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Termohidravlične razmere laminarnega toka tekočine v ozkih kanalih
Thermo-Hydraulic Conditions of Laminar Fluid Flow in Narrow Channels

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V članku obravnavamo termohidravlične razmere laminarnega toka tekočine v ozkih kanalih z metodo končnih prostornin. V nerazvitih tokovnih razmerah dosega toplotna prestopnost visoke vrednosti tudi pri laminarnem toku tekočine. Posebno pozornost smo posvetili vplivu udara vtočnega curka. Analizirali smo vpliv debeline kanala na toplotno prestopnost. Numerično dobljene rezultate smo primerjali z izkustvenimi obrazci za hitrostno in temperaturno nerazvite profile. Vrednosti toplotne prestopnosti lahko uporabimo kot robne pogoje za računanje temperaturne porazdelitve v kombinirano oljno-zračno hlajenem valju motorja.

The thermo-hydraulic situation of laminar fluid flow in narrow channels is studied by the control-volume approach. Due to the non-developed flow conditions, even in laminar fluid flow very high values of heat transfer coefficient are obtained. Special attention is given to the influence of the inflow jet. The influence of channel thickness on heat transfer coefficients is analysed. Numerically obtained results are compared with empirical expressions for velocity and temperature non-developed profiles. The results for heat transfer coefficients can be used as boundary conditions for temperature distribution computation in an oil-air cooling engine cylinder.

1 DEFINICIJA PROBLEMA

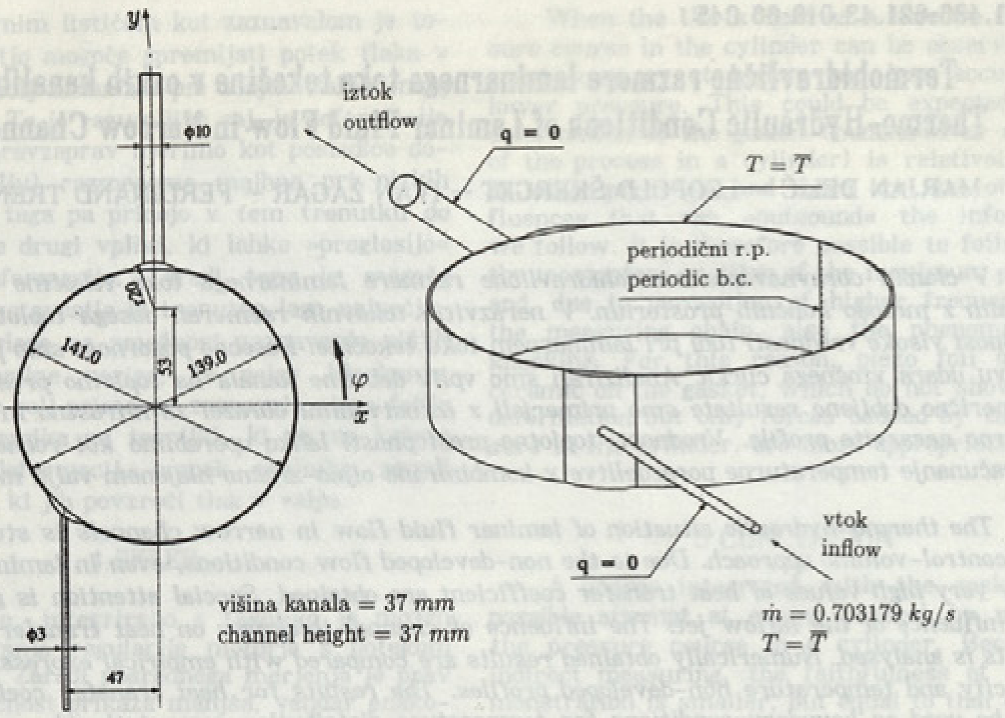
Zaradi različnih temperaturnih obremenitev valja motorja, ki dosegajo svojo največjo vrednost v področju izpušnega kanala, dobimo po obodu neenakomerno temperaturno porazdelitev v steni valja. Temperaturno ovalnost lahko zmanjšamo z intenzivnim lokalnim hlajenjem, ki ga dosežemo z vstavitvijo oljnega kanala pravokotnega prečnega prereza v zgornji, temperaturno najbolj obremenjeni del zračno hlajenega motorja. Za pravilno izvedbo hlajenja je treba poznati lokalne hitrostne in toplotne razmere v kanalu. Razmere v kanalu je mogoče določiti numerično ali eksperimentalno. Problem smo reševali numerično, ker je numerična rešitev cenovno ugodnejša in preprosteje izvedljiva. Pri numeričnem reševanju je tudi preprosteje spreminjati robne pogoje in izpeljati geometrijske spremembe. Geometrijska oblika kanala in robni pogoji, ki smo jih povzeli po literaturi [9], so prikazani na sliki 1.

Pri vtoku smo podali masni pretok, ustrezen prostorninskemu pretoku 5 l/min, pri iztoku smo predpisali referenčni tlak, po stenah kanala pa je hitrost enaka nič. Na ceveh smo podali adiabatne robne pogoje, na vseh drugih trdnih površinah pa smo predpisali konstantno temperaturo stene. V prvih izračunih smo podali temperaturo stene $T = 110\ ^{\circ}C$, v nadaljnjih pa smo na notranji steni podali temperaturo $T = 165\ ^{\circ}C$ in na zunanji $T = 115\ ^{\circ}C$, kar je bližje realnim razmeram v valju.

1 PROBLEM DEFINITION

Due to different thermal loads of the engine cylinder with peak value in the vicinity of the exhaust channel, there is pronounced asymmetric circumferential temperature distribution in the cylinder wall. This temperature ovalty can be minimized by controlled intensity of the local cooling, which can be obtained by introducing a curved square cross-section oil channel in the upper, thermally most loaded part of an air cooled engine cylinder. For correct realisation of cooling it is necessary to know the local thermo-hydraulic conditions in the channel. Conditions in the channel can be defined numerically or experimentally. The problem was solved numerically, because the numerical solution is economicaly more favourable. With the numerical solution it is also easier to change boundary conditions and introduce geometrical changes. The geometry of the channel and boundary conditions, summarized from reference [9], is shown in Figure 1.

On the inflow, the mass flow — which is equal to a volume flow of 5 l/min — is given, on the outflow, the reference pressure is prescribed, and on the walls the velocity vanishes. On the pipes, adiabatic boundary conditions are given, and on all other solid walls the constant temperature of the wall is prescribed. In the first calculations the constant surface temperature of $T = 110\ ^{\circ}C$ is given, and in the next calculations the temperature of the inner wall was $T = 165\ ^{\circ}C$, and on the outer one $T = 115\ ^{\circ}C$, which is closer to the real conditions in the cylinder.



Sl. 1. *Geometrijska oblika oljnega kanala in robni pogoji*
Fig. 1. *Geometry and boundary conditions of oil channel*

Temperatura vtočnega olja je 100 °C. Prav tako smo analizirali tudi vpliv temperaturne odvisnosti snovskih lastnosti, zato se izračuni razlikujejo tudi po upoštevanju ali neupoštevanju temperaturne odvisnosti snovskih lastnosti. Pri izračunih, pri katerih nismo upoštevali temperaturne odvisnosti snovskih lastnosti, smo te vzeli pri T = 100 °C za prvi in pri T = 140 °C za drugi primer. Za reševanje problema je uporabljen programski paket TASCflow 2.3. Konvergenco rezultatov smo proučili na več mrežah, od katerih je najgostejša imela 261 460 vozlišč.

Pomemben vpliv na toplotne razmere v kanalu ima debelina kanala, ker se s tem bistveno spremenijo tokovne razmere v kanalu. Zato smo dodatno analizirali vpliv debeline kanalčka na toplotno prestopnost. V ta namen smo sestavili mrežo oblike kvadra dimenzij *debelina* × 37 × 300. Mreža je imela 115 661 vozlišč (31 × 91 × 41), debelino kanala pa smo nastavili na 0,5, 1, 2 in 4 mm. Pri vtoku v kanal smo podali masni pretok in vtočno temperaturo fluida T = 100 °C, pri iztoku točkovni referenčni tlak in na stenah kanala temperaturne robne pogoje T_S = 110 °C. Masni pretok smo linearno povečevali s površino prečnega prereza kanala, tako da je bila srednja hitrost fluida v kanalu vedno enaka 1 m/s. Pri izračunu smo upoštevali temperaturno odvisnost snovskih lastnosti.

The temperature of the inflow fluid is given at 100 °C. Influences of the temperature dependence of the material properties were also analysed, therefore the calculations are also distinguished by consideration or non-consideration of the temperature dependence of the material properties. In the calculations where influence of the temperature dependence of the material properties is not considered, they are given at T = 100 °C for the first, and T = 140 °C for the second case. To solve the problem, a TASCflow 2.3. computer code was used. Because of the convergence, more meshes were generated, the finest of which has 261 460 nodes.

Modification of the channel thickness represents a significant influence on the functionality of it because of which the conditions in the channel essentially change. Therefore, the influence of the channel thickness on the heat transfer coefficient was additionally analysed. For this purpose a mesh with dimension *thickness* × 37 × 300 was created. The mesh has 115 661 nodes (31 × 91 × 41), and the thickness of the channel at 0.5, 1, 2 and 4 mm is given. At the inflow, the mass flow and inflow fluid temperature T = 100 °C are given. At the outflow reference pressure is given, and on the walls of the channel constant temperature boundary conditions T_S = 110 °C are given. The mass flow was linearly increased with a cross section area of the channel, so that the fluid mean speed in the channel is equal to 1 m/s. In the calculations the influence of the temperature dependence on the material properties was considered.

Snovske lastnosti fluida (motorno olje SAE 15 W/40) smo podali z naslednjimi izrazi, povzetimi iz literature [11], npr. dinamična viskoznost:

The material properties of the fluid (motor oil SAE 15W/40) are given by the following expressions [11], e.g. the dynamic viscosity:

$$\eta(T) = (172{,}37\, e^{-0{,}0242(T-288{,}15)} + 13)\,10^{-3} \quad [\mathrm{Pa\,s}]$$

specifična toplota:

specific heat:

$$c_p(T) = 622{,}72 + 4{,}187\, T - 273{,}15 \quad \left[\frac{\mathrm{J}}{\mathrm{kg\,K}}\right]$$

toplotna prevodnost:

thermal conductivity:

$$\lambda(T) = 0{,}16094 - 0{,}000057143\,T \quad \left[\frac{\mathrm{W}}{\mathrm{mK}}\right]$$

in gostota:

and density:

$$\rho(T) = 890{,}1\,((1 - 0{,}00065\,(T - 293{,}15)) \quad \left[\frac{\mathrm{kg}}{\mathrm{m}^3}\right]$$

kjer je temperatura T podana v K.

where temperature T is given in K.

2 REZULTATI

Numerično dobljene vrednosti toplotne prestopnosti smo primerjali z vrednostmi, dobljenimi iz izkustvenih kriterijskih enačb za Nusseltovo število (*Nu*), in sicer z izrazom, povzetim iz literature [6] za povprečno *Nu* število od vstopa do dolžine *x*:

2 RESULTS

The numerical results obtained for heat transfer with empirical expressions for Nusselt number (*Nu*) [6] were compared. For average *Nu* number, the expression:

$$\overline{Nu} = 6{,}998 + \frac{0{,}0668\left(\frac{d_h}{x}\right) Re\ Pr}{1 + 0{,}04\left[\left(\frac{d_h}{x}\right) Re\ Pr\right]^{\frac{2}{3}}} \tag{1}$$

oziroma z izrazom, povzetim iz literature [9] za lokalno *Nu* število:

was used, and for local *Nu* number [9]:

$$Nu_x = 1{,}077\left[\frac{Re\,Pr}{\frac{x}{d_h}}\right]^{\frac{1}{3}} \tag{2}$$

V izraz (1) smo uvedli korigirano konstanto, ki upošteva razmerje stranic prečnega prereza kanala (37 : 1). V obeh obrazcih je upoštevan hidravlični premer ($d_h = 4A/O$), kjer sta A in O površina in obseg prečnega prereza kanala. *Re* ($Re = v\, d_h\, \rho/\eta$) in *Pr* ($Pr = c_p \eta/\lambda$) sta Reynoldsovo in Prandtlovo število.

Vpliv ukrivljenosti kanala smo zajeli z odvisnostmi [8]:

The constant in equation (1) is changed to consider the relation between the edges of the channel cross-section (37:1). In both expressions the hydraulic diameter ($d_h = 4\,A/O$) is considered, where A and O are area and circumference of the channel cross-section. *Re* ($Re = v\, d_h \rho/\eta$) and *Pr* ($Pr = c_p \eta/\lambda$) are Reynolds and Prandtl numbers.

Influence of the channel curvature was considered by relations [8]:

$$\overline{Nu}_R = \overline{Nu}\,\epsilon_R, \quad Nu_{xR} = Nu_x\,\epsilon_R \tag{3}$$

kjer je ϵ_R korekcijski faktor relativne ukrivljenosti kanala, ki ga računamo po [1]:

where ϵ_R is the correction factor for the relative curvature, which was solved from [1]:

$$\epsilon_R = 1 + 1{,}77\ \frac{d_h}{R} \qquad (4),$$

pri čemer je R polmer krivine kanala. Numerične rezultate smo obdelali z izrazoma:

where R is the radius of the curvature. Numerical results were treated with expressions:

$$\overline{Nu} = \frac{\overline{\alpha}\, d_h}{\lambda} \quad \text{in / and} \quad Nu_x = \frac{\alpha_x d_h}{\lambda} \qquad (5),$$

kjer je:

where:

$$\overline{\alpha} = \frac{1}{l}\int_0^l \alpha_x\, dx \qquad (6),$$

in

and

$$\alpha_x = \frac{q_x}{\Delta T_m} \qquad (7).$$

Temperaturna razlika je podana z izrazom:

Temperature difference is given by the equation:

$$\Delta T_m = T_s - T_m \qquad (8),$$

kjer je T_s temperatura stene in T_m srednja temperatura po prečnem prerezu. Srednjo vtočno (T_{mv}) in iztočno (T_{mi}) temperaturo iz kanala izračunamo z izrazom:

where T_s is the wall temperature and T_m is the mean temperature at the cross section. The mean inflow (T_{mv}) and outflow (T_{mi}) temperature from the channel was determined by:

$$T_m = \frac{\int A_c\, \rho\, v\, c_p\, T\, d A}{\dot{m}\, c_p} \qquad (9).$$

2.1 Ukrivljen kanal

Na sliki 2 je prikazan potek tokovnic skozi ukrivljen kanal. Pri dani geometrijski obliki in pretoku olja (5 l/min) teče skozi krajšo vejo kanala 66,7 % in skozi daljšo vejo 33,3 % masnega pretoka fluida. Na sliki je jasno vidno intenzivno mešanje tekočine pri vstopu v kanal in vračanje dela toka v daljšo vejo zaradi zvečanja hidravličnih uporov v krajši veji kanala.

Na sliki 3 je iz vzdolžnega prereza jasno razviden vpliv vtočnega curka in zožitve kanala. Zaradi udarca curka prihaja do cirkulacije fluida v vzdolžni in prečni smeri, kar dodatno poveča toplotne tokove skozi stene kanala.

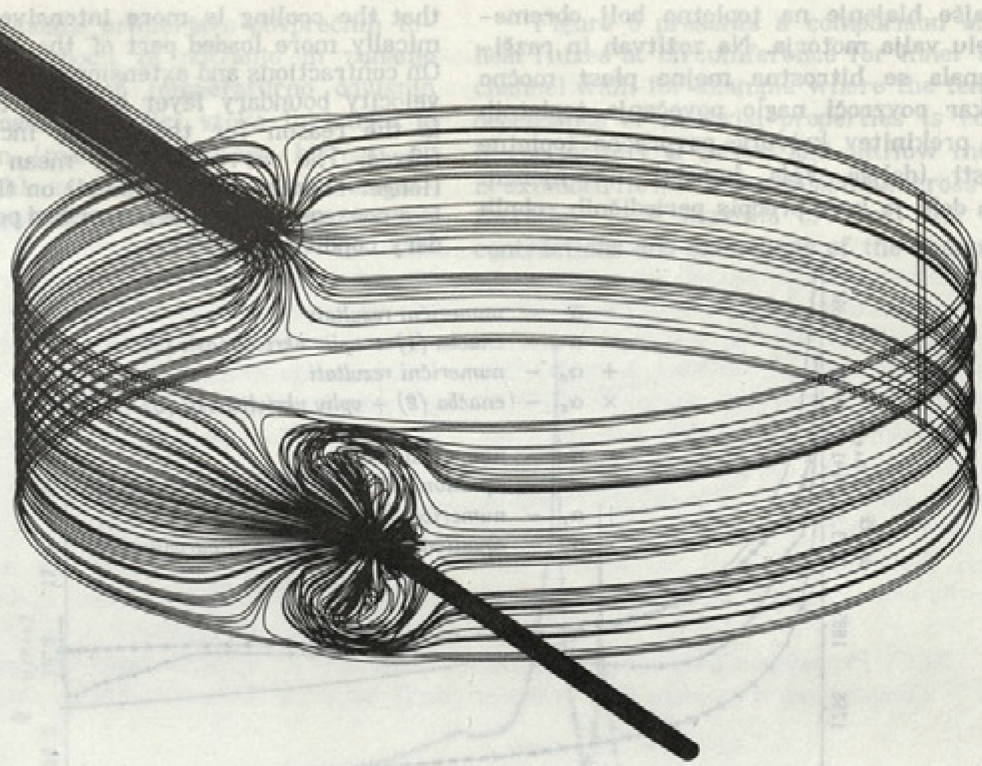
Na slikah 4 in 5 je prikazan potek povprečne prečne toplotne prestopnosti vzdolž obeh vej kanala. Vidimo, da se numerično dobljene vrednosti dobro ujemajo z vrednostmi, dobljenimi iz obrazcev 1 in 2, ki smo jih izbrali za primerjavo z numeričnimi rezultati. Največje vrednosti dobimo na vtoku v veje, nato pa prestopnost počasi pada

2.1 Curved channel

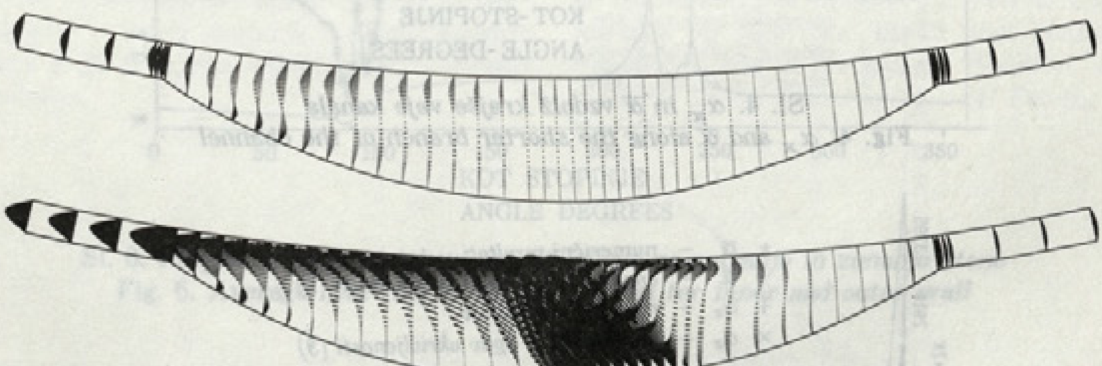
Figure 2 presents streamlines in the curved channel. At given geometry and oil flow (5 l/min) through the shorter branch the mass flow of the oil is 66.7 %, and through the longer one 33.3 %. On the figure there is clearly evident intensive fluid mixing at the inflow of the channel and the returning part of flow in the longer branch because of growing hydraulic dissipation in the shorter branch of the channel.

On Figure 3, the influence of impinging jet and channel contraction from longitudinal section is clearly evident. Because of the impinging jet fluid, circulation is impeded in longitudinal and cross-section, which additionally increases heat flux through the wall of the channel.

In Figures 4 and 5, the average cross heat transfer through both branches of the channel is shown. It is evident that the numerical results are in good agreement with the results obtained from equations (1) and (2), which were selected for comparison with numerical results. The greatest values are on the inflows in the branches,



Sl. 2. *Tokovnice*
Fig. 2. *Streamlines*

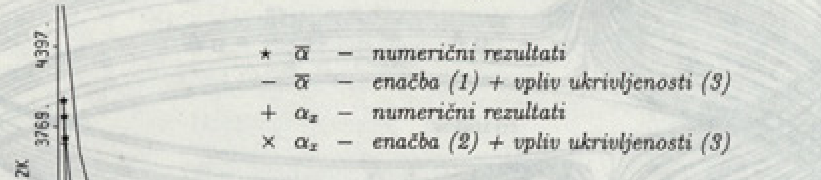


Sl. 3. *Hitrostna porazdelitev v vstopnem območju (1/4, 2/4, 3/4 višine kanala)*
Fig. 3. *Velocity distribution in inlet region (1/4, 2/4, 3/4 height of the channel)*

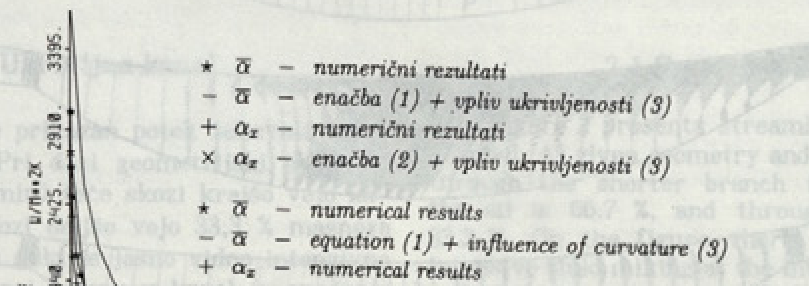
do samega izstopa, kjer se, zaradi razširitve kanala (gl. sl. 1) in s tem stanjšanja mejne plasti, spet sunkovito poveča. Vzdolž krajše veje kanala dosega toplotna prestopnost višje vrednosti zaradi večjega masnega pretoka in manjše dolžine kanala (razvoj mejne plasti), s čimer dosežemo

and then the heat transfer decreases due to outflow, where - because of extension of the channel (see Fig. 1) and with the thinning of the boundary layer - heat transfer instantly increases. Through the shorter branch of the channel, heat transfer has higher values because of the greater mass flow and the shorter branch of the channel (boundary layer development), and with intenzivnejše hlajenje na toplotno bolj obremenjenem delu valja motorja. Na zožitvah in razširitvah kanala se hitrostna mejna plast močno stanjša, kar povzroči naglo povečanje toplotnih tokov. Za prekinitev krivulje povprečne toplotne prestopnosti (daljša veja kanala) na njenem osrednjem delu je kriv predpis periodičnih robnih pogojev.

that the cooling is more intensive on the thermically more loaded part of the engine cylinder. On contractions and extensions of the channel, the velocity boundary layer becomes thinner, which is the reason for the strong increase of heat fluxes. The curve break of mean heat transfer (longer branch of the channel) on the middle part is a consequence of the prescribed periodical boundary conditions.



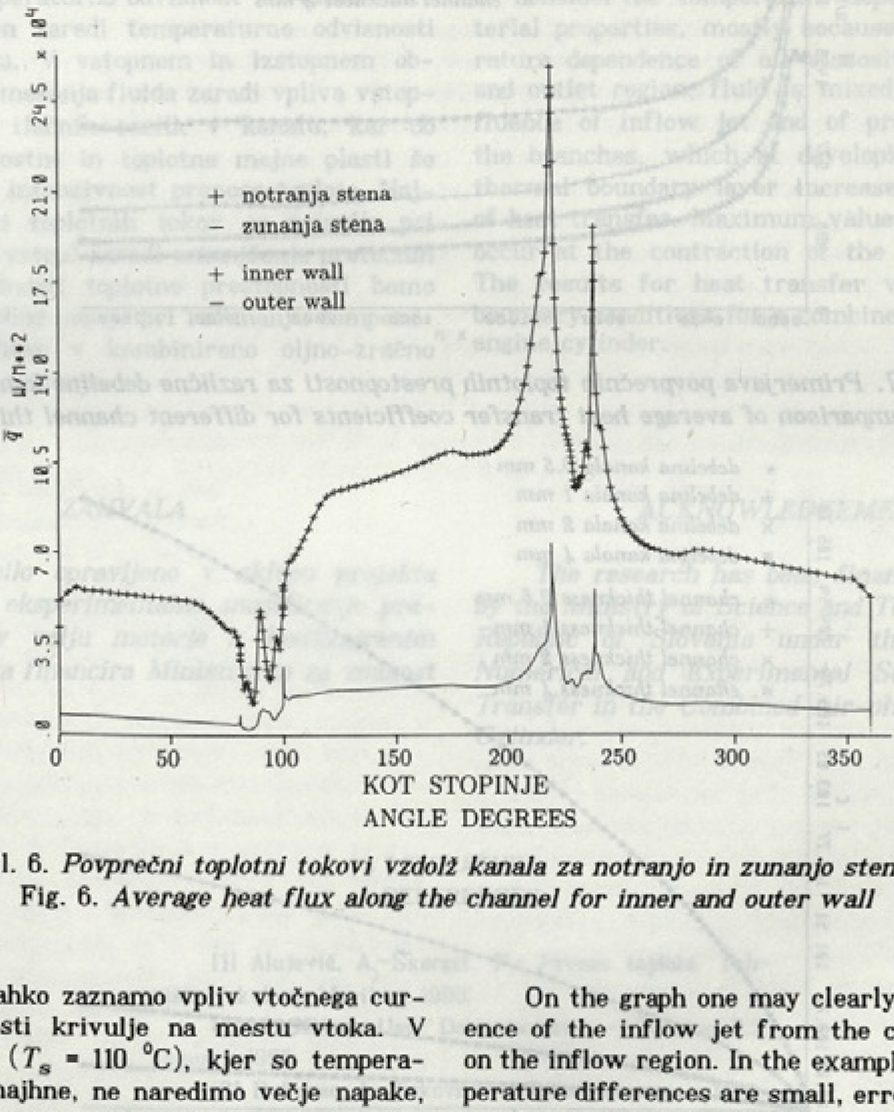
Sl. 4. α_x *in* $\overline{\alpha}$ *vzdolž krajše veje kanala*
Fig. 4. α_x *and* $\overline{\alpha}$ *along the shorter branch of the channel*



Sl. 5. α_x *in* $\overline{\alpha}$ *vzdolž daljše veje kanala*
Fig. 5. α_x *and* $\overline{\alpha}$ *along the longer branch of the channel*

Slika 6 prikazuje primerjavo povprečnih toplotnih tokov po obodu za notranjo in zunanjo steno kanala za primer temperaturno odvisnih snovskih lastnosti. V okolici vtoka in iztoka je kanal razširjen (manjše povprečne hitrosti po prečnih prerezih), zaradi česar so lokalni maksimumi toplotnega toka na zožitvah in razširitvah kanala.

Figure 6 presents a comparison of average heat fluxes at circumference for inner and outer channel wall, for example where the temperature dependence of material properties is considered. In the vicinity of inflow and outflow the channel is extended (lower mean velocity at cross-section), therefore local maxima of heat flux occur on contractions and extensions of the channel.



Sl. 6. *Povprečni toplotni tokovi vzdolž kanala za notranjo in zunanjo steno*
Fig. 6. *Average heat flux along the channel for inner and outer wall*

Na grafu lahko zaznamo vpliv vtočnega curka po izbočenosti krivulje na mestu vtoka. V prvem primeru (T_s = 110 °C), kjer so temperaturne razlike majhne, ne naredimo večje napake, če ne upoštevamo temperaturne odvisnosti snovskih lastnosti, kajti ob majhnih temperaturnih razlikah se te zelo malo spremenijo. V primerih, kjer imamo opraviti z večjimi temperaturnimi razlikami (kakršni so tudi realni robni pogoji — valj motorja z notranjim zgorevanjem), pa z zanemaritvijo temperaturne odvisnosti snovskih lastnosti naredimo že večjo napako.

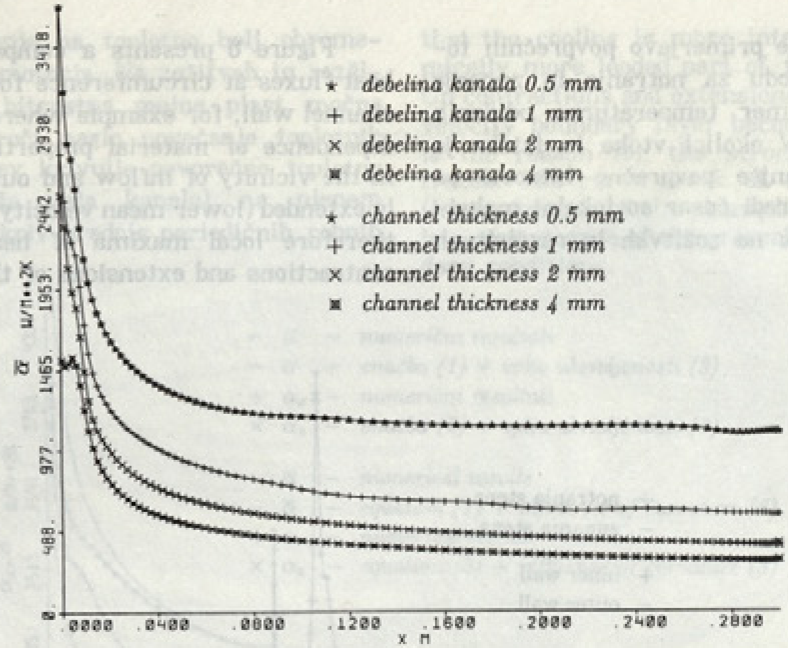
On the graph one may clearly see the influence of the inflow jet from the curve convexity on the inflow region. In the examples where temperature differences are small, errors are not large, if the temperature dependence of material properties is not considered, because with small temperature differences they are nearly constant. In the examples where the temperature differences are larger (as at the real boundary conditions - cylinder of internal combustion engine) a greater error occurs due to neglect of material properties temperature dependence.

2.2 Raven kanal

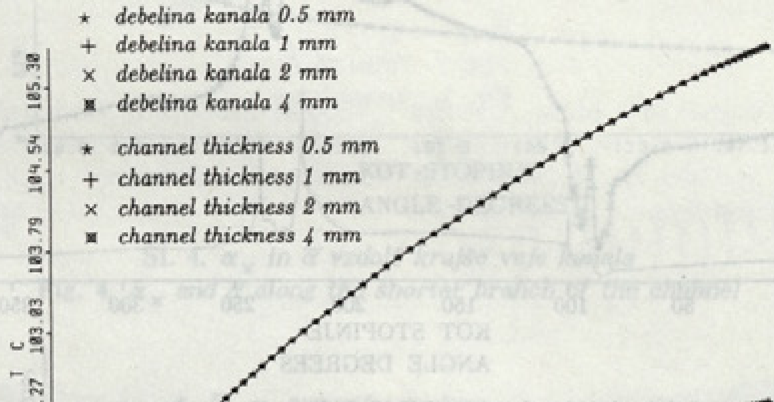
S slike 7 je razvidno, da je toplotna prestopnost precej večja pri ožjih kanalih. Slika 8 prikazuje segrevanje fluida v kanalih različnih debelin.

2.2 Straight channel

From Figure 7 it is evident that heat transfer is greater at the thinner channel. Figure 8 presents fluid heating in the different thick channels.



Sl. 7. *Primerjava povprečnih toplotnih prestopnosti za različne debeline kanala*
Fig. 7. *Comparison of average heat transfer coefficients for different channel thicknesses*



Sl. 8. *Srednja temperatura olja po prečnih prerezih za različne debeline kanala*
Fig. 8. *Mean oil temperature at cross-sections for different channel thicknesses*

V ožjih kanalih je prenos toplote intenzivnejši, kljub manjšemu masnemu pretoku tekočine. Večja toplotna prestopnost v ožjih kanalih je posledica strmejših prisilno oblikovanih profilov, ki so odvisni od debeline kanala (manjša stranica prečnega prereza). S tem povezani so tudi veliko večji tlačni padci v ožjih kanalih, ki so za računane primere v razmerjih 1,00 : 3,69 : 14,16 : 54,32 glede na najdebelejši kanal.

In the thinner channels the heat flux is bigger in spite of the lower mass flow of the fluid. Higher heat transfer in the thinner channel is the consequence of larger velocity gradients, which are conditioned by channel thickness (lower edge of cross section). The pressure drop is larger in thinner channels, which are for the calculated examples in proportions 1.00 : 3.69 : 14.16 : 54.32 in relation to the thickest channel.

3 SKLEP

Z dodatnim oljnim hlajenjem zmanjšamo temperaturne obremenitve in njihovo obodno neenakomernost na valju dieselskega motorja. Dobljeni numerični rezultati se dobro ujemajo z izkustvenimi obrazci. V nadaljnjih izračunih pri realnih robnih pogojih (valj motorja) bo treba upoštevati temperaturno odvisnost snovskih lastnosti, predvsem zaradi temperaturne odvisnosti viskoznosti olja. V vstopnem in izstopnem območju pride do mešanja fluida zaradi vpliva vstopnega curka in tlačnih razlik v kanalu, kar ob razvijanju hitrostne in toplotne mejne plasti še dodatno poveča intenzivnost prenosa toplote. Največje vrednosti toplotnih tokov se pojavijo pri zožitvi kanala (vstop) zaradi zmanjšanja pretočnih prerezov. Vrednosti toplotne prestopnosti bomo uporabili kot robne pogoje pri računanju temperaturne porazdelitve v kombinirano oljno-zračno hlajenem valju motorja.

3 CONCLUSION

With additional oil cooling, temperature loads and their circumferential ovalty on Diesel engine cylinder are reduced. The numerical results are in good agreement with the empirical expressions. Computations with real boundary conditions (engine cylinder) are required in order to consider the temperature dependence of material properties, mostly because of the temperature dependence of oil viscosity. In the inlet and outlet region, fluid is mixed due to the influence of inflow jet and of pressure drops in the branches, which at developing velocity and thermal boundary layer increases the intensity of heat transfer. Maximum values of heat fluxes occur at the contraction of the channel (inlet). The results for heat transfer will be used as boundary conditions for a combined oil-air cooling engine cylinder.

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