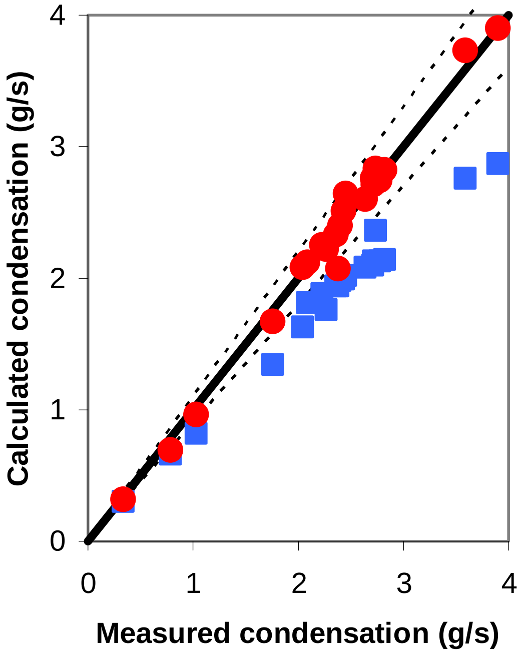
RESEARCH REPORT

VTT-R-01503-10



MELCOR Simulations of Steam Condensation in a Condenser Tube

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Confidentiality: Public

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MELCOR Simulations of Steam Condensation in a Condenser Tube

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Summary

Passive condenser systems have been proposed for removing heat from the containment in various new reactor concepts. To simulate the operation of a passive condenser, one must be able to model two phenomena: the condensation rate in the condenser, and the flow rate through the system. This report concentrates on the first issue, simulating steam condensation in a tube in the presence of air, with MELCOR 1.8.6 YV. Experiments that were conducted at Purdue University were chosen for the simulations because they were simple and they had well-defined boundary conditions.

Two conclusions were reached. First, one control volume was adequate for the modeling of the active length of the condenser tube. Dividing the tube into three control volumes had a negligible effect on the results. Second, with the default parameters, MELCOR underestimates the condensation rate by about 20 %, both with pure steam and with steam–air mixtures. Reason for the underestimation was found to be wrong Reynolds number limits for classifying the condensate film flow as laminar or turbulent. The film Reynolds number limits were changed to values found in a heat and mass transfer text book by modifying sensitivity coefficients in MELCOR. The average deviation of the modified MELCOR model is only 3.5 %, which is close to the measurement uncertainties in the experiments.

The tests were limited to air concentrations below 8.9 % in the inlet (max. 16 % in the outlet). The modified model should be validated also with higher non-condensable gas concentrations and possibly with other condenser tube diameters.

Confidentiality	Public

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Contents

1 Introduction

Various new reactor concepts utilize passive condensers for removing heat from the containment atmosphere in the case of an accident. There are two basic types of the condensers. In the other, water circulates in tubes that are located inside the containment, and steam condenses on the outer surfaces of the tubes (Fig. 1). This concept is used in Areva's Kerena reactor design (former SWR-1000) (Stosic et al. 2008) and in Atomstroyexport's AES-2006 design (STUK 2009). In the other condenser type, steam from the containment flows in tubes that are located in a water pool outside the containment (Fig. 2). In this case the steam condenses inside the tubes. This concept is used in Toshiba-Westinghouse's ABWR design (STUK 2009) and in GE Hitachi's ESBWR (Beard 2006). The present report deals with the latter condenser type.

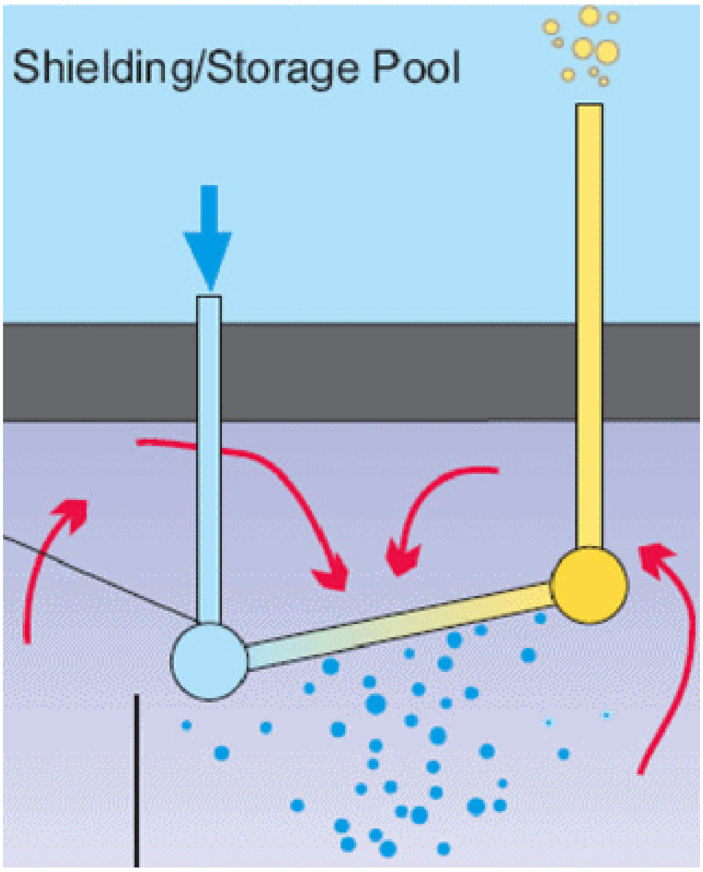

Fig. 1. Passive condenser in Kerena reactor concept, steam condensing outside the tube (Stosic et al. 2008).

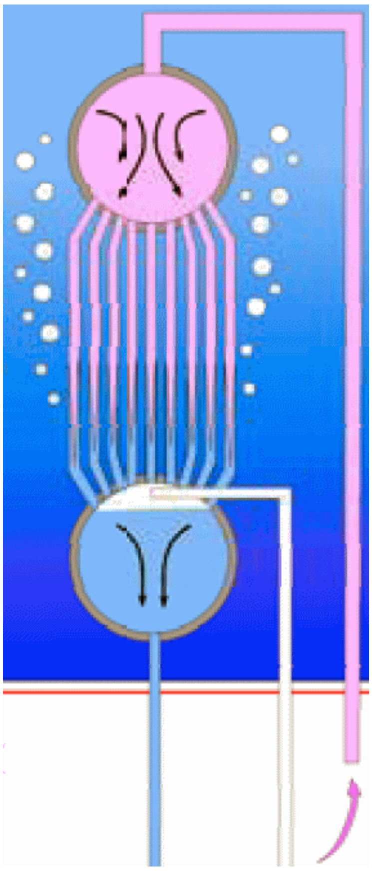

Fig. 2. Passive condenser in ESBWR, steam condensing inside the tube (Beard 2006).

Simulating the operation of a passive condenser is challenging. Two main phenomena have to be modeled: the flow of fluids in the tubes, and steam condensation. These phenomena are interlinked because the flow velocity affects the heat and mass transfer rates, and because the heat transfer rate affects the natural circulation gas flow. This report is related to modeling the second phenomenon, steam condensation, with the MELCOR code.

Experiments performed at Purdue University (Oh 2004; Oh & Revankar 2005) were chosen for simulations because they were simple and they had well-defined boundary conditions. The tests involved feeding steam and air into a single condenser tube that was immersed in water. The most important measured quantity was the steam condensation rate. In most other condenser experiments, subcooled water was flowing outside the tube. This makes the analysis more difficult because the heat transfer coefficient from the tube outer surface to the water significantly affects the results. The Purdue experiments used boiling water boundary condition on the outer surface. This makes them more suitable for testing condensation models.

2 A Description of the Experiments

The simulated experiments were performed at Purdue University in the United States. The results have been published by Oh (2004) and by Oh & Revankar (2005). The former reference includes the test parameters and results also as numbers, not only in pictures.

The test section is illustrated in Fig. 3. Steam and air flowed downwards in a vertical condensing tube. The inner diameter of the tube was 26.6 mm and wall thickness 3.38 mm. It was made of stainless steel 304. The active condensing length was 984 mm. The pressure, steam flow rate and air fraction were varied. The inlet gas temperature was a few degrees above saturation. The steam condensation rate was measured.

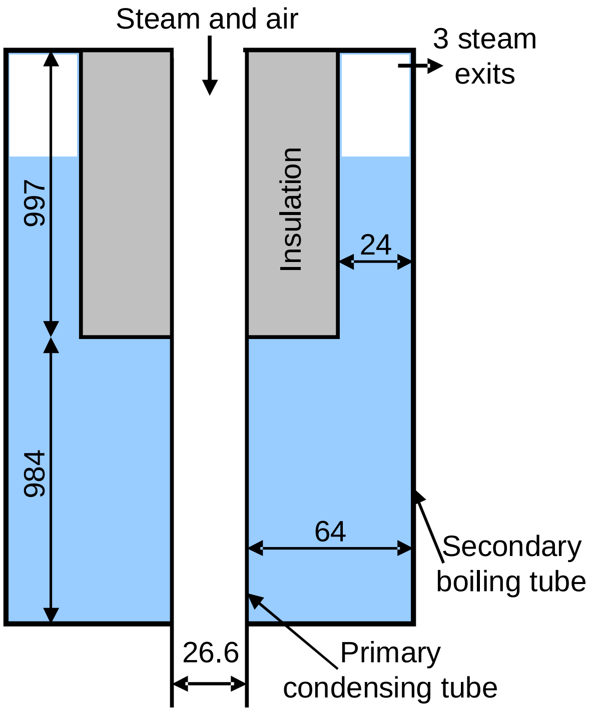

Fig. 3. *The test section. The dimensions are in millimeters. The drawing is not in the right scale.*

The inner diameter of the secondary boiling tube was 161 mm. The annulus between the pipes was filled with water at atmospheric pressure and saturation temperature. The water was boiling, and the generated steam flowed out from the test section through three steam exits. The upper part of the primary tube was thermally insulated from the secondary side in order to keep the active condensing length constant despite the decreasing water level on the secondary side. A small uncertainty arises from the heat conduction vertically along the primary tube. Oh (2004) has calculated that the extended condensation length due to the conduction was less than 1.5 % of the active length.

Experiments were performed in three modes. In the complete condensation mode, a valve was kept closed, preventing gas outflow from the primary tube and causing all the steam to condense. In the cyclic venting mode, the valve was periodically opened to simulate the plant situation where the vent line occasionally clears when the pressure difference between the drywell and the suppression pool exceeds the head due to the submergence of the vent line. In the through flow mode, the valve was kept open and the gas was free to flow through the condensing tube. Only through flow tests were simulated with MELCOR and reported in the present report. Oh (2004) gives 127 data points from the through flow tests, each corresponding to steady state conditions. The measurement error limits are given in Table 1.

Table 1. *Measurement errors in the experiments (Oh 2004, appendix A).*

Temperature	± 2 °C
Pressure	± 0.1 %
Steam flow rate	± 1 %
Air flow rate	± 5 %
Condensation rate	± 1.4 %

3 Modeling the Experiments with MELCOR

3.1 Steam Condensation Physics in MELCOR 1.8.6

The physical models for steam condensation in MELCOR 1.8.6 are described in the Heat Structure package reference manual (Gauntt et al. 2005). Condensation occurs when the structure surface temperature is below the dew point of the atmosphere. The following text is limited to condensation inside vertical tubes.

MELCOR evaluates the condensate film thickness from

$$\delta_f = \begin{cases} 0.909\left(\dfrac{\mu_f^2}{\rho_f^2\, g}\right)^{1/3} \mathrm{Re}_f^{1/3}, & \text{if } \mathrm{Re}_f < 1000 \\ 0.115\left(\dfrac{\mu_f^2}{\rho_f^2\, g}\right)^{1/3} \mathrm{Re}_f^{0.6}, & \text{if } \mathrm{Re}_f > 3000 \end{cases} . \tag{1}$$

Here μ_f is the dynamic viscosity and ρ_f is the density of the film, and $g = 9.8\ \mathrm{m/s^2}$. Re_f is the Reynolds number of the film, based on the film thickness. The upper expression is related to laminar and the lower to turbulent film flow. When the Reynolds number is between the laminar and turbulent regimes, the thickness is determined by linear interpolation.

MELCOR evaluates the heat transfer coefficient through the condensate film from

$$h_f = \begin{cases} \dfrac{k_f}{\delta_f}, & \text{if } \mathrm{Re}_f < 1000 \\ \dfrac{k_f}{\left(\dfrac{\mu_f^2}{\rho_f^2\, g}\right)^{1/3}} \left(\mathrm{Re}_f^{-0.44} + 5.82 \cdot 10^{-6}\, \mathrm{Re}_f^{0.8}\, \mathrm{Pr}_f^{1/3}\right)^{0.5}, & \text{if } \mathrm{Re}_f > 3000 \end{cases}. \tag{2}$$

Here k_f is the thermal conductivity and Pr_f is the Prandtl number of the film. When the Reynolds number is between the laminar and turbulent regimes, h_f is determined by linear interpolation.

In a nearly pure steam atmosphere, the condensation rate is limited by heat transfer through the condensate film and through the structure. Introduction of non-condensable gases to the atmosphere restricts the mass transfer because the non-condensable gases accumulate near the film surface. In the presence of non-condensable gases, MELCOR evaluates the condensation mass flux from

$$\dot{m}_c = h_D\, \rho_v \ln\left(\frac{P_{tot} - P_{srf}}{P_{tot} - P_{steam}}\right). \tag{3}$$

Here ρ_v is the density of vapor at the saturation temperature corresponding to the total pressure P_{tot}. P_{srf} is the saturation pressure of steam at the surface temperature, and P_{steam} is the steam partial pressure in the control volume. The mass transfer coefficient h_D is evaluated from

$$h_D = \mathrm{Sh}\frac{D}{L_c}, \tag{4}$$

where D is the gas diffusivity, L_c is the characteristic length of the surface, and Sh is the Sherwood number, also called the mass transfer Nusselt number.

MELCOR evaluates the Sherwood number from

$$\mathrm{Sh} = \mathrm{Nu}\ \mathrm{Sc}^{1/3}\ \mathrm{Pr}^{-1/3}, \tag{5}$$

where the Schmidt number Sc is defined as

$$\mathrm{Sc} = \frac{\mu}{\rho D}. \tag{6}$$

μ is the dynamic viscosity of atmosphere at average of surface and atmosphere temperatures, and ρ is the density of atmosphere.

In accordance with the heat and mass transfer analogy, MELCOR evaluates the Nusselt number Nu in equation (5) from heat transfer correlations related to the particular geometry and flow regime. For the current experiments, the forced convection correlations for internal flow in a cylinder apply:

$$\mathrm{Nu} = \begin{cases} 48/11, & \text{if } \mathrm{Re} < 2000 \\ 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{1/3}, & \text{if } \mathrm{Re} > 10000 \end{cases}. \tag{7}$$

The first expression is related to laminar flow, and it can be derived analytically. The second expression is the Colburn equation for turbulent flow in a tube (Incropera & DeWitt 2002). In both cases, the appropriate characteristic length is the diameter of the tube. In the transition flow regime, the Nusselt number is a linear interpolation between the laminar and turbulent regimes. In the experiments, transitional and turbulent flows occurred.

For calculating the Reynolds number in equation (7), the flow velocity in the control volume is needed. MELCOR is a lumped parameter code, where three-dimensional volumes and flows are modeled effectively zero-dimensionally. Therefore the local flow velocities are not known. MELCOR approximates the flow velocity v in a gas-only or liquid-only volume by

$$v = \frac{1}{2}\frac{\sum_i |J_i|}{A}, \tag{8}$$

where J_i is the volumetric flow rate through flow path i, and the sum is evaluated over all flow paths that are connected to the control volume. The factor 1/2 arises because, in steady state, each flow is calculated twice: once when it enters the volume and once when it leaves. External mass sources and sinks do not contribute to the flow velocity. A is the flow area of the control volume, and its default value is the volume divided by the height.

3.2 MELCOR Model of the Test Section

Fig. 4 shows the nodalization of the test section in the MELCOR model. Control volume CV 1 is the active condensing tube. CV 10 is an inlet volume. The incoming steam and air are introduced into CV 10 as mass and enthalpy sources. If the mass source was placed directly into CV 1, MELCOR would calculate more than 50 % too small flow velocity in the CV because the summation in equation (8) does not include external sources. This would result in too small heat transfer coefficients. For correct calculation of the flow velocity, the inflow and outflow to the active condenser tube have to be modeled as flow paths (FL 10 and 1), not as mass sources and sinks.

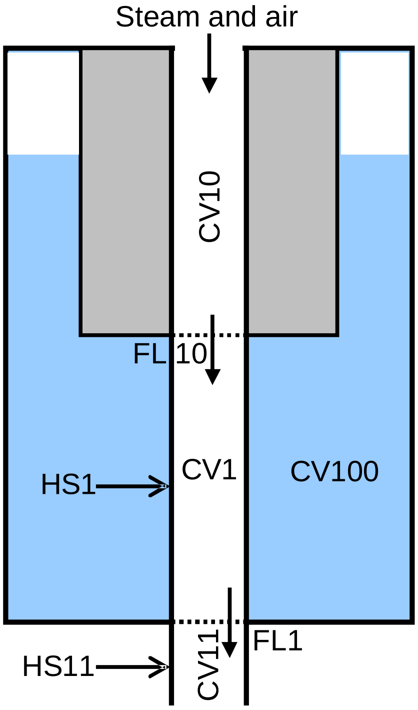

Fig. 4. Nodalization of the test section in MELCOR.

CV 11 is an outlet volume. It is defined as a time-independent control volume, and its duty is to keep the test section pressure at the right value. CV 100 is the secondary volume. It is a time-independent control volume containing saturated water.

There are two heat structures in the model. HS 1 is the active condensing tube. HS 11 is an auxiliary heat structure for receiving the drained condensate film from HS 1 by the film tracking model and dumping the liquid into CV 11. Without the auxiliary structure, the drained condensate would be placed into the pool of CV 1. This would cause a tiny pool to appear and disappear in the control volume, and the result would be oscillation. The inside, condensing surface of HS 1 is interfaced with CV 1 and the outside with CV 100, where nucleate boiling occurs. The inside surface of HS 11 is interfaced with CV 11, and the outside is insulated. The structures are modeled with three nodes.

The pipe in the experiments was made of type 304 stainless steel. In the steady state experiments, the only important material property is thermal conductivity. According to (Incropera & DeWitt 2002, table A.1), the thermal conductivity of stainless steel 304 at 400 K is 16.6 W/m·K. MELCOR has two stainless steel materials, *Stainless Steel* and *Stainless Steel 304*. Their thermal conductivities at 400 K are 15.8 and 14.6 W/m·K, respectively (Gauntt et al. 2005). Thus, the thermal conductivity of *Stainless Steel* material in MELCOR is closer to the correct value, and this material was used in the heat structures.

In addition to the control volumes shown in Fig. 4, there was an auxiliary CV 999. Its duty was to evaluate the enthalpy of the inlet steam at the specified temperature and pressure. This was necessary because steam enthalpy depends on both the temperature and the pressure. Thus it is not possible to model water or steam sources with MELCOR's TE type enthalpy source, which is intended for ideal gas sources and specifies only the temperature.

For setting the correct boundary conditions, the incoming steam and air temperatures were set to the value that was measured by the TS1 thermocouple, which was placed in the primary tube at 44 mm below the start of the active condensing length. The secondary volume CV 100 was defined to contain saturated water at the measured average pool temperature.

4 Results

There are a total of 127 steady state data points for the through flow experiments in (Oh 2004, appendix C). 21 of these were simulated with MELCOR 1.8.6 YV.

4.1 Effect of Nodalization

Fig. 4 presents the MELCOR nodalization where the active condenser tube was modeled with one control volume and one heat structure. The effect of nodalization was examined by dividing CV 1 into three equal control volumes and HS 1 into three equal heat structures. A film tracking network was defined to drain the film along the pipe and eventually to HS 11.

Six tests, with widely varying parameters, were chosen for this investigation. Default heat and mass transfer modeling parameters were used in MELCOR. Table 2 shows the test parameters, along with the measured and calculated condensation rates. Two conclusions can be drawn from the results. First, with the default parameters, MELCOR systematically underestimates the condensation rate by about 20 %. Second, the difference between the one CV and three CV models is negligible, about 2 %.

Table 2. *Effect of using three control volumes instead of one to model the condensing tube.*

		Inlet conditions				Condensation rate (g/s)		
Test #	Pressure (MPa)	Steam flow rate (g/s)	Air mass fraction (%)	Temperature (°C)	Secondary temperature (°C)	Measured	1 CV model	3 CV model
9	0.260	5.59	3.63	138.88	103.22	2.27	1.76	1.78
28	0.268	2.90	0.00698	138.87	102.67	2.45	2.02	2.06
62	0.359	5.04	0	143.38	102.38	3.90	2.87	2.97
79	0.339	4.14	3.33	144.78	102.82	2.74	2.36	2.39
85	0.340	2.76	3.27	140.34	103.05	2.38	1.94	1.94
108	0.194	1.59	8.93	121.03	102.99	0.786	0.659	0.659

A mass transfer scaling factor can be set for a heat structure in the input record HSCCCCC400. The mass transfer coefficient in equation (4) is multiplied by the scaling factor. Its default value is 1. The effect of the scaling factor was tested but found to be very small. This is because the condensation rate is limited by the heat transfer rate through the condensate film and the structure itself, at least with relatively small concentrations of non-condensable gases.

4.2 Effect of Film Reynolds Number Limits

In the previous section it was shown that, with the default heat and mass transfer modeling parameters, MELCOR significantly underestimates the condensation rate. This happens both in pure steam tests and in the presence of air. Liao & Vierow (2006) have made a similar observation.

In order to find out the reason for the underestimation, MELCOR's condensation models were compared with those recommended by Incropera & DeWitt (2002) for pure steam. The models are very similar, but one major discrepancy was found. In equations (1) and (2), MELCOR uses the laminar film correlations when the film Reynolds number $\mathrm{Re}_f < 1000$ and turbulent film correlations when $\mathrm{Re}_f > 3000$. In (Incropera & DeWitt 2002, section 10.8), the corresponding film Reynolds number limits are 30 and 1800.

In MELCOR, the film Reynolds number limits are coded as sensitivity coefficients 4253(5) and 4253(6). MELCOR Users' Guide claims that these sensitivity coefficients are applied only when the film tracking model is active, but this is incorrect. In reality, modifying the coefficients has the same effect also when the film tracking model is inactive, at least for the inside surface of a vertical cylindrical heat structure.

The film Reynolds number limits were modified to those recommended by Incropera & DeWitt (2002), and the six tests were recalculated. The one CV model was used. Table 3 shows the results with the MELCOR default and modified parameters. The test parameters were already given in Table 2 and are not repeated here. It is seen that the results are significantly improved by modifying the Reynolds number limits to those recommended by Incropera & DeWitt (2002). The input deck for simulating test #79 with the modified parameters is given in appendix A. The three CV model with the modified Reynolds number limits was also tested, but the difference to the one CV model was again found to be negligible.

Table 3. *Effect of modifying the film Reynolds number limits.*

Test #	Condensation rate (g/s) Measured	MELCOR default	MELCOR modified
9	2.27	1.76	2.22
28	2.45	2.02	2.64
62	3.90	2.87	3.90
79	2.74	2.36	2.83
85	2.38	1.94	2.07
108	0.786	0.659	0.694

To see the effect of the non-condensable gases on the condensation, a series of nine experiments with 0.28 MPa pressure and 3.6 g/s steam flow rate was calculated with both the default and the modified MELCOR models. Fig. 5 shows the results. The condensation rate decreases with increasing amount of air. MELCOR with default parameters significantly underestimates the condensation. MELCOR with modified film Reynolds number limits gives almost perfect results, with only one point outside the error bars related to measurement uncertainties.

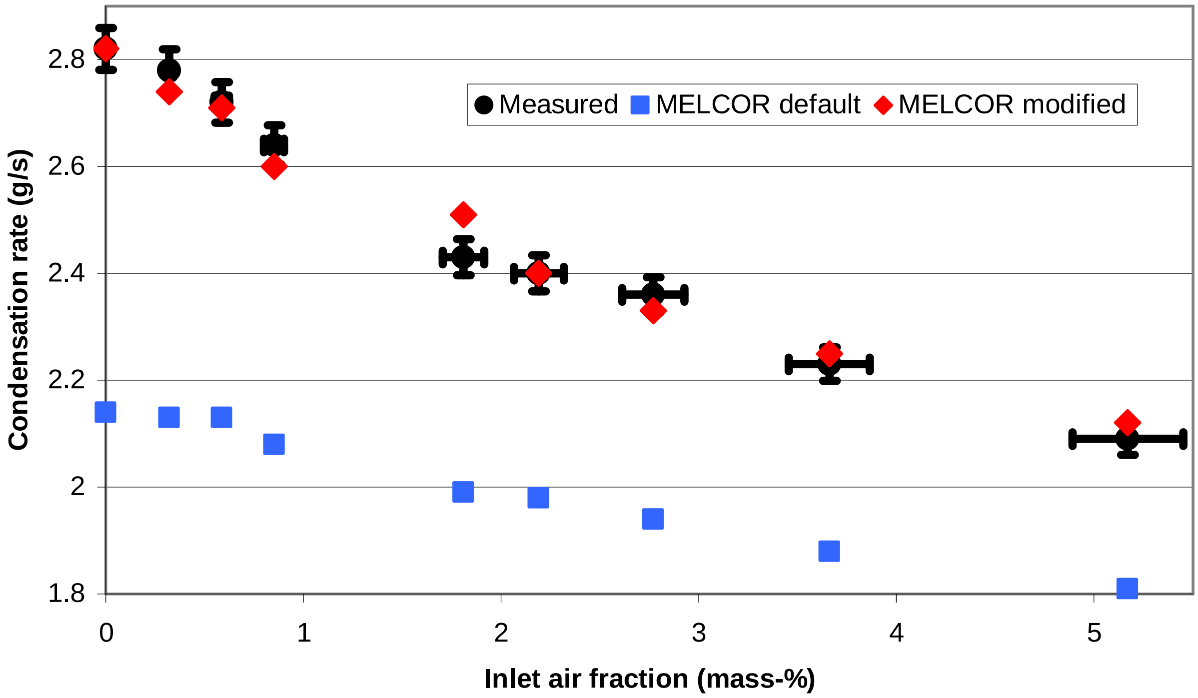

Fig. 5. *Steam condensation rate at 0.28 MPa pressure and 3.6 g/s steam flow rate. Measured values with error bars, compared with MELCOR results with default parameters and with modified film Reynolds number limits.*

A special case is pure steam condensation. A series of six tests with 4.96 g/s steam flow rate at different pressures was calculated. Fig. 6 shows the results. The condensation rate increases almost linearly with increasing pressure. Again, MELCOR with default parameters significantly underestimates the condensation, while MELCOR with modified film Reynolds number limits gives almost perfect results.

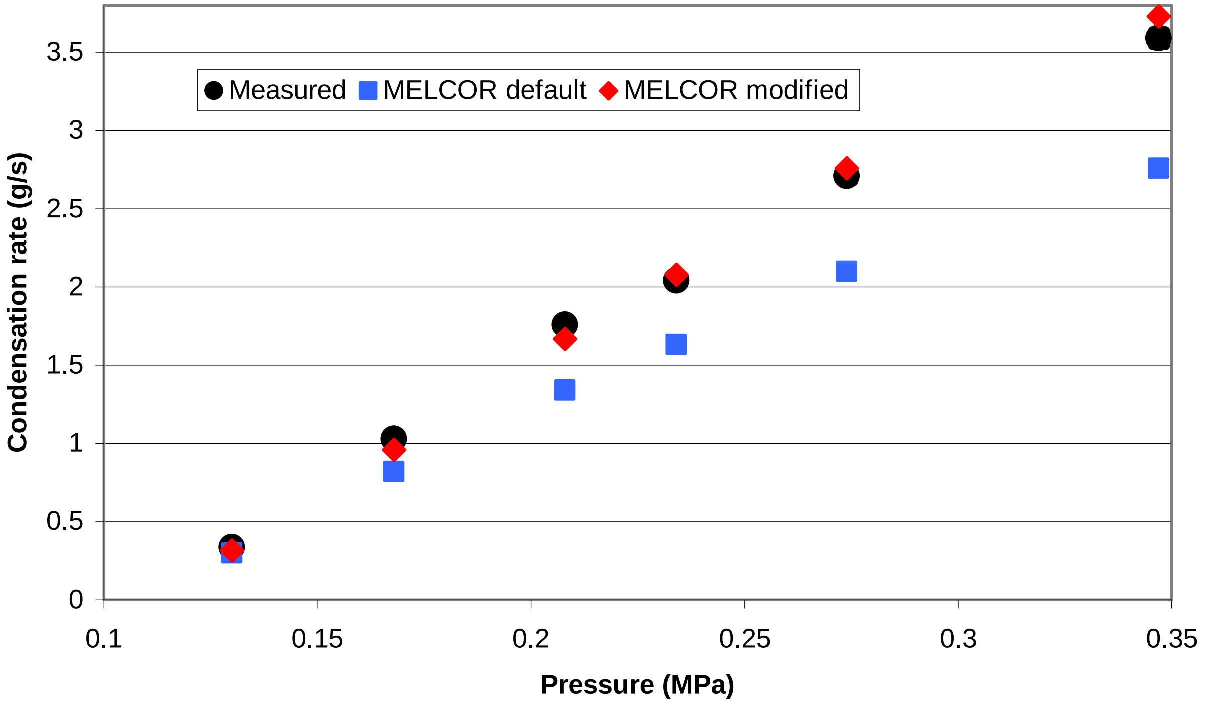

Fig. 6. *Steam condensation rate in pure steam experiments with 4.96 g/s steam flow rate. Measured values compared with MELCOR results with default parameters and with modified film Reynolds number limits.*

The measured and calculated condensation rates in all the 21 simulated experiments are plotted in Fig. 7. MELCOR with default parameters underestimates the condensation rate by 19 % on the average, while in MELCOR with modified film Reynolds number limits the underestimation is only 1.2 %. The average of the absolute values of the relative deviations is 19 % with the default and 3.5 % with the modified MELCOR parameters. Considering that the measurement uncertainty of the condensation rate is 1.4 %, and there are also measurement uncertainties in the test boundary conditions, the modified MELCOR model is very good. For reference, the test parameters and results of all the calculated experiments are given in Table 4.

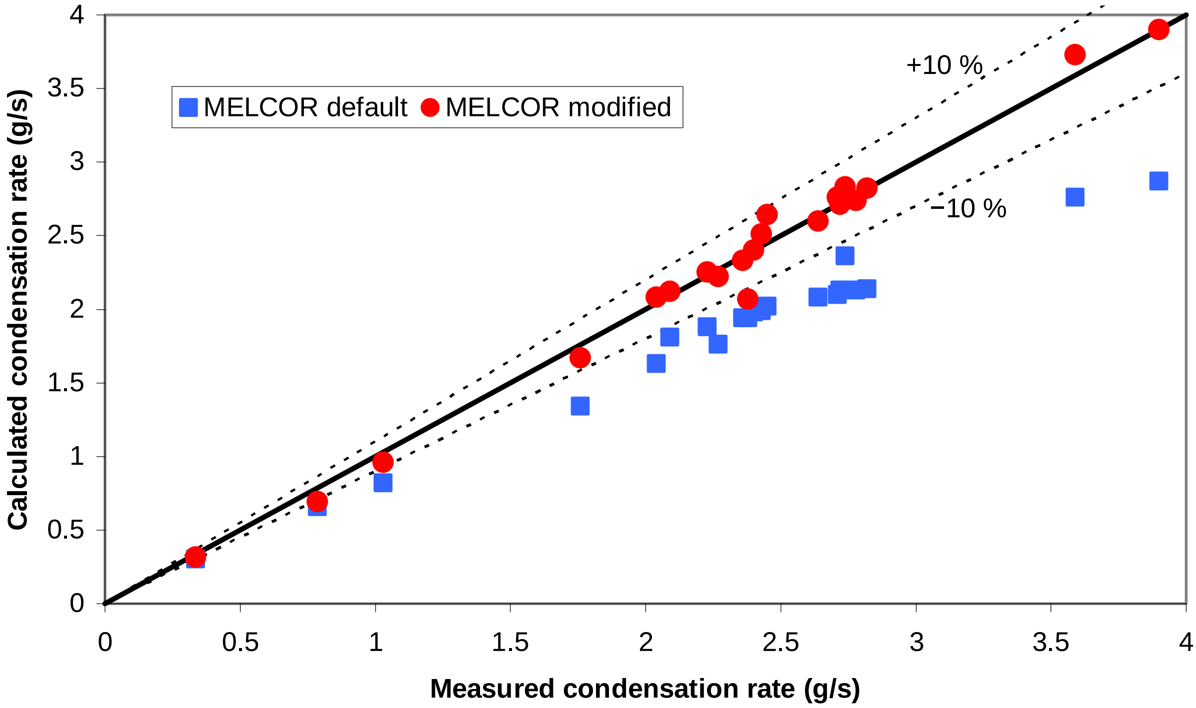

Fig. 7. Comparison of measured and calculated condensation rates in 21 experiments, using MELCOR default parameters and modified film Reynolds number limits.

Table 4. Test parameters and results from all the calculated experiments.

		Inlet conditions				Condensation rate (g/s)		
Test #	Pressure (MPa)	Steam flow rate (g/s)	Air mass fraction (%)	Temperature (°C)	Secondary temperature (°C)	Measured	MELCOR default	MELCOR modified
9	0.260	5.59	3.63	138.88	103.22	2.27	1.76	2.22
28	0.268	2.90	0.00698	138.87	102.67	2.45	2.02	2.64
33	0.284	3.62	0.587	141.97	102.35	2.72	2.13	2.71
35	0.283	3.61	2.19	140.31	102.83	2.40	1.98	2.40
37	0.282	3.59	5.17	138.12	102.83	2.09	1.81	2.12
38	0.280	3.61	3.66	139.87	102.61	2.23	1.88	2.25
39	0.279	3.61	2.77	140.39	102.44	2.36	1.94	2.33
40	0.282	3.61	1.81	141.33	102.96	2.43	1.99	2.51
41	0.282	3.61	0.851	141.85	102.75	2.64	2.08	2.60
42	0.282	3.61	0.322	141.59	102.56	2.78	2.13	2.74
44	0.281	3.61	0	141.46	102.83	2.82	2.14	2.82
62	0.359	5.04	0	143.38	102.38	3.90	2.87	3.90
63	0.347	4.96	0	143.89	102.59	3.59	2.76	3.73
64	0.274	4.96	0	141.62	102.29	2.71	2.1	2.76
65	0.234	4.96	0	141.05	102.74	2.04	1.63	2.08
66	0.208	4.98	0	139.85	102.41	1.76	1.34	1.67
67	0.168	4.96	0	139.34	101.87	1.03	0.821	0.960
68	0.130	4.96	0	138.66	100.45	0.335	0.301	0.316
79	0.339	4.14	3.33	144.78	102.82	2.74	2.36	2.83
85	0.340	2.76	3.27	140.34	103.05	2.38	1.94	2.07
108	0.194	1.59	8.93	121.03	102.99	0.786	0.659	0.694

5 Conclusions

21 experiments on steam condensation in a vertical tube were simulated with MELCOR 1.8.6 YV. The objective was to test MELCOR condensation models both for pure steam and for mixtures of steam and air. The pressure in the experiments varied between 0.13 and 0.36 MPa and the air mass fraction in the inlet between 0 and 8.9 %.

Two conclusions were drawn. First, the one meter long straight condenser tube can well be modeled as a single control volume and a single heat structure. Dividing the tube into three control volumes and heat structures had only a negligible effect on the results. Second, when using the default heat and mass transfer parameters, MELCOR underestimates the condensation rate by about 20 %, for both pure steam and steam–air mixtures.

The reason for the underestimated condensation was found in the Reynolds number limits that MELCOR uses to classify the condensate film flow as laminar or turbulent. The default film Reynolds number limits are 1000 and 3000, while Incropera & DeWitt (2002) recommend 30 and 1800. The film Reynolds number limits are coded as sensitivity coefficients 4253(5) and 4253(6). Changing these to the lower values improved the MELCOR results significantly. The modified MELCOR model simulates the experiments with an average deviation of 3.5 %, which is close to the measurement uncertainties in the experiments.

In the experiments the passive condenser tube was used in the active mode, so that steam and air were fed directly into the condenser. Therefore the present simulations deal only with the heat transfer aspects of the passive condenser system. Modeling the condensation and boiling phenomena correctly is required for proper simulation of passive condensers. The other aspect, not covered in the present report, is modeling the natural circulation flow caused by the pressure and temperature differences.

The experiments were limited to inlet air mass fractions below 8.9 %, corresponding to outlet air mass fraction below 16 %. Another limitation is that only one tube diameter, 26.6 mm, was used. The modified MELCOR model should be validated with higher non-condensable gas concentrations and other tube diameters, too.

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Appendix A. MELCOR Input for Test 79 with Modified Reynolds Number Limits

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*
* Purdue condensation test input for MELCOR 1.8.6
*
* One control volume, change Reynolds number limits for film
*
* Author:  Tuomo Sevon, VTT
* Date:    22 January 2010
*
*******************************************************************
*
*             FILE and JOB definition input
*
********************************************************************
*
TITLE      'Purdue'
JOBID      'Purdue'
*
DIAGF      MEGDIA.txt   * Diagnostics file
OUTPUTF    MEGOUT.txt   * Tabular output file
RESTARTF   RESTART      * Restart file
*
DTTIME 0.05              * Initial time step
*
********************************************************************
*
* NON CONDENSABLE GAS INPUT - definitions of materials
*
********************************************************************
*
NCG001      N2      4            * Material 4 is N2
NCG002      O2      5            * Material 5 is O2
*
********************************************************************
*
* CONTROL VOLUMES - CVH input
*
********************************************************************
*
* Inlet volume
CV01000 Inlet 2 2 1
CV010A1 PVOL 3.39E5  * Pressure
CV010A2 PH2O 3.39E5  * Initially pure steam
CV010A3 TATM 417.93  * Temperature
*       Altitude Volume
CV010B1  0.984   0.0
CV010B2  1.984   5.557E-4
* Steam source
CV010C0 MASS.3 001 3  * Steam source rate (kg/s) from CF001
CV010C1 AE 002 3  * Steam source enthalpy (W) from CF002
* Air source
CV010C2 MASS.4 006 3  * Nitrogen source rate (kg/s) from CF006
CV010C3 TE 007 9  * Air temperature from CF007
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CV010C4 MASS.5 008 3  * Oxygen source rate (kg/s) from CF008
CV010C5 TE 007 9  * Air temperature from CF007
*
* Primary volumes
*
CV00100 Primary1 2 2 1
CV001A1 PVOL 3.39E5  * Pressure
CV001A2 PH2O 3.39E5  * Initially pure steam
CV001A3 TATM 417.93  * Temperature
*       Altitude Volume
CV001B1  0.0     0.0
CV001B2  0.984   5.468E-4
*
* Outlet volume
CV01100 Outlet 2 2 1
CV01101 0 -1  * Time-independent
CV01103 5.557E-4  * Area, used for velocity calculation
CV011A1 PVOL 3.39E5  * Pressure
CV011A2 RHUM 1.0  * Pure steam
*       Altitude Volume
CV011B1   -1.0   0.0
CV011B2    0.0   5.557E-4
*
* Secondary volume
CV10000 Secondary 2 2 1
CV10001 0 -1  * Time-independent
CV100A1 ZPOL 1.5  * Pool surface elevation (approximate)
CV100A2 TPOL 375.97  * Pool temperature
CV100A3 VOID 0.1  * Void fraction (approximate)
CV100A4 RHUM 1.0  * Atmosphere pure steam
*       Altitude Volume
CV100B1  0.0     0.0
CV100B2  0.984   0.0192
CV100B3  1.981   0.0293
*
* Steam source, only for evaluating source enthalpy
CV99900 SteamSource 2 2 1
CV99901 0 -1  * Time-independent
CV999A1 PVOL 3.39E5  * Pressure
CV999A2 PH2O 3.39E5  * Pure steam
CV999A3 TATM 417.93  * Inlet steam temperature
*       Altitude Volume
CV999B1   0.0     0.0
CV999B2   1.0     1.0
*
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*
* CONTROL FUNCTIONS - CF input
*
********************************************************************
*
* Steam source rate (kg/s)
CF00100 SteamRate EQUALS 1 1.0 0.0
CF00110 0.0 4.14E-3 TIME
*
* Steam source enthalpy (W)
CF00200 EnthalpySource MULTIPLY 2 1.0 0.0
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CF00210 1.0 0.0 CVH-H.3.999  * Steam enthalpy (J/kg)
CF00211 1.0 0.0 CFVALU.001   * Steam source rate (kg/s)
*
* Air mass fraction
CF00300 AirFraction EQUALS 1 1.0 0.0
CF00310 0.0 3.33E-2 TIME
*
* Numerator for calculating air source rate
CF00400 Numerator MULTIPLY 2 1.0 0.0
CF00410 1.0 0.0 CFVALU.003  * Air mass fraction
CF00411 1.0 0.0 CFVALU.001  * Steam source rate (kg/s)
*
* Air source rate (kg/s)
CF00500 AirSource DIVIDE 2 1.0 0.0
CF00510 -1.0 1.0 CFVALU.003  * Denominator 1-Wair
CF00511  1.0 0.0 CFVALU.004  * Numerator Wair*Msteam
*
* Nitrogen source rate (kg/s)
CF00600 N2source EQUALS 1 1.0 0.0
CF00610 0.79 0.0 CFVALU.005  * 79 % of air mass
*
* Air temperature (K)
CF00700 AirTemperature EQUALS 1 1.0 0.0
CF00710 0.0 417.93 TIME
*
* Oxygen source rate (kg/s)
CF00800 O2source EQUALS 1 1.0 0.0
CF00810 0.21 0.0 CFVALU.005  * 21 % of air mass
*
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*
* FLOW PATHS - FL input
*
********************************************************************
*
* Inlet to test section
*              From To  Zfrom Zto
FL01000 InFlow 010  001 0.984 0.984
*         Area   Length FrOpen
FL01001 5.557E-4 0.492   1.0
FL01003 0.0 0.0   * Form loss coefficients zero because straight pipe
*         Area   Length  Dhyd
FL010S0 5.557E-4 0.492  0.0266
*
*             From To  Zfrom Zto
FL00100 Flow1 001  011  0.0  0.0
*         Area   Length FrOpen
FL00101 5.557E-4 0.492   1.0
FL00103 0.0 0.0   * Form loss coefficients zero because straight pipe
*         Area   Length  Dhyd
FL001S0 5.557E-4 0.492  0.0266
*
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*
* HEAT STRUCTURES - HS input
*
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*
* Change Reynolds number limits for film
SC42535 4253 30.0 5  * Laminar flow for Re < 30
SC42536 4253 1800.0 6  * Turbulent flow for Re > 1800
*
HS00001000 3 2  * 3 nodes, cylindrical
HS00001001 Structure1
HS00001002 0.0 1.0  * Bottom at 0 m, vertical
HS00001100 -1 2 0.0133 * Inner radius 13.3 mm
HS00001101 0.00169 2  * Two mesh intervals, 1.69 mm each
HS00001201 STAINLESS-STEEL 2
HS00001300 0  * No internal power source
HS00001400 1 001 INT 0.0 1.0  * Internal boundary CV1
*          Ignored CharL  AxLength
HS00001500   1.0   0.0266  0.984
HS00001600 1 100 EXT 0.0 1.0  * External boundary CV100
*          Ignored CharL  AxLength
HS00001700   1.0   0.0334  0.984
*
* Auxiliary heat structure for receiving the film drainage
HS00011000 3 2 -1  * 3 nodes, cylindrical, no steady-state
initialization
HS00011001 PipeEnd
HS00011002 -0.1 1.0  * Bottom at -0.1 m, vertical
HS00011100 -1 2 0.0133 * Inner radius 13.3 mm
HS00011101 0.00169 2  * Two mesh intervals, 1.69 mm each
HS00011201 STAINLESS-STEEL 2
HS00011300 0  * No internal power source
HS00011400 1 011 INT 0.0 1.0  * Internal boundary CV11
*          Ignored CharL  AxLength
HS00011500   1.0   0.0266   0.1
HS00011600 0  * External boundary insulated
HS00011800 -1  * Initial temperature distribution is given
HS00011801 409.5 3  * Initial temperature for all nodes
*
* Film-tracking input
HSFT00000 2  * Number of structures in the network
HSFT00100 00001 1 0.0 0 0.0 0  * HS1 drains to one structure
HSFT00101 00011 1.0 0.0  * Receiving structure is HS11, inner
surface only
HSFT00200 00011 0 0.0 0 0.0 0  * HS11 drains to zero structures
*
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