HEAT TRANSFER IN HELICAL TUBE SODIUM HEATED STEAM GENERATORS

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ABSTRACT

It is generally recognized that fast breeder reactors may play an important role for the energy production in the not too distant future. One of the key problems for the success of these reactors is the development of a reliable sodium heated steam generator.

For the 1200 MWe Super Phenix project, the CEA jointly with EDF, has carried out a research and development program for an helically coiled tube steam generator to be manufactured by the Fives-Cail Babcock Company.

In the 5 MW CEA test facility at Grand Quevilly we have performed an experimental program in order to gain the basic knowledge on heat transfer and pressure drop for boiling water flowing in four helically coiled tubes heated by a counter flow liquid metal, which was badly needed for the design.

For the experimental coil to tube diameter ratios of 630/20, 810/20, 1800/20, 2700/20 correlations for the predictions of the D.N.B., and the friction coefficient in the high Reynolds number range were obtained.

A comparison between experimental results and predictions at various heat ratings shows a fair agreement.

In addition it is demonstrated that experimental results compare very well with computer code predictions for a 45 MWt mock-up simulating a liquid metal heated once-through steam generator, with 24 tubes arranged in three layers, which has been tested at the EDF Renardières facility.

All correlations used in our computer code are reported in the paper.

INTRODUCTION

Super Phenix, 1200 MWe sodium cooled fast breeder power plant, is currently under development in France. This reactor will be equipped with four secondary sodium loops, each comprised of one once-through helically coiled tube steam generator.

Beginning in 1970, this steam generator concept has been developed by the manufacturer Fives-Cail Babcock jointly with CEA and EDF for commercial plant. In each unit rated at 750 MWt, the water flows upwards in a vertical helically coiled tube bundle heated by a counter flow of sodium. The 357 tubes are made of incoloy 800 and arranged in 17 concentrical layers whose coil diameters measured from tube axis to tube axis range from 1.17 m to 2.61 m, corresponding to D/d_j ratios from 58.5 to 130.5 respectively with an internal tube diameter of 0.020 m and a thickness of 2.6 x 10^{-3} m.
While helically coiled tubes appear to be attractive for compact boiler it seemed to us that there existed a lack of experimental data especially on dryout conditions with heat flux varying along the tubes length, and on the pressure losses at high Reynolds number. It was also necessary to verify the choice and the validity of heat transfer correlations retained for the heat transfer surface and partial loads calculations. In order to hit the target Fives-Cail Babcock build 4 helically coiled monotubes whose main dimension are listed in table 1; their curvature D/id ratios are 31.5 - 40.5 - 90 - 135, the smallest and the biggest diameter coil enclosed all tube layers diameters of commercial steam generators now foreseen.

| TABLE 1 |

<table>
<thead>
<tr>
<th>Helically coil diameter D</th>
<th>m</th>
<th>Tube outside diameter and thickness</th>
<th>4 helically Coiled Monotubes</th>
<th>45 MWt Steam Generator 24 tubes</th>
</tr>
</thead>
<tbody>
<tr>
<td>9.650</td>
<td>0.010</td>
<td>1.0</td>
<td>2.7</td>
<td>0.63-0.72-0.81</td>
</tr>
<tr>
<td>Tube outside diameter and thickness (material Inconel 600)</td>
<td>10^{-3} m</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>25.1 x 2.49</td>
<td>24.9 x 2.35</td>
<td>25.2 x 2.66</td>
<td>24.7 x 2.5</td>
<td>24.76 x 2.3</td>
</tr>
<tr>
<td>Curvature D/i</td>
<td>31.5</td>
<td>40.5</td>
<td>90</td>
<td>135</td>
</tr>
<tr>
<td>Tube I.D. roughness</td>
<td>10^{-6} m</td>
<td>0.51</td>
<td>1.0</td>
<td>1.79</td>
</tr>
<tr>
<td>Shell diameter and thickness (material 2.25 Cr 1 Mo)</td>
<td>10^{-3} m</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>70 x 16.3</td>
<td>70 x 16.3</td>
<td>70 x 16.3</td>
<td>70 x 16.3</td>
<td></td>
</tr>
<tr>
<td>Economiser-Evaporator length</td>
<td>m</td>
<td>48.511</td>
<td>48.298</td>
<td>48.327</td>
</tr>
<tr>
<td>Superheater length</td>
<td>m</td>
<td>18.82</td>
<td>18.798</td>
<td>18.944</td>
</tr>
<tr>
<td>Economiser-Evaporator pitch</td>
<td>m</td>
<td>0.315</td>
<td>0.408</td>
<td>0.940</td>
</tr>
<tr>
<td>Superheater pitch</td>
<td>m</td>
<td>0.191</td>
<td>0.343</td>
<td>0.600</td>
</tr>
</tbody>
</table>

We obtained a good agreement on the 4 coils in steady state conditions between the calculated and the experimental results. This leads us to compare our computer code model with results obtained on a 45 MWt steam generator mock-up comprising three concentrical layers which have 7, 8 and 9 helically coiled tubes whose main characteristics are listed in table 1. This bundle is located between two cylindrical shells and heated by sodium flowing downwards on this annular space. We shall give all heat transfer and pressure drop correlations utilized in our computer code model including the relation for predicting the dryout condition and the pressure drop obtained in the high Reynolds number range, we shall present :
- a comparison between measured and calculated temperature profiles performed at different partial loads on the 4 helically coiled monotubes and on the 45 MWt mock-up
- and also some diagrams showing a good agreement between heat transfer and pressure drop calculated and measured on the whole of tests.

TESTS FACILITIES

The 4 helically coiled tubes were tested at the CEA 5 MW Heat Transfer Facility installed at Grand Quevilly. As can be seen on figure 1, it simulates a L.M.F. R heat transfer system with a primary sodium loop, a secondary NaK loop, a tertiary water loop and a river water cooling circuit.
The 45 MWt steam generator mock-up has been tested at the EDF Heat Transfer Facility (C.G.V.S.) at Les Renardières. The schematic diagram (figure 2) shows a primary sodium loop heating a secondary water steam loop by means of the steam generator under test. The steam is depressurised and desuperheated at three levels and cooled in the condenser by a closed circuit of water equipped with two refrigerating towers.

HELCALLY COILED STEAM GENERATOR TESTS
HELCALLY COILED MONOTUBE DESCRIPTION:

The 4 heat insulated coils are installed vertically as shown on fig. 3, the 0.63 m coil being located concentrically within the 1.8 m one.
Each coil consists of two concentric tubes, the water flowing upwards in the inner tube and the NaK downwards in the annular space surrounded by the outer tube. As can be seen in Figure 4, the inner tube is centered by means of several sets of three radial fingers spaced at 0.12 m with 120° lag and welded on the outer tube, the interval from one set to the next is about 1 m along the coil length.

The 4 coils are fed in parallel on both water and NaK sides. Manually actuated throttle valves at the water inlet and at the NaK outlet allow flow control. Moreover, the water valves act as stabilizing orifices during two-phase heat transfer experiments so that hydrodynamic stability can be achieved even at low flow rates.
The NaK flow rate at the outlet of each steam generator was measured by permanent magnet e.m. flowmeters with an accuracy of $\pm 3\%$. The water flow rate was obtained from venturi nozzles calibrated at $\pm 2\%$. The absolute water and steam pressures were measured at the water inlet to header and at the steam outlet header by strain gage transducers calibrated to $\pm 1\%$. The inlet throttle valve pressure drop was metered with an error of $\pm 3\%$ by differential transformer transducers.

The fluid temperatures were measured at the inlet and outlet of each steam generator. Moreover, the NaK temperature profile was measured by chromel alumel TP 304 sheathed thermocouples located in the wall of the outside tube every two meters along each coil. The electrically insulated hot weld of the thermocouples protrudes $4 \times 10^{-3}$ m inside the NaK stream. The estimated error depends on the temperature range: $\pm 0.5\^\circ C$ from 100 to 200 $^\circ C$; $\pm 1\^\circ C$ from 200 to 400 $^\circ C$ and $\pm 1.5\^\circ C$ until 600 $^\circ C$.

For each run all measurements were recorded in five minutes by a 400 channel data logger.

- **STEADY STATE TESTS**

The hundred runs which were performed separately or simultaneously on the 4 coils, covered the following variable ranges.

- Pressure levels 45 - 70 - 100 - 135 - 175 bar (1)
- $37.5 < G < 350$ g/cm$^2$s (2)
- $31 < q$ dryout $< 150$ W/cm$^2$ (3)

The large variation of these parameters showed clearly their influence on the critical steam quality prevailing on the tube wall dryout zone and were used to look for a pressure drop correlation at high Reynolds number and to compare heat transfer predicted by our computer code with experimental results at full and partial loads.

- **45 MWT HELICALLY COILED TUBE STEAM GENERATOR MOCK-UP DESCRIPTION**

The steam generator sketch, on figure 5, shows three layers of tubes made of incoloy 800: their internal diameter is $20 \times 10^{-3}$ m with a thickness of $2.5 \times 10^{-3}$ m. The inner diameter coil is 0.63 m, the middle is 0.72 m and the outer diameter coil is 0.81 m. They have respectively 7, 8 and 9 tubes. The 24 tubes are fed by water from two manifolds with orifices located at the inlet of each tube. The water flows upwards in the bundle tubes and the steam goes out into two manifolds. Inlet and outlet temperatures on two half bundles and water flow rates on each manifold were measured. Moreover the following were measured: the 24 tube steam outlet temperatures, the 12 tube pressure drops as measured between the inlet to the steam generator and the outlet manifold and the pressure drops as measured for the two half bundles between the inlet header and outlet steam manifold.

Sodium is passed into the steam generator by two pipes opening below the free sodium level and it flow downwards in an annular space in counter-current to the water and goes out by holes through the internal pipe at the bottom. Inlet and outlet temperatures, and flow rate were measured. Moreover the sodium temperature profile was obtained from 144 thermocouples located at 9 different levels along the length and distributed on two perpendicular diameters, each radius being equipped with 4 thermocouples set at half space between tube layers.
The accuracy of the whole measures is about of the same magnitude as for the 4 helical coil experiments.

- STEADY STATE TESTS

The 40 runs have been performed under the following conditions:

Phénix's steam cycle  \( P = 164 \text{ bar} \quad T = 512^\circ \text{C} \)

range thermal rating from 20% to 100%

Super Phénix's steam cycle  \( P = 184 \text{ bar} \quad T = 490^\circ \text{C} \)

at 10% - 60% and 100% load.

An experimental static stability curve has been established for an inlet sodium temperature of 521°C and an outlet pressure of 171 bar at a thermal rating of about 60%.

EXPERIMENTAL AND PREDICTED HEAT TRANSFER AND PRESSURE DROP COMPARISONS

In order to compare experimental heat transfer performances and pressure drop with theoretical predictions, the sodium and steam outlet temperatures and steam outlet pressure were computed by our ZEBALUR Code on a CDC 7600 computer. The input data is the sodium flow rate and inlet temperature, the water flow rate and inlet temperature. Depending on the prevailing local mode of heat transfer, four zones are considered, namely:

1. subcooled water heating (economizer)
2. nucleate boiling
3. film boiling
4. steam superheat (superheater)

To achieve a sufficient accuracy, each zone is divided into 20 segments with constant fluid properties, and the calculation proceeded step by step. The properties of water and steam in S.I. units given by Schmidt [1] have been retained for the calculation.
- HEAT TRANSFER CORRELATIONS

The forced convection liquid metal film coefficient in helically coiled monotube steam generator is calculated from our correlation [2] obtained in an heat exchanger NaK to NaK with NaK flowing in an annular space

\[
Nu = (6.15 + 0.02 \, Pe^{0.8}) \, F
\]

where the Nusselt and Peclet's numbers are computed from the hydraulic diameter \( d_{h} = d'_{j} - d_{o} \) and the liquid metal physical properties are taken at the bulk average temperature. The NaK thermal conductivity as a function of temperature varies according to the following equation

\[
k = 20.255 + 0.0288 \, T - 3.018 \times 10^{-5} \, T^2
\]

\( F \) is the factor to increase the heat transfer coefficient in the thermal and hydraulic establishment length with

\[
\frac{Pe \, dh}{L} \geq 28 \quad F = 0.194 \left( \frac{Pe \, dh}{L} \right)^{0.49}
\]

and

\[
\frac{Pe \, dh}{L} < 28 \quad F = 1
\]

where \( L \) is the length from inlet sodium.

The forced convection liquid metal coefficient in the 45 MWt steam generator mock-up is calculated from the Dwyer correlation [3]

\[
Nu = 1.086 \left( 5.44 + 0.228 \, Pe^{0.614} \right) \left( \frac{\sin \theta + \sin^2 \theta}{1 + \sin \theta} \right)^{1/2}
\]

\( 0 < \theta < 90^\circ C \)

The film coefficient for water-forced-convection heat transfer is calculated from Mori and Nakayama [4] correlation

\[
Nu = \frac{1}{41} \, Re^{5/6} \, Pr^{0.4} \left( \frac{d_{i}}{D} \right)^{1/12} \left( 1 + \frac{0.061}{Re \left( \frac{d_{i}}{D} \right)^{2.5}} \right)^{1/6}
\]

\( d_{i} \) is the inside diameter tube, \( D \) is the coil diameter, the water physical properties are taken at the bulk average temperature.

In the nucleate boiling zone we use A. Owhadi, K.J. Bell and B. Crain's experimental results [5], where the heat transfer coefficient is correlated with Lockart-Martinelli's parameter \( 1/X_{tt} \) but at low steam quality, namely for \( 1/X_{tt} \) below 2, the discrepancy is large. Therefore it seems to us we should have a better accuracy by using the Rosenhow correlation [6] although this does not take into account the centrifugal effect which should be very low in this nucleate boiling area. In this way we discerned 2 zones:

a) Zone \( 1/X_{tt} < 2 \)

In the nucleate boiling zone, the Rosenhow's method is applied; the water-side heat transfer coefficient results from the addition of a boiling coefficient \( h_{b} \) that are expressed as follows:
\[ h_b = \frac{\mu_L \Gamma}{\left[ \sigma/g \left( \rho_L \rho_v \right) \right]^{1/2}} \left( \frac{C_L}{0.013 \Gamma \Pr_L^{1.7}} \right)^{3} \left( \frac{t_w - t_{\text{sat}}}{t_w - t_b} \right)^{3} \] (10)

and

\[ h_c = 0.019 \frac{k_L}{d_i} \left( \frac{G d_i}{\mu_L} \right)^{0.8} \Pr_L^{1/3} \] (11)

where the following properties we taken at saturation temperature \( t_{\text{sat}} \) for the prevailing pressure:

- \( \mu_L \) = dynamic viscosity of water
- \( \Gamma \) = latent heat of vaporisation of water
- \( \rho_L, \rho_v \) = specific mass of water and steam
- \( C_L, C_v \) = specific heat of water and steam
- \( \sigma \) = surface tension of water
- \( k_L \) = thermal conductivity of water
- \( \Pr_L \) = Prandtl number of water
- \( g \) = gravity acceleration
- \( G \) = mass flux
- \( d_i \) = tube internal diameter
- \( t_w \) = wall temperature
- \( t_b \) = fluid bulk temperature

b) Zone \( 1/X_{tt} > 2 \)

The two-phase heat transfer coefficient in the nucleate boiling zone is computed from the Ohwadi, Bell and Crain experimental results [5] which can be fairly well fitted by the equation:

\[ \frac{h_{\text{TPF}}}{h_L} = A \exp \left( 4.436 - 29.72X_{tt} + 141.237X_{tt}^2 - 325.34X_{tt}^3 + 272.58X_{tt}^4 \right) \] (12)

where

\[ X_{tt} = \left[ \frac{1 - \rho_v}{\rho_L} \right]^{0.9} \left( \frac{\rho_v}{\rho_L} \right)^{0.5} \left( \frac{\mu_L}{\mu_v} \right)^{0.1} \] (13)

\[ A = 1 \text{ when } 0.05 < X_{tt} \] (14)

\[ A = \left( \frac{D}{d_i} \right)^{0.25} \text{ when } X_{tt} \leq 0.05 \] (15)

The liquid heat transfer coefficient is computed from:

\[ h_L = 0.023 \frac{k_L}{d_i} \text{Re}^{0.85} \Pr_v^{0.4} \left( \frac{d_i}{D} \right)^{0.1} \] (16)
The water physical properties are taken at saturation temperature.

The departure from nucleate boiling prevails when the steam quality becomes equal to its critical value.

\[ x_{\text{crit}} = 1.39 \times 10^{-4} \quad Q^{0.732} \quad G^{-0.209} \quad \exp 0.00246 \quad P \]  

where \( Q \) = the heat flux, in W/m\(^2\)
\( G \) = mass flux, in kg/m\(^2\) sec
\( P \) = fluid pressure, in bar

This correlation has been obtained from the 4 helically coiled monotube experimental tests at a previous stage. The method used is described in [7] and our results are given on figure 6.

In the film boiling zone, the quality effect on the heat transfer is taken into account by the Miropolisky's correlation [8]

\[ Nu_{x} = Nu_{1} \left[ 1 - 0.1 \left( \frac{\rho_{L}}{\rho_{v}} - 1 \right)^{0.4} (1 - X)^{0.4} \right] \left[ X + \frac{\rho_{v}}{\rho_{L}} (1 - X) \right]^{0.8} \]  

where the Nusselt number, \( Nu_{1} \) for the saturated steam \( X = 1 \) is computed from correlation [9]

The thermal resistance of the tube wall is computed by the usual equation

\[ R_{w} = \frac{d_{1}}{2 \cdot k_{w} \cdot \log \frac{d_{o}}{d_{1}}} \]  

where \( k_{w} = 11.73 + 1.557 \times 10^{-2} \quad T \cdot \frac{W}{m \cdot ^{\circ}C} \) drawn from [9]

**PRESSURE DROP CORRELATIONS**

The pressure drop of water and steam flowing in helically coiled tube are computed from the following equation:
\[ P_1 - P_2 = \frac{\Delta G^2}{2 \Delta i} \sum v_i \Delta L + \frac{\Delta L}{v_i} + g \sum \frac{\Delta z}{v_i} \]  

where

- \( d_i \) = inside diameter, in m
- \( g \) = gravitational acceleration, in m/sec\(^2\)
- \( G \) = mass flux, in kg/m\(^2\) sec
- \( \Delta L \) = length of coiled tube segment of calculation, in m
- \( P_1 - P_2 \) = pressure drop between 2 tubes sections, in N/m\(^2\)
- \( \nu \) = specific volume, kg/m\(^3\)
- \( \Delta Z \) = vertical distance of segment L, in m
- \( \mu \) = Darcy coefficient computed from Mori and Nakayama correlation [4] which is:

\[ \lambda = \left( \frac{d_i}{D} \right)^{0.5} \left[ \frac{0.192}{\text{Re} \left( \frac{d_i}{D} \right)^{2.5}} \right]^{1/6} \left[ 1 + \frac{0.068}{\text{Re} \left( \frac{d_i}{D} \right)^{2.5}} \right]^{1/6} \]  

On figure 7 we give the \( \lambda \) values computed for our helical coil diameters and confirmed by Ito's experimental results. At high Reynolds number the pressure drop measured on the 4 helically superheaters fitted well with this equation:

\[ \lambda = \left( \frac{d_i}{D} \right)^{0.5} \left[ \frac{0.1614}{\text{Re} \left( \frac{d_i}{D} \right)^{2.815}} \right]^{1/6.63} \left[ 1 + \frac{0.002}{\text{Re} \left( \frac{d_i}{D} \right)^{2.815}} \right]^{1/6.63} \]  

which was used for Reynolds number above the value given by the equation

\[ \log_{10} \text{Re} = 4.77 + 0.0535 \left( \frac{D}{d_i} \right)^{0.5} \]  

where, \( D \) = coil diameter.

The curves slope founded are slightly lower than Mori and Nakayama's predictions but they are nevertheless in fairly good agreement with them and with Ito's experimental results.

The roughness effect on helically coiled tube is assumed to be the same than in a straight tube, so that:

\[ \frac{\lambda_{RC}}{\lambda_{SS}} = \frac{\lambda_{RS}}{\lambda_{SS}} \]  

where:

- \( RC \) = rough coil
- \( SC \) = smooth coil
- \( RS \) = rough straight tube
- \( SS \) = smooth straight tube.

The two-phase pressure drop in the forced-circulation boiling region was calculated according to the Martinelli-Nelson method [10].
Neither the usual overall heat transfer coefficient nor the thermal efficiency is well suited to determine the effectiveness of a heat exchanger where a phase change takes place in the heated fluid, so we propose the following effectiveness definition, computed from water enthalpy changes:

\[
\varepsilon = \frac{H(t_o, p_o) - H(t_i, p_i)}{H(T_i, p_o) - H(t_i, p_i)}
\]

It represents the ratio of the water enthalpy increase to the maximum possible enthalpy gain that would be obtained from an exchanger with an infinite heat transfer area. In this latter case the steam temperature would be identical to the sodium inlet temperature \( T_i \).

The computer code ZEVALUK allows the calculation of steam pressure and temperature \( p_o \) and \( t_o \) at the outlet, with \( p_i \) and \( t_i \) being the inlet conditions; \( \varepsilon_{\text{cal}} \) can be predicted for given fluid flow rate of water and sodium. \( \varepsilon_{\text{exp}} \) can be also calculated as well with steam outlet temperature and pressure measured \( t_o \), \( p_o \).

The agreement between prediction and experiment may be evaluated by the relative effectiveness differential:

\[
\frac{\varepsilon_{\text{exp}} - \varepsilon_{\text{cal}}}{\varepsilon_{\text{cal}}} \times 100
\]

**Figure 8** is a comparison of calculated and experimental effectiveness of the 4 helically coiled monotube steam generators and of the 45 MW steam generator mock-up. The general agreement is very good since the average deviation is about 0.1% for the entire load range and for the whole of the tested steam generators. The standard deviation \( \pm 1.1\% \).
Therefore our computer code may oversize very slightly the heat exchange area because gap value is positive but probably not if we compare it at the standard deviation value.

It should be remembered that the effectiveness characterizes only the gross behavior of a heat exchanger. A much finer analysis results in the comparison of the experimental and computed pressure and temperature profiles along the heat transfer surface. This analysis is necessary, particularly in view of partial load conditions.

TEMPERATURE AND PRESSURE PROFILES

The adequacy of a steam generator calculation code can be evaluated under the most stringent conditions when comparing the predicted parameter profiles to the experimental data obtained at various thermal loads.

Figures 9 - 10 - 11 and 12 are comparisons of the calculated and experimental temperature and pressure profiles as observed on the 4 helically coiled monoboke steam generators at 100 % thermal rating with a pressure of 180 bar, and at 25 % thermal rating with a lower steam pressure of 45 bar. The computed wall temperature and the heat flux were added as supplementary information. The excellent agreement between the calculated and measured liquid metal temperature along the heat exchange length is evident. Moreover the experimental steam outlet superheated temperature differs from the predicted value less than 1 °C and the steam outlet pressure is better than 0.5 bar from the calculated pressure.

The general agreement between code predictions and experiments performed on the 45 MW steam generator mock-up is also very good at full and low thermal loads. Figure 13 gives comparisons at 100 % and 80 % thermal ratings for the steam outlet pressure of the Phenix cycle.

Figure 14 gives the calculated temperature profile which in spite of its very high gradient agrees with the temperature measured along the length.
HEAT TRANSFER IN HELICAL TUBE SODIUM HEATED STEAM GENERATORS

FIG. 9. HELICALLY COILED MONOTUBE STEAM GENERATOR
12.7 cm COIL DIAMETER

FIG. 10. HELICALLY COILED MONOTUBE STEAM GENERATOR
LINE COIL DIAMETER

FIG. 11. HELICALLY COILED MONOTUBE STEAM GENERATOR
LINE COIL DIAMETER

Figure 15 represents a run with 100% mass flux performed at the steam pressure cycle of Super Phenix. The 45 MW steam generator mock-up has been sized for the Phenix steam cycle. It represents a lower heat exchange surface than the steam cycle of Super Phenix, therefore we have dropped the first part of the water heating zone and the last part of the steam superheating zone and this leads us to compare theoretical and experimental results on the boiling area, the steam being only slightly superheated.

- PRESSURE DROP COMPARISON

A general comparison between water and steam side calculated pressure drops and experimental results obtained on the 4 helically coiled monotube steam generator mock-up is shown in figure 16. The average value of the difference between calculated and experimental results is 3.5% with a standard deviation 2σ equal to ±11%. It seems that the calculated pressure drop is a little too high, but this is just to be considered as a safety margin for industrial units.

- Static stability curve

As an example we can see on the figure 17 the static instability curve given by the pressure drop measured and calculated between the water inlet and the steam outlet at a thermal rating of 60% with a high steam outlet pressure of 171 bar. In this test, the sodium mass flow rate and the inlet temperature of both sodium and water were kept constant. Within the considered operation range, the steam generator static stability can be fairly well predicted like the outlet temperature of both fluids showed on figure 17. This instability obtained at a thermal rating of 60% shows the sensitiveness of the helically coiled tube toward the static instability.

CONCLUSIONS

The amount of thermal tests performed, (about a hundred on 4 helically coiled monotube steam generator and forty on the 45 MWT mock-up steam generator with 24 tubes heated by sodium) has demonstrated that our computer code ZEBALUR can fairly well predict any steady state operation for a commercial steam generator with a coiled tube bundle.

It is also possible to calculate the theoretical static instability curves at
FIG. 16. CALCULATED AND MEASURED STEAM TUBE PRESSURE DROP COMPARISON

FIG. 17. 45 MW STEAM GENERATOR STABILITY - (THERMAL RATING 60% - STEAM OUTLET PRESSURE 71 bar)
any conditions and to deduce the appropriate orifice to be installed at the water inlet in order to obtain the required static stability.

REFERENCES


