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INFLUENCE OF THE YAW ANGLE ON HEAT TRANSFER AND PRESSURE DROP OF HELICAL TYPE HEAT EXCHANGERS

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Abstract

The influence of the flow angle φ between the tube axis and the flow direction on heat transfer and pressure drop has been studied at $15° \le \varphi \le 90°$ (cross flow). The Reynolds number was varied in the range $2 \cdot 10^3 \le Re \le 10^5$. Experimental data are given for a number of tube banks of straight tubes as well as a helical type heat exchanger with a large thread pitch of the tubes.

1. INTRODUCTION

Thermalhydraulics of helical type heat exchangers normally are calculated by using the relations for cross flow. The exactness is sufficient for steam generators with a small thread pitch of the tubes if the tubes are coiled all in the same direction. In other case when coiling direction is different for neighboured tube cylinders one have to reduce the values for heat transfer and pressure drop. But how much? This question, of course, arises as greater the thread pitch of the tubes is. For example, the effect of yaw must not be neglected for a helium/helium intermediate heat exchanger the tubes of which are contrariwise coiled and having a thread pitch of 25°. For oblique flow only insufficient and partly inconsistent data are to be found in the literature. For instance there is no reference to the influence of the tube pitches and tube arrangement (in-line or staggered) on heat transfer and pressure drop of yawed tube bundles. Furthermore there is almost no information about the range of Reynolds number for which the data referred to are valid.

The influence of the flow anlge on the flow over a tube bundle can be investigated succesfully only by experiments. Therefore tests were carried out at a number of in-line and staggered tube arrangements with different tube pitches. The angle φ (see Fig. 1) between the flow direction and the tube axis was varied in the range $15° \le \varphi \le 90°$ (cross flow). Straight tubes were used for this fundamental part of work. Finally results were proved by testing a helical type heat exchanger. The investigation of this large and expensive component was supported by the firms L. & C. Steinmüller, Sulzer Brothers and Interatom.

Although the influence of oblique flow is a field for experiments a hint to its effects can be obtained from the boundary layer equations for the inclined cylinder. The flow pattern around a yawed cylinder is a three dimensional one. However, a theoretical estimate is essentially simplified by the fact that the flow is independent of the axial extension of the cylinder. As a consequence the boundary layer equations allow to calculate the velocity component normal to the tube axis independent of the axial velocity component. This fact is called the independene principle. The boundary layer equations for the component normal to the tube axis are the same as those for a cylinder in cross flow. From the independece principle follows, for example, that the point of boundary layer separation of a yawed cylinder should be uneffected by the axial velocity component /1/. In the same way the pressure distribution around the cylinder should not be influenced by the flow angle. A main feature of this paper is to answer the question if the principle of independence is appropriate to describe the thermalhydraulic behaviours of inclined tube bundles. Experiments should clarify down to which flow angle the flow through an inclined tube bundle can be related to a bundle in pure cross flow. For that reason the velocity component normal to the tube axis is used to describe the flow through the inclined tube bundle in this paper.

2. EXPERIMENTAL SET UP

The experiments at the straight tubes were designed as to represent the flow conditions of tube bundles in the subcritical Reynolds number range $2 \times 10^3 \le Re \le 10^5$. The Reynolds number as well as the hydraulic resistance

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coefficient ζ are based on the velocity component u normal to the tube axis in the narrowest cross section. The relation between u and the velocity in the empty channel u_0 is $u/u_0 = S_T/(S_T-1)$. For S_T see Fig. 1. As the characteristic length in the Reynolds number as well as the Nusselt number Nu the tube diameter was chosen. Air at ambient conditions was used as fluid. The dimensions of the test section were determined by the data of the blower available and the necessity to arrange tube bundles with a representative number of tubes across as well as along the flow direction. The test set-up is sketched in Fig. 1. The cross section of the channel was 300 times 300 mm^2, the tube diameter 25 mm⌀. The dimensions chosen allowed to arrange 8 tubes per row normal to the flow direction at a transversal tube pitch of S_T = 1.5 and 9.5 tubes at S_T = 1,25. In flow direction 10 rows were used. Pressure holes in front of and behind the tube bundle were used to determine the pressure loss over the bundle.

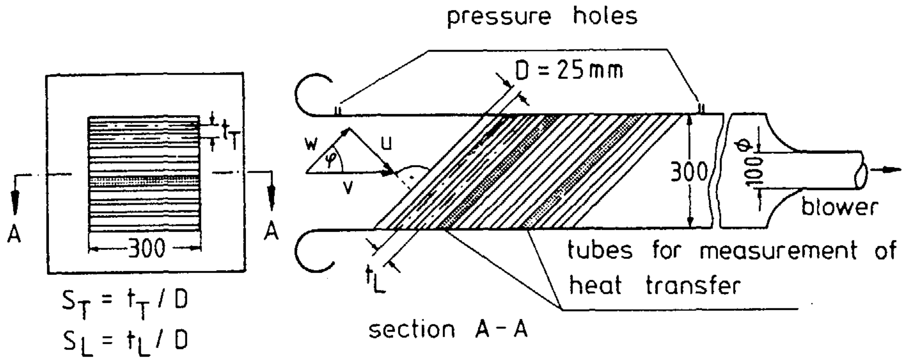

Fig. 1: Experimental setup

For heat transfer measurements electrically heated tubes were applied. A coax heater was inserted into a copper rod. It was pressed into one of the steel tubes of the bundle. Thus the test tube had the same surface conditions as the remainder dummy tubes. The temperature of the test tube was measured at the inner wall of the steel tube because of the tube diameter was relatively small and the surface of the tube should not be hurt. The decrease in temperature through the tube wall was calculated. The temperature conditions were similar to those of tubes through which water runs.

Three thermocouples were distributed over the active length of 0.1 m of the heater and four over the circumference. The heaters were threedivided. The outer parts acted as guard heaters while the inner piece was representing the active element. The particular sections were thermally insulated from each other and separately heated.

The helical type heat exchanger tested was a cut-out of a helium/helium-intermediate heat exchanger. It consisted of 252 tubes of a diameter of D = 22 mm⌀ coiled on seven tube cylinders between 612 mm⌀ and 1116 mm⌀ in diameter. The coiling direction was contrariwise for neighboured tube cylinders. The thread pitch of the tubes was 25° (φ= 65°). The bundle had tube pitches of S_T = 1.64 D and S_L = 1.48 D (definition as in Fig. 1). 45 tube rows were arranged in flow direction. The heat exchanger was tested in a high pressure wind tunnel. Air and helium of 200 °C at a pressure of up to 40 bars were used as flow mediums outside the tubes. Inside ran water.

3. INCLINED TUBE BANKS WITH PARALLEL TUBE LAYERS

3.1 Heat Transfer

The overall heat transfer data were taken from the eighth row of the ten row deep bundle. So the data should be representative also for larger tube bundles. Fig. 2 shows the dimensionless heat transfer coefficient in dependence on the Reynolds-number for an in-line tube bank with the tube pitches S_T = 1.5 and S_L = 1.5. The flow angle φ is parameter in the representation. The diagram reveal a good agreement between the data for cross flow and oblique flow. That means that the heat transfer data of yawed tube bundles can be calculated in the subcritical Reynolds number range by means of the relation for cross flow tube arrangements if the velocity component normal to the tube axis is used for the Reynolds number.

This result was obtained for all tube arrangements tested /2/. As an second example Fig. 3 shows the data for a tube bank with an extremely large longitudinal pitch. When regarding the heat transfer data for different flow angle one has to account that the Reynolds number is formed with the velocity component normal to the tube axis. With respect to a constant mass flow rate the heat transfer decreases with smaller flow anlge. The same is valid for the flow resistance considered below.

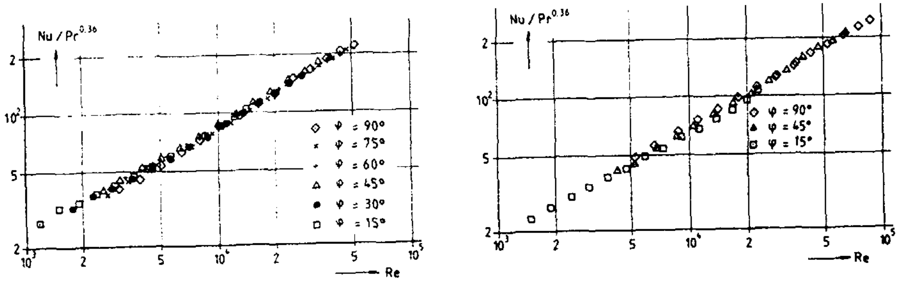

Fig. 2: in-line, S_T=1.5, S_L=1,5 Fig. 3: staggered, S_T = 1.25, S_L =3.25

Fig. 2 and 3: Heat transfer of inclined tube banks at several tube arrangements

3.2 Hydraulic Resistance

In opposite to the heat transfer data the description of the pressure drop of yawed tube banks is not sufficiently when related to the velocity resp. the dynamic pressure component normal to the tube axis, Fig 4 and 5. The representation doesn't consider the frictional part of the flow resistance which increase with smaller flow angle because of the velocity parallel to the tube axis becomes larger (~cos φ) and the tube length increase (~1/sinφ). The increase of the frictional part of the flow resistance with smaller flow angle leads to an increase of the hydraulic resistance coefficient in the representation chosen. Assuming nearly the same frictional flow resistance for all tube bundles tested the increase of the overall flow resistance coefficient in dependence on the flow angle is expected to be the greater the smaller the flow resistance at cross flow is. Accordingly the measured data for the in-line tube bundle with the tube pitches S_T = 1,5; S_L = 1,5 reveal clearest the influence of the flow angle on hydraulic resistance, Fig. 4. Besides the increase of the absolute value of the hydraulic resistance coefficient the dependence on the Reynolds number becomes more evident for smaller flow angle. The Reynolds number decreases for which the hydraulic resistance coefficient becomes dependent on the Reynolds number. The smallest influence of the flow angle on the hydraulic resistance coefficient was observed for the staggered tube arrangement with the tube pitches S_T = 1,25; S_L = 3,25. With the exception of the in-line tube bank of Fig. 4 the change in the slope of the curves between the cross flow data and the values for φ = 15° is the same for all tube bundles tested.

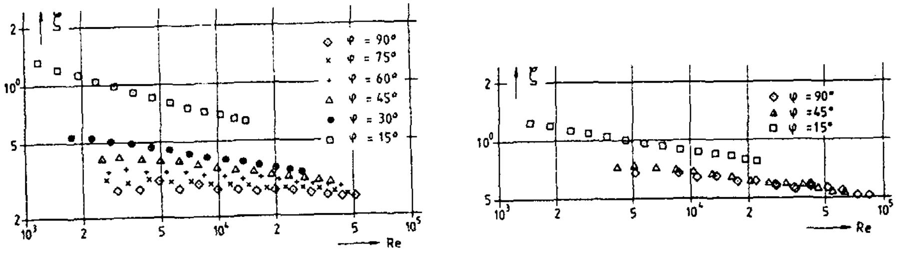

Fig. 4: in-line, S_T=1.5, S_L = 1,5 Fig. 5: staggered, S_T = 1.25, S_L=3.25

Fig. 4 and 5: Hydraulic resistance of inclined tube banks at several tube arrangements

3.3 Comparison with Literature

In the literature the influence of the flow angle on heat transfer and pressure drop is accounted for by factors like $\alpha_\varphi / \alpha_{90°}$ = f (φ) resp. $\Delta p_\varphi / \Delta p_{90°}$ = g (φ) without any regard to the Reynolds number. That means for the application of such factors to the experimental data of this paper that curves through the data points for all flow angle have to be parallel. This premise is given for the heat transfer data. Accordingly a good agreement to predictions of other authors /3/ can be expected, Fig. 6. The heat transfer coefficient of a tube bundle with axial flow is also given. The value was calculated with Pressers's /6/ prediction. An approximation between the heat transfer coefficient for a yaw angle of φ = 15° and axial flow should be possible without a significant error. From the experimental result a factor for the heat transfer of yawed tube bundles can be deduced as follows. Using the fact that the dimensionless heat transfer coefficient is nearly independent of the flow angle and introducing the usual power relation for pure cross flow

$$Nu = C\,Re^{d} \qquad C,d \text{ constants}$$

as well as the relation between the velocity components normal to the tube axis u and in main flow direction v one obtains

$$\alpha_{\varphi}/\alpha_{90°} = (\sin\varphi)^d$$

The equation (full line in Fig. 6) is in excellent agreement with the experimental data down to φ = 15°. However, it doesn't describe the transition to longitudinal flow.

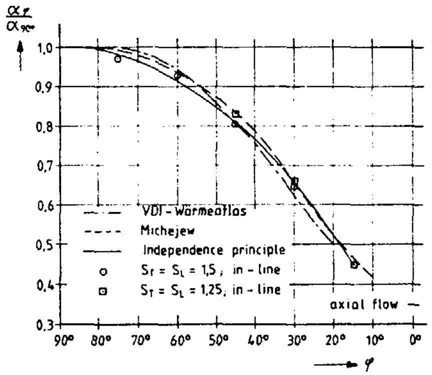

Fig. 6:
Comparison of experimental heat transfer data with several predictions

Ratios for the pressure drop of yawed tube banks are given by Michejew /3/ Idelchik /5/ or the VDI-Wärmeatlas /4/. However, the change of the slope of the curves for the hydraulic resistance coefficient in dependence on the flow angle indicates the restricted usefulness of such partition factors. This will be obvious by the comparison of the predictions /3, 4, 5/ with the experimental data of Fig. 4 at two different mass flow rates, Fig. 7. A constant mass flow rate is described by a constant Reynolds number Re_v which is formed with the velocity v in main flow direction. Re_v is connected with Re by the ratio $Re/Re_v = \sin\varphi$. The difference between the partition factors at the two different mass flow rates arises with smaller flow angle. It is about 80 % at φ = 15° if related to the value at $Re_v = 5 \times 10^4$. The deviation from Michejews /7/ prediction is + 55 % resp. - 16 %.

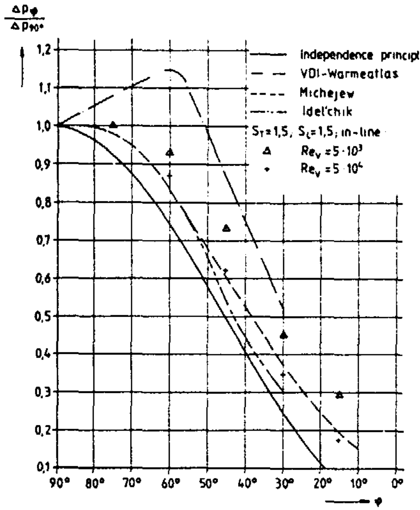

Fig. 7: Influence of Re-number

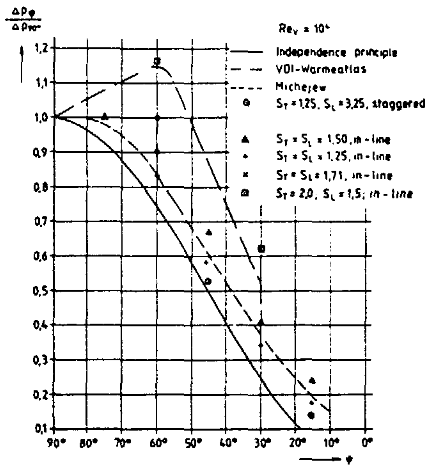

Fig. 8: Influence of tube pitches

Fig. 7 and 8: Comparison of experimental pressure drop data with several predictions

And there is a second reason which speaks against such simple relations for $\Delta p_\varphi/\Delta p_{90°}$ as given in /3, 4, 5/. They don't consider the tube pitches. In Fig. 8 predictions mentioned are opposed to the measured data. For the in-line tube bank S_T = 2.0 S_l = 1.5 the pressure drop increase when the flow angle decrease. This is in accordance with the prediction given by the VDI-Wärmeatlas. However, for the staggered tube arrangement S_T = 1.25, S_L = 3.25 the pressure drop is nearer to the prediction of Michejew. The fact is that the influence of the flow angle on the hydraulic resistance is different. It depends strictly on the pressure drop of the cross flow tube arrangement having the same tube pitches as the inclined tube bundle has.

4. IN-LINE TUBE BANKS WITH CROSSED TUBE LAYERS /8/

If neighbouring tube layers of a tube bank are contrariwisely inclined the flow through the tube arrangement changes definitely. For the parallel tube arrangement the maximum velocity u exists over the whole width of the channel. Therefore the velocity in each point of the cross section will be directed parallel to the main flow direction. For crossing tube layers the narrowest cross section exists only at the crossing points of the tubes.

Between the crossing points the cross section is much larger. The fluid will prefer this section to minimize its pressure loss and pass the bundle on a way as sketched in Fig. 9. The smaller the transversal tube pitch the stronger the zigzag way of the flow. That means that the maximum velocity u in the definition given above is not suitable to characterize the flow through tube banks with crossed tube layers. Nevertheless it will be maintained in the following diagrams for comparison with data of tube banks with parallel tubes.

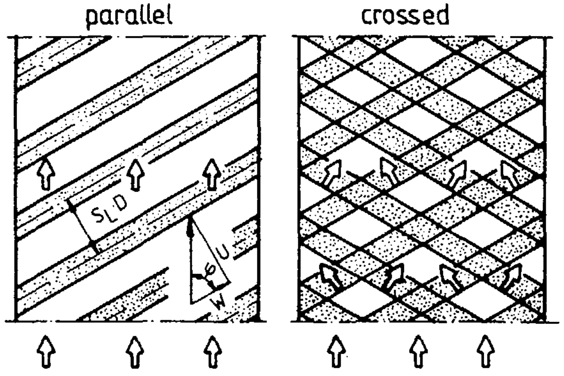

Fig. 9: Sketch of flow through yawed tube banks with parallel and crossed tube layers

From the flow sketch in Fig. 9 it can be expected that heat transfer and pressure drop are smaller for crossed tube layers than for parallel tube arrangements. The difference should be depended on the flow angle as well as on the transversal tube pitch.

The influence of the flow angle on the thermalhydraulic behaviour of a crossed tube arrangement can be seen from Fig. 10. Data are given for the same tube pitches S_T x S_L = 1.5 x 1.5 for which Fig. 2 shows the results when tubes are parallel arranged. In opposition to the parallel tube arrangement the heat transfer data don't agree for different flow angle. With smaller flow angle the heat transfer coefficient decreases. The diminution is about 12 % for φ = 60° and 24 % for φ = 30°. A more significant difference is obtained for the hydraulic resistance. When the tubes are parallel the ζ-value for $\varphi < 90°$ is greater than the cross flow value. For the crossed tubes the hydraulic resistance coefficient decreases with smaller flow angle. Furthermore, the dependence on the Re-number is stronger. The ζ-value for φ = 60° is about 55 - 65 % and for φ = 30° about 40 - 50 % of that of the parallel tube arrangement. A usual efficiency coefficient for heat exchangers is defined as the ratio of transferred heat to the expended pumping power. This efficiency coefficient is considerably greater for the crossed tube layers than for the parallel tube arrangement. The improvement reaches its maximum at φ = 45° with $\varepsilon_{crossed}/\varepsilon_{parallel} \approx 1.8$. But also for the more practicable flow angle of φ = 60° the ratio is yet 1.5. The thermalhydraulic advantages of crossed tube arrangements increase with decreasing transversal tube pitches. As an example Fig. 11 shows the data for the tube pitches S_T x S_L = 1.25 x 1.25 at a flow angle of φ= 60°. The corresponding data of the parallelly arranged tubes agree with the cross flow data which are shown in Fig. 11, too. The percentage diminution of heat transfer is nearly the same as for the tube pitches 1.5 x 1.5. The

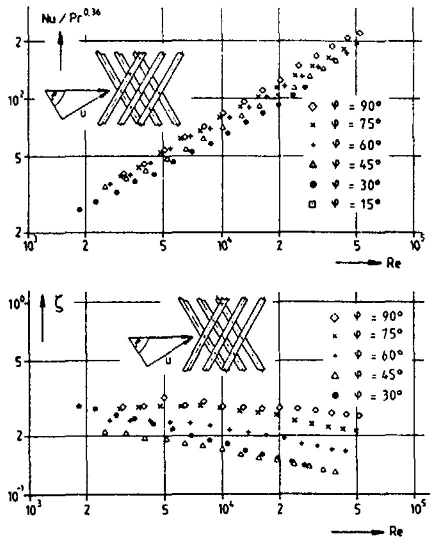

Fig. 10: S_T = 1.5, S_L = 1.5

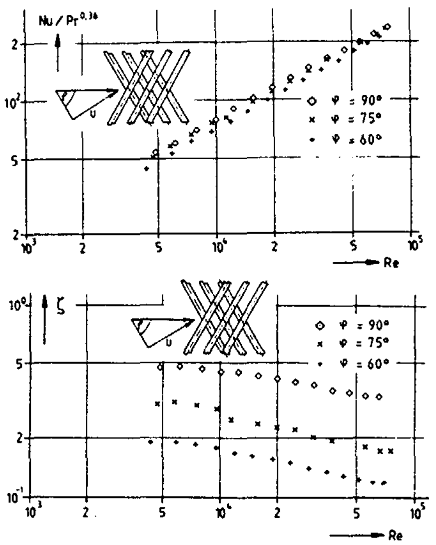

Fig. 11: S_T = 1.25, S_L = 1.25

Fig. 10 and 11: Heat transfer and pressure drop of yawed tube banks with crossed tube layers hydraulic resistance coefficient, however, is only about one third of that of the parallel tube arrangement. This means, that the efficiency coefficient is greater by a factor of 2.5 for the crossed tube arrangement than for the same bank with parallely arranged tubes.

5. HELICAL TYPE HEAT EXCHANGER /9/

As described in chapter 2 the helical type heat exchanger tested is characterized by a large thread pitch of the tubes of 26° and by its contariwise coiling of neighboured tube cylinders. Heat transfer and pressure drop were measured in the range $2.5 \cdot 10^3 \leq Re \leq 5 \cdot 10^5$. Data are shown in Fig. 12 and 13. For comparison the heat transfer correlation of Grimison /7/ was inserted. As mentioned above the heat transfer coefficient for a tube bundle with crossed tube layers is expected to be up to 10 % below that of the cross flow tube arrangement. The difference measured is about 6 % and therefore in good agreement with the prediction.

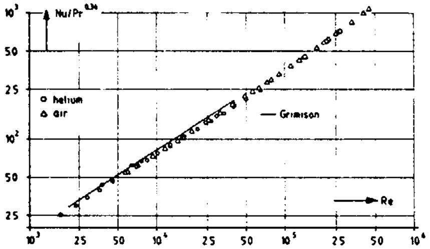

Fig. 12: Heat transfer

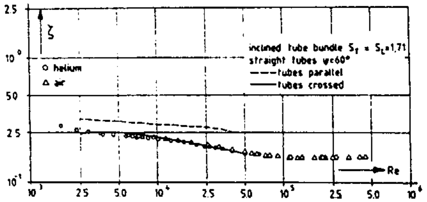

Fig. 13: Hydraulic resistance

Fig. 12 and 13: Helical type heat exchanger $S_T = 1.64$, $S_L = 1.48$

A valuation of the pressure drop data is more circumferential because we didn't test a tube arrangement of straight tubes at different flow angles before having the same tube pitches as the helical type heat exchanger had. On the other hand we had learned that the diminuation of the pressure drop by crossing the tube layers is decisively influenced by the pressure drop value of the cross flow tube arrangement. But we had tested a tube bank with $S_T = S_L = 1.71$. Following Grimisons's recommandation this tube bank has the same flow resistance as a heat exchanger with the tube pitches of our helical tube bundle has. So we chose this tube bank for comparison. The prediction is in very good agreement with the experimental data, Fig. 13.

6. CONCLUSION

At subcritical Re-numbers the independence principle is appropriated to describe the heat transfer of inclined tube banks down to an angle beetween the flow direction and the tube axis of $\varphi = 15°$. That means that the cross flow relations can be used if the velocity component normal to the tube axis is taken for the Re-number. The indepence principle is unsuitable for describing the hydraulic resistance of inclined tube banks. With respect to the dynamic pressure component normal to the tube axis the hydraulic resistance coefficient increases with smaller flow angle. The usefulness of predictions found in the literature is restricted.

Contrariwise inclination of neighboured tube layers of in-line tube banks lead to a small diminution in heat transfer while the hydraulic resistance decreases substantially. The thermalhydraulic advantages become more evident with smaller transversal tube pitch.
As shown at a helical type heat exchanger with a large thread pitch of the tubes the data collected for banks of straight tubes are transferable without any restriction.

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BOILER REFERRULING ON THE HARTLEPOOL AND HEYSHAM 1 ADVANCED GAS-COOLED REACTORS

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Abstract

The Hartlepool and Heysham I reactors each use eight cylindrical boilers having nineteen rows of helical tubes. The advantages of this design are partially offset by the relatively poor radial gas mixing. Some rows of tubing may have an imbalance between heat imput from the gas and the flow of feedwater, causing a temperature profile at the upper transition joints. The thermal/hydraulic behaviour meant that the metallurgical constraints limited output.

Analysis of the behaviour of these boilers required a new two-dimensional mathematical model, known as PODMIX. This describes the thermal hydraulics in each of the rows of tubing and also in the gas between the rows. Not all of the parameters for the model can be determined from first principles. However, two out of the thirty two pods have thermocouples at some of the upper transition joints and these made back calculation possible.

In order to translate this model to other boiler pods, a novel thermocouple rake system was designed for sampling superheated steam temperatures in selected tubes.

A result of this analysis was to show that different, individual ferrule patterns were needed for each pod. The characteristics could, in general, best be met using twin orifice ferrules.

Unfortunately, the installed system did not permit the replacement of orifices, so that a completely new system had to be developed. In the course of designing this, the opportunity was taken to over come susceptibilities to erosion/corrosion and crevice corrosion.

Removal of the old ferrules and replacement with the new ones necessitated the development of high precision, programmable machines to operate under difficult site conditions. These carried out drilling, boring, grinding and polishing operations as well as making face welds and tube bore welds.

Modifications have already achieved substantial improvements in performance and output, but an extended, iterative programme still lies ahead.