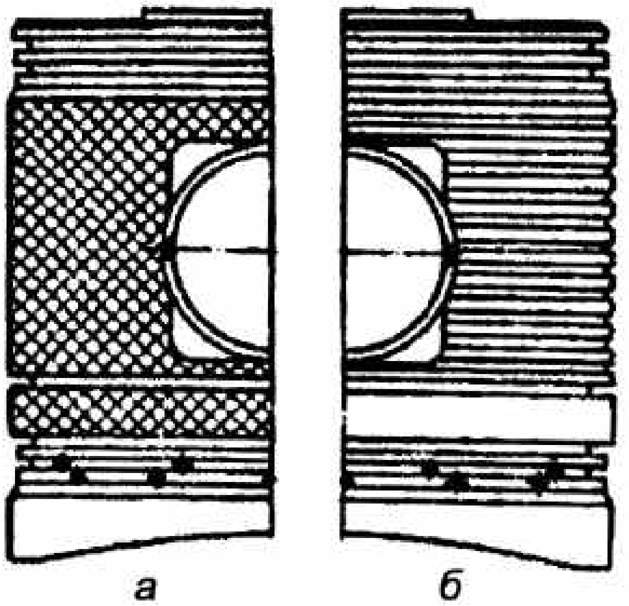


Figure 12 – Pistons: a – intake piston; b – exhaust piston

injection of fuel by fuel spraying nozzles there are four pits in the side surfaces of the shields. There are tightening bolts 25 pressed in the shield. To compensate thermal strains, shields and piston are bolted pair wise via springs 24.

Nuts 24 are locked by clenching cylindrical shoulders on square end of bolt. Bores drilled for bolts fitting are sealed by fluoroplastic-rings. These rings are bundled by washers 26 and bolts. Bolts misalignments that arise because of strains of shield or piston are compensated by spherical washers 23.

Spacer serves as support for the heat-barrier ring 6 and also reduces the heat flow from shield to piston. The spacer is manufactured from steel ***40X10C2M*** (***ЭИ 107***).Heat-barrier ring jointly with two upper rings provides the leak tightness of combustion chamber. The unsplitted thin-walled heat-barrier ring provides accurate opening and closing intake and exhaust ports. All piston rings are manufactured from steel ***4X5MФ1C***. Surfaces of heat-barrier ring and compression rings that are in contact with cylinder walls are covered with hard-wearing chrome-tungsten and running-in ability molybdenum disulfide coverings.

The chrome-tungsten covering is porous for better oil absorption.

There is an oil-retaining micro relief (helical groove) on the contact surface of heat-barrier ring designed for fitting in the “exhaust” piston (Fig. 13). Oil control ring are covered with hard-wearing chrome covering. Piston pin 9 (see Fig. 11) serves as inner roller race 27. It is manufactured from steel ***20X2H4A-Ш***. The working surface of piston pin is carbonized. The inner cavity of piston pin is hermetically sealed by steel blank covers 10.

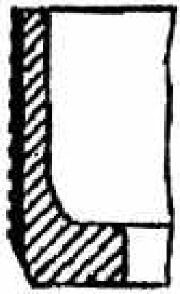


Figure 13 – Heat-barrier ring of “exhaust” piston

The axial fixing of the piston is done by snap rings 8, which are fitted in the grooves of piston bushes.

Piston transfers the force through piston pin and needle bearing. The needle bearing consists of two rows of needle rollers that are separated by ***rings 11*** and limiting ***rings 12***.

Piston of ROTAX-582 RE is molded from aluminum alloy and then machined from outside and partially from inside (Fig. 14).The piston crown is convex, spherical. There are two grooves for fitting the rings turned in the upper part of piston. There are pins pressed into the grooves for constraining their angular shift. The upper ring 2 is self-sealing. The base of the upper ring is trapezoidal with rectangular shoulder. The lower ring is rectangular. There are two bosses 4 from the opposite sides of the piston. There are piston bores drilled in these bosses. Floating piston pin 5 is hollow. It joins the piston with connecting rod via needle cageless bearing 7. The piston pin is constrained from axial displacement by snap rings 8. Piston skirt has booster bore 10 machined for partial bleeding of air/fuel mixture from inside the casing volume. Air/fuel mixture provides cooling of upper and lower parts of the piston and lubrication of piston pin bearing.

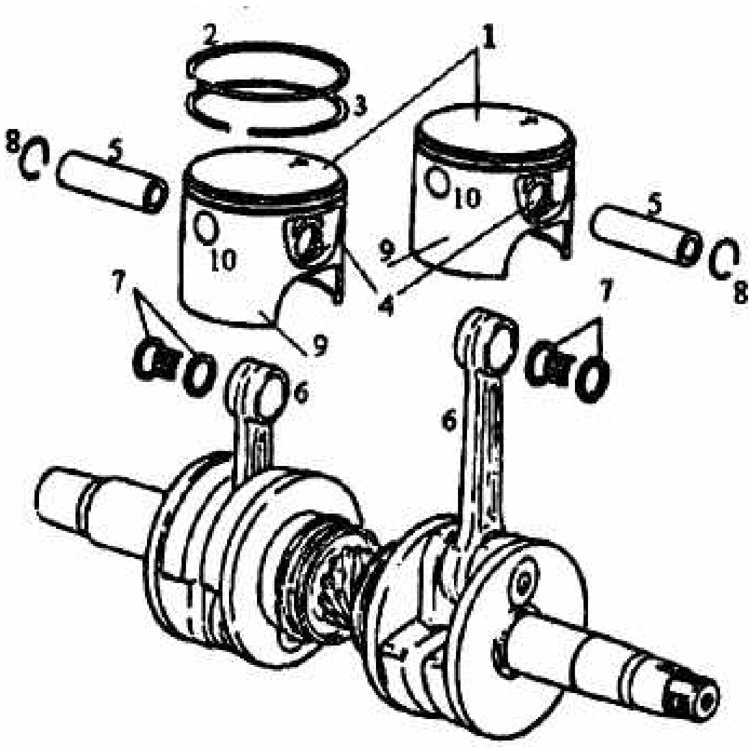


Figure 14 – Piston-cylinder-unit of ROTAX-582 RE

There is ***VOLVO D12/D13 RE*** piston (the modification of 1998) shown in Fig. 15. The piston consists of piston head 1 with integrated ring 2 that jointly form the volume for cooling, and piston skirt 3. Cylinder head and skirt are fastened to the piston pin 4. The piston pin stands pressure loads that come directly from piston head and side force that comes directly from piston skirt. Such a design allows providing better conditions for lubrication.


Figure 15 – Piston of VOLVO D12/D13

There is a drawing of piston of SMD-60 RE in Appendix.

The precise analysis of drawing reveals that the surfaces of piston head and skirt are of complex form. This is because of loading conditions. There are three main reasons of such a piston form:

– non-uniform heating (see Fig. 1) must be compensated choosing the correct initial form of the piston;
– piston must stand the normal load that appears as a result of interaction between piston and cylinder barrel.
– there is necessary to keep lubrication oil wedge between piston and cylinder barrel at any operating mode.

To provide shock-free piston transposition, the piston is of smaller diameter in the bottom part of the skirt.

The change of piston profile at the part A (see Appendix) in transversal cross-sections (perpendicular to M axis) is presented in the Table 2.

Table 2 – Coordinates of piston profile

A, mm	α°										H, mm
	0	10	20	30	40	50	60	70	80	90	
		170	160	150	140	130	120	110	100		
	180	190	200	210	220	230	240	250	260	270	
	360	350	340	330	320	310	300	290	280		
47	0,000	0,000	0,000	0,000	0,000	0,050	0,083	0,092	0,094	0,094	
53	0,000	0,002	0,014	0,049	0,100	0,154	0,200	0,227	0,235	0,235	
61	0,000	0,003	0,017	0,053	0,109	0,167	0,219	0,251	0,267	0,267	
69	0,000	0,003	0,017	0,055	0,111	0,171	0,224	0,258	0,277	0,279	
78	0,000	0,003	0,017	0,055	0,112	0,173	0,227	0,261	0,282	0,285	
88	0,000	0,003	0,017	0,056	0,113	0,174	0,229	0,264	0,285	0,290	
98	0,000	0,003	0,018	0,057	0,114	0,175	0,231	0,267	0,289	0,294	
108	0,000	0,003	0,018	0,059	0,117	0,176	0,232	0,269	0,291	0,297	
120	0,000	0,003	0,018	0,070	0,128	0,177	0,233	0,272	0,294	0,300	
128	0,000	0,003	0,018	0,077	0,133	0,177	0,233	0,272	0,292	0,298	
139	0,000	0,003	0,018	0,073	0,129	0,175	0,231	0,266	0,286	0,292	
149	0,000	0,003	0,018	0,062	0,118	0,174	0,229	0,264	0,283	0,285	
155	0,000	0,003	0,017	0,054	0,110	0,169	0,222	0,255	0,272	0,275	

The specifications made to piston of SMD-60 RE

1. Holes are not allowed on the machined surfaces of the piston:
   – the surfaces of grooves for fitting the piston ring: holes of diameter more than ***1,5 mm***, depth – more than ***1 mm***, number – more than ***2 holes***;
   – surfaces of piston bore: holes of diameter – more than ***2 mm***, depth – more than ***1 mm***, number – more than ***1 hole***, if they are at the distance less than ***5 mm*** from boss face;
   – surface of the piston skirt: holes of diameter – more than ***2 mm***, depth – more than ***1 mm***, number – more than ***2 holes***, if they are closer than ***5 mm*** from each other and from edge;
   – surface of top land;

– inner unmachined surface of piston: diameter – more than ***2 mm***, depth – more than ***1,5 mm***, number – more than ***2 holes***, if they are closer than ***20 mm*** from each other; the maximum number of holes on one piston must not exceed three.

2. Number of pores per the specific area (***cm***^{2}) must not exceed:
   – in the zone of piston bore – ***10 pores*** (***80 %*** of pores must be of diameter lower than ***0,1 mm*** and ***20 %*** of pores must be of diameter ***0,2 mm***); this zone is limited by radius ***E***, which is circumscribed round axis of piston bores;
   – other zones must have less than ***15 pores*** (***80 %*** of diameter must be lower than ***0,3 mm*** and ***20 %*** – lower than ***0,5 mm***).
3. Nicks, machining marks, scorings on the face surfaces are not permitted. The presence of nicks, machining marks, dents and rubbings must confirm approved master forms.
4. The angular shift of oil returning holes relative to their nominal positions is ***±5°***.
5. The volume of combustion chamber must be (***87±1***) ***cm***^{3}.
6. Variation in wall thickness in opposite sides of the piston skirt must not exceed ***0,5 mm***. It must be measured at the distance Я.
7. Having been heated to temperature ***200 – 220°C*** and being kept at this temperature for ***10 h***, the diameter of the piston must not increase for more than ***0,03 mm*** (this size is bigger than the one specified in the drawing). This size must be measured at piston crown and ensured by the manufacturing technology.

Questions for self-testing

1. What are application purposes of RE?
2. What are requirements made to piston construction?
3. Construction and materials of piston rings.
4. What are the application purposes of piston pin, its construction and specific features of its fitting in piston?
5. What are application purposes and sizes of clearances between piston and facing it cylinder barrel?
6. What are specific features of engine pistons: ***AM-38(AM42)***, ***VK-105***, ***ASH-82***, ***ROTAX-582***.
7. What are the reasons of such a complex design of the piston of ***6TD RE***?
8. Name main reasons outer surface of piston skirt is cone-ellipsoidal or barrel-ellipsoidal?

Laboratory activity report

1. Draw the constructive schemes of aviation piston engines.
2. Draw the schemes of piston rings and the scheme that describes the pumping effect of the ring.

3. Identify the emergent constructive zones of the piston.
4. Name materials for manufacturing elements of piston assembly. Give their short characteristic.

4 STRENGTH ANALYSIS

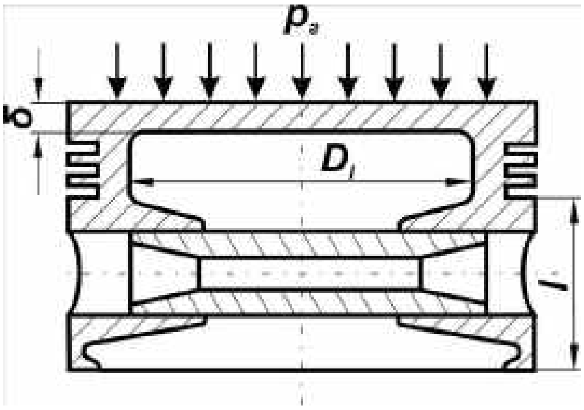


Figure 16 – Illustration to strength analysis

Strength analysis of the piston (Fig. 16).The form of the piston crown and its trimming side walls must be designed in the way to guide the heat follow to the biggest possible number of rings. The most heat is removed via rings to cylinder barrel. The intensity of heat flow increases from center to sides of the piston crown. For this reason, thickness of piston crown must also increase from center to its periphery. Some pistons may have ribbings aimed to improve strength and rigidity of the piston. Ribs also intensify heat removal. The precise analysis of the piston crown is very complex; but modern ***CAD/CAM*** systems enable an opportunity to make such calculations.

For an approximate analysis piston crown can be considered as round plate, which is constrained form opposite sides. This plate is loaded with uniform pressure of gasses. Effect of ribbings is usually neglected. Bending stresses of piston crown can be estimated as follows:

$$\sigma_{bend} = 6,668 \frac{p_{gas\,max} \cdot D_1^2}{4 \cdot \delta^2} \cdot 10^{-4}, \qquad (3)$$

where $p_{GAS\,max}$ is maximum excessive pressure of gases, N/m^2; D_i is an inner diameter of piston crown, m; δ is the width of the piston crown, m; σ_{bend} are bending stresses, which must lie within the limits ***40-60 MPa***.

The height of the side surface l (which is the total height of the piston skirt) is determined by allowable specific loads, which are caused by side force ***N***:

$$k = \frac{N_{max}}{l \cdot D}, \qquad (4)$$

where N_{max} – maximum side force, N; k – allowable specific load, which is equal to ***0,7–1,1 MPa***; D – diameter of the piston, m.

An analysis of the piston pin usually comprises (Fig. 17): strength analysis under bending moments action, maximum allowable strain analysis to prevent jamming of the pin in the rod bushing, an analysis to identify the specific pressure acting interacting surfaces of the piston pin.

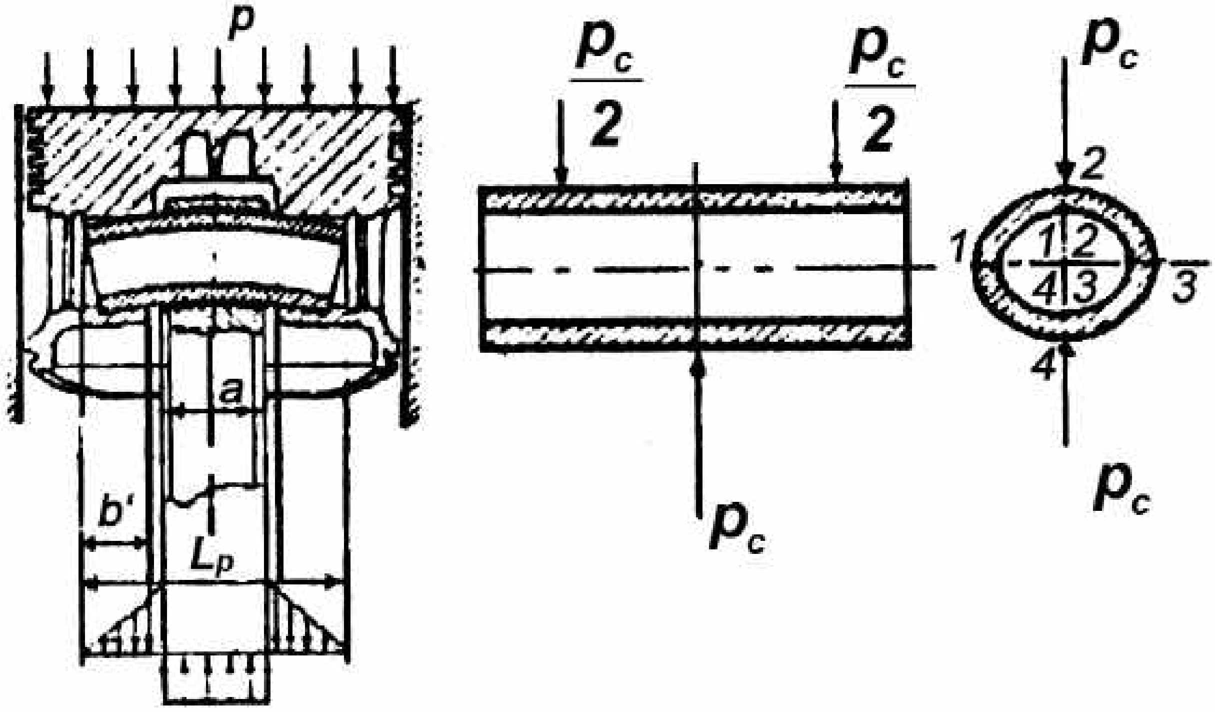


Figure 17 – Illustration to strength analysis of the piston pin

Piston is acted by the variable in time loads. Hence it is reasonable to use fatigue limit as limiting value in the strength analysis. In the analysis piston is considered as a two-support beam, which is loaded by distributed force from both sides and in the central part. The length of piston pin part acted by the distributed force is determined by the length of bosses and rod bushing (see Fig. 17, b). The maximum bending moment in the central part of the pin equals

$$M_{bend} = \frac{p_c}{2}\left(\frac{L}{2} - \frac{a}{4}\right), \qquad (5)$$

where L is a distance between centers of supporting parts in piston bosses, m; a is the length of rod bushing, m; p_c is the force acting the pin that appears as a result of interaction between piston and piston pin, N.

The bending moment, calculated by the eq. (5) is higher than experimentally measured. The bending moment changes with crankshaft angular position, because force p_c itself depends on crankshaft angular position.

Fixed piston pin is acted by the force that changes from its maximum positive value to maximum negative value. The maximum positive value equals

$p_{Gmax} - p_{j\ pist}$ ($p_{j\ pist}$ is the inertia of “pinless” piston) at the moment of explosion and the maximum negative value corresponds to piston being in the top dead center in the end of exhaust stroke.

When considering floating piston pin, it may occur that the maximum strains may appear in zone of the most tensed or compressed fibersat the moment of maximum loading. The maximum loading usually equals $p_{G_{max}} - p_{j\ pist}$. Hence, stresses in each point of design section may vary depending on loading within the limits $p_{pist\ max}$ $p_{pist\ min}$, where $p_{pist\ max} = p_{G\ max} - p_{j\ pist}$.

This case of symmetric loading is the most strength unsafe for the piston pin. Hence it is chosen for the analysis.

The corresponding bending moment (5) is changed with the limits $+M_{bend\ max}$ $-M_{bend\ max}$, where

$$M_{bend\ max} = \frac{p_{g\ max} - p_{j\ max}}{2}\left(\frac{L}{2} - \frac{a}{4}\right). \qquad (6)$$

Forces $p_{g\ max}$ and $p_{j\ max}$ can be calculated as:

$$p_{j\ pist} = -M_{pist} \cdot R \cdot \omega^2 \left[\cos(\alpha) + \lambda \cdot \cos(2\alpha)\right]; \qquad (7)$$

$$p_g = F_n\left(p - p_{H\ calc}\right), \qquad (8)$$

where $p = p_{max}$ (pressure in cylinder).

The mean stresses σ_m of the considered symmetric loading equal to zero, and the stress amplitude of the cycle is

$$\sigma_a = \sigma_{max} = \frac{M_{bend\ max}}{W_{bend}}. \qquad (9)$$

The second bending moment of piston pin area:

$$W_{bend} = \frac{\pi\left(D_{pp}^2 - d_{pp}^2\right)}{32 \cdot D_{pp}}, \qquad (10)$$

where D_{pist} and d_{pist} are outer and inner diameters of the design (mean) station.

The bending safety factor can be calculated as

$$n_{\sigma_{-1}} = \frac{\sigma_{-1}}{\sigma_a \cdot \dfrac{k_\sigma}{\varepsilon_\xi}}, \tag{11}$$

where $\sigma_{-1} = 400 - 450\ MPa$. The size factor ε_ξ for carbonized or nitrated, grinded or polished piston pins can be chosen from Table 3.

Table 3 – The dependence of size factor from the diameter of the piston pin (bending analysis)

D_{pist}, mm	20	40	80	120	200	300
ε_ξ	0,94	0,8	0,66	0,6	0,57	0,57

The safety factor of the piston pin, calculated by eq. (11), is lower than real safety factor. The reasons are:

– increased value of calculated bending moment;
– made assumption that frequency of load variation is multiple to frequency of piston turning (the multiplicity is necessary condition for symmetrical loading provision).

Loading conditions and the shape of the piston pin in different engines are very similar, so safety factor is quite a reliable criterion to estimate the strength of pins.

The safety factor $n_{\sigma_{-1}}$ from eq (11) significantly varies within the limits ***1,0 – 2,2***. When choosing the needed safety factor from the specified frame it is necessary to take into account the technical features of each particular piston pin. So, for example, the strength of the piston can be improved by accurate machining and nitrating its surfaces (inner surface of the piston pin is also nitrated). And inversely, the intense machining (without thermal treatment) of the inner surface reduces a lot the strength of the piston pin.

Being acted by the force p_c piston pin becomes oval as it is shown in Fig. 18. The maximum ovalization is in the central part of the piston pin. The diameter of the piston becomes smaller in the plane of p_c action. And in the perpendicular plane it becomes bigger. The ovalization of the piston pin results in stresses in inner and outer surface layers. The nature of stresses is determined by the change of curvature. Tensile stresses appear at points ***1*** and ***3***, and compression stresses at points ***2*** and ***4***. These points belong to outer surface of the piston pin. And inversely, compression stresses appear at points $1'$ and 5, and tensile stresses at points $2'$ and $4'$. These points belong to the inner surface of the piston pin. There is a diagram of stresses that appear in the outer surface layer of the pin is presented in Fig. 18.

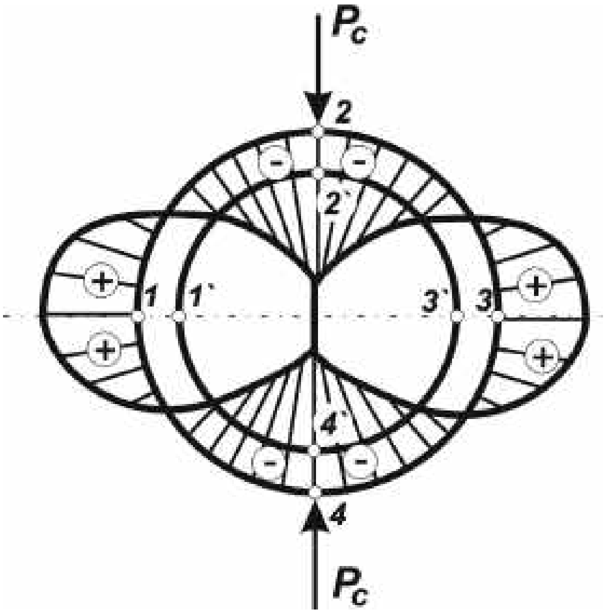


Figure 18 – Diagram of stresses in the piston pin at ovalized state

The researches of R.S. Kinasoshvili demonstrated that the biggest stresses appear at points ***2'*** and ***4'***. They equal:

$$\sigma_{OV} = \frac{p_c}{L_{pp} \cdot d_{pp}} \cdot \xi ,$$

where $\xi = f(\alpha)$ and α in its turn is a coefficient that equals $\frac{d_{pp}}{D_{pp}}$.

The dependence between ξ and α is presented in Fig. 19. The ovalization degree is usually determined as increasing of external diameter in direction perpendicular to the direction of applied force. The increase of diameter can be calculated y Kinasoshvili equation:

$$\Delta d_{pp} = 0{,}09 \cdot \frac{p_{pp}}{E \cdot L_{pp}} \left(\frac{1+\alpha}{1-\alpha} \right)^3 \cdot \left[1{,}5 - 15 \cdot (\alpha - 0{,}4)^3 \right] = \frac{p_{pp}}{E \cdot L_{pp}} \cdot \alpha , \qquad (13)$$

where E is Young's modulus, L_{pp} – the length of piston pin.

The value of Δd_{pp} is within the limits ***0,02 – 0,07 mm*** (the value may be even more than ***0,07*** for some piston pins). When designing the new engine it is appropriate to choose the piston pin geometry that will provide the ovalization degree Δd_{pp} less than ***0,05 mm***. Engines with high power output have σ_{ov} being within the limits ***110–230 MPa***.

The external diameter of the piston pin is limited by the specific loads in piston bosses and rod bushing. The specific load in rod bushing can be calculated as

$$k_{bush} = \frac{p_{gas\,max}}{D_n \cdot a} . \qquad (14)$$

The specific load in bronze rod bushing must be within the range ***40 – 70 MPa***.

The specific load in piston bosses can be calculated by the transformed eq. (14):

$$k_{boss} = \frac{p_{gas\,max}}{2 \cdot b \cdot D_n}, \tag{15}$$

where ***2b*** is common length of two piston bosses.

The specific load in aluminum piston bosses must be within the range ***40 – 50 MPa.***

An analysis of the piston rings (Fig. 20). The piston ring must be compressed during operation and apply certain pressure at cylinder walls. The pressure piston ring acts cylinder wall depends as on elastic strain of the ring having been compressed in the cylinder, so on sizes of ring profile. The deformations of the ring result in bending stresses.

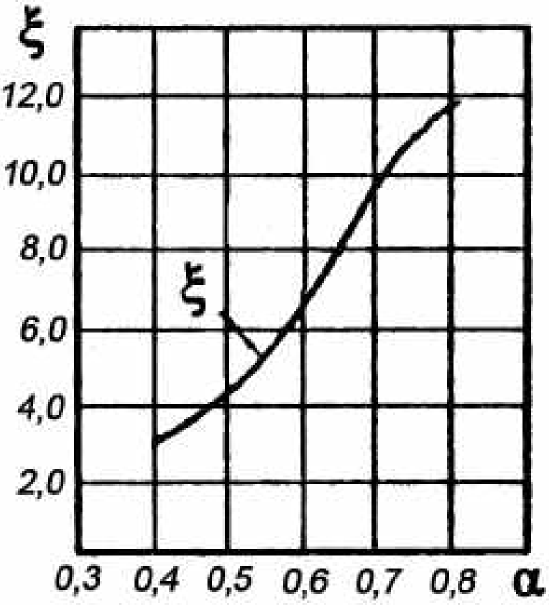


Figure 19 – The dependence $\xi = f(\alpha)$

The mechanics of materials gives the equation to relate curvature with bending moment M_{bend}:

$$\frac{1}{\rho'} - \frac{1}{\rho} = \frac{M_{bend}}{E \cdot J}, \tag{16}$$

where ρ' and ρ are respectively final and initial radiuses of curvature of the middle ring fiber, E, J are Young's modulus and second area moment, m^4 respectively.

Let`s designate the width (see Fig. 20) of the ring as δ, then bending stresses equals to

$$\sigma_{bend} = \frac{M_{bend} \cdot \delta}{2 \cdot J}. \tag{17}$$

So, eq. (16) is transformed to

$$\frac{1}{\rho'} - \frac{1}{\rho} = \frac{2 \cdot \sigma_{bend}}{\delta \cdot E}. \tag{18}$$

To calculate ring width according to given stresses, the change of ring curvature must be known. The curvature is of opposite signs in case when ring is fitted the piston (the ring is tensed) and in case ring is compressed inside the cylinder barrel.

Generally, piston ring is considered to be fitted in the way to enfold piston by its inner surface (internal diameter). Then external diameter equals $D+2\delta$, and radius of middle ring fiber – $\rho'=R+\frac{\delta}{2}$ (see Fig. 20). If ρ is the curvature radius of the ring at standalone state, then stresses appear when fitting the piston that can be calculated as:

$$\frac{1}{\rho}-\frac{1}{R+\frac{\delta}{2}}=\frac{2\cdot\sigma_{bend}}{\delta\cdot E}, \tag{19}$$

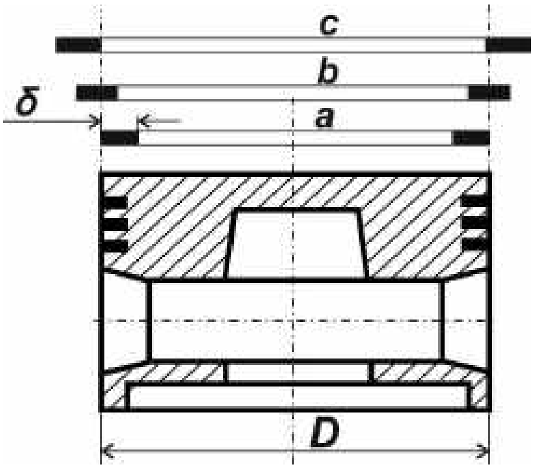


Figure 20 – The sizes of the piston ring: a –working state; $\rho''=R-\frac{\delta}{2}$; b – standalone state; $\rho=R$; c – when fitting the piston, $\rho''=R+\frac{\delta}{2}$

where σ_{bend} are stresses that appear in the middle ring part during fitting the piston.

The external diameter of the ring being fitted in the cylinder (working state) equals D, and radius of middle fiber – $\rho=R-\frac{\delta}{2}$. For this case: $\frac{1}{R-\frac{\delta}{2}}-\frac{1}{\rho}=\frac{2\cdot\sigma'_{bend}}{\delta\cdot E}$, where σ'_{bend} are stresses that appear in the middle ring part in compressed state.

The stresses that appear in the ring in both cases depend on ring profile in standalone state. The bigger radius ρ of the ring at standalone state, the bigger strains and stresses appear in the ring at compressed state and the lower strains and stresses appear during fitting the ring in the piston. If to make an assumption that middle radius at mean state is equal to the radius of external surface of the piston (see Fig. 20) then $\rho=R=\frac{D}{2}$, $\frac{1}{R-\frac{\delta}{2}}-\frac{1}{R}=\frac{2\cdot\sigma'_{bend}}{\delta\cdot E}$ or

$$\sigma'_{bend}=\frac{\delta^2\cdot E}{4\cdot R\cdot\left(R-\frac{\delta}{2}\right)}.$$

As $\frac{\delta}{2}$ is generally 25–30 times lower than R then bending stresses equal to

$$\sigma'_{bend} = \frac{\delta^2 \cdot E}{4 \cdot R^2} = \left(\frac{\delta}{2 \cdot D}\right)^2 \cdot E . \quad (20)$$

If $\rho = R = \frac{D}{2}$ then

$$\sigma'_{bend} \approx \left(\frac{\delta}{D}\right)^2 \cdot E . \quad (21)$$

Bending stresses of cast iron rings ($E \approx 82500\ MPa$, $\frac{\delta}{D} = \frac{1}{25} - \frac{1}{30}$, $\rho = R$) are within the limits $\sigma_{bend} = 92 - 135\ MPa$.

The pressure ring acts cylinder bearing surface can be calculated according to given bending moment M_{bend}. Let`s suggest that ρ is constant along the whole ring surface, then bending moment caused by pressure acting section **s-t** (see Fig. 21) can be estimated taking into account that the resultant force of pressure acting the arc, equals to resultant of pressures acting the chord subtending this arc. Resultant of pressures is applied to the center of chord and directed to the center of ring.

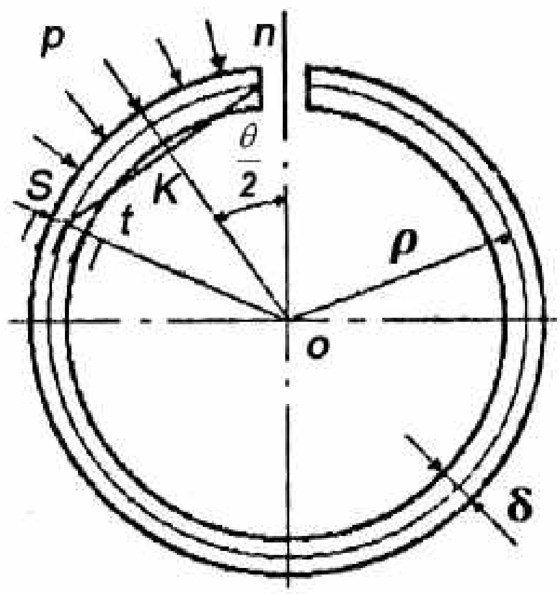


Figure 21 – Illustration to strength analysis of piston ring

If section **s-t** forms an angle θ with central line of the lock **o-n**, then neglecting the lock (the difference between radius of middle fiber and cylinder radius is neglected), bending moment can be calculated as

$$M_{bend} = p \cdot y \cdot l_{mn} \cdot R \cdot sin\left(\frac{\theta}{2}\right) = 2 \cdot p \cdot R^2 sin^2\left(\frac{\theta}{2}\right), \quad (22)$$

where h is the ring height.

But

$$\boldsymbol{M_{bend}} = \frac{\boldsymbol{2 \cdot l} \cdot \sigma_{\boldsymbol{bend}}}{\delta} = \frac{\boldsymbol{2 \cdot h} \cdot \delta^{\boldsymbol{3}}}{\boldsymbol{12} \cdot \delta} \cdot \sigma_{\boldsymbol{bend}} = \frac{\boldsymbol{h} \cdot \delta^{\boldsymbol{2}}}{\boldsymbol{6}} \cdot \sigma_{\boldsymbol{bend}}, \tag{23}$$

consequently,

$$\frac{\boldsymbol{h} \cdot \delta^{\boldsymbol{2}}}{\boldsymbol{6}} \cdot \sigma_{\boldsymbol{bend}} = \boldsymbol{2 \cdot p \cdot R^2 sin^2}\left(\frac{\theta}{\boldsymbol{2}}\right), \tag{24}$$

or

$$\rho = \frac{\delta^{\boldsymbol{2}} \cdot \sigma_{\boldsymbol{bend}}}{\boldsymbol{12 \cdot R^2 sin^2}\left(\frac{\theta}{\boldsymbol{2}}\right)} = \frac{\boldsymbol{1}}{\boldsymbol{3 \cdot D^2}} \cdot \frac{\delta^{\boldsymbol{2}} \cdot \sigma_{\boldsymbol{bend}}}{\boldsymbol{sin^2}\left(\frac{\theta}{\boldsymbol{2}}\right)} \tag{25}$$

In the center of the ring ($\theta = \boldsymbol{180°}$) pressure equals $\boldsymbol{p} = \frac{\delta^{\boldsymbol{2}} \cdot \sigma_{\boldsymbol{bend}}}{\boldsymbol{3 \cdot D^2}}$.

Taking into account eq. (21) for the case when $\boldsymbol{p} = \boldsymbol{R}$:

$$\rho = \frac{\boldsymbol{E}}{\boldsymbol{3}} \cdot \left(\frac{\delta}{\boldsymbol{D}}\right)^{\boldsymbol{4}}. \tag{26}$$

So, pressure piston ring acts cylinder bearing surface does not depend on ring height $\boldsymbol{h}$. The higher ring width δ , the bigger pressure $\boldsymbol{p}$ is. But stresses are increased if ring width increases. Hence, pressure, the piston ring acts cylinder bearing surface, is practically limited by allowable operation stresses $\sigma_{\boldsymbol{bend}}$ and may vary within the limits ***0,06 – 0,1 MPa*** for modern engines.

The above presented analysis of piston ring was carried out under the assumption that the pressure equally acts cylinder walls. But theoretical and experimental researches of the piston ring operation revealed that that it is profitable to design piston rings with non-uniform circular pressure profile. It is particularly reasonable to increase pressure close to ring lock (at the ends of the ring), as it is in the majority of modern designs. Chosen circular pressure profile is ensured by the corresponding ring profile in standalone state. Manufacturing such rings requires special technologies.

The clearance in ring lock must provide free bending during operation (thermal strains of the cylinder are also taken into account for choosing the proper clearance). The clearance equals $\Delta \boldsymbol{p} = \boldsymbol{0,5} - \boldsymbol{1,0\ mm}$. If middle radius of piston ring changes from $\boldsymbol{p} = \boldsymbol{R} - \frac{\delta}{\boldsymbol{2}}$ during operation to $\boldsymbol{p}$ in standalone state, then the lock increases by

$$2\pi - 2\pi\left(R - \frac{\delta}{2}\right) = 2\pi\left(\rho - \frac{D}{2} + \frac{\delta}{2}\right). \quad (27)$$

So, clearance in standalone state is

$$\Delta_{STAND} = \Delta_p + 2\cdot\pi\cdot\left(\rho - \frac{D}{2} + \frac{\delta}{2}\right). \quad (28)$$

In the particular case when $\rho = R = \dfrac{D}{2}$

$$\Delta_{STAND} = \Delta_p + \pi\cdot\delta = \pi\cdot\delta + (0,5 - 1,0)\ mm. \quad (29)$$

Questions for self-testing

1. Name forces acting piston surfaces.
2. Name strains of components of piston assembly.
3. Piston pin loading. The strength analysis of the piston pin.
4. The specific features of strength analysis of piston rings

Laboratory activity report

1. Draw the loading schemes that illustrate the strength analysis of the piston, piston pin. Study and understand basic mathematical equations used in these calculations.
2. Make an analysis of stress profile while ovalization.
3. Draw loading scheme that illustrates strength analysis of piston pin. Study and understand basic mathematical equations used in the analysis.

5 MATERIALS AND FRAGMENTS FORM MANUFACTURING TECHNOLOGY

Basic requirements to materials, applied for **manufacturing pistons** are:

– high mechanical strength that provides high reliability of operation under shock loads;
– high fatigue limit and high immunity to cyclic loads;
– sufficient elasticity that provides piston skirt shaping cylindrical without plastic strains;
– sufficient heat strength and heat resistance that allows piston operation at temperatures ***300 – 400°C*** without significant reduction of mechanical properties;

– minimum thermal expansion factor;
– good heat conductivity that provides intensive heat removal; piston is heated because it contacts hot gases that are in above piston volume;
– high corrosion-resistance, which is needed to stand aggressive gases and acids that are formed in above piston volume;
– high antifriction properties preventing scores during engine operation.

The most abundant materials for manufacturing pistons are graphitic pig iron and aluminum based alloys. The more high-speed engine is, the higher inertia is. Hence, the above enumerated requirements to pistons became prevalent. That is why, only aluminum based alloys will be considered next.

The most abundant method to get the aluminum cast is gravity die casting. The outer surface of the piston is formed by the walls of the open die, and the inner surface is formed by steel bars. The compound and properties of certain casted alloys are given in Table **Ошибка! Источник ссылки не найден.**.

Number of bars (3-5) depends on the shape of the piston internal surface. There also applied aluminum diecasting dies with anodized inner surfaces. This type aluminum diecasting dies are applied to cast pistons with uniform fine-grained structure. This gives increasing in fatigue limit σ_{-1} by approximately ***30 %***. To obtain the necessary hardness of aluminum pistons (more than ***90 – 100 HB***), it is reasonable to control the cooling rate of the cast in the diecasting die. If casted piston is of lower hardness, then it will be hardened with followed ageing. Eutectic aluminum-silicon alloys are hardened at temperatures ***510 – 530°C***. If possible, it is better to escape hardening because it may result in micro cracks at the most loaded sides of the piston. Aluminum pistons are tend to ageing that results in enlarging the piston. Pistons are exposed to artificial ageing at temperature ***190 – 200°C*** during ***8 – 10 hours*** to prevent scoring. To stabilize sizes of the piston, it is exposed to tempering three times (***5 hours*** each) with interim cooling at cooling rate ***50°C/hour***. Being thermally treated in above considered way pistons reliably operate at temperatures ***350 – 360°C***.

Uprated engines (aircraft and sport engines) require stamped pistons manufactured from wrought, aluminum and copper based alloys (see Table 5). These alloys are of good mechanical properties as at room temperatures so at increased temperatures. Their properties can be significantly improved by hardening (Table 6).

Table 4 – Compound and properties of some casted alloys

Country	Mark	Elements concentration in aluminum alloys, %								σ_{ultim}, MPa	Hardness HB
		Cu	Si	Fe	Mn	Mg	Ni	Zn	Ti		
								Less than			
USA	SAE332	2 – 4	8,5 – 10,5	max 1,5	max0,8	0,2 – 0,6	max 0,5	1	0,25	310	105
USA	SAE334	1,8 – 2,8	11,13	max 1,5	max 0,5	0,7 – 1,3	1	1	0,25	310	110
England	ZM – 13WP	0,5 – 1,3	11,13	0,8	0,5	0,8 – 1,5	0,7 – 2,5	0,1	0,2	280	100 – 150
FRG	AlSi12	max 0,05	11,13,5	0,6	max 0,5	–	max 0,2	0,5	0,15	200 – 260	55 – 70
FRG	AlSi10M gCu	max 0,2	9 – 11	max 0,6	0,2 – 0,5	max 0,5	max 0,1	0,2	0,1	200 – 260	65 – 85
FRG	K1275	0,8 – 1,5	11 – 13	0,7	0,2	0,8 – 1,3	0,8 – 1,3	0,2	0,2	200 – 250	90 – 125
FRG	KS282	0,8 – 1,5	23 – 26	0,7	max 0,2	0,8 – 1,3	0,8 – 1,3	0,2	0,2	180 – 220	90 – 125
Japan	AC8B	2,4	8,5 – 10,5	1,2	0,5	0,5 – 1,5	0,5 – 1,5	0,5	0,2	220	90
CIS	АЛ25	1,5 – 3	11 – 13	0,8	0,3 – 0,6	0,8 – 1,3	0,8 – 1,3	0,5	0,05 – 0,2	190	90
CIS	АЛ26	1,5 – 2,5	20 – 22	0,7	0,4 – 0,8	0,4 – 0,7	1 – 2	0,3	max 0,2	160	90
CIS	АЛ30	0,8 – 1,3	11,13	0,7	max 0,2	0,8 – 1,3	0,8 – 1,3	0,2	max 0,2	200	90

Table 5 – Compound and properties of some aluminum and copper based alloys

Country	Mark	Elements concentration in aluminum alloys, %								σ_{ultim}, MPa	Hardness HB
		Cu	Si	Fe	Mn	Mg	Ni	Zn	Ti		
								Less than			
USA	SAE34	9,2 – 10,8	max 2	max 0,5	max 0,5	0,25	max 0,05	0,8	0,25	300	115
USA	SAE39	3,5 – 4,5	max 0,7	max 1	max 0,35	1,2 – 1,8	1,7 – 2,3	0,35	0,25	340	105
England	LM – 14	3,5 – 4,5	max 0,6	max 0,6	max 0,6	1,2 – 1,7	1,8 – 2,3	0,1	0,2	240	65 – 85
Japan	AC5A	3,5 – 4,5	max 0,6	max 0,8	0,1	1,2 – 1,8	1,7	0,1	0,2	305	90
CIS	AK2	4	0, 75	0,75	–	0,6	2	–	–	370	95
CIS	AK4 – 1	2,2	max 0,35	1,3	–	1,6	1,2	–	–	390 (450)	103

Table 5 – Mechanical properties piston allow

Temperature T, °C	Mechanical properties			
	Ultimate strength σ_{ultim}, MPa	Long-term strength σ_{-1},MPa	Young`s modulusE, MPa	Coefficient of thermal expansion α, 1/°C
150	400	290	-	19,6
200	340	170	0,63	-
250	280	100	-	-
300	170	40	0,51	24,8

Aluminum-copper pistons are greatly exposed to thermal expansion, that is why they are designed to have great clearances inside the piston barrel. This results in strikes and intense dynamic loads during initial period of engine operation (“cold” operation). Choosing the form of internal cavity for stamped pistons is determined as by design considerations, so by manufacturing technology:

- the vertical walls must be inclined at ***5 – 10°*** for convenient pulling out puncheon;
- stamp must have no complex forms that are complex for stamping.

There are series of strict requirements made to machining the pistons. It is necessary to provide the precise distance between piston pin axis and piston crown, because this distance determines the compression ratio. The axis of the piston pin and face surfaces of piston grooves turned for piston rings fitting must be strictly perpendicular to piston axis. To reduce friction and wearing out the pistons, their outer surfaces are machined extremely carefully. The outer surface of the piston is specially coated to prevent it from wearing out during breaking-in and short-time extreme loadings. The most widely used methods are tin-electroplating and contact tinning by pure tin, graphitization and parkerising of the piston skirt.

The analysis of the maintenance of the uprated engines have revealed that the groove for upper ring fitting is tend to crumpling and early wear out. To cope with this problem designers usually add strengthening insert made of Ni-resist cast iron or apply remelting of topmost compression ring zone with adding alloying additions during remelting. Fitting Ni-resist cast iron insert and potting it with the alloy must last less than ***30 seconds***. Otherwise, metallic binding will not happen between insert and piston material.

Requirements to materials of piston rings:

- high fatigue strength;
- high static bending strength;
- high wear resistance and good antifriction properties;
- ability to stand gas and electrochemical corrosion;
- short-time run-in;

– ability to keep the lubricant in the surface layer;
– good heat conductivity and resistance;
– good cutability;
– absence of adhesion between materials of the piston and the cylinder bearing surface.

Piston rings are casted from alloyed graphitic pig iron of pearlite structure (with small lamellar graphite), that has high antifriction properties and high-temperature strength. Oil control rings are mainly manufactured from steel. Piston rings are casted individually or by centrifugal casting.

The quality of piston rings significantly depends on carbon distribution in the ring. The limiting values of fixed carbon (***0,45 – 0,80 %***) are generally specified in engineering specifications. Increased content of fixed carbon results in increased fragility of piston rings, and reduced content – in reduced hardness and elasticity of rings that in their turn cause intense wear out. The distribution of free carbon (graphite) at the cross-section of piston ring is also of great importance. Non-uniform distribution of free carbon is one of the main reasons piston rings lose their elastic properties.

Piston rings are significantly affected by the technologies used for billets and machining the rings. Chemical compound and mechanical properties of piston rings are presented in Table **Ошибка! Источник ссылки не найден.**.

Table 6 **– Chemical compound and mechanical properties of piston rings**

Chemical composition						E, MPa	σ_{ultim}, MPa	σ_{bend}, MPa	Hardness HB
C	Si	Mn	P	Cr	other				
3,7 – 3,9	2,7 – 2,9	0,6 – 0,8	0,3 – 0,5	0,25 – 0,35	0,3 – 0,5 Mo	70000 – 90000			
2,9 – 3,3	1,2 – 1,8	0,6 – 1,0	0,2 – 0,5	–	–	105000 – 12500	250 – 350		197 – 229
3,6 – 4,0	2,5 – 3,0	0,5 – 0,8	0,4 – 0,6	0,3	0,3 Cu	85000 – 115000		350 – 450	
3,5 – 4,0	2,3 – 2,9	0,2 – 0,5	0,3	0,2	0,3 Cu, 0,4 V, 1,0 Ni	150000 – 170000		1300	

To improve the resistance of piston rings their surfaces are covered with coatings. The most used is chromium – plating, especially for topmost piston ring. Coating width is ***0,12 – 0,18 mm***. Chrome coating works well with cast iron or steel cylinder. To keep lubricant on the ring surface the chrome coating of the ring is manufactured porous. The hardness of chrome is much higher than the hardness of structural components used to manufacture cylinders. Hence, chromium – plated rings are beneficial to be used with more rigid cylinders. Higher requirements are made to manufacturing tolerance of such rings because hard chrome coating is very complex to be runin.

Another variant is to phosphatize ring surfaces in phosphatic – acid salt mortars of manganese and iron. This chemical treatment forms the film (film width ***8 – 15 mcm***) on the surfaces of the piston ring. The film is formed by orthodiphosphates and orthotriphosphates of metals. Phosphatized surface is porous that improves lubricant absorption. Other chemical treatments can be applied: tinning, ironing, oxidation, etc. Generally, chemical treatments improve run – in and reduce friction coefficients.

Piston pins are manufactured from carburizing or nitralloy steels with high mechanical properties (e.g. ***15XM***, ***38XA***, ***12XH3A*** и ***18X2H4BA***, which ultimate strength is $\sigma_{ult} = \boldsymbol{120 - 150\ MPa}$). Mechanical properties of materials are: $\sigma_{ult} = \boldsymbol{480 - 1200\ MPa}$, $\delta = \boldsymbol{10 - 25}\%$, $\boldsymbol{HB = 130 - 220}$. The hardness of carbonized surface is $\boldsymbol{HRC = 59 - 61}$ (carbonized layer is ***0,8 – 2 mm*** depth).

The surface of the piston is machined very carefully (it is grinded and polished) to reduce the wear out and improve the fatigue strength. The outer surface of the piston is of fifth (sixth) grade according to current ITG (international tolerance grade).

Piston is fitted in bosses and rod bushing at different tolerances. Hence, Machining tolerances are specified by shaft tolerance, because using hole tolerance results in stepped piston pin. This makes no constructive and technological benefits.

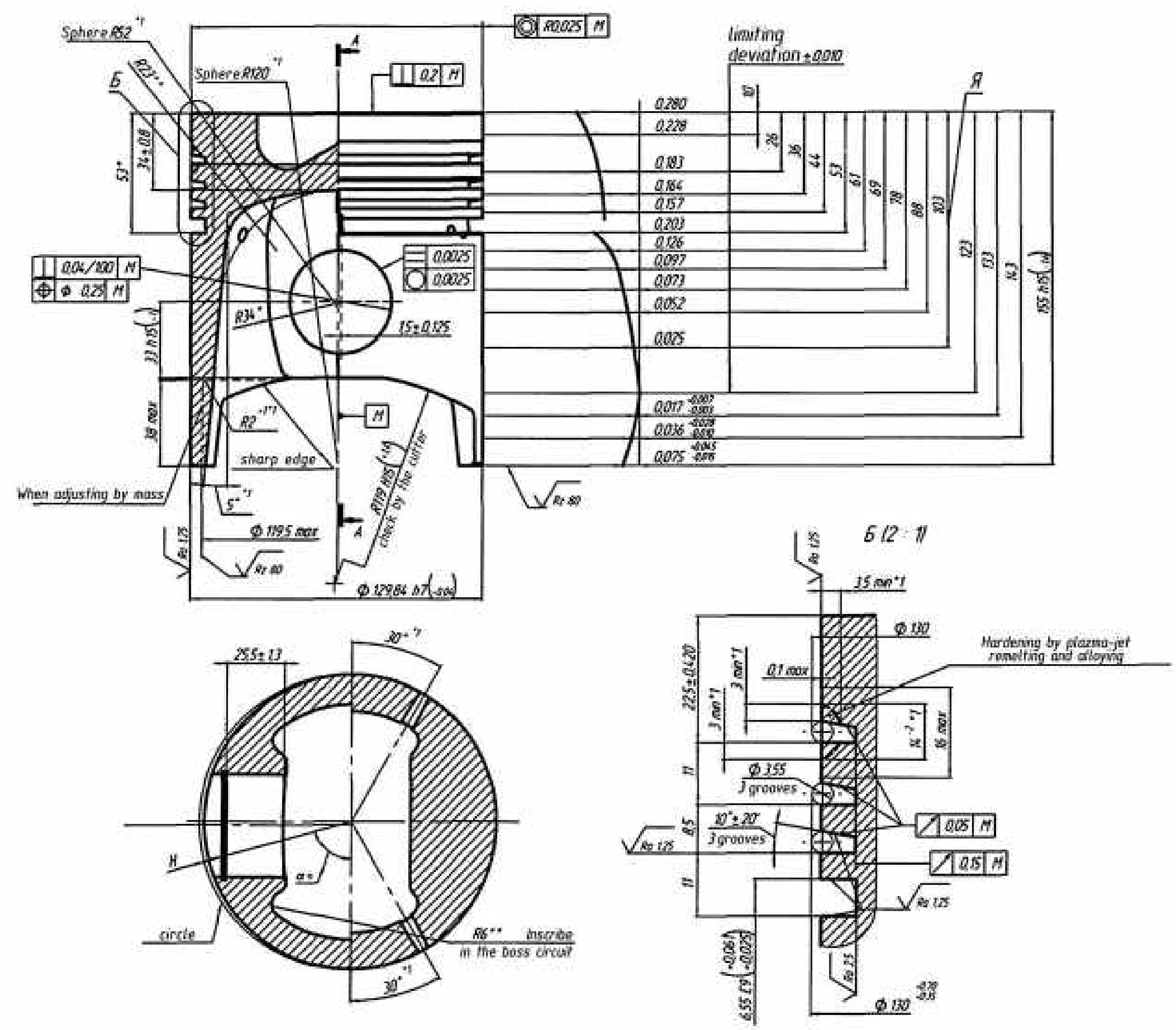
Questions for self – testing

1. What are main requirements made to materials used to manufacture piston, piston rings and pins?
2. Main specific features of manufacturing process of piston assembly components.
3. What coatings are applied in piston assembly?

REFERENCES

1. Масленников, М. М. Авиационные поршневые двигатели [Текст]: учеб. пособие для авиац. вузов / М. М. Масленников, М. С. Рапипорт. – М.: Оборонгиз, 1951. – 848 с.
2. Двигатели внутреннего сгорания. Конструирование и расчеты на прочность поршневых и комбинированных двигателей [Текст]: учеб. для вузов / под ред. А. С. Орлина, М. Г. Круглова.– М.: Машиностроение, 1984.–383 с.
3. Штода, А. В. Авиационные двигатели. Динамика, конструкция и расчет на прочность. [Текст]: учеб пособие для вузов / А. В. Штода., – М.: ВВИА им. Н. Е. Жуковского, 1959. – Ч. 2. – 432 с.
4. Материалы для карбюраторных двигателей [Текст]: справ. пособие / А. В. Лакедемонский, Ю. Е. Абраменко, Е. А. Васильев, А. Г. Возлинский.– М:. Машиностроение.– 1969.– 223 с.
5. Конструирование двигателей внутреннего сгорания [Текст]: учеб. для вузов / Н. Д. Чайнов, Н. А. Иващенко, А. Н. Краснокутский, Л. Л. Мягков / под. ред. Н. Д. Чайнова.– М.: Машиностроение, 2008. – 496 с.

The drawing of the piston of СМД-60 RE (fragment)



Навчальне видання

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ПОРШНІ ДЛЯ ДВИГУНІВ ВНУТРІШНЬОГО ЗГОРЯННЯ

(Англійською мовою)

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