7. Thermohydraulic Calculation Code DISTEMP

The computer code DISTEMP was designed to perform thermohydraulic calculations of gas heated spiral type steam generators. In its present form it is not suitable for sizing calculations, but only for simulation calculation, i.e. the steam generator thermal performance is worked out based on a given tube bundle. The output mainly consists of gas-, water/steam- and tube wall temperatures along the bundle representing mean values across the bundle.

DISTEMP is a linear code, that determines the heat flow from the gas to the water or steam using conventional heat transfer correlations. The calculations are started at the "cold end" of the steam generator and repeated stepwise along the heating surface using the results of the previous step as input for the actual step, until the end of the surface is reached.

Some special features implemented in DISTEMP are to be mentioned:

- Option to include a hot spot factor, taking into consideration the nonuniform gas side heat transfer coefficient along the tube circumference.
- Option to upgrade or downgrade the theoretical heat transfer and pressure drop coefficients.
- Option to enter different fouling resistances in economizer, evaporator and superheater section.
- Temperature plot subroutine available.

8. Thermohydraulic Calculation Code DERZ

The thermohydraulic calculation code DERZ is used by HRB for determining the heating surface area of a helically wound steam generator. Hereby, gas side heat transfer is calculated according to the slightly modified Grimison correlation including a flow factor, whereas waterside the Gnielinski correlation as described in the "VDI-Wärmeatlas 1974" is used.

The input consists of geometry, material properties, gas inlet and outlet temperature, pressure and mass flow rate. On the water side, also inlet and outlet temperatures and steam pressure must be specified.

The program calculates the required heating surface, pressure drops and additional geometrical data as e.g. number of tubes in the individual cylinders.

EXPERIMENTAL INVESTIGATIONS OF HEAT EXCHANGE AND HYDRODYNAMICS ON MODELS OF A VG-400 STEAM GENERATOR TUBE BUNDLE MADE UP OF SMALL DIAMETER HELICOILS

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Abstract

Features of HTGR steam generators having heat exchange surface made up of small diameter helicoils are discussed in the paper. A general approach to optimization of thermohydraulic characteristics of ВГ-400 steam generator design backed by calculation and experiment are given. Main results of steam generator assembly's model aerodynamic test are presented. Data of thermohydraulic tests of a single tube model in a helium heated test rig are discussed.

At present when designing HTGR steam generators (SG) the USSR considers the possibility of using tube bundles made up of small diameter helicoidal coils. Such coils have the ratio of the mean coil diameter D_{cp} to the tube outer diameter d_{H} from 3 to 14. The minimum value of the ratio is defined by manufacturing considerations while the maximum value is defined by the coils' diameter effects on its thermohydraulic characteristics and by the coil's overall dimensions. The lower value is also limited by the increasing tubes' ovality when winding and by changes in wall thickness at the outer and inner generatrices.

The application of such coils is due to the features combining advantages of straight tube bundles and coils, i.e.

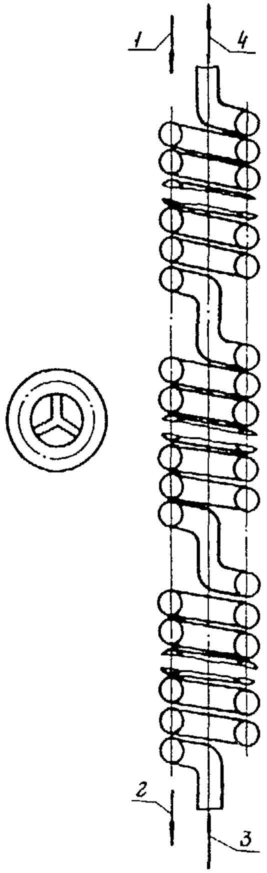
ease of manufacture and bundle compactness, interchannel mixing, self compensation of thermal deformations, improved heat transfer between inner flow and tube's wall and shifting of degraded heat transfer boundary to the end of evaporator section ($X^{o}_{rp} = 0.7...1$). The disadvantages of such coils are increased number of defects at tube winding (as compared to large diameter coils), stratification of two-phase flow inside small diameter coils due to centrifugal forces which may lead to the degradation of heat transfer and resulting temperature fluctuations and overheating of tube's material at the coils inner generatrix, see ref. /1/; insufficient knowledge of heat exchange and hydrodynamic processes in such coils.

Lack of experimental data concerning the problems of thermal physics and peculiarities of heat exchange in tube bundles made up of small diameter helicoils predetermined the following general approach to development of the SG designs:

- investigation of tube bundle geometry influence (within supposed dimensional range) on distribution of coolant flowrate over the intertube cross section and of pressure loss;
- investigation of heat exchange feature and influence of coolant flow redistribution over the tube bundle cross section on the effective use of heat exchange surface and on temperature distribution over tubes' perimeter in the boiling crisis zone in the evaporator section;
- choice of optimum tube bundle dimensions for the whole SG power range and verification of adopted decisions on a large scale model.

According to this approach, the experimental substantiation of thermohydraulic characteristics is performed on the VG -400 SG variant where the heat exchange surface is constituted by small diameter coils.

Seven coils butt-welded at their straight sections form a heat exchange element, Fig. 1. There is a filler inside the coils consisting of three steel strips, 19 heat exchange elements are grouped to hexagonal assemblies each with its own casing.



1 - helium inlet; 2 - helium outlet;
3 - feed water inlet; 4 - steam outlet.

FIG. 1. VG-400 Steam Generator Heat Exchange Element.

Aerodynamic tests of isothermal models of the GS tube bundle have been performed according to the experimental program.

Investigations have been carried out on models with rhombic casing, Fig. 2, in which 16 coils are placed on a triangular pitch. The models differed in casings' dimensions, coils' diameters and longitudinal pitches. Some models had 3-ribbed fillers inside the coils. The coils were wound of 8 mm outer diameter tubes or copper bars of the same diameter. In the process of winding the tubes acquired ovality $a = \frac{d_H^{max} - d_H^{min}}{d_H} \cdot 100\% = 12.5\%$, where d_H^{max} and d_H^{min} - maximum and minimum oval axis. Cross section of bars after winding remained circular.

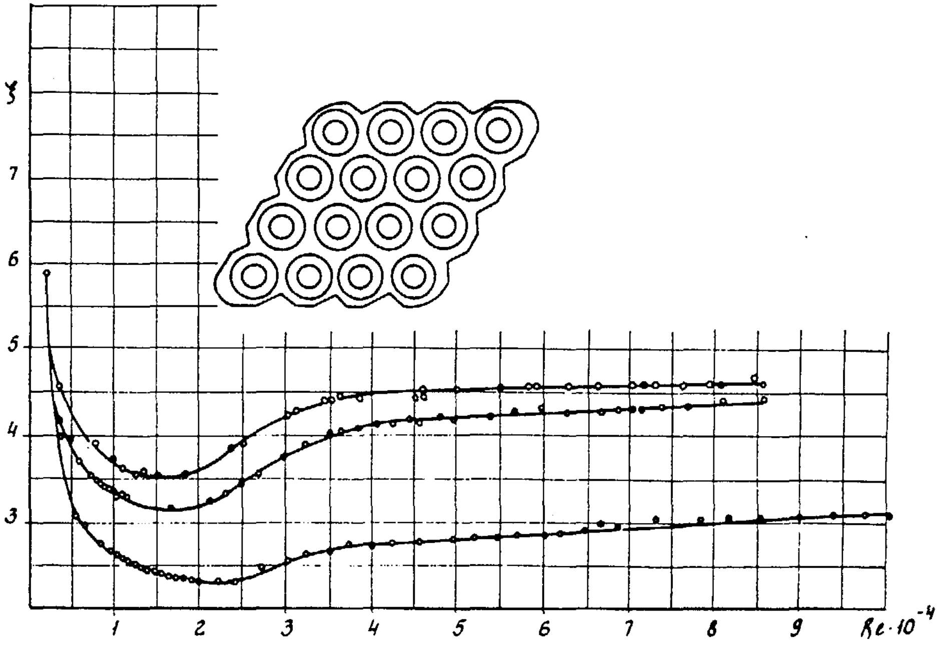


Fig. 2. Models' Hydraulic Resistance Factor versus Re number and Assemblies' Cross Section.

Hydraulic resistance of models was determined by the static pressure drop along the assembly. The measurements were made with 2.5 mm diameter cylindrical probe which could be moved axially inside the assembly.

Air flow distribution across the bundle was determined by flow velocity profile at the model's outlet. A comb of four Pitot tubes was used for this purpose.

Hydraulic resistance factors of models related to velocity V at the assembly's inlet cross section obstructed with coils were calculated versus Reynolds number $Re = \frac{V \cdot d_r}{\nu}$, where hydraulic diameter was determined by relation $d_r = \frac{4 \cdot F}{\Pi_{3M} + 0.15\,\Pi_{r\Lambda}}$, where F - assembly's obstructed flow cross section, Π_{3M} - coils' perimeter, $\Pi_{r\Lambda}$ - smooth surfaces' perimeter.

Plots of resistance factor variation versus Re number for three models are given in Fig. 2, /2/. The experimental results for all models have been approximated by the relation in which determining parameters are Re number, relative transversal $S_1/\,d_H^{min}$ and longitudinal $S_2\ /\ {}_H^{max}$ coil pitches (transversal coil pitch was determined by analogy with in-line flow-across bundles of straight tubes: $S_1 = \frac{F}{\pi D_{CP}} + d_H^{min}$, d_H^{min} and d_H^{max} are the minimum and maximum axes of oval).

The dependence resistance factor for the single turn:

$$\xi_1 = \frac{\xi}{Z} = 0.032\,\frac{S_2/d_H^{max} - 0.25}{S_1/d_H^{min} - 1.73} \cdot \frac{S_1/d_H^{min}}{1 + \frac{8434}{Re}}\left(1.02 - 1.4 \cdot 10^{-6} \cdot Re\right)^{S_2/d_H^{max} - 1.12}$$

where Z - number of coil's turns in the model.

The expression is true for Re numbers' range $Re = (17...130) \cdot 10^3$ with relative pitches between turns $S_1/\,d_H^{min} = 2.09...3.9$ and $S_2/\,d_H^{max} = 1.12...1.82$.

Difference between calculated and experimental values does not exceed ± 10%.

Taking into account variation of the flow density along the coil the hydraulic resistance factor of a single turn may be corrected using the following expression:

$$\xi_{cp1} = \xi_1 \cdot (1 - 0.8 \cdot 10^{-5} \xi_1 \cdot Re).$$

Distribution of heating coolant flowrate in the tube bundle considerably affects the heat exchange efficiency and to a great extent defines the coils' temperature state. Fig.3 depicts some investigation results of air flowrate distribution over the assembly cross section, namely air flowrate Q_{BH} inside the coils to the flowrate Q_H outside the coils versus Re number for five models. As shown, the fraction of

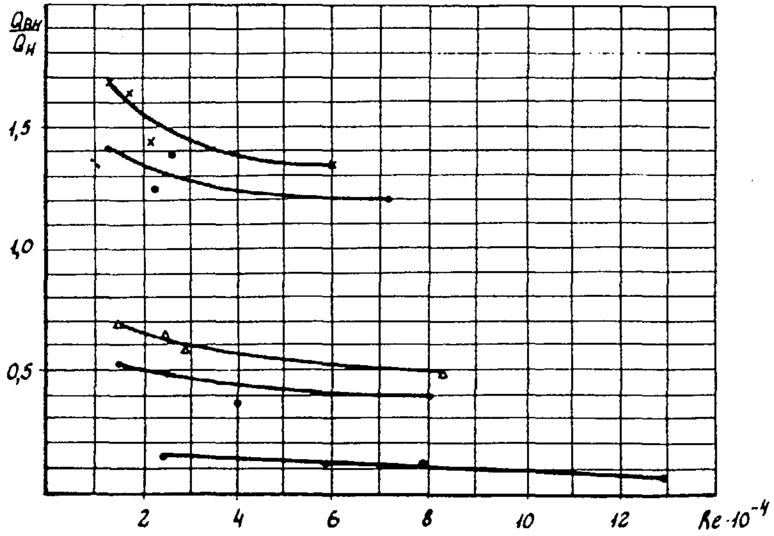


Fig. 3. Air Flow Ratio Out- and Inside the Coils versus Re number.

air flowrate inside the coils goes down with the increase of Re number. Flowrate ratios versus Re number were investigated on 9 models and summarized as follows:

$$\frac{Q_{BH}}{Q_H} = \frac{K^{0.844}}{0.866 + 3.731 \cdot 10^{-6} \cdot Re},$$

where $K = \frac{F_{np}^{BH}}{F_{np}^{H}} \sqrt{\frac{d_r^{BH}}{d_r^{H}}}$,

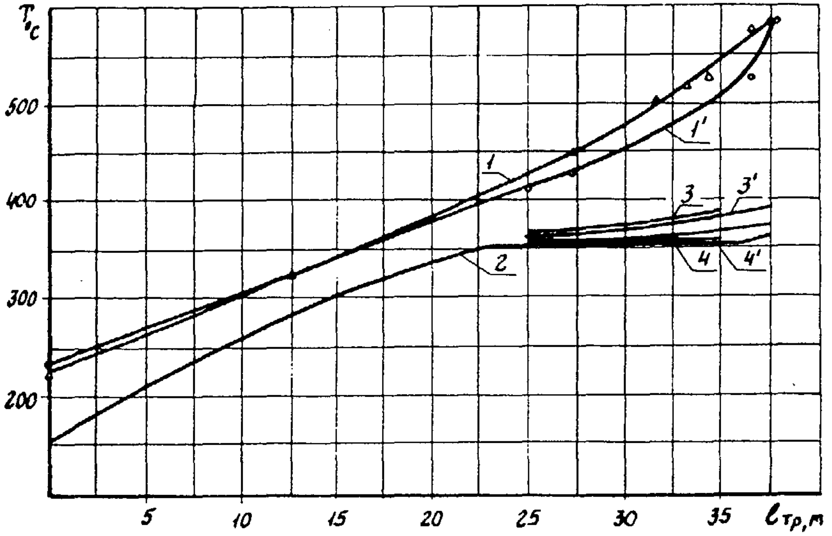
F_{np}^{BH} and F_{np}^{H} - air flow cross section in- and outside the coils;

d_r^{H} and d_r^{BH} - hydraulic diameter of the flow cross section out- and inside the coils.

The relationship is true for variation of $K = 0.1...1.6$.

At present experimental investigation of heat exchange, hydrodynamics and temperature regimes of small diameter coils are performed on a single tube models with actual parameters of helium and steam water circuits. Each model imitates one SG heat exchange element and consists of 6 coils connected in-series and arranged vertically in a cylindrical tube. The models differ in transversal dimensions of the filler placed inside the coils. Thus in different models various ratios of helium flowrates are created along the external and internal generatrices. Measurements of a number of thermohydraulic characteristics of models are provided during the tests: flowrates, pressure differences, helium and water (steam) temperatures, steam generating tube temperatures. During the tests helium flowrates and temperatures at the model's inlet were chosen such as to provide at a supposed place of boiling crisis in the evaporator section heat fluxes in the model equal to those of the actual SG.

Fig. 4 presents experimental obtained curves of temperature variation of helium flow inside and outside the coil, water (steam) and heat exchange tube (along the tube) temperatures. As shown, at the end of evaporator section (at helium inlet into the model) there is nonuniformity of helium temperature over the cross section (the temperature of helium flowing inside the coil is less than the temperature of helium flowing outside the coil). Such helium temperature distribution over the model's cross section affects the tube's temperature (wall's temperature at the outer generatrix is a bit higher than at the inner one). The helium temperatures in- and outside the coil became progressively equal and over the major part of model's length helium temperatures over the cross section are practically the same. It stands for the effective use of SG tube heat exchange surface.



1 and 1′ – helium temperature out- and inside the coil;
2 – water (steam) temperature;
3 and 3′ – tube outer surface temperature at the outer and inner coil's generatrices;
4 and 4′ – tube inner surface temperature at the outer and inner coil's generatrices.

Fig. 4. Helium, Water (Steam) and Heat Exchange Tube Temperature Distribution along a Single Tube Element.

Aerodynamic and single tube models' tests in combination with the following full-size assembly experiments on a helium test rig of increased heat power will enable to ultimately specify geometrical dimensions and heat engineering parameters of the SG made up of small diameter helicoils.

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