

HEFAT2010
7th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics
19-21 July 2010
Antalya, Turkey

VAPOUR CONDENSATION IN PRESENCE OF A NON CONDENSING GAS AND AEROSOLS

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ABSTRACT

In the study of growth/evaporation of particles due to (water) vapour in the atmosphere, it is usual to assume that the vapour is diluted and the temperature is uniform enough in the environment. These assumptions cannot be applied to heat exchanger condensers, where the gas mixture is enriched in water vapour (vs. the concentration of an incondensable). An added difficulty is the large value of water latent heat. Therefore, the equations that describe the phenomenon in the atmosphere should be corrected.

We present a model for vapour condensation in a double pipe heat exchanger. The main physical processes in the gas/aerosol atmosphere flowing between coaxial cylinders, being the internal one cooled by a refrigerant, are modelized. The model takes into account: conduction and convection; film vapour condensation; the aerosol particle size distribution (PSD); heterogeneous nucleation in non-diluted conditions; some particle phenomena: gravitational and turbulent deposition, thermophoresis and difussiophoresis; and mass, momentum and energy balances in the heat exchanger.

In the model, the particles (liquid water coming from condensed vapour with a log-normal PSD) are injected in the system at the heat exchanger inlet. Future model improvements will include particles formation by homogeneous nucleation.

Results of the model application are truly plausible. For example, the mean size of the particles grows as the flow runs towards the end of the exchanger, but some of them disappear eventually. As it could be expected, the main condensation occurs at the liquid film interface of the interior due to the greater temperature difference at these points. The presence of particles leads to a diminishing of the condensed vapour as compared with the case without particles (only vapour in gas phase).

NOMENCLATURE

b	[m]	Diffusiophoresis and thermophoresis regions widths
D	[m]	Particle diameter
d	[m]	Cylinder diameter
f	[-]	Friction factor
G	[kg/s/m ²]	Radial mass flux
h	[W/m ² /K]	Heat transfer coefficient
I	[kg/s]	Growth rate for a particle
i	[J/kg]	Specific enthalpy
P	[Pa]	Pressure
Q	[W/m ²]	Radial heat flux
T	[K]	Temperature
t	[s]	Time
U	[W/m ² /K]	Thermal transmittance
u	[m/s]	Flow velocity
v	[m/s]	Particle deposition velocity
x	[kg/s]	Longitudinal mass flow
Y	[-]	Molar fraction in the gas
z	[m]	Longitudinal axis direction

Special characters

Δ		Increment
ε	[-]	Ackerman coefficient
η	[1/m]	Deposition rate
κ	[m/s]	Coefficient in the Chilton-Colburn analogy
ρ	[kg/m ³]	Density
ψ	[1/s]	Discretized flow particle size distribution (PSD)

Superscripts and subscripts

'		end of a longitudinal computational step
c		carcase (external tube)
con		condensation toward the particles
cp		condensed phase at the internal tube
di		difussiophoresis
g		gas
gr		gravitational
l		condensed liquid
n		incondensable
o		outside of the internal cylinder
p		particle
r		refrigerant
tra		transmitted by conduction and convection
th		thermophoresis
tu		turbulence and Brownian diffusion
v		vapour

INTRODUCTION

During condensation in presence of incondensable gases in shell and tube heat exchangers aerosols can be present. Their influence in thermal efficiency has a growing interest. Some decades ago, two-phase flux models had been developed, which took into account vapour nucleation. In this regard, Hill designed a one-dimensional flux model for Laval nozzles, accompanied with some experiments [1]. Very recently, 3-D models have been developed with increasing complexity for drops treatment, with computational fluid dynamics (CFD) and two-phase approximation for the liquid-gas interaction [2].

Aerosol particles originate from two phenomena: homogeneous nucleation and external sources. In the first, particles form directly from vapour and a high level of supersaturation is required. This process, also called homogeneous condensation (auto-condensation in the mass) is favoured in presence of other compounds that are different from a pure vapour (other vapours, sulphate, ammonia, etc.).

In these cases, the supersaturation ratio, S , the quotient between the actual vapour pressure and the saturation pressure, can be close to one. Then, the lack of purity in the vapour cycle water in power plants favours the creation of aerosol particles. However, this is a metastable phenomenon and it requires, in practice, that S be somewhat greater than one condensation to begin [3].

Particles present in the system are influenced by several phenomena: deposition, diffusiophoresis, thermophoresis, coagulation and heterogeneous nucleation (condensation-evaporation to-from particles). Heterogeneous nucleation occurs on previously present particles, which in vapour cycles are possibly due to defects in the purification system or to impurities carried in the air infiltrations, at points in where the pressure is lower than the atmospheric one (v.g., some pre-heaters and in the condenser).

In the case of turbines, the studies based on CFD are beginning to have a relevant role in the impact evaluation of humidity formation in the low pressure stages. Throughout the entire expansion process, the vapour enters superheated into the turbine and it leaves saturated, forming humidity as a result of various phase transition processes. From a design point of view, it is important to have CFD models that can accurately include these transitions (including non-stationary behaviours). Therefore, multiphase models that include particle behaviour are very interesting, because particle phenomena depend a lot on the particle size and on the PSD itself. For the flow in the last stages of turbines, very recently a poly-disperse PSD has been taken into account [4].

In vapour condensers, condensation occurs mainly on the tube surface. This mechanism has been and is currently being amply studied. This condensation is produced in presence of an incondensable gas (coming from air and other gases filtrations) and other contaminants with potential to become condensation nuclei. The first experiments on fog formation in condensation have been noted by Colburn and Edison [5].

Recently, some researchers are focusing their work on the study of fog drops in gas (aerosols). For example, in [6] and [7] the evolution of aerosols is analyzed by using restricted size ranges (even mono-disperse PSD). Aerosol particles perturb heat exchange because some of the energy for film condensation is wasted, in this case, in drop growth. On the other hand, aerosol formation and drop growth cause loss of treated water because the fluid drags particles.

The main motivation for this paper is an interest in obtaining a model for improving the understanding of the vapour-gas-condensed system, while taking into account film condensation in a double tube exchanger in presence of particles in the gas. Particles can grow due to vapour condensation, but the main condensation is on the refrigerated (interior tube) surface. The conditions for applying the model are similar to the ones that are produced in the condensers of a thermal power plant, and we want to analyze thermal phenomena that could be derived from all these processes.

SIMULATION MODEL

The equations that govern aerosol evolution, both in closed ducts and in the atmosphere, are very complex. For this reason, authors have developed simplified evolution models. Specifically, the various contributions in the differential equations that govern the multiple phenomena can be classified into three contributions:

- Coagulation.
- Condensation/Evaporation, also called heterogeneous nucleation.
- Source terms: homogeneous nucleation, gravitatory and turbulent depositions, diffusiophoresis and thermophoresis.

For the model presented in this paper, we have developed these equations taking into account both the fluid (gas, composed of vapour and incondensable) as well as the aerosol (solid and liquid particles in suspension). These equations include all of the relevant phenomena indicated before: fluid and particle transport, heat exchange between the fluid and the particles, and between the fluid and the walls of the duct, the formation of new particles, interaction between particles, diffusiophoresis, thermophoresis, deposition, etc.

Coagulation can be dismissed, because it is relevant only with high particle concentrations and long residence times (which are not issues in this case) [3]. Besides, we shall dismiss homogeneous nucleation, which depends a lot on the impurities present in the system (in this first approximation to the solution of the problem we do not include it).

Our model is one-dimensional, relating the z coordinate of longitudinal advance inside the double tube with time, t , through the flow gas velocity, u_g . The advance along this coordinate is performed in slabs of width Δz , with a small value to decrease the numerical calculation errors. At each step balances of mass, energy and momentum are done, taking into account the incondensable, n , water vapour, v , and particles, p , present in liquid form.

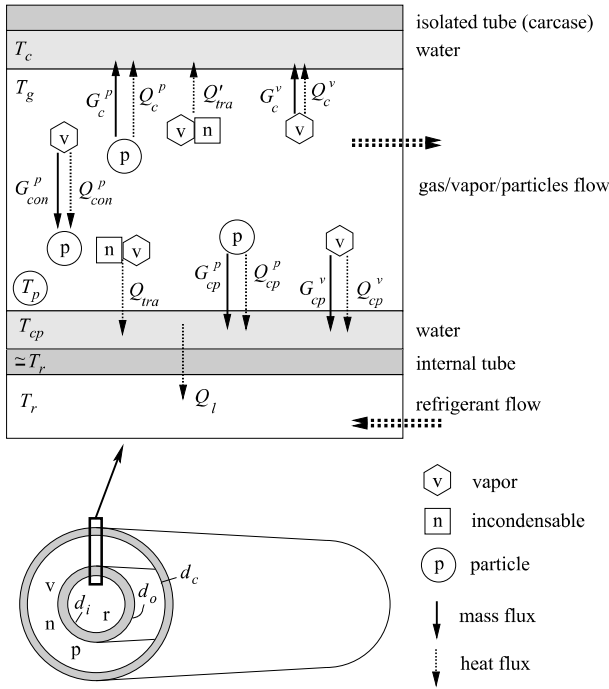


Figure 1 Diagram of the cylindrical exchanger.

Following fig. 1, at the beginning of each slab we pose a stationary equilibrium: film condensation is supposed occurs in such way that the interface gas/condensed phase temperature is the same as the saturation temperature at this point. Thus, we calculate the heat, Q_{cp}^v and Q_c^v , and mass, G_{cp}^v and G_c^v , fluxes.

In the calculation, the heat flux towards the condensed phase, Q_{tra} and Q'_{tra} , which include convection and conduction in gas phase, are determined. The heat transmitted to the refrigerant from liquid film, Q_l , is calculated. Other contributions are due to the gravitatory and turbulent deposition, diffusiophoresis and thermophoresis of the particles to the walls of the exchanger. They are Q_{cp}^p and Q_c^p for the heat, and G_{cp}^p and G_c^p for the mass. The heat flux corresponding to the condensation in the particles, Q_{con}^p , does not appear in the total balance because it is mainly wasted in modifying the particle temperature [3]. The corresponding mass flux, G_{con}^p , does appear in the total mass balance of the vapour in the system.

We have the following contributions:

(i) *Mass and heat transfer in film condensation* [8]. For the mass flux:

$$G_{cp}^v = \kappa \frac{\rho_v}{Y_v} \ln \left(\frac{P_g - P_{cp}}{P_g - P_v} \right), \quad (1)$$

being P_{cp} the pressure of vapour in equilibrium at the condensed-phase interface temperature. A remark is appropriate: in the original equation (see [8]) the density factor outside the logarithm is the density of the condensing vapour at the total pressure (which can be expressed as $\rho_v Y_v$).

For the heat flow:

$$Q_{cp}^v = G_{cp}^v i_{cp}, \quad (2)$$

$$Q_{tra} = h_g \frac{\varepsilon}{1 - \exp(-\varepsilon)} (T_g - T_{cp}), \quad (3)$$

$$Q_l = U (T_{cp} - T_r), \quad (4)$$

being U the thermal transmittance, combined conduction-convection coefficient between the refrigerant and the interface liquid-gas.

This permits us to determine the mass and heat passes from the vapour to the walls per unit of tube length by film condensation towards the internal tube and the carcase. Also, we determine the equilibrium temperatures at the liquid-gas interface, both in the central zone (T_{cp}) and in the carcase (T_c), by solving two non linear systems, by using Ridders method [9].

(ii) *Gravitatory and turbulent (towards the duct walls) depositions, diffusiophoresis and thermophoresis*. These depends strongly on the particle size, through the corresponding deposition velocity (we do not show the expressions).

The deposition rates (shown only for the internal tube for brevity) per tube unit length are the following ones [10].

Gravitatory:

$$\eta_{gr} = \frac{v_p^{gr} d_o}{\pi u_g (d_c^2 - d_o^2)}, \quad (5)$$

Turbulent diffusion to the walls:

$$\eta_{tu} = \frac{4v_p^{tu} d_o}{u_g (d_c^2 - d_o^2)}, \quad (6)$$

where the deposition velocity, v_p^{tu} , includes the inertial and the Brownian components.

Thermophoresis (only of application to the aerosol fraction present in a circular slab of width b_{th} near the internal tube wall):

$$\eta_{th} = \frac{4v_p^{th} d_o}{u_g b_{th} (2d_o + b_{th})}, \quad (7)$$

Diffusiophoresis (only of application to the aerosol fraction present in a circular slab of width b_{di} near the internal tube wall):

$$\eta_{di} = \frac{4v_p^{di} d_o}{u_g b_{di} (2d_o + b_{di})}, \quad (8)$$

2 Topics

The values of b_{th} y b_{di} are calculated from the diffusivity and the conductivity of the gas [8]. The deposition velocities for thermophoresis v_p^{th} , and diffusiophoresis, v_p^{di} , depend on the temperature and pressure gradients, respectively, near the external wall of the internal tube [10].

G_{cp}^p and G_{cp}^p are related to these deposition coefficients, η . The corresponding expressions are not shown by brevity.

(iii) *Vapour condensation/evaporation towards/from the particles.* The particle growth rate, I_p , is determined following Wagner [11], who includes in his model the non-dilute vapour case. In the present model, some additional modifications for taking into account the surface energy of the drops are included. Because the drop sizes are smaller than about 10 μm , we can dismiss convection and radiation in the heat exchange between particles, gas and walls [11]. During condensation or evaporation the total number of particles remains constant, varying only their size. The resulting expressions are not shown for the sake of brevity.

Once we know the heat and mass flows, and the equilibrium temperatures, the advance of gas in a slab of width Δz is posed. In each slab we know the variables T_g , P_g , T_r , x_n , x_v , x_r and ψ at the beginning. With the radial fluxes of heat and mass in equilibrium cited above we do the balances in the slab with the following steps:

1. Mass balance.

$$x'_n = x_n, \quad x'_v = x_v - \Delta x_v. \quad (9)$$

Δx_v includes both the pass of vapour to the particles and to the walls film.

2. Energy balance.

$$i_l \Delta x_v + \pi d_o Q_l \Delta z = x_n i_n + x_v i_v - x'_n i'_n - x'_v i'_v. \quad (10)$$

3. Momentum balance.

$$(P'_g - P_g) \frac{\pi}{4} (d_c^2 - d_o^2) = - \left[x'_v + \frac{f x'_v \Delta z}{2(d_c - d_o)} + \frac{1}{2} \Delta x_v \right] u' + \left[x_v + \frac{f x_v \Delta z}{2(d_c - d_o)} - \frac{1}{2} \Delta x_v \right] u \quad (11)$$

4. Particle diameter evolution.

$$D' - D = \frac{2 I_p \Delta z}{\pi u_g \rho_l D^3}. \quad (12)$$

5. PSD evolution (for each diameter):

$$\psi' = \psi \exp(-[\eta_{gr} + \eta_{tur} + \eta_{th} + \eta_{di}] \Delta z). \quad (13)$$

Steps 1, 2, y 3 have been obtained following the methods appearing in [12], and the other two derived from [3].

The PSD is discretized in diameter ranges (bins), which initially are logarithmically distributed between 0.01 μm and 10 μm . Since in the evolution we consider particles growing (by

condensation) or shrinking (by evaporation), the limits of each bin are modified in accordance with the distance to the inlet of the exchanger, following expression (12). If particles evaporate then $I_p < 0$ and particle diameter diminishes. In this case, when the diameter is less than 0.001 μm it is assumed that the particle evaporates totally, and its mass passes to the vapour phase.

With the balances shown above we calculate the values T'_g , P'_g , T'_r , x'_n , x'_v , x'_r and ψ' at the end of the slab of width Δz . In order to do this, we have to solve a non-linear system of equations (in this case a simple iterative method is enough). These variables are, in the subsequent step, the initial ones for the next slab. This process needs to use a value for Δz that is small enough for the treatment of the particles in the system (about 0.05 mm). As our model is able to be applied without particles (taking zero particles in the system is enough), in this specific case, an integration step of 10 mm suffices.

The liquid that appears in each slab in the internal tube and in the carcase is eliminated from the system, and does not have any effects in later slabs.

RESULTS

We have analyzed a realistic case of heat and mass transfer, whose parameters are shown in table I. We want mainly to analyze the behaviour of the system with and without particles. In order to do this, three related cases are compared:

- The one that corresponds to the parameters appearing in table I.
- The same parameter as case A, with the addition of a log-normal drops PSD with mean geometrical diameter of 0.1 μm , a standard geometrical deviation of 1.8, and a mass flow in particles (pure water) of 1.2 g/h.
- A case identical to B but with all the particles initially evaporated (148.7 g/h of total vapour).

Thus we compare the effect of adding the same amount of water mass as particles and as vapour.

Table I Main parameters of the exchanger.

Tubes material	steel 316L
Carcase diameter	66.93 mm
Interior diameter of the internal cylinder	28 mm
Exterior diameter of the internal cylinder	32 mm
Duct length	3.0 m
Absolute roughness of the duct	0.01 mm
Refrigerant	mineral oil
Refrigerant temperature at their exit	293 K
Refrigerant flow	43 kg/h
Gas pressure at the inlet	133.3 hPa
Gas temperature at the inlet	311.5 K
Incondensable (nitrogen) flow	206.7 g/h
Water vapour flow	147.5 g/h
Computational slab width	0.05 mm
Number of bins in the PSD	300
Minimum initial particle diameter	0.01 μm
Maximum initial particle diameter	10 μm

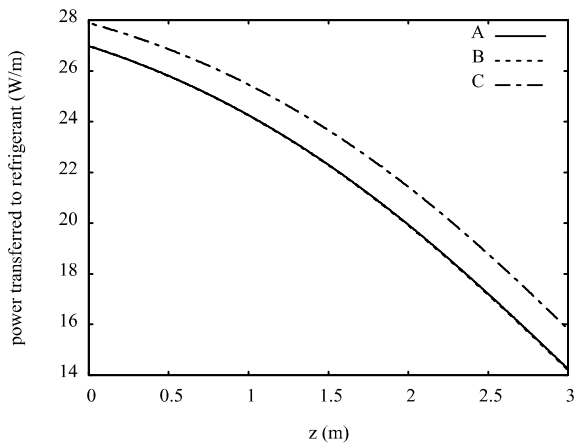


Figure 2 Power transferred to refrigerant per unit length as function of the distance to the exchanger inlet.

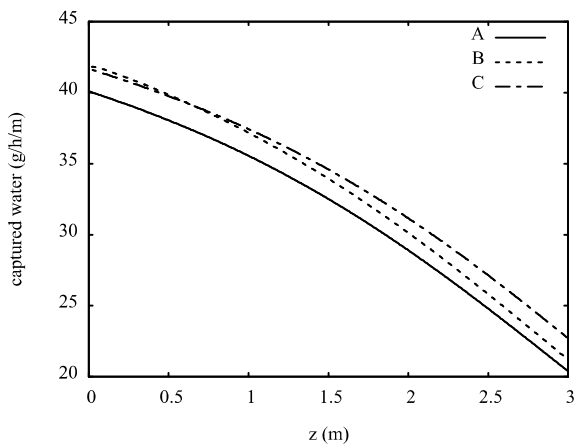


Figure 3 Captured water at the internal cylinder per unit length as function of the distance to the exchanger inlet.

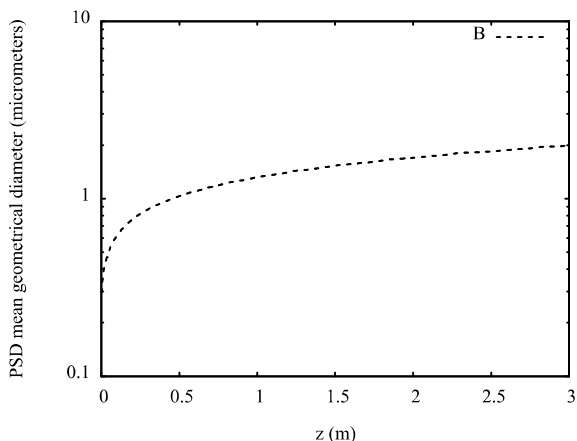


Figure 4 Mean geometrical diameter of the PSD as function of the distance to the exchanger inlet.

We have analyzed other cases, with similar behaviour. We do not show results for brevity.

Firstly we analyze the flux of heat towards the refrigerant, shown in fig. 2. For cases A and B this flux is about the same, being higher in the case C. This is due to the greater amount of vapour in gas phase. Therefore, water temperature in the condensed phase in the internal tube is also higher. The almost equality of the cited parameter for cases A and B is easily explained: practically all the heat involved in the condensation/evaporation in the particles is consumed in changing the particles temperature, having little effects on the environmental gas [3].

On the other hand, the total condensed mass in the internal tube does depend on the presence of particles in the system. We can see this in fig. 3. This is due to the fact that particles are deposited on the walls by several of the phenomena described above. Liquid passes to the walls by both film condensation and drops deposition. The presence of particles implies that the amount of condensed water recuperated in the exchanger has diminished (for a nearly equal amount of heat transferred to the refrigerant).

In case B, particles grow very fast (some milliseconds [11]). This effect is practically instantaneous in our model. Once the stationary equilibrium is reached, as the gas advances with the particles in the exchanger, the smaller sizes will disappear by evaporation. This implies an increase of the mean geometrical diameter of the particles remaining in the gas flow, as we can see in fig. 4.

In fig. 5 the saturation ratio of the water vapour in the gas phase is shown. For cases A and C without particles the system begins with $S > 1$, but after some time the saturation ratio decreases below one. This is possible because the internal tube is colder than the gas, and gets much colder as the gas approaches the outlet of the exchanger. This forces more film condensation. In case B, right from the beginning the presence of particles forces the water vapour pressure in gas phase to be adjusted to a value near the equilibrium (saturation), because initially particles grow due to the vapour excess. The saturation rate slowly diminishes along the exchanger, maintaining a value only slightly greater than one. Actually, particles act as a water “buffer” that controls the system behaviour.

In fig. 6 the amount of water in drop form along the exchanger is shown. At the beginning there existed a flow of 1.2 g/h in particle form. However, in the first milliseconds, due to the supersaturation, about 9 g/h of water vapour initially present in gas phase condenses in the particles. From this moment on, some particles evaporate slowly and the others are deposited on the walls, resulting in a diminishing water flow in particles form. Nevertheless, as we can see in the cited fig., at the end of the exchanger a lot of particles remain.

IMPROVEMENTS OF THE MODEL

Although our model includes important effects that were not considered previously, there are possibilities for improving it.

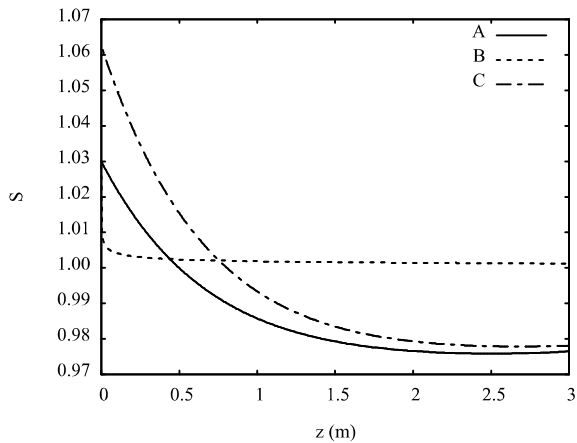


Figure 5 Saturation ratio of water vapour as function of the distance to the exchanger inlet.

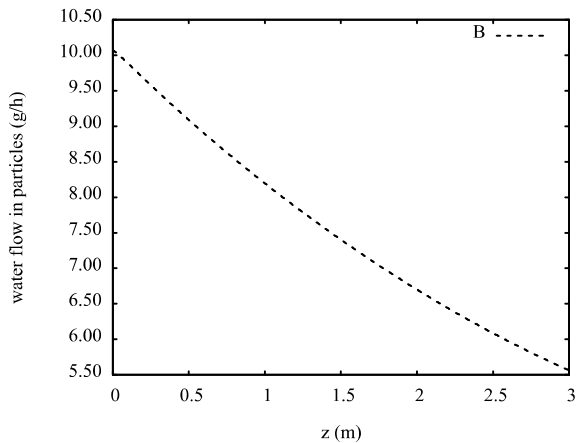


Figure 6 Water mass flow in the particles as function of the distance to the exchanger inlet.

The first one consists of the radial subdivision of the duct into several concentric regions, each of which has its own set of control variables (vapour pressure, temperature, etc.), that are related to one another along the boundaries between the regions. This would allow us to analyze transitory effects (with the present model this cannot be done because of the stationary character of the model), since the points that are far away from the walls will be affected not instantaneously (the steady state condition will be posed for the regions adjacent to the walls).

Another improvement consists of including homogeneous nucleation, with the presence of anions and cations into the system. This will create small particles, which will obviously behave as condensation nuclei.

Finally we expect to test the model experimentally by building a specific device to measure the number and distribution of particles in several points along the heat exchanger.

CONCLUSION

A model for vapour condensation in a double pipe heat exchanger has been developed. It takes into account: conduction and convection; film vapour condensation; aerosol particle size distribution; heterogeneous nucleation in non-diluted conditions; some particle phenomena: gravitational and turbulent depositions, thermophoresis and difussiophoresis; and mass, momentum and energy balances in the heat exchanger.

The model has been applied to several cases (only one shown), with and without particles. In all of the cases, the presence of particles leads to a decrease of the condensed vapour in comparison with the case without particles (only vapour in gas phase), which remain at the outlet of the heat exchanger. However the heat flux transferred to the refrigerant is not much sensitive to the presence of particles.

ACKNOWLEDGEMENT

We thank the “Plan de Ciencia y Tecnología e Innovación 2006-2009 del Principado de Asturias” for financial support under the grant BP07-121 and the project PC06-039.

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