HEAT AND MASS TRANSFER IN HWGCR TYPE A-1 FUEL ASSEMBLIES

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SUMMARY

Gas cooled fuel assemblies of rod type designed for use in Czechoslovakian heavy water reactors are characterized by relatively high intensity of heat flux and extensive finning of heat surface. High intensity of heat flux gives rise, beside the finning, to an intensification of fuel assembly cooling. The original concept of fuel assembly was based on large number of smooth rods (up to 200) with fuel diameter of 4 mm. Relatively high coolant velocity in "smooth" fuel assembly had an unfavorable influence on dynamic stability of the fuel assembly. As a consequence of experimental investigations, the fuel diameter had to be increased from 4 mm to 6.3 mm. Such a change of fuel diameter gives rise to use of fuel rod with finned surface.

In paper the results of experimental investigation of pressure drop coefficient, heat and mass transfer coefficient, velocity and temperature distribution in bundles of smooth and finned rods are described. Uniformly heated rods represented model of either central part or full cross section of HWGCR type fuel assembly. Further, some results of investigation on other types of fuel assemblies carried out by collaborating laboratories are presented.

INTRODUCTION

The questions concerning a heat removal from fuel assemblies of gas cooled power reactors are very important due to relatively
low level of a heat transfer coefficient. One of the main features of HñR type A-1 represents possibility of achieving high specific power density and this fact demands the structure of plant having high unit power, however the sufficient fuel heat removal must be guaranteed. The nuclear reactors heat power increasing leads to an improvement of economy factors and is desirable from the point of view of reactors to be competitive. The increasing of reactor unit power gives rise to new requirements concerning heat removal from surfaces from the point of view of efficiency, heat transfer intensification, production technology and compactness.

The analysis of the state of art of theoretical and experimental works concerning rod bundles cooled in-line specified necessity of further improvement of the theoretical solution of flow and heat transfer matters /31/ and experimental investigation of pressure drop and heat transfer characteristics and the investigation of the flow picture in smooth and finned rod bundles /23, 24, 27 + 30/. There is a large amount of theoretical and experimental works /1 + 20/, studying pressure drops and heat transfer in smooth rod bundles, however, relatively of a few works concern the problems of the pressure drops and heat transfer in finned rod bundles /21 + 30/. The analytical solution in a such complicated channel geometry is nearly impossible. The necessary experiments for our fuel assemblies were carried out on different types of the rod bundles, which were specified on the basis of future concept aspects. Further, some experimental results are described.

**BASIC HYDRODYNAMIC AND THERMAL PARAMETERS**

Problem of fuel assembly pressure drop plays an important role particularly in gas cooled nuclear reactors because of considerable influence of pumping power on total efficiency of this type of nuclear power plant. From pilot project of first
Czechoslovakian nuclear power plant fuel assembly, which consists of bundle of rod fuel elements, follow main features of investigations in this field. Both maximum allowable fuel temperature and high power density leads to considerable extension of heat transfer surface. Beside the extension of fuel surface itself, the use of cladding material with high coefficient of thermal conductivity enables the extension of heat transfer surface by intensive finning. A comprehensive research concerning hydrodynamic parameters of finned rod bundles was carried out. Main goal of investigation was to determine basic parameters of finned heat transfer surfaces with regard to its use in HENGR.

The motion of viscous compressible fluid for non-isothermal flow may be described by means of fundamental differential equations - the equation of motion, the equation of continuity and the energy equation. Further we have to apply the equation of state and equations which characterise dependence of thermal properties of coolant on pressure and temperature. These fundamental equations in non-dimensional form represent a closed system of 9 equations for 9 unknown parameters. Solving this system of equations and boundary conditions one should derive relation

$$Eu = \varphi(\text{Re}, \text{Pr}, \beta, \frac{\lambda}{\text{Re}}, \frac{\delta}{\text{Re}}, \eta, \rho, \beta, \Theta)$$

Experimental investigations were carried out with regard to variable parameters from equation (1) except of parameter which represents the influence of temperature coefficient.

In order to assure direct comparison of measurement to that of other authors, D"uler number was replaced by coefficient of friction $\lambda$ derived from relation

$$\lambda = \frac{2 \cdot \text{Re}}{L} \left[ \frac{P_e^2}{G^2} \cdot \Delta p \cdot q_{av} - \left( \frac{\Delta P}{\rho_{av}} - \frac{\Delta t}{T_{av}} \right) \right]$$

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From criterion (1) follows the range of both fin and bundle parameters under investigations. With regard to technology of fuel element cladding the main research was concentrated on helical fins. The fin pitch influences turbulisation of the flow and intensification of the heat transfer along the fuel element as well, so that the mentioned parameter, which is characterised by the angle of fuel element axis and the plane of helical fin, was investigated in large range.

Fig. 1 shows as an example measured values of friction coefficient as depends on Re number and different fin pitch for one type of fins.

Fig. 2 a 3 show the influence of helical fins with different number and height of fins on relative increasing of friction factor in comparison with that of straight fins. There is evident that the influence of helical fins with given height and number of fins takes place for the case of fin pitch being less than 200 mm.

Similarly to the case of friction factor one should derive Nu criterion.

\[ Nu = \varphi(Re, Pr, Bi, \beta, \frac{A}{d_e}, \frac{\delta}{d_e}, n_f, \beta_f, \Theta) \]  

Experiments were carried out on two types of finned rods /27, 28, 29, 30/ which were chosen as most convenient ones from the point of view of further development of fuel assembly.

First type of finned rods has 3 high fins which play double role: extension of heat transfer surface and spacing of fuel elements. The second one has 12 an 15 fins low fins respectively; fuel elements at the same time have special grids.

Heat transfer experiments were carried out on models which represented central part of actual fuel assembly. Number of different types of rods was cut comparing to that of hydrodynamic investigations. Fig. 4 show cross sections of 35 mm o.d. and 70 mm o.d. bundle respectively.

Calculations of heat transfer coefficient were based on the
equation

\[ \text{Nu} = \frac{\frac{Ne \cdot de}{2o \cdot S_c \cdot \delta t \cdot \lambda_v}}{t} \]  \quad (4)

Thermal properties of air were related to average parameters in measured section. Equivalent diameter in a 1 bundles was calculated from the whole wetted perimeter including fins and free flow passages of the bundle.

Mutual comparison of finned surfaces under investigation is based on the use of power coefficient, which represents the ratio of the energy output to frictional power output, related to the unit of surface per 1°C

\[ E = \frac{\alpha}{N} \]  \quad (5)

where

\[ N = 2 \cdot \frac{L}{2 \cdot de} \cdot \frac{G^3}{F^2 \cdot \tau_{av} \cdot S_c} \]

Rods with three fins

Fig. 5 shows the friction coefficient \( \tau \) for the case of different fin pitch in bundle of 7 rods with 3 fins placed in 36 mm i.d. channel. There is evident that in the fin pitch range \( \infty \) to 600 mm \( \tau = \text{const}. \) and the curve for large Re numbers follows Frenkel relation for circular tube taking into account the influence of roughness

\[ \tau_{\infty} = \left\{ 0.068 \ln \left[ \frac{\Delta}{3.7 \cdot de} + \left( \frac{6.81}{Re} \right)^{0.9} \right] \right\}^{-2} \]  \quad (6)

The friction coefficient increases with square order for the case of fin pitch \( s = 600 \text{ mm} \). Such increase can be described
\[ \lambda_s = \lambda_\infty \left[ 1 + 0.0165(\beta - 2.4)^{1.6} \right] \quad (7) \]

and above mentioned relation is valid starting from the value \( s = 600 \text{ mm} \) which corresponds to the angle of fin pitch \( \beta = 2.4^\circ \). Fig. 7 shows dependency Nu against Re. Heat transfer coefficient in the range \( s = \infty \pm 600 \) is constant. The result is similar to that of pressure drop coefficient. Measured values can be expressed analytically as

\[ \text{Nu}_\infty = 0.018 \text{Re}^{0.8} \cdot \text{Pr}^{0.43} \quad (8) \]

Relation (8) differs from that for heat transfer coefficient calculation in smooth circular tube having the coefficient of multiplication approximately 15% lower. Lower values follow from the use of equivalent diameter calculation in which the whole fin perimeter is considered as a part of wetted perimeter. For \( s = 600 \text{ mm} \) the increase of heat transfer coefficient takes place. The increase is linear (taking into account possible inaccuracy of measurement) and can be approximate by analytic relation

\[ \text{Nu}_s = \text{Nu}_\infty \left[ 1 + 0.052(\beta - 2.4) \right] \quad (9) \]

Optimum fin pitch was determined using power coefficient of heat transfer surface. Power coefficient is linear function of the pumping power. In the range \( s = \infty \pm 600 \text{ mm} \) the power coefficient is constant, for \( s = 600 \text{ mm} \) the power coefficient increases.

Rods with 12 and 15 low fins respectively - Fig. 8 shows friction coefficient of six bundles of heated rods with 12 and 15 fins respectively placed in 70 mm i.d. channel with \( s = 550 \text{ mm} \). There
is evident the influence of fin number on the friction coefficient. Values of measurement in the case of 15 fins are generally lower than those of 12 fins. Such a result takes place due to the flow in intermediary fin space. In the case of heat transfer surface with greater number of fins the intermediary fin space decreases, the flow velocity decreases, so that the viscous sublayer increases.

On the base of a comparison of measured friction coefficient with circular tube formula involving actual roughness influence flows that the measured curves have contradictory behaviour than calculated ones. The friction coefficient calculated in accordance with above method is increasing in contrary to actual course. Such a behaviour is typical for all curves. The number of fins has the same influence on the heat transfer coefficient as it had on the pressure drop coefficient. Fig. 9 shows Nu number against Re number. Rods with 15 fins have 7% lower value of Nu number than those of 12 fins. In the case of the bundle of 21 rods the influence of non-uniform fuel cross section to free flow passage ratio is evident. The values of heat transfer coefficient are lower than those calculated according the analytic formula, probably because of the fact that the nonuniform rod distribution gives rise to a flattening of the velocity distribution in large flow area and consequently larger part of a coolant is passing through it.

The influence of fin pitch on hydrodynamic and thermal parameters of rods with low fins was investigated on four bundles of 7 and 6 rods in 36 mm i.d. channel. Fig. 11a and 11b respectively show the heat transfer coefficient and the pressure drop coefficient against the angle of fin pitch. The values of both coefficients are constant in the range of \( s = 300 \text{ mm} \). In the case of \( s = 300 \text{ mm} \) (angle of fin pitch \( \phi = 3^\circ \)) both heat transfer coefficient and pressure drop coefficient are increasing.

On the base of said analyse of geometrical variables the general relation for heat transfer coefficient and pressure drop
Pressure drop coefficient can be calculated as follows

\[ \lambda_\infty = \left\{ -0.868 \ln \left[ \frac{\Delta}{3.7 \Delta e} \cdot C_F + \left( \frac{\gamma}{Re} \right)^{0.9} \right] \right\}^{-2} \]  

Equation (10) can be completed by relation which characterises influence of fin pitch in the case of \( s = 300 \text{ mm} \)

\[ \lambda_s = \lambda_\infty \left[ 1 + 5.44 \cdot 10^{-3} (\beta - 3) \right] \]  

where

\[ C_F = 9.055 - 7/\beta_F \quad \text{and} \quad C_Z = 1.65 \text{ rods with 12 fins} \]

\[ C_F = 1.33 - 3.25/\beta_F \quad \text{and} \quad C_Z = 1.5 \text{ rods with 15 fins} \]

The following expression is convenient for heat transfer coefficient calculations

\[ N\!u_\infty = C_N \cdot Re^{0.9} \cdot Pr^{0.4} \]  

This expression is completed by following one which considers the influence of \( s \geq 300 \text{ mm} \)

\[ N\!u_s = N\!u_\infty \left[ 1 + C_V (\beta - 3) \right]^{16} \]  

where

\[ C_N = 6.13 \cdot 10^{-3} \quad \text{and} \quad C_V = 2.04 \cdot 10^{-3} \text{ rods with 12 fins} \]

\[ C_N = 5.12 \cdot 10^{-3} \quad \text{and} \quad C_V = 1.15 \cdot 10^{-3} \text{ rods with 15 fins} \]

The expressions given above are in good accordance with experimental measurements. Fig. 12 shows mutual comparison of power coefficients corresponding to all bundles under investiga-
In order to find the influence of finned surface geometry and roughness on basic hydrodynamic and heat transfer parameters the investigation with flat models were carried out /32, 33, 34/. The same result was obtained: the closer fins, the worse heat transfer coefficient. Fig. 13 and 14 respectively show the results of heat transfer and pressure drop measurements. The comparison of different finned annuli from the point of view of power efficiency was carried out by Maláč /35/. The detail results concerning friction coefficient and heat transfer coefficient were obtained in the case of channel with o.d. 81, 70, 3 and 64 mm having inside longitudinal fin. One of the main author's conclusion is that the Re number loses its role of determining criterion for the case of closely finned heat transfer surface.

**FLOW PATTERNS IN ROD BUNDLES**

Heat removal from nuclear fuel assemblies consisting of in-line cooled rod bundles is specified by certain peculiarities which make the calculation more complicated than usually, for example in the case of ordinary heat exchanger:

a) In the fuel assembly the heat flux distribution with limited maximum temperature is given. In the case of insufficient cooling of some fuel element the overheating could take place and consequently the failure of the whole assembly is more probable;

b) Replacing of failed fuel assembly is much more complicated and expensive.

From said above follows the importance of detail knowledge of flow and temperature patterns in rod fuel assembly for optimization of hydrodynamic and thermal design and for achieving of reliable and safe operation.
Design of fuel assembly included investigation of velocity distribution in the bundles of smooth rods. Main object of these experiments was to find the influence of bundle parameters and spacers on the velocity distribution in fuel assembly cross section. Fig. 15 shows the form of velocity field in radial direction before and beyond the spacer. Fig. 16 shows the velocity distribution across the bundle. From both figures the influence of non-uniform rod distribution in different parts of bundle on the velocity distribution is evident. From the distortion of velocity distribution caused by grids follows that the cross flow takes place in the fuel assembly. If one assumes the influence of turbulent diffusion, there is clear that the partial mixing of fuel assembly flow exists and considering of this mixing in calculation might have a favorable effect from the point of view of local overheating coefficient. That was the reason for carrying out investigation concerning the flow mixing in smooth rod bundles. As a criterion of mixing phenomena the eddy diffusivity coefficient was used. The mentioned coefficient was measured using method of scalar quantity propagation from the point source in uniform flow which could be described in cylindrical coordinates by the equation

\[ \frac{\partial C}{\partial z} = \varepsilon D \left[ \frac{1}{r} \cdot \frac{\partial}{\partial r} \left( r \frac{\partial C}{\partial r} \right) + \frac{\partial^2 C}{\partial z^2} \right] \]  

(14)

Assuming \( \varepsilon D \), \( Q \) and \( W \) constant, the solution of equation (14) is as follows

\[ C = \frac{Q}{4 \pi E D} \cdot S \cdot e^{-\frac{W}{2 \varepsilon D} (S - z)} \]  

(15)

where \( S = \sqrt{r^2 + z^2} \)

After simplification and transformation to logarithmic form we obtain
If one determines concentration of scalar quantity for some values of radius and plots the diagram of function \( \ln(Sc) = f(S - x) \), the gradient of line is equal to \(-\frac{W}{2E_D}\) and the eddy diffusivity coefficient \( E_D \) can be determined in such way.

Freon 12 \((CF_2Cl_2)\) as a scalar quantity source was used applying above method to the rod fuel assembly investigation. After Freon injection into main air flow the concentration of it on certain distance from the point of injection was measured. Fig. 17 shows curves of constant concentration in a part of the bundle for one case of Freon injection. After subtraction of concentration marked by dot-and-dash lines the obtained values were processed by the use of equation (16) and the average values of eddy diffusivity coefficient were found. In this particular case the eddy diffusivity coefficient reached the value of \(1.16 \times 10^{-5}\) cm\(^2\)/sec. Knowledge of \(E_D\) and gas temperature difference in adjacent flow sections of bundle enables to find \((\frac{E_D}{E_D} = \frac{E_D}{E_D})\) the heat fluxes between sections and so the influence of flow mixing on fuel assembly temperature field.

Experimental investigation of temperature field of the coolant in smooth rod bundles was carried out by means of electrically heated fuel assembly model with 37 rods \(/31/\). The model was placed in open air loop and tested up to \(Re = 2 \times 10^5\). Fig. 18 shows the measurement results. There is evident that the generated heat from centrain row of rods is concentrated in the very neighbourhood of heated rods and only a small part of heat is transferred in adjacent sections. After superimposition of temperature fields originated due to a local rod heating we obtain the temperature field which is very near in fact to the actual temperature field when overall heating of the bundle.
There was carried out detail theoretical calculation of flow and heat transfer in square and triangular array of smooth and finned rod bundles respectively. The results are presented in [31].

The local parameters of fins and flow sections of rod bundles were investigated by means of enlarged models [32]. Relatively considerable differences of local values of shear stress in circumferential direction of the fin and the main cross section were found. Results are shown on fig. 19, 20.

**TEMPERATURE FIELD OF FUEL RODS**

From the point of view of safe and reliable operation of nuclear reactors fuel assemblies the detail information concerning the temperature field in fuel rods is necessary for two reasons:

- there is given the maximum allowable temperature,
- mechanical stress of the rod due to temperature gradient could effect the long-term strength of fuel rods.

In the case of gas cooled reactor a considerable distortion of temperature field in the spacers area is taking place. There was worked out the method of temperature field simulating by means of electrically heated rods with following mathematical treatment which enable to find the temperature field in actual fuel element rod [14, 29, 31]. In the spacers area the substantial intensification of heat transfer was found and there is possibility to utilize this effect. Fig. 21 shows results of temperature field measurements expressed by means of dimensionless Nu number as function of dimensionless distance for some types of spacers. Detail informations about the method of simulation of both steady and unsteady fuel rod temperature fields by the means of electrical analogy are given in [37, 31].
CONCLUSION

The hydrodynamic parameters measurements carried out on different types of fuel assemblies enabled to obtain the clear picture of flow in fuel rod bundles and estimate the influence of heat transfer surface parameters under investigation from the point of view of hydrodynamic and heat transfer.

The calculating formulas valid for given heat transfer surface geometry were found. Analysing these relations one can conclude the conventional equation for smooth rod bundles using $d_a$ as the characteristic dimension cannot be used. Application of Freon method enable the investigation of flow mixing in complicated fuel assemblies. There is necessary to know detail flow picture in fuel rod bundles from the point of view of the optimisation of the fuel assembly as a whole. Future development in this field is desirable for the calculating methods to be improved.

Besides that, the method of simulation of temperature field in spacers area has been worked out and experimentally proved. Such a method seems to be very useful from the point of view of fuel rod temperature field in various fuel assembly arrays to be found.

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SYMBOLS

C \( /\text{kg/m}^3/ \) - mass concentration
\( d_e \) \( /\text{m/} \) - equivalent diameter, 4 F/O
\( E \) \( /\text{l/deg/} \) - energy coefficient
\( F \) \( /\text{m}^2/ \) - free-flow area of the bundle
\( G \) \( /\text{kg/s/} \) - air mass flow
\( L \) \( /\text{m/} \) - length of the working section
\( n_f \) \( - \) number of fins
\( n_i \) \( - \) number of rods at the given radius of the bundle
\( N_e \) \( /W/ \) - electric power output in the working section
\( N_f \) \( /W/m^2/ \) - frictional power output from the unit of surface
\( O \) \( /\text{m/} \) - wetted perimeter
\( P_{av} \) \( /\text{N/m}^2/ \) - average pressure
\( Q \) \( /\text{kg/s/} \) - injected quantity of Freon
\( \Delta P \) \( /\text{N/m}^2/ \) - pressure drop in the working section
\( \Delta \) \( /\text{m/} \) - roughness of the surface
\( \rho_{av} \) \( /\text{kg/m}^3/ \) - density of coolant in the working section
\( s \) \( /\text{mm/} \) - fin pitch
\( S_c \) \( /\text{m}^2/ \) - heat transfer surface
\( \delta T \) \( /\text{deg/} \) - average temperature difference between the surface and the coolant, \( (t_w - t_{av}) \)
\( t_w \) \( /\text{C/} \) - average surface temperature of the bundle
\( t_{av} \) \( /\text{C/} \) - average temperature of the coolant in the working section
\( \Delta t \) \( /\text{deg/} \) - air temperature rise in the working section
\( \dot{m} \) \( /\text{m/s/} \) - flow velocity
\( \theta \) - dimensionless temperature, \( \frac{T - T_0}{T_w - T_0} \)
\( z \) \( /\text{m/} \) - coordinate
\( \alpha \) \( /\text{m}^2/\text{deg/} \) - average heat transfer coefficient
\( \beta \) \( /\text{degrees/} \) - angle of the helix of the fins
\( \beta_f \) \( /\text{degrees/} \) - total rod cross section /to channel cross section ratio

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\( \lambda \) - coefficient of friction of the bundle

\( \lambda_v \) - thermal conductivity of air

\( \varepsilon_0 \) - efficiency of the heat transfer surface,

\( \varepsilon_c / m^2/s/ \) - eddy diffusion coefficient

\( Bi \) - Biot number

\( Zu \) - Euler number, \( \lambda L/2d_e \)

\( Nu \) - Nusselt number, \( \rho \mu c_p / L \)

\( Pr \) - Prandtl number

\( Re \) - Reynolds number, \( \rho \mu c_p / F \rho D \)

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7 RODS, 15 FINS
6 RODS, 15 FINS
7 RODS, 12 FINS
6 RODS, 12 FINS

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ϕ 36 mm MODEL

6 RODS, 15 FINNS
6 RODS, 12 FINNS
7 RODS, 15 FINNS
7 RODS, 12 FINNS
7 RODS, 3 FINNS

ϕ 70 mm MODEL

19 RODS, 12 FINNS
21 RODS, 12 FINNS
24 RODS, 12 FINNS
19 RODS, 15 FINNS
21 RODS, 15 FINNS
24 RODS, 15 FINNS

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