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Analiza učinkovitosti kombiniranega hlajenja valjev sodobnih dizelskih motorjev

Analysis of Combined Air-Oil Cooling Effectiveness of Diesel Engine Cylinders

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Toplotne obremenitve valja — predvsem z zrakom hlajenih dizelskih motorjev omejujejo specifično moč motorjev, obenem pa so po valju razporejene zelo neenakomerno. Da bi ta učinek vsaj omilili, je bil v preteklosti razvit učinkovit način dodatnega hlajenja valja z oljem za mazanje motorja.

V prispevku je prikazana analiza primerjav rezultatov izračunov in meritev temperatur v steni kombiniranega, zračno-oljno hlajenega valja za različne konstrukcijske izvedbe kanala. Podana sta tudi potek in velikost gostote toplotnih tokov skozi zanimivejše površine valja in ocenjeni pogreški meritev posameznih parametrov, ki določajo toplotne tokove.

Thermal loads in the cylinder wall, especially in air-cooled internal combustion engines, are often limiting parameters, in relation to engine output increase. Their size and distribution can be levelled by an additional engine lubrication oil jet, flowing through a horizontal channel in the upper part of the cylinder wall.

Analysis of measured and computed results was performed for various design solutions of the cooling channel. Temperature, as well as heat-flux distribution in the cylinder wall is presented in the paper, together with analysis of measuring errors that influence the heat flux calculation.

0 UVOD

Neenakomeren potek temperature in temperaturnih gradientov ter njihova velikost, zlasti v zgornjem delu z zrakom hlajenih valjev nemalokrat omejujejo možnosti nadaljnjega varnega povečanja specifične moči motorjev. Z dodatnim hlajenjem zgornjih površin valja lahko raven temperatur in temperaturnih gradientov znižamo, neizenačenost temperaturnih potekov pa zadovoljivo uravnotežimo. V bližnji preteklosti je bila objavljena vrsta prispevkov [1] do [11], ki so obravnavali značilnosti in karakteristike kombiniranega hlajenja valjev in njihov vpliv na delovne ter ekološke karakteristike motorja. Rezultati dosedanjih raziskav so pokazali, da je kombinirano hlajenje učinkovito. Učinkovitost hlajenja je odvisna tudi od načina — konstrukcije kanala, zato je bilo preizkušenih in analiziranih doslej več izvedb dodatnega oljnega hlajenja [1], [2], [3], [9]. Za optimalno rešitev problema toplotnih obremenitev valja so potrebne tudi numerične analize procesa prenosa toplote skozi valj; numerične modele je treba opremiti z ustreznimi robnimi pogoji [2], [5], [6], [7], [8], ki jih navadno določimo s preizkusi. Pri tem mislimo predvsem na temperature, snovne in toplotne tokove, ki obremenjujejo posamezne površine valjev. Da bi bila uporabnost rezultatov čim večja, morata biti tudi točnost in zanesljivost eksperimentalno dobljenih podatkov ustrezni. Analiza pogreškov pri meritvah različnih parametrov, ki določajo toplotne tokove, je zato neogibna.

0 INTRODUCTION

Non-uniform temperature distribution, large temperature gradients and their peak values are often limiting parameters in relation to further and safe increase of the engine specific output. By additional and very intensive cooling of the upper part of the air-cooled cylinder, peak temperatures and gradients can be satisfactorily reduced and levelled. Some papers have been published in the recent past [1] to [11] describing the characteristics and features of combined cooling and its impact on engine performance and ecological parameters. Results have proved combined cooling to be very efficient. This efficiency generally depends on the cooling channel design; different versions have so far been analyzed numerically and experimentally [1], [2], [3], [9]. Numerical computations of the heat transfer are substantially supported by experimentally defined boundary conditions [2], [5], [6], [7], [8]; temperatures, heat fluxes, coolant mass-flows are used for this purpose. The accuracy of the measured results is of crucial importance for calculated heat transfer results. Analyses of measurement errors must therefore continuously be applied.

1 PRIMERJAVA UČINKOVITOSTI RAZLIČNIH KONSTRUKCIJSKIH IZVEDB DODATNEGA HLAJENJA VALJA Z OLJEM

V preteklosti je bilo ugotovljeno [2], [11], da je za optimalno hlajenje zgornjega dela valja zračno hlajenega motorja treba v steno valja vgraditi vodoravni kanal pravokotnega prečnega prereza, v katerem se vtekajoči curek olja razcepi v dva delna tokova, ki obtečeta valj in združena zapustita valj. Pri tem je pomembno, da je vstopno ustje kanala izvedeno tako, da sta delna masna tokova olja neenaka: večji tok teče po področju, ki je toplotno bolj obremenjeno in pri tem odvede več toplote, in manjši delni tok, ki obteka hladnejše površine valja, in prenaša s seboj manj toplote. Tudi mesto vtoka olja je zelo pomembno: curek olja vteka v kanal v področju najvišjih temperatur valja [1], [2], [3], [4], [9]. Oblika (razmerje dolžin stranic prereza kanala) vodoravnega kanala prav tako močno vpliva na količino prenesene toplote na olje; kanali kvadratnega prereza [2] so primernejši za intenziven prestop toplote v dolgih ceveh, kjer je tok popolnoma razvit. Visoki in ozki kanali zagotavljajo višje lokalne vrednosti toplotne prestopnosti že na samem vtočnem delu krajših cevi.

Tehnološki postopki izdelave valjev, razpoložljiv prostor za vgradnjo ter količina prenesene toplote z oljem določajo oblike in velikost kanala. Na sliki 1 so prikazane tri razvojne stopnje konstrukcijskih različic kanala za dodatno hlajenje valja z oljem. DOH 3 je prva izvedba, pri kateri je bil uveden dvotočni, nesimetrični način pretakanja olja [1], [2], [3],[9]. Zaradi sorazmerno velike širine kanala in mogočih kasnejših tehnoloških problemov izdelave kanala v steni valja (tanke stene ob kanalu pomenijo potencialno nevarnost prepuščanja olja zaradi poroznosti snovi).

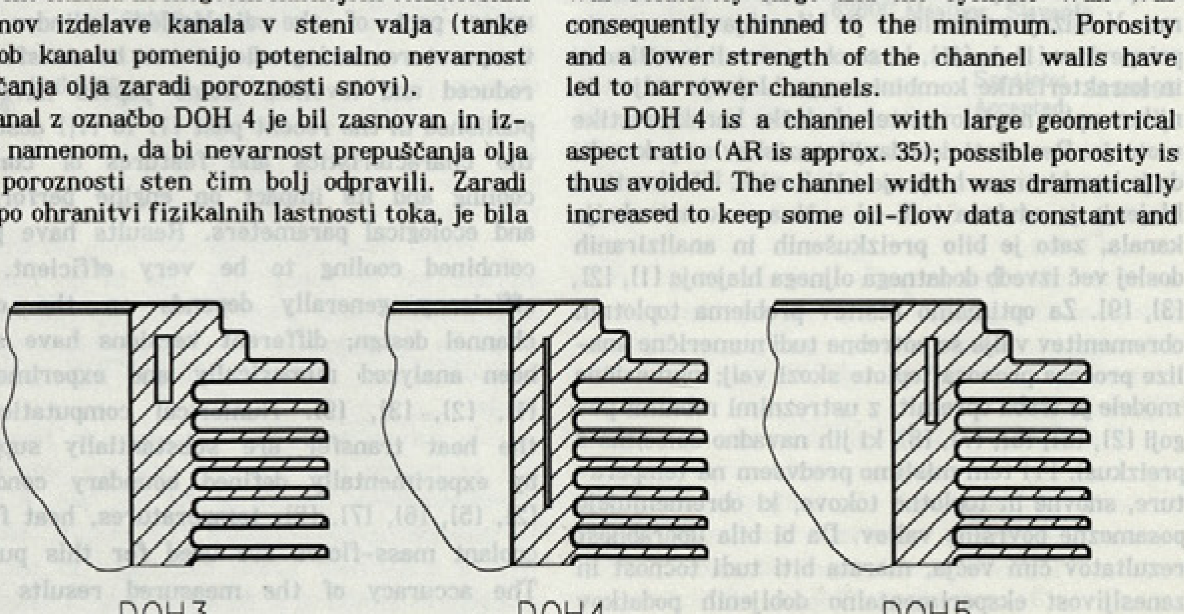
Kanal z označbo DOH 4 je bil zasnovan in izdelan z namenom, da bi nevarnost prepuščanja olja zaradi poroznosti sten čim bolj odpravili. Zaradi težnje po ohranitvi fizikalnih lastnosti toka, je bila

1 COMPARISON OF DIFFERENT OIL-CHANNEL DESIGNS

A horizontal, curved channel fed with engine lubrication oil was introduced in the upper part of the air-cooled cylinder wall to obtain optimum temperature distribution [2], [11]. The cooling oil jet is initially split into two asymmetrical branch flows which leave the channel through a common outlet. Partial oil flows are not identical; the larger portion flows through the region of pronounced local cylinder temperatures and takes away more heat. The smaller oil-flow cools down cooler part of the cylinder wall. The location of the oil inlet is also very important: the oil jet enters the channel where the maximum temperatures are expected [1], [2], [3], [4], [9]. The geometrical aspect ratio (height to width ratio) of the channel transverse section also greatly affects oil heat transfer: a squared channel is more suitable for longer channels with fully developed laminar flow, whereas channels with bigger aspect ratios result in higher local heat transfer coefficients and thus better heat transfer in the vicinity of the oil inlet [2].

The manufacturing technology of an air-cooled cylinder and its wall thickness define the size and shape of the cooling channel. Figure 1 shows three of the latest design solutions of the channel and its position in the cylinder wall. DOH 3 represents the first channel design, where asymmetrical split-oil-flow was successfully applied [1], [2], [3], [9]. The width of the channel was relatively large and the cylinder wall was consequently thinned to the minimum. Porosity and a lower strength of the channel walls have led to narrower channels.

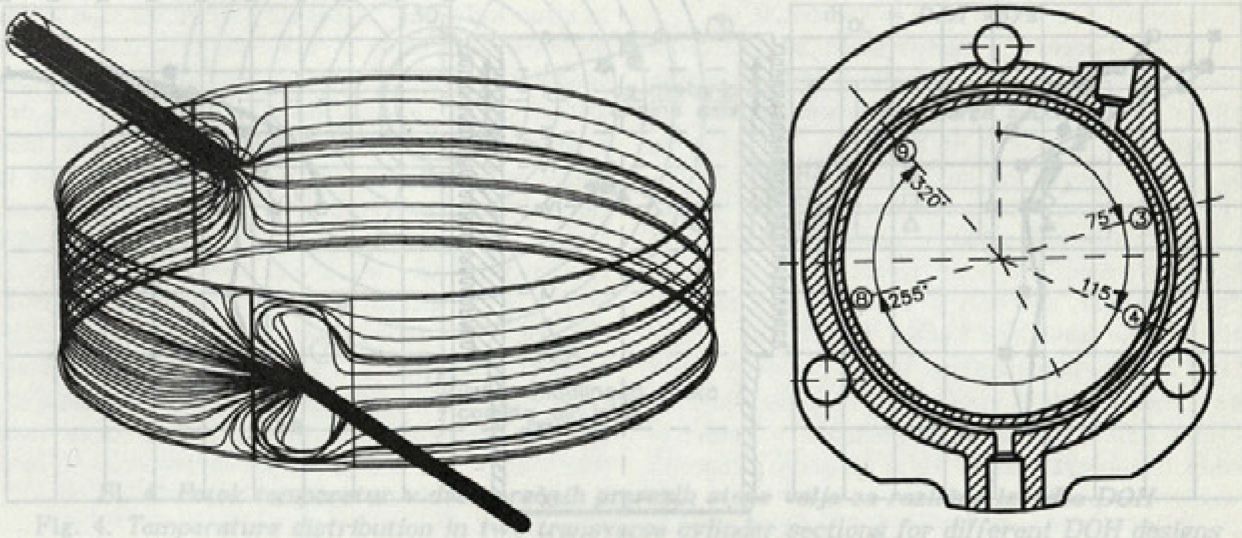
DOH 4 is a channel with large geometrical aspect ratio (AR is approx. 35); possible porosity is thus avoided. The channel width was dramatically increased to keep some oil-flow data constant and



Sl. 1. *Konstrukcijske izvedbe kanala za dodatno hlajenje*
Fig. 1. *Design solutions of the cooling channel*

višina kanala močno povečana (tako da sega v drugo četrtino višine valja [9].Izvedba DOH 5 je oblikovni kompromis med izvedbama 3 in 4; širina kanala je nekoliko večja, višina kanala pa pokriva področje lege prvih dveh batnih obročkov (kadar je bat v zgornji mrtvi legi).

almost 25% of the total cylinder length was therefore additionally oil-cooled [9]. DOH 5 is a compromise between DOH 3 and DOH 4; its channel is high enough to cover the critical cylinder wall between the first and the second piston rings (when the piston is in the top dead centre).



Sl. 2. *Izvedba nesimetričnega toka olja v kanalu in izračunan potek tokovnic*
Fig. 2. *Asymmetrical channel split oil-flow, calculated stream line distribution*

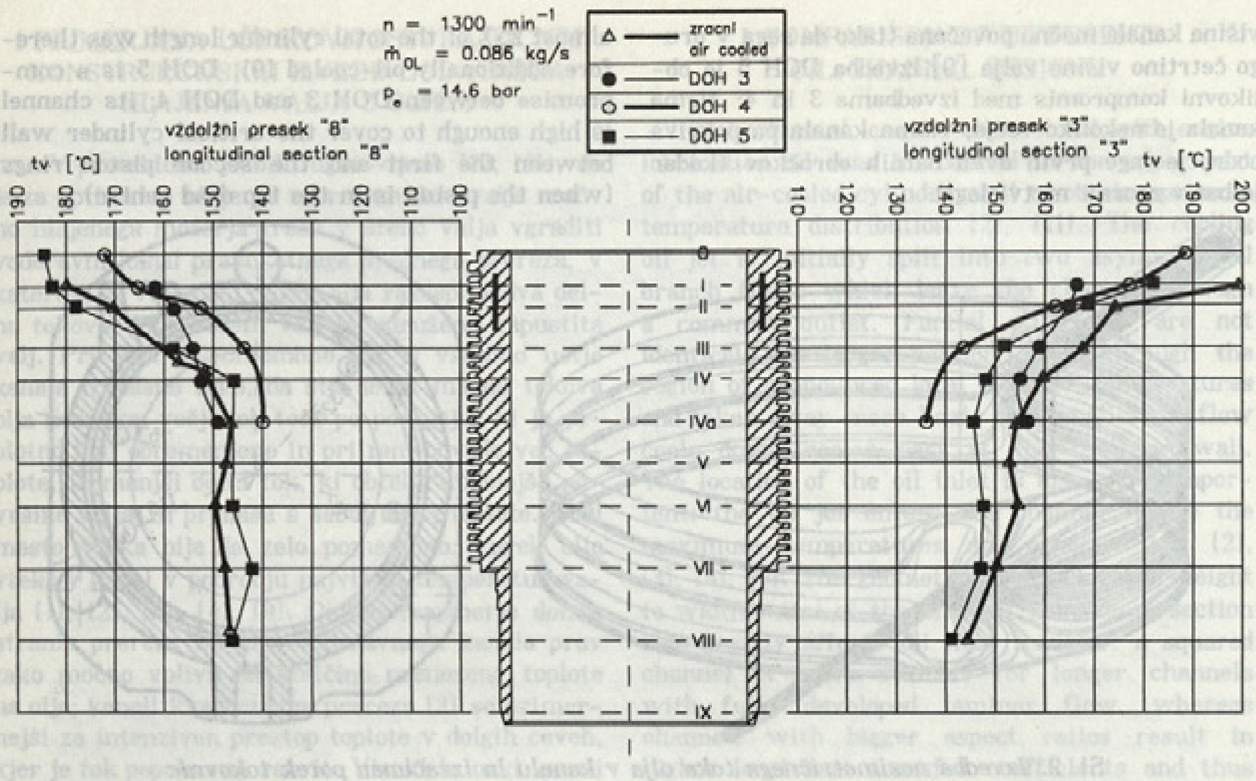
Prečni prerez kanala in dvodelni, nesimetrični tok olja je prikazan na zgornji polovici slike 2. Obenem so v isti skici prikazane lege vzdolžnih prerezov valja, v katerih so bile opravljene meritve temperatur, temperaturnih razlik in gostote toplotnih tokov (prerezi 3, 4, 8 in 9). Na isti sliki je prikazan tudi rezultat numeričnega modeliranja pretakanja olja skozi kanal DOH 4. Na sliki so prikazane tokovnice v kanalu, izračuni pa so opravljeni s programskim paketom TASCFLOW na računalniku CONVEX 3860 [5]. Na sliki 3 je podan potek temperatur v vzdolžnih prerezih valja: »3« — najtoplejšem (lege prerezov 3, 4, 8 in 9 so prikazane na sliki 2) in »8« - najhladnejšim, za konstrukcijske izvedbe kanalov DOH 3, DOH 4, DOH 5 in osnovno — zračno hlajeno izvedbo valja. Izbrana je največja obremenitev motorja, oziroma toplotna obremenitev valja pri vrtilni frekvenci največjega navora motorja. Za vse dodatno hlajene valje je bil uporabljen enak masni tok olja za hlajenje.

Izmerke — vzdolžni potek temperatur lahko strnemo v naslednje sklepe :

— Najučinkovitejši (najintenzivnejši) odvod toplote je bil dosežen s sistemom DOH 3, kar potrjuje teoretična izhodišča, ki smo jih omenili v uvodu. Ker je kanalček sorazmerno nizek, je njegov vpliv na spodnji del valja manjši; temperature so tam višje, vendar ne previsoke. V zgornjem delu valja so pri tem načinu hlajenja doseženi najmanjši povprečni vzdolžni temperaturni gradienti (0,3 °C/mm).

A transverse section of the cylinder, together with the split oil-flow outline, is shown in figure 2. In addition the locations of the cylinder longitudinal sections where temperatures, temperature differences and heat fluxes were measured are also shown in the same figure (sections 3, 4, 8 and 9). The results of numerical computations of the channel oil-flow by means of flow streamlines are also presented in figure 2. TASCFLOW programme together with a CONVEX 3860 computer were used for this purpose. Figure 3 presents cylinder wall temperature distribution in the coolest (section 8) and the hottest (section 3) longitudinal sections of the cylinder (according to Fig. 2) for DOH 3, DOH 4, DOH 5, and the baseline air-cooled cylinder. The results were obtained at peak torque engine speed and load. One can conclude from the results, that:

— The most effective heat transfer was achieved with the DOH 3 version; the lower channel only effectively cools down the upper part of the cylinder, whereas the lower cylinder part keeps higher but still favorable temperatures. Longitudinal temperature gradients have minimum values with the DOH 3 version (0.3 °C/mm average).



Sl. 3. *Vzdolžni potek temperatur za različne izvedbe dodatnega hlajenja*

Fig. 3. *Longitudinal wall temperature distribution for diverse DOH design solutions*

— Pri sistemu DOH 4 opazimo v zgornjem delu valja (kritični so prečni prerezi 0, I in II) močno znižanje temperature notranje stene valja, še posebej med prerezi II in III (povprečni temperaturni gradient znaša tam približno 1 °C/mm). Potek temperatur valja je po velikosti in obliki precej podoben ustreznemu poteku pri motorju, hlajenem z vodo [2]. Na nizke temperature v prerezih III, IV in IVa vpliva predvsem zelo visok kanalček, ki sega (geometrijsko) skorajda do prereza III.

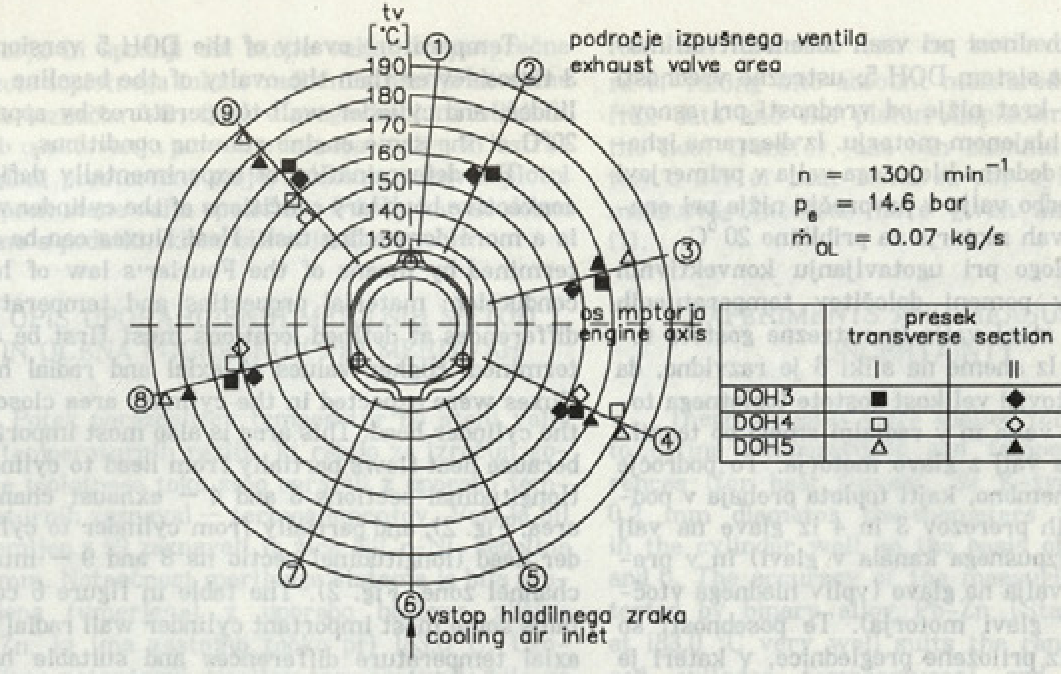
— DOH 5 je geometrijsko, kakor tudi glede prenosa toplote kompromis, med izvedbama DOH 3 in DOH 4. Temperature v prečnem prerezu I so nekoliko višje, temperature spodnjih dveh tretjin valja pa znašajo 140 do 150 °C. Povprečni vzdolžni gradient temperatur med prerezoma II in III znaša približno 0,8 °C/mm. Pogled na potek temperatur v »hladnem« vzdolžnem prerezu 8 kaže za DOH 5 višje vrednosti, ki so podobne tistim v toplem prerezu 3. Uravnoteženost temperatur po obodu valja je zato v primeru izvedbe DOH 5 zelo dobra.

Na sliki 4 je prikazan potek temperatur po obodu valja za vse tri omenjene izvedbe DOH in prečna prereza I in II. V tem pogledu vidimo, da je temperaturna »ovalnost« pri sistemu DOH 5 skoraj popolnoma izginila. Zaradi višjih temperatur ob notranji steni valja pri izvedbi DOH 5 bi bilo primerno povečati masni tok olja.

— DOH 4 is represented by a very intensive longitudinal wall temperature decrease — especially between transverse sections II and III, where temperature gradients exceed 1 °C/mm. Lower parts of the cylinder (sections III, IV, and IVa) have substantially lower temperatures as a result of the higher cooling channel. The shape of the temperature distribution is similar to the wall temperature distribution of a typical water cooled engine [2].

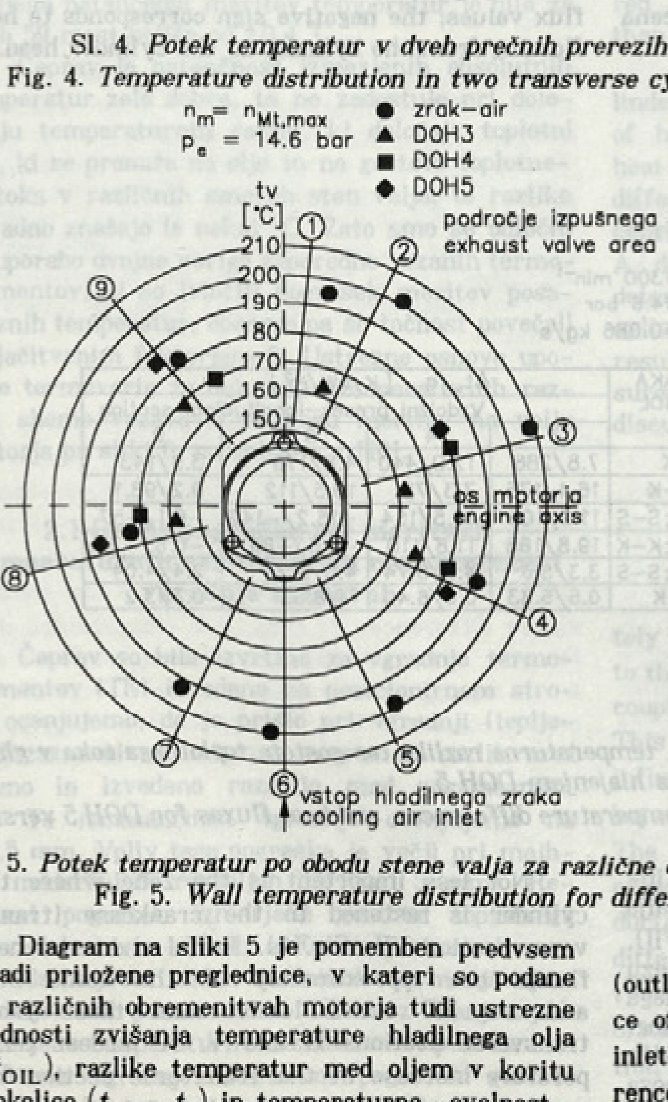
— DOH 5 is a compromise between DOH 3 and DOH 4. The average longitudinal temperature gradient amounts to 0.8 °C/mm. Temperatures in the coolest section 8 are higher compared to those measured with versions DOH 3 and DOH 4, but better leveled with the temperatures in the hottest section 3. The circumferential wall temperature ovalty was therefore decreased from 40°C (air- cooled cylinder) to less than 10°C (for DOH 5 version).

The circumferential cylinder wall temperature distribution for the three mentioned DOH designs and transverse cylinder sections I and II are shown in figure 4. The temperature ovalty of DOH 5 nearly disappeared. Lower temperatures of the cylinder liner wall and DOH 5 version can be obtained by slightly higher cooling oil-flows.



Sl. 4. *Potek temperatur v dveh prečnih prerezih stene valja za različne izvedbe DOH*

Fig. 4. *Temperature distribution in two transverse cylinder sections for different DOH designs*



Hlajenje Cooling	p_e [bar]	$\Delta t_{v,max}$ [°C]	Δt_{OIL} [°C]	$t_{MO}-t_0$ [°C]
ZRAK-AIR	10	28	–	–
DOH3	10	7	2.5	75.4
DOH4	10	8	1.4	72.9
DOH5	10	4.3	3.01	75.2
ZRAK-AIR	12	28	–	–
DOH3	12	9	3.3	75.9
DOH4	12	9	1.9	76.7
DOH5	12	7.2	5.22	79.0
ZRAK-AIR	14.6	34	–	77.4
DOH3	14.6	12	4.4	81.5
DOH4	14.6	13	3.0	77.1
DOH5	14.6	10.9	6.61	83.8

Sl. 5. *Potek temperatur po obodu stene valja za različne obremenitve motorja in različne izvedbe DOH*

Fig. 5. *Wall temperature distribution for different engine loads and DOH designs*

Diagram na sliki 5 je pomemben predvsem zaradi priložene preglednice, v kateri so podane pri različnih obremenitvah motorja tudi ustrezne vrednosti zvišanja temperature hladilnega olja (Δt_{OIL}), razlike temperatur med oljem v koritu in okolico ($t_{MO} - t_0$) in temperaturna »ovalnost«, tj. razlika med najvišjo in najnižjo temperaturo po obodu stene valja in prečnem prerezu I.

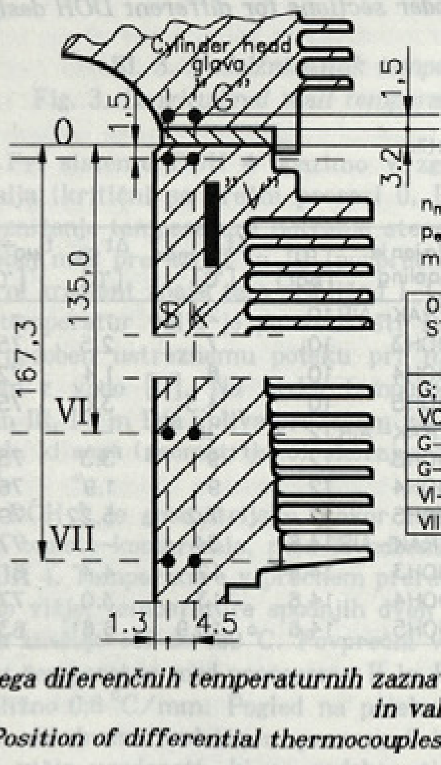
Temperature differences of the cooling oil (outlet to inlet; Δt_{OIL}), temperature difference of the engine lubrication oil (oil pan to air inlet; $t_{MO} - t_0$), temperature ovalty — difference of peak and minimum wall circumferential value ($\Delta t_{v,max}$) are presented in the table in figure 5.

Najnižjo ovalnost pri vseh obremenitvah motorja zagotavlja sistem DOH 5; ustrezne vrednosti so za skoraj 3-krat nižje od vrednosti pri osnovnem – zračno hlajenem motorju. Iz diagrama izhaja, da so stene dodatno hlajenega valja v primerjavi z osnovno izvedbo valja v povprečju nižje pri enakih obremenitvah motorja za približno 20 °C.

Težko nalogo pri ugotavljanju konvektivnih robnih pogojev pomeni določitev temperaturnih razlik v sami steni valja in ustrezne gostote toplotnega toka. Iz sheme na sliki 6 je razvidno, da smo želeli ugotoviti velikost gostote toplotnega toka v smeri osi valja in v radialni smeri ob tesnilki, ki povezuje valj z glavo motorja. To področje je izjemno pomembno, kajti toplota prehaja v področjih vzdolžnih prerezov 3 in 4 iz glave na valj (vpliv bližine izpušnega kanala v glavi) in v prerezih 8 in 9 iz valja na glavo (vpliv hladnega vtočnega kanala v glavi motorja). Te posebnosti so razvidne tudi iz priložene preglednice, v kateri je gostota toplotnega toka iz valja na glavo označena z negativnim predznakom.

Temperature ovalty of the DOH 5 version is 3 times lower than the ovalty of the baseline cylinder, and cylinder wall temperatures by approx. 20°C at the same engine running conditions.

The determination of experimentally defined convective boundary conditions of the cylinder wall is a more demanding task. Heat fluxes can be determined by means of the Fourier's law of heat conduction; material properties and temperature differences at defined locations must first be determined. Higher values of axial and radial heat fluxes were expected in the cylinder area close to the cylinder head. This area is also most important because heat flows partially from head to cylinder (longitudinal sections 3 and 4 – exhaust channel area, Fig. 2), and partially from cylinder to cylinder head (longitudinal sectio ns 8 and 9 – intake channel zone, Fig. 2). The table in figure 6 contains some most important cylinder wall radial and axial temperature differences and suitable heat flux values; the negative sign corresponds to heat flowing from the cylinder to the cylinder head.



n_m = 1300 min^{-1}
p_e = 14.6 bar
m_{OL} = 0.086 kg/s

OZNAKA SYMBOL	$\Delta t_v / q$ [K/kW/m²]			
	Vzdolžni presek–longitudinal section			
	3	4	8	9
G; S–K	7.8/286	12.0/440	4.8/176	3.9/143
VO; S–K	16.4/175	7.3/78	10.5/112	9.2/98.1
G–VO; S–S	11.2/105	16.5/154	−15.2/−142	−6.1/−57
G–VO; K–K	19.8/185	11.8/110	−9.5/−89	−0.8/−7.5
VI–VII; S–S	3.3/5.6	4.0/6.74	4.0/6.74	2.4/4.04
VII; S–K	0.5/5.33	0.6/6.40	– /8.53	0.3/3.2

Sl. 6. *Lega diferenčnih temperaturnih zaznaval, temperaturne razlike ter gostote toplotnega toka v glavi in valju s hlajenjem DOH 5*

Fig. 6. *Position of differential thermocouples, temperature differences, and heat fluxes for DOH 5 version*

Prav tako zanimiv, vendar malo manj intenziven prenos toplote poteka blizu naseda valja na okrov motorja (področje prečnega prereza VII). Rahlo zvišanje temperatur stene valja v prerezu VI pripisujemo prenosu toplote s prvega batnega obročka na valj (ko je bat v spodnji mrtvi legi SML); temperatura v prerezu VIII (na mestih 4, 8 in 9) se najverjetneje zviša zaradi prenosa toplote na valj prek plašča bata, ki v bližini SML spreminja smer naleganja in ima višjo temperaturo kakor okrov motorja, olje, plini v okrovu

Not less important is the zone where the cylinder is fastened to the crankcase (transverse section VII, Fig. 6). Radial and axial heat fluxes are approximately of the same size and are 100 times lower than those from transverse sections G and V. A modest temperature increase in the transverse section VI, as a result of better contact between the piston body and cylinder wall, as well as of a longer period (piston stagnation at the piston BDC) motorja in spodnji del stene valja. Iz povprečne gostote toplotnega toka v radialni smeri prek stene valja (označba VO; S-K v preglednici na sliki 6) in ob upoštevanju površine (plašča) valja, ki ustreza gibni prostornini valja, lahko ocenimo toplotni tok prek stene valja sprbližno 6 kW, kar se dobro ujema s podatki, ki so bili objavljeni v [1].

2 OPIS UPORABLJENIH MERILNIH METOD IN OCENA POGREŠKOV PRI MERITVAH

Potek porazdelitev temperatur po steni valja ter temperaturnih razlik, ki rabijo za izračun gostote toplotnega toka smo opravili z uporabo temperaturnih zaznaval — termoelementov. Valj je bil opremljen z 59 zaznavali iz žice NiCr-Ni, debeline 0,2 mm. Natančnost merilnega sistema je bila preverjena (umerjena) z uporabo binarne zlitine Pb-Zn, ki ima zastojno točko pri 183,0 °C. Ugotovljena natančnost meritev temperatur je bila za vseh 59 mest boljša od ±0,4 °C.

Čeprav je natančnost izmerjenih absolutnih temperatur zelo dobra, ta ne zadostuje pri določanju temperaturnih razlik, ki določajo toplotni tok, ki se prenaša na olje in na gostoto toplotnega toka v različnih smereh sten valja; te razlike navadno znašajo le nekaj °C. Zato smo se odločili za uporabo dvojne verige zaporedno vezanih termoelementov, ki so izločili pogrešek meritev posameznih temperatur, obenem pa so točnost povečali z ojačitvenim faktorjem 2. Ustrezne osnove uporabe termoverig za določanje temperaturnih razlik, sheme vezave v primeru meritev na valju motorja po sliki 6, so opisane v [10].

2.1 Ocena pogreška pri meritvah temperaturnih razlik zaradi nenatančnosti vgradnje zaznaval

Čeprav so bile izvrtine za vgradnjo termoelementov (TE) izvedene na pozicionirnem stroju, ocenjujemo, da je prišlo pri vgradnji (lepljenju) zaznaval na ustrezno mesto do razlik med želeno in izvedeno razdaljo med posameznimi TE. To nenatančnost vgradnje ocenjujemo na ± 0,5 mm. Vpliv tega pogreška je večji pri majhnih medsebojnih razdaljah TE (pod 5 mm). Relativni pogrešek izmerjene gostote toplotnega toka se veča (v skladu s Fourierjevim zakonom prevoda toplote skozi trdnino) z manjšanjem razdalje med posameznimi TE in z manjšanjem izmerjene temperaturne razlike med opazovanima mestoma vgradnje. Relativni pogrešek gostote toplotnega toka pri predpostavljenem pogrešku vgraditve N_L = 0,5 mm in temperaturni razliki (ΔT) smo izračunali po obrazcu:

for heat transfer can be noticed from figure 3. Taking into account measured average heat flux data and the piston displacement area for the heat transfer, one can calculate approximately 6 kW of heat removed per cylinder. Direct measuring methods have given similar results [1].

2 EXPERIMENTS AND MEASUREMENT UNCERTAINTY

Thermocouples were applied for experiments to define temperatures and temperature differences (for heat fluxes). 59 K-type (NiCr-Ni) 0.2 mm diameter thermometers were located in the cylinder wall on the basis of figures 2, 3 and 6. The accuracy of the measuring chain was tested by binary alloy Pb-Zn (Stagnation point at 183,0 °C very well suits the range of measured cylinder temperatures) and was better than ± 0.4 °C.

Although the accuracy of the measured cylinder temperatures is very good, determination of heat fluxes (through the cylinder wall or heat flow to the cooling oil) at low temperature differences (they are normally as low as a few centigrade Celsius) requires still better accuracy. A double thermocouple chain was applied to determine temperature differences at different cylinder wall locations (the accuracy of the results was thus doubled). Details of the measurements, according to figure 6, have been discussed in [10].

2.1 Measurement error due to uncertain location of the thermocouples

Thermocouple bores were positioned accurately on a drilling-machine. Mounting error refers to the small possible displacement of the thermocouples in their bore during the fastening period. This error was estimated to be ⁺ 0.5 mm. The influence of this error is bigger at low distances between particular thermocouples (under 5 mm). The relative error of the measured heat flux increases (according to Fourier' s law of heat conductance) with smaller measured temperature differences and with smaller distances between two adjacent measuring points. The relative error of the measured heat flux due to uncertain location of the thermocouple (estimating the displacement of two adjacent thermocouples to be N_L = = 0.5 mm) and temperature difference (ΔT) can be calculated by using the following formula:

$$n_R = \frac{N_L}{\Delta T} 100\ [\%] \tag{1}$$

Največji relativni pogrešek ocene gostote toplotnega toka napravimo (v skladu z označbami na sliki 6) v prerezu VII: S-K, kjer so gostota toplotnega toka in temperaturne razlike (na majhni medsebojni razdalji) majhne. Tam velja, da je izračunana gostota toplotnega toka v velikostnem razredu relativnega pogreška (ta pa znaša okrog 40 %) in moramo rezultate zato upoštevati z zadržkom. V prerezih zgornjega dela valja, kjer so temperaturne razlike velike in medsebojne razdalje večje, so relativni pogreški manjši in znašajo le nekaj odstotkov.

The biggest error of the above type was therefore introduced in the calculation of the heat fluxes in transverse section VII: S-K (according to Fig.6) where small temperature differences at small distances were measured. The size of the calculated relative error (approx. 40 %) corresponds to the size of the measured and calculated heat fluxes, so very careful handling with measured data is therefore recommended. The situation is more favourable and clearer in the upper part of the cylinder. Bigger distances and temperature differences result in a relative error that can amount to a few percent.

2.2 Ocena pogreška toplotnega toka s stene valja na hladilno olje

Pri določanju toplotnega toka, ki se prenese s stene valja na olje, je ocena pogreška meritve nekoliko drugačna. Zopet je uporabljena veriga TE, vendar so TE potopljeni v vstopni in izstopni curek olja. Ob upoštevanju ocene pogreška meritev masnega toka olja (tehtanje olja) ±3 % in upoštevanju napake pri linearizaciji temperaturne razlike (uporaba ustreznih preglednic) v znesku 0,25 °C, znaša skupni relativni pogrešek meritve toplotnega toka na olje — v skladu z zakonom o razširjanju pogreškov:

2.2. Measurement error of the heat flux to the cooling oil

The measurement error of the heat flux transferred from the channel wall to the cooling oil is based on the measurement accuracy of the oil temperature difference at the oil outlet and inlet, as well as on the measurement error of the oil mass-flow. The oil mass-flow error was estimated to be ± 3% (weighting of the time-average oil-flow) and the error connected to the linearization of the measured voltage (temperature difference voltage) to be 0.25 °C. The total relative error of the oil heat flow measurements can be expressed (taking into account the uncertainty of the two independent variables) with the following formula:

$$n_R = \left(m_{OLJA}\, n_{ABS}\, \Delta t + \Delta t\, n_{ABS}\, m_{OLJA}\right) \Big/ \left(m_{OLJA}\, \Delta t\right)\ [\%] \tag{2}$$

Pri tem pomenijo: m_{OLJA} — masni tok olja za hlajenje, $n_{ABS}\Delta t$ — absolutni pogrešek linearizacije temperaturne razlike, Δt — izmerjeno temperaturno razliko med dvema soležnima TE ter $n_{ABS}\, m_{OLJA}$ — absolutni pogrešek meritve masnega toka olja. Za srednje vrednosti toplotnih obremenitev motorja in izbrani masni tok olja 0,1 kg/s znaša relativni pogrešek meritve toplotnega toka olja približno 5—10 odstotkov.

Where: m_{OLJA} denotes measured cooling oil mass flow, $n_{ABS}\Delta t$ absolute error of the temperature difference, Δt measured temperature difference of two adjacent thermocouples, and $n_{ABS}\, m_{OLJA}$ absolute measurement error of the cooling mass oil-flow. For an oil-flow of 0.1 kg/s, the relative error of the measured oil heat flow amounts to 5—10 %.

3 SKLEP

V delu je opisana primerjava karakteristik različnih konstrukcijskih izvedb dodatnega oljnega hlajenja valja. Rezultati analize kažejo, da je višji in ožji kanal v skladu z omenjenimi osnovami prenosa toplote najučinkovitejši. Temperature notranjih sten valja so pri izvedbi kanala DOH 3 celo nižje kakor pri drugih dveh prikazanih izvedbah DOH 4 in DOH 5, ki imata ožji in višji kanal.

3 CONCLUSION

The paper presents comparisons of the characteristics for three different design solutions of additional oil-cooling for an air-cooled cylinder. Short and narrow channels provide, the most effective heat transfer to the cooling oil according to the general discussion in the second section. DOH 4 is therefore the best technical solution to reduce peak temperatures in the upper part of a

Vzrok lahko najdemo v nekoliko ugodnejših obratovalnih razmerah motorja pri izvedbi DOH 3, pri kateri sta bili temperaturi zraka za zgorevanje in hlajenje nižji. Poleg tega pa v spodnjem delu valja ne želimo vzdrževati prenizkih temperatur (zaradi korozije drsnih - notranjih sten valja), kar je značilno za izvedbo DOH 4. DOH 5 pomeni v celoti kompromisno rešitev. Tudi najboljšo polarno izenačenost temperatur pri enakem masnem toku olja za hlajenje zagotavlja sistem DOH 5. Za znižanje ravni vseh temperatur bi morali pri sistemu DOH 5 nekoliko povečati pretok olja za hlajenje.

Gostota toplotnega toka na različnih mestih valja in v različnih smereh je bila izračunana iz izmerjenih vrednosti temperaturnih razlik na različnih mestih valja - v skladu s Fourierjevim zakonom. Večjo natančnost izmerjenih temperaturnih razlik smo dosegli z uporabo verige termoelementov. Analiza pogreškov je pokazala, da je relativni pogrešek pri oceni gostote večjih toplotnih tokov na zgornjem delu valja manjši od 10 odstotkov ter da je največji pogrešek meritev temperaturnih razlik povezan z napako vgradnje (razdaljo med merjenimi mesti v steni valja) zaznavala. Relativni pogrešek vgradnje zaznavala lahko znese ob upoštevanju majhne temperaturne razlike na majhni razdalji v najslabšem primeru celo prek 40 odstotkov.

baseline air-cooled cylinder. DOH 3 results are as good or sometimes even slightly better (from the diagrams in figure 3, 4 and 5) than DOH 4 data. However, if we take into account the more favourable engine running conditions (lower engine inlet and cooling air temperatures), DOH 4 data gave the lowest temperature values. On the other hand, most intensive cooling of the bottom part of the cylinder is not recommended (corrosion problem of the cylinder running surface); the DOH 5 desing therefore provides a compromise solution. This is also the advantage of air or combined cooling over other types of liquid cooling. The increase in the mass oil-flow and design solution DOH 5 can in the end give the optimum results.

Heat fluxes in axial and radial direction at different places were calculated from measured temperature differences, according to Fourier's law. Better measurement accuracy was achieved by applying multiple thermocouples — thermopiles. Evaluation and analysis of errors gave values of less than 10 % for larger heat fluxes (larger temperature differences). The major portion of the relative error is caused by the uncertain location of thermocouples. The relative error of this type, taking into account lower temperature differences and small distances between particular thermocouples can, even sometimes amount to over 40 %.

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