OPERATION & ECONOMY

Identification of service failures of cylinder valves of ship piston combustion engines

Zbigniew Korczewski
Polish Naval University

ABSTRACT

This paper presents selected diagnostic problems of charge exchange system of ship piston combustion engines. Theoretical background of wear process of cylinder valves was highlighted in the aspect of identification and sources of known and identifiable states of unserviceability. The presented results of endoscopic examinations concern failures of cylinder valves of the engines installed on Polish Navy ships.

Keywords : technical diagnostics, ship diesel engine, valve timing

INTRODUCTION

Correct operation of self-ignition engine, which ensures expected performance and efficiency in steady and transient operational states, depends to a great extent on effectiveness of charge exchange in its cylinders. Quality of the process is demonstrated by values of coefficients of filling the cylinders with fresh charge, by which the so called filling efficiency is determined. Values of the coefficient are determined mainly by two factors :

♦ optimally selected distribution phases in the sense of ensuring the most favourable opening and closing angles of air inlet and exhaust gas ducts (cylinder filling and scavenging), in full range of possible engine load changes

♦ optimum velocity of air and exhaust gas flowing through inlet-outlet system, which ensures effective whirling the air flowing into cylinder.

To ensure optimum values of the parameters during ship engine service it is necessary to reach the full serviceability state of cylinder valves which constitute its structural parts most thermally and mechanically loaded. Specially sensitive elements are outlet valves washed by exhaust gas having the temperature above 1000 K. In such service conditions they are to fulfil additional requirements as regards heat exchange and resistance to abrasion and impact load in high temperature, as well as high demands concerning corrosion resistance.

MECHANISM OF GENERATING FAILURES OF CYLINDER VALVES

During its operation the cylinder valve is forced to move along its spindle in the guides which undergoes friction wear. The process goes at high temperature of the spindle whose additional task is to absorb heat from valve head. As a result an excessive increase of radial clearance is produced between the guide and spindle, which leads to an undesirable skew of the valve resulting in loss of cylinder tightness, gas eruption, lubricant leakage from the spindle – guide precision pair until intensive wear of the entire valve unit is reached. The phenomenon may especially intensively develop in the case of supplying the engine with fuel oil of high sulphur content. The cases are known of completely burned-out valves as a subsequent result of extensive wear of valve guide [7]. In extreme case, cracking the valve spindle, its falling down into cylinder space and subsequent failures of the „piston – piston rings – cylinder" system (TPC), including piston cracking, can happen. An observable symptom of worn valve guides are smoked valve springs, that indicates a lack of tightness of combustion chamber.

Another, often found failure of cylinder valves is a drop of elasticity of tightening springs and even their fatigue failures [4,7]. In such situation also loss of cylinder tightness can happen during its filling when the springs of outlet valves are cracked, and also during gas exhaust process when the springs of inlet valves are cracked.

THEORETICAL BACKGROUND OF WEAR PROCESS OF VALVE GUIDES AND SPINDLES

From the equilibrium condition of the forces acting on the valve at the initial instant of its opening (Fig.1) it results that the force, R, generated by distribution shaft cam pressure acting onto the valve spindle face, is balanced by the sum of the cylinder gas pressure force, P_g, acting onto valve head, the valve mechanism inertia force P_b and the spring tension force P_s, according to the equation :

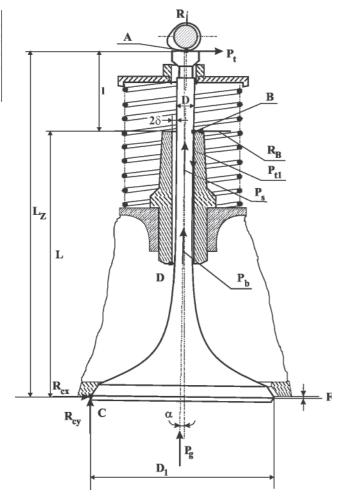$$R = P_g + P_b + P_s \qquad (1)$$

Fig. 1. Schematic diagram of the forces acting onto cylinder valve at the initial instant of its opening .

Due to cam sliding over valve spindle face the additional friction force P_t resulting in side pressure forces of the spindle moving inside the guide, is produced :

$$P_t = R\mu \qquad (2)$$

where : μ – sliding friction coefficient in the point A.

In the same time due to action of the force R in the point B, apart from the lateral force R_B, appears the friction force P_{t1} which becomes the direct cause of valve guide wearing :

$$P_{t1} = R_{B1}\mu_1 \qquad (3)$$

where : μ_1 – sliding friction coefficient in the point B.

Values of the coefficients μ and μ_1 depend on material properties of sliding elements, as well as on a degree of smoothness and dryness of contact surface. However, they do not depend on a size of contact surface (unless its area is so small that pressure force can deform it).

As in the spindle-guide precision pair always certain radial clearance δ dependent on the spindle diameter D appears, during valve opening certain deviation, α, of coaxiality of valve guide and spindle is generated. However in the case of an excessive wear of the guide and increased radial clearance at the instant when cam pressures on the valve spindle, before the valve head becomes separated from valve seat, the gap F appears on one side of the seat face, that leads to loss of valve tightness, but on its other side the valve head is pressed onto the valve seat in the point C, that leads to valve spindle bending in the direction of action of the force P_t (around the point C). Consequently through the developed gap hot cylinder gases

get out with high velocity, which results in local overheating the valve head and seat materials, as well as in forming local erosion pits in valve seat face, Fig.2.
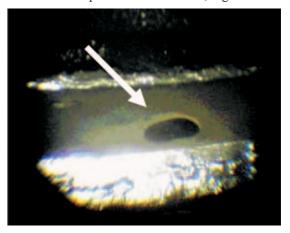

Fig. 2. M401A-1 engine − the outlet valve head of the cylinder no. 1 − traces of erosion wear of valve seat face .

Analysing the distribution of the forces acting onto the valve, shown in Fig.1 one can observed that due to action of the friction force P_t in the points B and C of the valve spindle - guide contact as well as the contact in the valve seat face, the reaction forces R_B and R_{Cx}, respectively, appear which generate the bending moments M_B and M_C in those points. It should be also observed that the force system is dynamic, i.e. that in which load is changing continually. In quasi-stationary approach the force resultants can be determined by using the condition of equilibrium of forces and moments, as follows :

$$\sum P_{ix} = 0 \Rightarrow R_{Cx} - R_B + P_t = 0 \qquad (4)$$

hence :

$$R_{Cx} = R_B - P_t \qquad (5)$$

$$\sum P_{iy} = 0 \Rightarrow R_{Cy} + P_g + P_b + P_s - P_{t1} - R = 0 \qquad (6)$$

Assuming that at a given instant the sum of the forces acting onto the system is constant one obtains the following :

$$P_w = P_g + P_b + P_s - R \qquad (7)$$

Then the sum of bending moments respective to the point C is as follows :

$$\sum M_{iC} = 0 \Rightarrow P_w \left(\frac{D_1}{2} + \delta \right) + $$
$$- P_{t1} \left(\frac{D_1}{2} + \frac{D}{2} + \delta \right) + R_B L - P_t L_Z = 0 \qquad (8)$$

hence :

$$P_w \left(\frac{D_1}{2} + \delta \right) - R_B \left[\mu_1 \left(\frac{D_1 + D + 2\delta}{2} \right) - L \right] - R\mu L_Z = 0 \qquad (9)$$

The reaction force in the point B, which decides on the rate of wear of the upper part of the valve guide, is then the following :

$$R_B = \frac{P_w(D_1 + 2\delta) - 2R L_Z \mu}{\mu_1(D_1 + D + 2\delta + 2L)} \qquad (10)$$

Making use of the relation (7) one finally obtains the following :

$$R_B = \frac{(P_g + P_b + P_s)(D_1 + 2\delta) - R(D_1 + 2\delta + 2L_Z\mu)}{\mu_1(D_1 + D + 2\delta + 2L)} \qquad (11)$$

When the relation (3) is taken into consideration one can determine the value of friction force in the upper part of the valve guide :

$$P_{t1} = \frac{(P_g + P_b + P_s)(D_1 + 2\delta) - R(D_1 + 2\delta + 2L_z\mu)}{D_1 + D + 2\delta + 2L} \quad (12)$$

Making use of the relation (5) one can also determine the reaction force in the point C, which decides on the rate of wear of the valve seat face :

$$R_{Cx} = R_B - R\mu \quad (13)$$

Considering the system of the forces acting onto the valve at the instant of separation of the valve head from the valve seat, Fig.3, one can observe that the spindle presses the lower part of the valve guide in the point D (the support point C disappears).
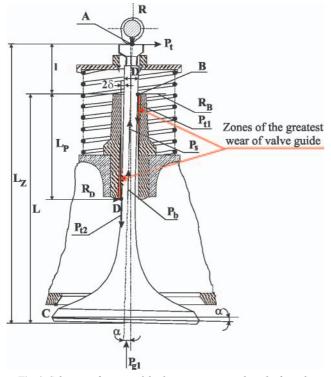

Fig. 3. Schematic diagram of the forces acting onto the cylinder valve at the instant of its full separation from the valve seat .

Under influence of the force R in the points B and D, apart from the transverse forces R_B and R_D, the respective friction forces P_{t1}, P_{t2} are induced, which become the direct cause of the valve guide wearing in its upper and lower part :

$$P_{t1} = R_B\mu_1 \quad (14)$$
$$P_{t2} = R_D\mu_1 \quad (15)$$

where :

μ_1 – sliding friction coefficient in the points B and D.

In such situation the reaction forces R_B and R_D, resulting from the friction force P_t, generate the respective bending moments M_B and M_D in the points B and D. Making use of the conditions of equilibrium of forces and moments one obtains the following :

$$\sum P_{ix} = 0 \Rightarrow R_D - R_B + P_t = 0 \quad (16)$$

hence :

$$R_D = R_B - R\mu \quad (17)$$

$$\sum P_{iy} = 0 \Rightarrow -P_{t1} + P_{g1} + P_b + P_s - P_{t2} - R = 0 \quad (18)$$

By assuming that at the given instant the sum of the forces acting onto the system is constant the following is obtained :

$$P_{w1} = P_{g1} + P_b + P_s - R \quad (19)$$

On insertion of (19) to the relation (18) the following is yielded :

$$P_{w1} - P_{t1} - P_{t2} = 0 \quad (20)$$

When the relations (14), (15) and (17) are taken into account the reaction force in the point B, which decides on the rate of wear of the upper part of the valve guide :

$$R_B = \frac{P_{w1} + R\mu_1\mu}{2\mu_1} \quad (21)$$

as well as the reaction force in the point D, which determines the rate of wear of the lower part of the valve guide, can be determined :

$$R_D = \frac{P_{w1} - R\mu_1\mu}{2\mu_1} \quad (22)$$

In Fig. 3 the zones of the greatest rate of wear of the valve guide are marked red.

Summing up the above performed considerations one can enumerate the design factors determining the reaction forces R_C, R_B and R_D, which detrimentally influence the design structure of valve spindle guides :

➢ the height of spindle above valve guide, l, which should be as small as possible
➢ the valve guide length L_p as well as the distance from the upper edge of the guide to the seat face of valve head, L, which should be as big as possible
➢ value of the friction force P_t generating side pressure forces of the spindle moving along the guide, which should be kept as small as possible.

The last design factor decreasing wear of cylinder valves is controlled by changing the character of valve rocker friction against the valve spindle face – from sliding friction to rolling one. As shown in Fig.4 it is possible by mounting the rollers at the rocker's end cooperating with the valve spindle.
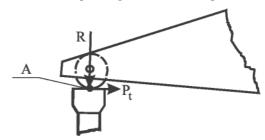

Fig. 4. Schematic diagram of the forces acting in the " valve rocker with roller – face of valve spindle" system .

Introducing simplifications, one can assume that the roller which substitutes the cam, generates only the pressure force R applied to the valve, in absence of any friction force P_t (its value is about ten times smaller than that in the case of sliding friction) [5]. Then no side components responsible for wearing the guide will be produced. The system will be in the state of equilibrium described by the relation (1). However because of the drop of elasticity (relaxation) of the valve spring, its service wear, wear of spindle face and valve head, certain transverse reaction leading to guide wearing will be always produced and the system will reach the equilibrium shown in Fig.1, except that in the point A no friction force P_t will be present.

From the condition of equilibrium of forces and moments the reaction forces are determined as follows :

$$R_{Cx} = R_B \quad (23)$$

$$R_{Cy} + P_g + P_b + P_s - P_{t1} - R = 0 \quad (24)$$

The friction force in the point B, which determines valve guide wearing, is described as follows :

$$P_{t1} = R_B \mu_1 \quad (25)$$

where : μ_1 – sliding friction coefficient in the point B.

Assuming as before that at a given instant the sum of forces acting onto the system is constant (7) one obtains the expression which describes the sum of bending moments respective to the point C :

$$\sum M_{iC} = 0 \Rightarrow P_w \left(\frac{D_1}{2} + \delta \right) +$$
$$- P_{t1} \left(\frac{D_1 + D + 2\delta}{2} \right) + R_B L = 0 \quad (26)$$

hence :

$$P_w \left(\frac{D_1}{2} + \delta \right) - R_B \left[\mu_1 \left(\frac{D_1 + D + 2\delta}{2} \right) - L \right] = 0 \quad (27)$$

Hence the reaction force in the point B, which decides on the rate of wear of the upper part of valve guide, is as follows :

$$R_B = \frac{P_w(D_1 + 2\delta)}{\mu_1(D_1 + D + 2\delta) - 2L} = R_{Cx} \quad (28)$$

It is quantitatively equal to the reaction force in the point C, which decides upon the rate of wear of valve seat face.

The expression which describes the sum of bending moments respective to the point B, obtains the following form :

$$\sum M_{iB} = 0 \Rightarrow -R_{Cy} \left(\frac{D_1 + D + 2\delta}{2} \right) +$$
$$+ R_{Cx} L - P_w \left(\frac{D + 2\delta}{2} \right) = 0 \quad (29)$$

Hence the vertical component of the reaction force in the point C is expressed by the following formula :

$$R_{Cy} = \frac{2R_{Cx} L - P_w(D + 2\delta)}{D_1 + D + 2\delta} \quad (30)$$

Applying the analogical considerations to the system shown in Fig. 3 where the roller is used instead the cam to neglect the friction force P_t in the point A, one can determine the equilibrium equations for the following forces :

$$\sum P_{ix} = 0 \Rightarrow R_D = R_B \quad (31)$$

$$\sum P_{iy} = 0 \Rightarrow -P_{t1} + P_{g1} + P_b + P_s - P_{t2} - R = 0 \quad (32)$$

Assuming that at a given instant the sum of forces acting onto the system is constant one obtains the following :

$$P_{w1} = P_{g1} + P_b + P_s - R \quad (33)$$

Inserting it into (32) one obtains :

$$P_{w1} - P_{t1} - P_{t2} = 0 \quad (34)$$

After taking into account the conditions :

$$P_{t1} = \mu_1 R_B \quad \text{and} \quad P_{t2} = \mu_1 R_D \Rightarrow P_{t1} = P_{t2} \quad (35)$$

one can determine the reaction forces in the points B and D, which decide upon the rate of wear of the upper and lower part of valve guide :

$$R_B = \frac{P_{t1}}{\mu_1} = \frac{P_{t2}}{\mu_1} = \frac{P_{w1}}{2\mu_1} = \frac{P_{g1} + P_b + P_s - R}{2\mu_1} = R_D \quad (36)$$

From comparison of the expressions (21), (22) and (36) which describe the reaction forces in the support points B and D, for the system fitted with the distribution shaft cam and that with the valve arm roller, respectively, it results that the use of the roller makes the rates of wear of upper and lower part of valve guide equal. And, the greater the radius of the roller the smaller the rate of wear. At the initial instant of valve opening the rate of wear of valve seat face will be also lower due to a significant decrease of the reaction force in the point C.

EXAMINATIONS OF CYLINDER VALVES OF SHIP DIESEL ENGINES DURING SERVICE

Novel methods of diagnostic tests are commonly introduced into operation process of ship diesel engines. The dynamically developing endoscopy which has been earlier applied only in medical examinations, now serves as a very useful, even indispensable tool for assessing the technical state of complex ship engines.

Endoscopy is a disassembling-free method for visual - optical inspection of interior of machines and devices by means of specular instruments (endoscopes).

For endoscopic examinations of the engines installed on Polish Navy ships the following instruments are used : IF8D4−15 fiberscope and a kit of borescopes of OLYMPUS and STORZ firms, differing to each other by the length of optical system, its diameter and angle of observation of a diagnosed element, namely : 90cm/8mm/90°, 55cm/8mm/90°, 45cm/8mm/90°, 50cm/6mm/90°, 30cm/4mm/0°, 30cm/10mm/120° – see Fig.5. The instruments make it possible to examine and prepare photographic documentation of engine's internal elements, through inspection openings having their diameter greater than 5 mm. A special digital photo-camera, Camedia C−2500L made by OLYMPUS firm, is used to perform dimensional analysis of detected failures, visualize them and record in a data base.
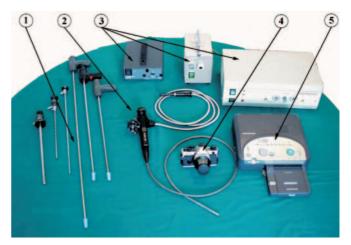

Fig. 5. The endoscopic diagnostic system of OLYMPUS firm : 1 − kit of borescopes, 2 − fiberscope, 3 − kit of light sources, 4 − digital photo-camera, 5 − photo-printer .

The camera is connected with a borescope or fiberscope by means of special links (adapters).

The length of the fiberscope's elastic light pipe whose controllable end makes observation in an arbitrary direction possible, is equal to 1500 mm. It has replaceable ends making observation within front and side sectors of different observation angles possible. Owing to this, to a great extent are increased manual possibilities of inspection of interior of air and exhaust gas flow passages of engine and turbo-compressor system.

Borescopes of different lengths and rigid optical system make it possible to carry out observations within front and side sectors in a wide range of variability of observation angle. The 30cm/10mm/120^{0}optical system is especially useful in diagnosing the engine combustion chambers, the valve seats fixed in lower plate of engine head in particular. Borescopes are also very useful during inspection of guide vanes and moving blades of turbo-blower.

In Fig.6 is presented a way of conducting the endoscopic examinations of ship engine cylinder systems by using a borescope and fiberscope. And, in Fig. 6 is presented a way of getting access to interior of the cylinder liner of the ship diesel engines : M401A-1(2) and 16V149TI Detroit Diesel, for endoscopic examinations.

a)

b)

Fig. 6. *Ways of inserting the end of borescope and fiberscope into cylinder interior of : a) M401A-1 diesel engine – through the holes remaining from dismounted injectors, b) 16V149TI Detroit Diesel engine – through inlet air windows in cylinder liner .*

When the injector is dismounted the borescope (fiberscope) makes it possible to assess technical state of piston head, cylinder liner surface, cylinder head and other units fixed in it such as spray nozzles of remaining injectors, inlet and outlet valves, starting valves etc (Fig.7). The endoscopic method of inspection is especially useful in diagnosing multi-block and multi-cylinder engines. For instance, in the case of radial engines, e.g. the M503 or M520 engine already installed in the engine room, an access to its lower mono-blocks and lower parts of reduction – reversing gear is very difficult. In order to perform inspection of the engines together with their gears they must be uncoupled from propeller shaft, next inclined and lifted and sometimes even rotated inside the engine room so as to make the first or seventh cylinder block accessible. From operational practice it results that a fiberscope of a sufficiently long light pipe makes it possible to avoid those inconveniences and thus to save execution time of overhauls and associated costs even by 25-30% [7].
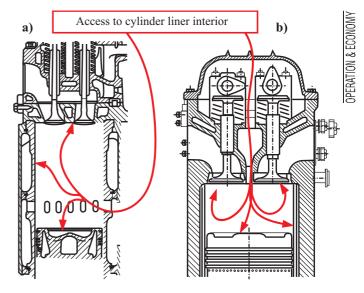

Fig. 7. *Endoscopic examination of ship engines – access to cylinder liner interior: a) 16V149TI Detroit Diesel engine, b) M401A-1(2) diesel engine .*

FAILURES OF CYLINDER VALVES OF SHIP ENGINES

Systematic endoscopic examinations of ship engines installed on Polish Navy ships are carried out in the following situations :
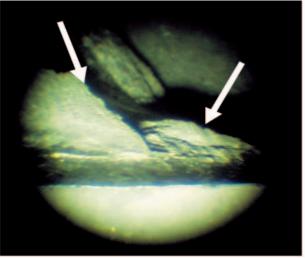
❖ during preventive surveys (once a year at least)
❖ at current assessment of technical state of an engine when to prolong the time between repairs is necessary
❖ in the case of an increased vibration level, found metal particles in lubricating oil, occurrence of sudden deviations from the trend line of mean indicated pressure (indicated power) in a cylinder, increased exhaust gas temperature, excessive smoke emission etc.
❖ when to dismount the engine head is difficult and time--consuming.

On the basis of multi-year endoscopic examinations of ship engines a procedure of their technical state assessment in service conditions was elaborated. It contains the necessary scope and scheduling principles of examinations of engine interior, aimed at detection of possible defects of particular elements of functional systems of the engine. For every type of the engines used on Polish Navy ships detail instructions for realization of diagnostic examinations by means of the fiberscope and kit of borescopes, were elaborated. The identified failures are photographically recorded to make documentation of the detected defects and to determine a tendency of their development. Results of the examinations are saved in a computer data base [1,2,3,7].
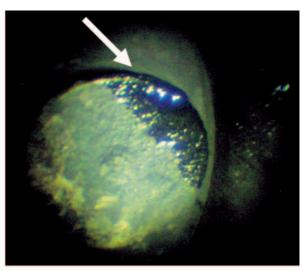
High effectiveness of the method in question, at a relatively simple use of the diagnosing instruments, has been confirmed during almost 15 years of experience in endoscopic examining the ship engines, gained by specialists from Polish Naval University. As a result of the performed examinations were identified many material defects which could be transformed into serious hazards to engine reliability in the case of their further development.

The selected failures of cylinder valves of ship diesel engines, identified during endoscopic inspections in service, are presented in Fig.8 and 9 [1,2,3 and 7].
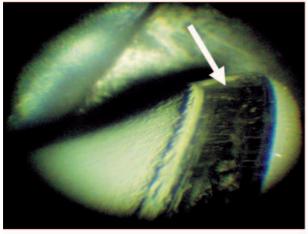
The heavy oiling-up of inlet and outlet valves goes to show that the valve spindle guides are excessively worn. In order not to allow for premature wearing-out the guides and for bending the valve spindles, that always leads to loss of tightness of
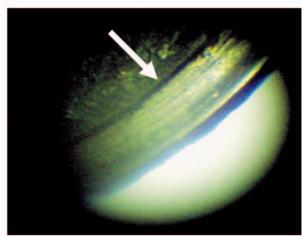
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Access to cylinder liner interior

a)   b)

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a) Inlet valve head in the cylinder head of 5BAH22 engine – deposit layer on valve head surface

b) Outlet valve of 16V149TI Detroit Diesel engine – – lubricating oil and tar compounds deposited on valve head
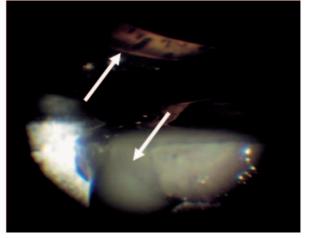

c) Valve head of opened inlet valve in the cylinder head seat of 75H12 WOLA engine – smooth valve seat face, without any traces of wear (reference state)

d) Valve head of opened outlet valve in the cylinder head seat of 75H12 WOLA engine taken out of service – characteristic wear sill on valve seat face

e) Outlet valve of M503A engine – loss of valve tightness due to bent valve spindle

f) Inlet valve head in the face of cylinder head of M401A-2 engine – oil layer (visible oil drop) on valve head surface, smooth valve seat face, without any traces of wear and corrosion

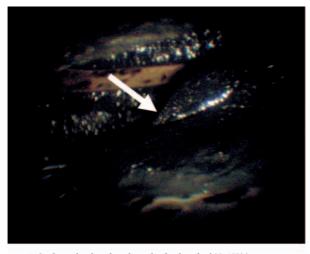**Fig. 8.** *Defects of ship diesel engines, identified during endoscopic examinations .*

cylinders, two values of radial clearance between the guide and spindle are provided :

☆ optimum one, i.e. that for assembling

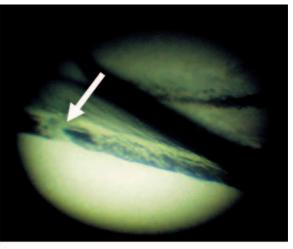☆ ultimate one, i.e. that qualifying a given valve for replacement or regeneration.

It is recommended to keep the assembling values of radial clearance of inlet valves, related to the valve spindle diameter D, within the range of (0.004 ÷ 0.006) D and their ultimate values – within the range of (0.01 ÷ 0.015) D, which means that the radial clearance value may increase even 2 ÷ 3 times during engine service. In the case of outlet valves the respective

values of radial clearance amount to : $(0.006 \div 0.01)D$ – for assembling, and $(0.015 \div 0.025)\ D$ – ultimate ones [6].
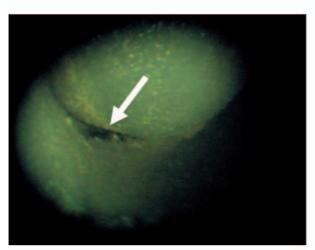
The ultimate value of radial clearance results from the permissible bend (curvature) of valve spindle moving within valve guide. A measure of the bend can be the angle α which determines coaxiality of the spindle and guide, or the gap F which appears on the valve seat face. (Fig. 1 and 3). As results from the system of forces acting onto cylinder valve, shown in

a) Outlet valve head in the cylinder head of 5BAH22 engine – – deposit layer on valve head surface
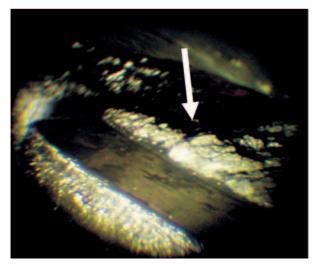

b) Inlet valve head in the cylinder head of 5BAH22 engine – smooth valve head surfaces, without any traces of contamination, visible surface roughness on valve seat face, resulting in exhaust gas blow-by
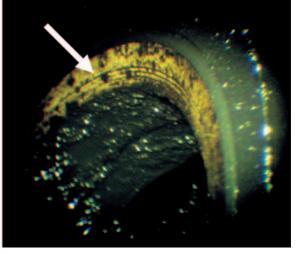

c) Outlet valve head in the cylinder head of Wartsila engine – – carbon deposit layer close to valve seat face

d) Outlet valve seat of Wartsila engine – carbon deposit layer close to valve seat face – exhaust gas blow-by through non-tight valve

e) Outlet valve head of M401A-1 engine – oily deposit layer on valve head surface

f) Outlet valve head of M401A-1 engine– carbon deposit and tar compounds on valve seat duct surface, valve seat face with some traces of carbon deposit

Fig. 9. *Defects of ship diesel engines, identified during endoscopic examinations .*

Fig. 3, the angle of valve spindle bend is equal to the angle of skew of valve head in compliance with the expression:

$$\alpha = \frac{2\delta}{L_P} = \frac{F}{D_1} \qquad (37)$$

The quantities are determined, at the same values of radial clearance, by the following design parameters :

⇨ guide length
⇨ valve spindle length
⇨ valve head diameter
⇨ mutual location of the spindle and guide

From the expression (37) it results that the dimensions of the gap of valve seat face, F, increase when the radial clearance between guide and spindle, δ, increases, the greater the valve head diameter D_1, and the smaller the guide length L_p :

$$F = \frac{2\delta D_1}{L_P} \qquad (38)$$

From the relation (38) it also results that in the case of the predetermined permissible value of the valve seat face gap, F, the greater the guide length L_p and the smaller the valve head diameter D_1, the greater the ultimate value of the radial clearance δ :

$$\delta_{gr} = \frac{FL_P}{2D_1} \qquad (39)$$

Additionally, worth mentioning that the ultimate value of the radial clearance does not depend on the valve spindle diameter D.

For the considered ship diesel engines the radial clearance values of inlet and outlet valves are presented in Table [6].

Table. *Assembling and ultimate values of the radial clearance between the spindle and guide of inlet and outlet valves of the engines : M401A-1(2) and 16V149TI Detroit Diesel .*

Type of engine	Inlet valves			Outlet valves		
	Valve spindle diameter [mm]	Assembling clearance value [mm]	Ultimate clearance value [mm]	Valve spindle diameter [mm]	Assembling clearance value [mm]	Ultimate clearance value [mm]
M401A-1(2)	18.00	0.094	0.144	17.93	0.099	0.149
16V149TI Detroit Diesel	Gap-valve distribution system			9.44	0.026	0.055

CONCLUSIONS

❍ Leakage of lubricating oil from valve guide is a symptom of dropping compression pressure in engine cylinder as a result of loss of tightness of combustion chamber.

❍ Rate of wear of cylinder valve guide depends on pressure force exerted by valve spindle onto valve guide wall, and on their mutual sliding.

❍ And, velocity of the sliding is directly proportional to the mean piston velocity and crankshaft rotational speed.

NOMENCLATURE

D – diameter of valve spindle
D_1 – diameter of valve head
F – height of valve face gap
l, L – height, length
M – moment of force
P, R – force
p – pressure

α – angle
δ – radial clearance between valve spindle and guide
μ – sliding friction coefficient in the point A
$μ_1$ – sliding friction coefficient in the points B_1 and D
τ – time

Indices and abbreviations

b – of inertia
g – of gases
gr – an ultimate quantity
p – of guide
s – of spring tension
t – of friction
w – a resultant quantity
x,y – a component quantity
z – of valve
A, B, C, D – points of contact (support) of valve system elements
TPC – the,, piston – piston rings- cylinder" system
ZS – self ignition

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CONTACT WITH THE AUTHOR
Assoc. Prof. Zbigniew Korczewski
Mechanic-Electric Faculty,
Polish Naval University
Śmidowicza 69
81-103 Gdynia POLAND
e-mail: zkorczewski@wp.pl