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Heat transfer in coiled type superheater of moisture separator-reheater of turbines at the nuclear power plant

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Abstract. The reheating part of the moisture separator-reheater (MSR) of nuclear power plant (NPP) steam turbine occupies a significant part of MSR volume and requires large consumption of metal. The superheating of low pressure steam provides by the high pressure steam condensation. This steam supplies partly from turbine inlet and partly from turbine stage bleeding because of two superheater stages. The significantly less heat transfer intensity at the single-phase side of reheater comparatively with condensation side requires intensification of heat transfer from the heated-steam. For such goal the different methods can be used: for instance the cross flow of the coiled tubes and finning of straight tubes bundles. Also is perspective the usage of artificial roughness in form of small "moons" and other types of artificial roughnesses.

Introduction

In saturated steam turbines of NPPs at the high pressure casing (HPC) humidity in the HPC steam humidity reaches 10-15 %. Under such conditions effectiveness and reliability is decreased; the last blades fall under erosion and liable to failure before the expiration of the planned lifetime. At NPP the interturbine moisture separation is combined with external separation and superheating in special apparatus MSR. The technical-economic indicators of NPP as a whole depends on the effectiveness and reliability of MSR. Their masses and occupied volumes are significant, and with the growth of NPP units output the masses and sizes of MSR's grow too. So, the increase reliability and effectiveness of MSR leads to an improvement in the overall characteristics of NPP. In the given paper the problem of steam superheating effectiveness MSR is considered for MSR operating at Russian Leningrad NPP (LAES).

Initially at the first unit of LAES the apparatus named asMSR-500 was used, which was developed in Polzunov institute and manufactured by Podolsk machinery works (ZIO). Later MSR-500 was removed and changed by MSR-500-1. Such replacement was performed after some failures SPP-500 elements, which were not related to the main functions – separation and overheating. MSR-500 was more effective in means of working process than MSR-500-1. We think that its main ideas can be used when perspective MSR develop for NPP of next generation. For new apparatus development new computation methods and operation experience appreciation will be required. These methods should be based on the modern investigations of separation and heat transfer. In the paper we presented new coiled type superheater of the same type as was used in MSR-500.

Method of reheating in moisture separator-reheater type MSR-500

In these apparatus the two stage superheating is realised. The steam heats in the spiral of helical tubes, assembled in bundles cross flowed condensing steam in two-step superheater (fig. 1).The tube bundle of first stage, which placed in the peripheral part of tube system, consists of 432 four paths spiral tubes. The heating steam is taken of stage bleeding (y<10 %; t=209°C, p=1,9 MPa [1]). The condensate outflow (t=205°C, p=1,8 MPa [1]) is performed from the tube bundle outer side. With this purpose the inner ends of the higher row of spirals, which are twisted clockwise, are welded with inner ends of low row spirals, which are twisted contraclockwise. The heating steam enters in the upper spirals, moves clockwise from the periphery to the center, passes to the low spiral row, moves through it from the center to the periphery. The tube bundle of second stage, which is placed closer to the apparatus center, is of the same construction, but consists from 216 tubes. The heat transfer is resulted by extracted inlet turbine steam condensation, which inlet temperature t=278°C, p=6,3 MPa and outlet condensate parameters equal to t=275°C, pressure p=6,1 MPa.

All coils are manufactured from stainless steel tubes 18x1,4 mm.

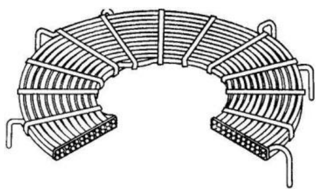

Figure 1. Two rows element of the coiled spiral tubes bundle of MSR-500.

Method of calculation of the heat transfer in a coiled type superheater

The character of cross flow in a bundle with tightly fixed tubes is shown in fig. 2. Flow and heat transfer analysis in such the bundle becomes more complicated for tubes, twisted by spirals (fig. 3).

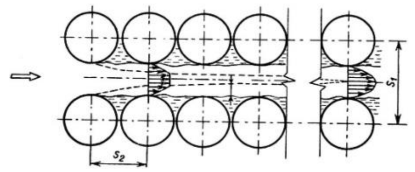

Figure 2. Sheme of flow in a “pressed” (tight) corridor-type tube bundle.

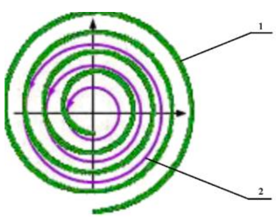

Figure 3. Sheme of actual coiled bundle geometry (1) and proposed calculation model (2).

In some works (for example, [1]) it is proposed to perform the calculations for the multirow bundle as a whole and uniform one, without taking into account the changing of heat transfer along rows. But only in multirow bundle the averaged heat transfer of whole bundle is close to heat transfer for depth row of bundle. In the construction considered here the number of rows along the depth of bundle is not a large – in MSR-500 such number is ~20 for the first stage of superheating and ~19 for the second stage. So, one can't neglect of first rows deposit to the heat transfer. Moreover, in a spiral coil the outer rows of tubes consist the larger part of heat transfer surface in comparison with deeply located rows (outer rows are located at maximal coil diameter).

We suggest to consider a spiral coil model in form of set of the concentric tube rings (fig. 3). Such a model takes into account a flow character in tight multirow bundle with simultaneously taking into account of coil geometry and deposit of outer rows. In this model in each radial section of coil is kept constant the distance between the axises of separate tubes of bundle along its depth and the distances between axises of separate tubes along bundle height is also unchangeable. So, the relative cross pitch along depth ($\sigma_1 = s_1/d$) and longitudinal (along height) pitches ($\sigma_2 = s_2/d$) are kept constant. Heat transfer coefficient α calculation are performed by the averaging of meanings of α method. Partial meaning α_1, α_2, … are obtained for separate rows (rings) with taking into account the income of each ring in overall heat transfer area,

$$\alpha_{\text{bundle}} = (\alpha_1 \cdot H_1 + \alpha_2 \cdot H_2 + \ldots\ \alpha_{\text{j}} \cdot H_{\text{j}})/(H_1 + H_2 + \ldots\ H_{\text{j}}) \tag{1}$$

that is the heat transfer coefficients of separate rows α_j install in formula (1) as multiplied ones by corresponding correction coefficient, which takes into account the heat transfer intensity by rows deep into the bundle.

During heat exchange process the temperature of flow changes and, consequently, meanings of the heated steam parameters change also. The account of the influence of physical properties of flow on heat transfer is connected with the choice of the conditioning temperature, by which the physical parameters are defined. Are known two methods for the taking into account the changings of physical properties with temperature. According to first one the meanings of physical properties of media are defined by the temperature of flow, and in the similarity equations are included as additional parameters. According to second method the physical properties are defined by the mean temperature between flow and wall temperature and a form of heat transfer coefficient correlation remains the same, as in case of constant meanings of physical properties. We have used the first metho*d.*

Assumptions

1. Heat losses to environment from MSR are absent. For MSR and similar apparatuses the relative heat losses are not larger, that parts of percent.
2. For non-isotermical conditions the designed thermophysical properties of heating and heated steams is each stage are taken by mean temperatures of these media.
3. Velocities of flows are equal to their averaged meanings in each stage of superheater.
4. The heat transfer surface does not have depositions; that is thermal resistance of system is constant.
5. Heat transfer from heating media by condensation inside tubes is calculated in full condensation assumption. It corresponds to the recommendation [4].

Heat transfer coefficients calculation for superheating steam by flow through coiled tubes.

We used several variants of calculations to choose the correlations corresponds conditions typical for MSR. For example, in works [10-12] are recommended to conduct heat transfer for multiflow bundle (n rows more than 10) as whole but not for rows separately; Prandtl number Pr_f is recommended to define by flow temperature, that is by meaning of bundle depth temperature. The recommended exponent at Pr_f in [10-12] is equal to 0.33 and somehow lower, than in [6] (0.35); the temperature correction factor is $(\mathrm{Pr}_f/\mathrm{Pr}_w)^{0.25}$.

It is known, that heat transfer intensity of the widely used in-line bundles is lower, than of staggered ones. But negligible difference between the values of heat transfer coefficients obtained when considering the tight bundle in-line and staggered approximations, confirms the validity of the assumption of the author that the true value of the heat of the investigated a dense beam should be considered to be in the range between the two values (in-line and staggered).

V.P.Isachenko in textbook [17] recommends to define heat transfer coefficient from row to row, using as determining one a mean temperature of flow. The proposed by him coefficient in the expression for Nusselt numbers is equal to C=0.26.

That remarkably increases the value of heat transfer coefficient is comparison with values, which have been used by us where C=0.2. Author [17] involves correction factor $e_s=\sigma_2^{-0.15}$, accounting an influence of relative placement of tubes in deeply located rows. It is indicated, that such a correction is received with investigation of the bundles with relatively large cross pitches (σ_1 = 1.3…2.6). From that we concluded, that for our pitch meaning (σ_1 = 1.14) correction factor e_s is not needed. For the determination of heat transfer coefficient for all bundle we are averaging the meanings of heat transfer coefficiens for separate row, using formula (1).

In the work [17] is indicated also, that by relative low extent of turbulence, particularly, by the absence of the artificial flow turbulization of incoming flow, the heat transfer intensity of the first (outer) row of in-line bundle consists of ~0.6, and of the second row ~0.9 from heat transfer intensity of the third and following rows. If a extent of turbulence of incoming flow is high the heat transfer of outer rows coincides with heat transfer of deep rows. In the work [17] are indicated also the correction coefficients for other rows, however there are not information about Reynolds numbers Re_f recommended diapason. In our calculations we have accounted, that numbers $\mathrm{Re}_{f\mathrm{I}} = 4.9\cdot10^4$ and $\mathrm{Re}_{f\mathrm{II}} = 5.7\cdot10^4$ correspond to low extent of turbulence, because the meanings of recommended in [17] correction coefficient are close to recommended by another authors [18-20] ones.

In the handbook [13] the correlation is proposed

$$Nu = 0.26\cdot\mathrm{Re}^{0.65}\cdot\mathrm{Pr}^{0.33} \qquad (2)$$

for the determination of heat transfer for the individual row of a bundle. This correlation coincides with the given in [17] correlation. However as a difference from [17], the dependence (2) in the work [13] is recommended for the turbulent flow with indication of Re and Pr numbers diapasons Re_f=(10^3…10^5) и Pr_f=(0,7…500). Also is recommended somewhat less, than in [17] correction coefficient, taking into account the changing of heat transfer for second row of a bundle equal to 0.8. Correction $e_s = \sigma_2^{-0.15}$ on the influence of relative placement of tubes in deep rows has been taken by us equal to unity by the same reasons which was mesented above.

In the textbook [2] was proposed dependence (2) for the determination of the mean heat transfer coefficient for the bundle as a whole. That dependence of accounting of changing of heat transfer intensity in outer rows. The correction factor $e_s = \sigma_2^{-0.15}$ at coincides with given in [17] for individual row [Re_f=(10^3…10^5)]. As a difference from [17] is an absence the influence of relative placement of tubes in deep rows is introduced without indication of the diapasone of relative cross steps σ_1 which characterises the bundles for which it was obtained (as was done in [17]). For analysed by us case this factor is equal e_s = 0.98. Prandtl number is defined bundle depth distance temperature of flow; the temperature factor equals $(\mathrm{Pr}_f/\mathrm{Pr}_w)^{0.25}$.

Textbook [21] authors propose to perform heat transfer calculation by the dependence (2), which was obtained by V.P.Isachenko et. al [17] and was recommended for the heat transfer coefficient for each separate row. Mean liquid temperature is used as characteristic. As in [2] correction $e_s = \sigma_2^{-0.15}$ is entered but witout indication the range of σ_1. According to data [21] the correlation for first row heat transfer for $\mathrm{Re}_f>4\cdot10^4$ and with higher exponent at Re: C=0.0266, m=0.80, n=0.33 [21]. According to early data [6] at Re~($5\cdot10^3$…$5\cdot10^4$), C=0.22, m=0.60, n=0.35.

Correlation [10] is proposed for usage in a rangle Re_f=(10^3…10^5) also in the work [18]. As in [13, 17], the calculation is proposed for individual rows of a bundle. As in [2, 21] correction $e_s = \sigma_2^{-0.15}$ has been introduced without indication of a range of σ_1. It were indicated correction coefficients for outer rows; their values concides with same ones in [17], but they are different from values of [13]. In this work is confirmed, that the flow around first row tubes do not differ practically from flow around single tube. For whole bundle is recommended the averaging the separate heat transfer coefficient, using formula (1). As a conditioning one is used mean flow temperature.

The textbook [22] has its own specific features – the difference from [2, 17, 18, 21] in coefficient e_s – it is recommended as $e_s = \sigma_2^{-0.16}$. However for our case the value of such correction practically the same as recommended ones in [2, 17, 18, 21], that is $e_s \sim 0.98$.

In [24] formula (3) is proposed to use for calculation of α for depth row,

$$Nu = 0.27 \cdot \mathrm{Re}^{0.65} \cdot \mathrm{Pr}^{0.36} \quad (3)$$

with multiplication by the coefficient c_z, accounting dependence of average heat transfer on number of rows. Multiplier in (3) is equal to 0.27, which higher, that in above considered works. In our case the increase will be equal to 35 %. The exponent at Pr in [24] equals to n=0.36, but at Re the exponetu is less (m=0.63). Recommended Re range is $10^3 \ldots 2 \cdot 10^5$.

In [24] it was marked, that for the majority of bundles the heat transfer stabilization begins from third-fourth rows. In [25] opinion offered that for the stabilized flow the heat transfer of depth rows increase with longitudinal pitch decrease (the increase 30…70 % is depended on longitudinal pitch).

For the mean heat transfer intensity in depth rows in [26] was recommended correlation (3). The recommended range of Reynolds number was expanded up to $2 \cdot 10^3 \ldots 2 \cdot 10^5$. It was marked, that in the first row the tubes are in conditions, similar single tube only if pitch is sufficiently large. In our case ($\sigma_1 = s_1/d = 1.14$ and $\sigma_2 = s_2/d = 1.11$) and correlations for a single tube in principle are not valid.

Finally we came to a conclusion, that mostly close to realised in MSR geometrical and regime parameteres are recommendations of Zhukauskas group [3, 24] – formula (4)

$$Nu = 0.308 \cdot \mathrm{Re}^{0.63} \cdot \mathrm{Pr}^{0.36} \quad (4)$$

which was obtained by data genelalization for the bundles with $\sigma_1 = (1.2 \ldots 1.4)$, $\sigma_2 = (1.09 \ldots 1.12)$ – that is mostly close to our case $\sigma_1 = 1.14$, $\sigma_2 = 1.11$. It should be marked, that the recommended in [3] value of C=0.308 leads to increase at ~14 % the heat transfer coefficient in comparison with works, where C=0.27 [20, 24, 26].

In fig. 4 are shown the heat transfer coefficients for flow of heated steam in stage of MSR SPP-500 in correspondence with [1-3, 5, 6, 9-14, 16-26].

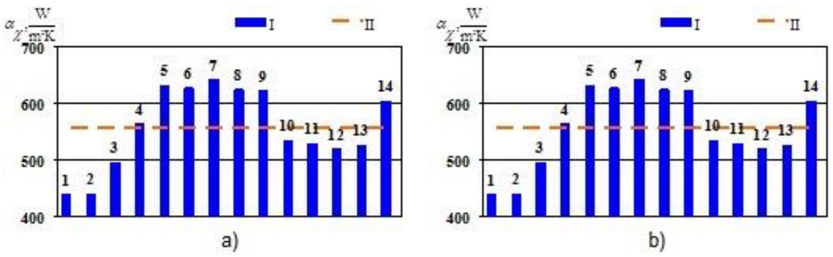

Figure 4. Heat transfer coefficient by flow of heated steam in the I (a) and II (b) stage of superheater: I–calculated values α_χ, index $\chi = 1 \ldots 14$ marks specific meanings of recommendations with correspondence to indexes and recommendations 1–Margulova T.H. [5], 2–Kutateladze S.S., Boryshanskii V.M. [6], 3–Andreev P.A.; Kirillov P.L.; Migai V.K. [10–14], 4–Budov V.M. [16], 5–Isachenko V.P. [17], 6–Kirillov P.L.; Isachenko V.P. [13, 17], 7–Isachenko V.P.; Baskakov A.P. [2, 17], 8–Isachenko V.P.; Baskakov A.P.; Kirillov P.L. [2, 17, 21], 9–Nashekin V.; Ilchenko O. [18, 22], 10–Mikheev M.A., Mikheeva I.M. [19], 11–Zhukauskas A., Ulinskas R. [3, 24], 12–Zhukauskas A.A.; Kutateladze S.S. [24, 26], 13–Zhukauskas A.A.; Tsvetkov F.F., Grigoriev B.A. [24, 20], 14–Zhukauskas A., Ulinskas R. [3]; II–averaged calculated value.

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After in-line approximation let us consider staggered and mixed approximation. As already was marked, data for staggered bundles of interesting for us geometry are few. In the work [27] have been investigated regular bundle, that is with the equal relative pitches ($\sigma_1 = \sigma_2 = \sigma$) rows $\sigma = 1{,}05$ and $\sigma = 1{,}1$. These data have been compared with data for regular staggered bundles ($\sigma = 1.027...1.19$) presented in [28-31]. It was confirmed in [27], that staggered bundles with $\sigma = 1.1...1.2$ are optimal in sense of maximum of heat transfer intensity.

Authors [27] generalized their experimental data and results of other works with a correlation for regular staggered bundles:

$$Nu_f = 0.36\cdot\{1+0.25\exp[-100(\sigma-1.18)^2]\mathrm{Re}_f^{0.6}\cdot\mathrm{Pr}_f^{0.36}\} \tag{5}$$

We have used the formula (5) and have received a Nusselt number for the first stage of MSR-500 $\mathrm{Nu}_{SP\,CI} = 281$ and for the second stage $\mathrm{Nu}_{SP\,C\,II} = 304$ that is practically the same as ones which have been obtained for in-line bundle formula (4) usage (for the first stage $\mathrm{Nu}_{SP\,C\,I} = 283$ and for the second stage $\mathrm{Nu}_{SP\,C\,II} = 303$). So we observe negligible small (less than 1 %) difference between the values, obtained by consideration of a tight bundle in-line and staggered approximations.

For actual mixed in-line staggered geometry we have taken meanings $\mathrm{Nu}_{SP\,MIX\,I} = 281$ and $\mathrm{Nu}_{SP\,MIX\,II} = 303$, as a mean values between in-line and staggered values. With such approach the heat transfer coefficients are $\alpha_{SP\,MIX\,I} = 458$ W/(m^2 K) and $\alpha_{SP\,MIX\,II} = 605$ W/(m^2 K), and calculated values of heat exchange surfaces are $H_{CALE\,MIX\,I} = 1423$ m^2, $H_{CALE\,MIX\,II} = 929$ m^2. The design constructive values of these surface areas were $H_{DES\,I} = 1920$ m^2, $H_{DES\,II} = 1218$ m^2 [1]. These meanings are 30 % higher. Usual values are 10-15 %.

1. Thermal calculations of superheater type of MSR-500 have next elements of the novelty in comparison with used during creation MSR-500 methodics:

1.1. The taking into account the variety of α and difference in areas of tubes each row by the definition of a mean heat transfer of a bundle.

1.2. Taking into account the temperature factor.

1.3. Introduction of corrections accounting the dependent of averaged het transfer coefficient on the quantity of row.

1.4. Comparison of the correlations, recommended by many authors for “corridor” and “chess” bundles and choice the proper correlation for actual geometry of the bundle in apparatus of MSR-500 type (mixed “corridor-chess” type).

2. As in the considered by us construction the number of rows in spiral tube coils is not a large one it is unproper to neglect the income of outer (first) rows in heat exchange, moreover that in spiral coil outer rows consists of the larger part of the heat transmission surface.

3. The difference in the quantative characteristics of a heat transfer, obtained by proposed by different authors, quite significant (up to 40 %). It means, that the conduction of additional experimental studies with real geometries of coils and preferable with steam of real parameters very desirable.

4. For conditions of the heat exchange in the superheater of SPP-500 type for the heat transfer calculations for side of heated steam (with actually defines overall heat transfer coefficient) can be recommended the correlation (fig. 5)

$$Nu_f = 0.308\cdot\mathrm{Re}_f^{0.63}\cdot\mathrm{Pr}_f^{0.36}(\mathrm{Pr}_f/\mathrm{Pr}_w)^{0.25} \tag{6}$$

5. Calculation results testify about possibilities to reduce a reserve of heat exchange surface in MSR-500 comparatively with designed one.

6. It looks as not justified the conducted in works [2-9, 11-18, 21, 22, 24] expansion of application boundaries of V.P.Isachenko [17] correlation beyond the boundary of a range of bundles studied.

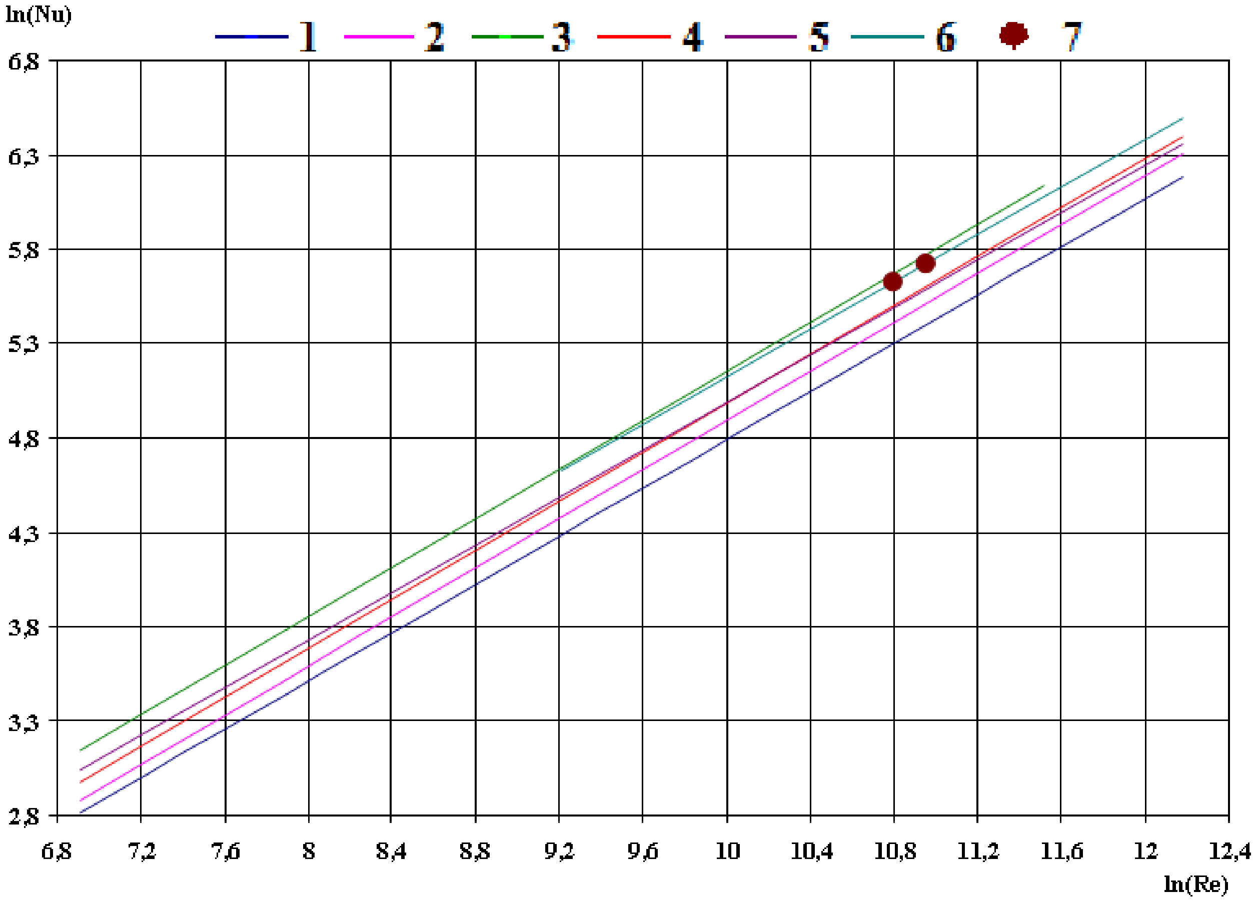

Figure 5. Non-dimensional heat transfer used for calculations: 1–$0.2c_z\,Re^{0.64}\,Pr^{0.35}$ without temperature factor, Re>$6\cdot10^3$; 2–$0.2c_z\,Re^{0.65}\,Pr^{0.33}(Pr_f/Pr_w)^{0.25}$ for bundle, Re=($10^3..2\cdot10^5$); 3–$0.26c_z\,Re^{0.65}\,Pr^{0.33}(Pr_f/Pr_w)^{0.25}$ by rows, Re=($10^3…10^5$); 4–$0.22c_z\,Re^{0.65}\,Pr^{0.36}(Pr_f/Pr_w)^{0.25}$ by rows, Re>10^3; 5–$0.27c_z\,Re^{0.63}\,Pr^{0.36}(Pr_f/Pr_w)^{0.25}$ for bundle, Re=($10^3…2\cdot10^5$); 6–(6) for bundle, Re=($10^4…2\cdot10^5$), σ_1=(1.2…1.4), σ_2=(1.09…1.12); 7–calculated values of I, II stages of SPP-500 type superheater.

Acknowledgments

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Conclusions

There are different ways for the improving of the heat transfer effectiveness of a MSR’s superheater: the usage of coiled type tube bundles, finnings and artificial roughnsses.

In the given paper are presented the methods of a calculation of the heat transfer of coiled tube bundles of a specific geometry (intermediate between corridor and chess type), based of the analysis of numerous experimental correlations by different authors.

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