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DEVELOPMENT OF A WET VAPOR HOMOGENEOUS LIQUID METAL MHD POWER SYSTEM

FINAL REPORT
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ABSTRACT

A feasibility study for the approval of liquid metal seeds recovery from a liquid metal vapor-inert gas mixture was conducted and presented in this report. The research activity included background studies on processes relating to mixing stream condenser performance, parametric studies and its experimental validation.

The condensation process under study includes mass transfer phenomena combined with heat transfer and phase change. Numerical methods were used in order to solve the dynamic equations and to carry out the parametric study as well as the experimental data reduction. The MSC performance is highly effected by droplet diameter, thus the possibility of atomizing liquid metals were experimentally investigated.

The results are generalized and finally used for a set of recommendations by which the recovery of seeds is expected to be feasible.

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NOMENCLATURE

a	-	constant
A	-	area
B _M	-	spalding mass transfer number
B _T	-	spalding heat transfer number
C _D	-	discharge coefficient
C _p	-	heat capacity
d	-	droplet diameter
D	-	diameter
D _G	-	binary diffusion coefficient
g	-	gravitational acceleration
h _{fg}	-	heat of evaporation
J	-	condensation volumetric rate
K	-	thermal conductivity, atomizer losses coefficient
l, L	-	length
Le	-	Lewis number
m	-	mass
$\dot{m}$	-	mass flow rate
N	-	number of drops
Nu	-	Nusselt number
P	-	pressure
Pr	-	Prandtl number
Q	-	liquid volume fraction, heat transfer rate
r	-	droplet radius
Re	-	Reynolds number
S	-	standard deviation of the mean
Sc	-	Schmidt number
Sh	-	Sherwood number
SMD	-	Sauter mean diameter
t	-	time
T	-	temperature
U	-	velocity

We	-	Weber number
x	-	area ratio
Y	-	mass fraction
Z	-	longitudinal coordinate

Subscripts

a	-	ambient, air
eq	-	equivalent
G	-	gas
i	-	inlet, index number
I	-	internal
j	-	index number
k	-	droplet
L	-	liquid
m	-	maximal
o	-	orifice, outlet
p	-	port
R	-	relative
s	-	surface, swirl chamber
sat	-	saturation
0	-	minimal
V	-	vapor

Greek Symbols

σ	-	surface tension
ρ	-	density
δ	-	dispersion index
μ	-	viscosity

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1. INTRODUCTION

The ultimate goal of the Strategic Defence Initiative Organization (SDIO) is to provide security for the United States and its allies. This goal could be promoted by the adoption of the MHD multimewatt power generation concept, as has been already demonstrated. This concept has numerous advantages, among them the following: The absence of moving parts, the use of high-temperature working fluids, minimal maintenance requirements, high specific power, rapid start up and shut down, the possibility of using various working fluids, and compatibility to a variety of components connected to the MHD generator. Thus the Department of Energy (DOE) had initiated the Program Research and Development Announcement for Innovative Engineering Research Approaches to Support the Development of Space Based Multimewatt MHD Power Systems (PRDA RA22-87 PC 90271) that complements the "Feasibility Assessment Program for Spaced Based Multimewatt MHD Power Systems" Ref. No. DE-RP22-86PC90270"

As an outcome of this DOE initiative, the proposal entitled "Development of a Wet Vapor Homogeneous Liquid Metal MHD Power System," submitted by Solmecs Ltd. (Israel)/Center for MHD Studies at Ben Gurion University, Israel, has been selected for award, under Project No. 22-88PC88653.000

In this project the feasibility of the separation of vapor from a gas stream by means of a mixing stream condenser (MSC) has to be studied. Figure 1.1 shows a schematic of this condenser, which operates as follows. A hot stream of H_2 seeded with C_s vapor flows downstream into the MSC entrance zone. Subcooled liquid C_s is atomized into the gas-vapor mixture at this zone. The C_s droplets serve as condensation nucleation centers and C_s concentration in the gas-vapor bulk stream is decreased due to vapor mass transfer. The seeds are thus recovered into the liquid droplets and at the exit from the MSC, liquid C_s is be separated from the gas-vapor mixture.

In the process analysis along the MSC, one should first consider the thermodynamics of the process: heat and mass balances of the adiabatic process are to be carried out.

In addition, the transient phenomena are to be studied carefully, in order to evaluate the length of the MSC which will recycle a given quantity of C_s seeds. Two mechanisms

Figure 1.1 Schematic of mixing stream condenser (MSC)

govern this process. One is heat transfer rate from the hot gas-vapor mixture to the subcooled C_s stream including condensation heat transfer. The second mechanism is the mass transfer from high C_s concentration at the gas-vapor bulk flow to low C_s concentration around the droplets, where condensation takes place. For the gas-liquid separation process at the MSC exit, one should remember that the device is to be operated in non-gravity conditions, and the separator design should be planned accordingly.

The literature survey presented in this report covers the following topics: direct contact phase change, two-phase gas-liquid flow patterns, spray formation and separators.

A parametric study was conducted in two stages. First, a thermodynamic analysis of the process was carried out. The liquid metal and gas flow rates were calculated, and different alternatives for heat sinks were studied. Then a dynamic model was selected for the transient phenomena along the MSC. A single droplet model which was previously developed for direct contact evaporation, was modified and employed to the case under study. The integration of the single droplet model is the basic mechanism of the homogeneous mist flow condensation model used to calculate the MSC characteristics, among them the length required for the completion of the process, and its dependence on droplets diameter.

The experimental phase of the project was conducted in two stages. First the theoretical model was validated, thereafter experiments with liquid metal were carried out. The latter are extremely important for the feasibility assessment, due to the lack of practical experience in this area. The findings are presented in the report, and are summarized as recommendations toward the run of this kind of condenser at the defined conditions.

2. LITERATURE SURVEY

2.1 Direct-Contact Heat Transfer

Direct-contact heat exchangers have several advantages over surface type ones, in cases where contamination of one stream on the other do not present a problem.

The advantages include the lack of surfaces to corrode and/or foul, the potentially superior heat transfer for a given volume of heat exchanger due to large heat transfer surface and the ability to transfer heat at much lower temperature differences, lower pressure drop and lower capital cost due to very simple construction (Jacobs, 1988).

The different alternatives for direct contact condensers could be divided into four groups (Keith and Boehm, 1988), which are shown in Fig. 2.1.1. These are:

- Drop-type condensers, which refer to condensation on sprays or drops of liquid coolant that are injected into a chamber filled with vapor gap or vapor mixture. (Fig. 2.1.1a)
- Jet and sheet-type condensers, where the condensation occurs on the intermediate surface between the coolant liquid and the vapor stream. (Fig. 2.1.1b)
- Film-type condensers, which are strongly related to jet and sheet condensers in that the condensation takes place on a thin film of coolant. In this case the coolant flows over a solid substrata, while in the former case the jet or sheet is unsupported. (Fig. 2.3.3c)
- Bubble-type condensers, refer to the case where vapor is injected as a jet or bubbles into a continuous stream or a pool of coolant. (Fig. 2.1.1d)

It is likely that the advantages of direct-contact condensation mode is best presented in the drop-type condenser.

Early work dealt primarily with the condensation of saturated pure vapor, where the liquid and the vapor are of the same substance. The existence of non-condensable gases in the vapor stream was less dealt with in the literature. Evaporation of drops moving through a gas-vapor mixture is based on the same physical principles as condensation, and in the area of gas quenching and combustion systems the process includes substances which do

Figure 2.1.1 Direct-contact condensers

a) spray-type b) jet-type c) film-type d) bubble-type

not undergo a phase change. Many publications relate to this case and in the current study the model which is presented by Abramzon and Sirignano (1987, 1989) on spray combustion is adopted. In Chap. 4 their model is presented in detail.

2.2 Two-Phase Flow Pattern

The topic of flow pattern prediction in two-phase flow is of great importance for the feasibility assessment of MSC. As discussed earlier condensation will be more efficient as droplets become smaller, and thus for a given liquid flowrate, the heat transfer will be higher. One must be sure that under a given set of flow conditions, a desired two-phase flow pattern will exist. In our case we are interested in a mist-flow pattern where tiny droplets are surrounded by continuous gas flow.

Two-phase flow parameters are used to predict the flow pattern by means of flow pattern maps. It is most common to define the coordinates of these maps either by liquid and gas superficial velocities (Taitel et al., 1980; McQuillan and Whalley 1985; Taitel and Dukler, 1976) or by liquid and gas-superficial momentum fluxes (Hewitt and Roberts, 1969). The transition mechanics from one flow pattern to another studied by the above mentioned authors (and others) and equations for transition curves are presented. The employment of superficial velocities for maps presentation has also found applications for liquid metal flows (Unger, 1987; El-Boher et al., 1988) is discussed herein.

In vertical two-phase flow, in accordance with observations (Taitel et al.), a bubble flow pattern exists at low liquid and gas superficial velocities. See Fig. 2.2.1. An increase in the gas flow rate will change the pattern into slug or churn flow patterns as a result of gas bubble coalescence and agglomeration. At still higher gas flow rates, when an equilibrium between drag force on the droplet surface and gravity is achieved, a mist flow pattern exists, and discrete droplets will be "held-stable" in the gas flow. This pattern will exist as soon as the required gas velocity will be achieved, no matter what the liquid flow rate is. An increase of liquid flow rate at a bubble flow pattern will cause a finely dispersed bubble-flow pattern where turbulent fluctuations of the liquid prevent gas bubbles from agglomeration, and the gas distribution could homogeneously be related. It is logical to assume that gravity acting on the mixture will not destroy the homogeneity after it has been reached.

Figure 2.2.1 Flow patterns in vertical two-phase flow

Figure 2.2.2 Flow patterns in horizontal two-phase flow.

In horizontal two-phase flow pattern studies, the transition from one pattern to another also occurs in conjunction with increasing phase flow rates.

At low superficial velocity values, a stratified flow pattern is observed, (see Fig. 2.2.2), and as the gas-flow rate is increased, the intermediate surface becomes wavy, until liquid forms a bridge over the entire pipe area. The transition from a wavy flow pattern to a mist flow pattern occurs, according to Taitel and Dukler, when gravitational forces on the liquid are less than the Bernoulli effect on the pressure in the gas phase over the wave.

Many other authors have also suggested other mechanisms and other equations for these transitions; however, it should be noted that the models presented by Taitel are generally accepted for two-phase flow pattern prediction.

The transition from one pattern to another due to a change in flow parameters is considered in these models. The gas and liquid entrance zone geometry are not reported as one of the parameters governing the downstream flow pattern. Thus, the use of these models for a liquid spray which mixes with a gas stream is limited. Furthermore, gravitational effects do play a major role in these models.

The formation and stabilization of sprays is addressed in the following section.

2.3 Atomization

Spray Formation

The formation of sprays relates to a number of different mechanisms. A static drop is formed at the end of a vertical burette of diameter D_0 when the gravitational force pulling the drop downward is equal to the surface force due to the surface tension. At equilibrium.

$$mg = \pi D_0 \sigma \quad 2.3.1$$

and the drop diameter d is

$$d = \left(\frac{6D_0\sigma}{\rho_L g} \right)^{1/3} \quad 2.3.2$$

After leaving the burette, under equilibrium conditions the internal drop pressure P_i at any point on the surface is equal to the external pressure P_a plus the surface tension pressure P_σ . For a spherical drop

$$P_\sigma = \left(\frac{4\sigma}{d} \right) \quad 2.3.3$$

According to Giffen and Muraszew (1953), if P_a is large compared to P_σ , an appreciable change in the external pressure may deform the drop and finally result in rupturing it into a smaller one.

Among three types of deformations which cause drop break up in flowing gas, Hinze (1955) identifies the flattening into an oblate ellipsoid as a result of aerodynamic pressure or viscous stress as the one which is produced by parallel and rotating flows (see Fig. 2.3.1). (The other two types of deformations are a result of plane hyperbolic or Couette flows, and irregular flow patterns).

For low-viscosity liquids of the deformation of a drop is determined primarily by the ratio of the aerodynamic forces, represented by $0.5\rho_a U_R^2$ and the surface tension forces which are related to σ/D . These two opposing forces form a dimensionless group, the Weber number:

$$We = \rho_a U_R^2 d / \sigma \quad 2.3.4$$

The higher the Weber number, the larger are the deforming external pressure forces compared with the reforming surface tension forces.

A critical relative velocity U_R could be determined for drop break-up. In many practical situations, such a high relative velocity does not exist; however, it is logical to assume that in turbulent flows, dynamic pressure forces of the turbulent motion will determine the size of the largest drops.

Drops are also formed when a liquid jet emerges from a nozzle as a continuous flow and reaches instability. Here oscillations and perturbations are the result of the competition

Figure 2.3.1 Basic types of globule deformation

Figure 2.3.2 Schematics of pressure swirl atomizers

between cohesive and disruptive forces. The fundamental principle consists of increasing the jet surface area until it becomes unstable and disintegrates.

Rayleigh compares the surface energy of a disturbed configuration of a cylindrical jet cross section with that of an undisturbed column and calculated the potential energy of the disturbed one. The conclusion of this analysis is that all disturbances with wavelengths greater than its circumference will grow for nonviscous liquids, under laminar flow conditions. This conclusion has found general acceptance in later theories including turbulent and viscous liquid jets (Lefebvre, 1989).

According to Reitz (1978), the break-up of jets is accelerated, and the drop radius is decreased as the liquid Reynolds number increases.

Liquid flat sheets are also disintegrated into tiny droplets, and according to York et al. (1953), the major factors for the breakup are instability and wave formation at the interface between the continuous and discontinuous phases.

Conical sheets are produced by pressure swirl or simplex atomizers. A schematic of this kind of atomizer is shown in Fig. 2.3.2. The liquid is forced into the atomizer at a certain axial velocity, enters the swirl chamber, rotates and thereafter leaves the atomizer as a conical hollow sheet of disintegrated fine droplet spray. The relatively high pressure drop along this path limits the available flow rates. In order to overcome this barrier, sometimes a part of the inlet stream is driven directly to the exit nozzle, through a second centric port. In this configuration the discharged spray assumes a solid cone geometry with larger drop diameters at the center and tiny droplets at the circumference.

The theory of attenuating conical sheets of this kind has yet to be developed. However, according to York, disintegration of the sheet is basically described by the Rayleigh mechanism. Despite the lack of a full theoretical presentation, the wide range of applications of this kind of atomization allowed the publication of a large number of experimental studies. Among the other advantages of that kind of atomizer, are their simplicity, availability of wide range of cone angle and relatively low price. Based on these it was suggested to adopt these atomizers for the purpose of the current study. Further discussion of the atomizer characteristics is presented in the section to follow.

Drop size distribution

Practical nozzles do not produce sprays of uniform drop size at any operating conditions. Sprays can be regarded as a spectrum of drop sizes distributed about some arbitrary defined mean value. According to Fraser and Eisenklam (1956), Fig. 2.3.3 indicates the range of drop sizes as they occur in some natural phenomena and also as commonly produced by atomizers. A typical histogram of drop-size is shown in Fig. 2.3.4. Here, all drop diameters of this typical spray were divided into classes or groups in the range of 17 μm . Another method of presentation of drop distributions is by their volume. Each increment in the histogram represent the total volume of drops in a defined range of diameters. ΔQ_i , the volume of drops of the i 'th class is:

$$\Delta Q_i = \Delta N_i \frac{\pi}{6} [0.5(d_i + d_{i+1})]^3 \quad 2.3.5$$

The distribution function or frequency function is presented as a mathematical function in the absence of any fundamental mechanism or model on which to build a theory of drop-size distribution. One example is the normal distribution function, which is based on totally random occurrences of a given drop size:

$$\frac{dN}{dd} = \frac{1}{\sqrt{2\pi}S} \exp \left[-\frac{1}{2S^2} (d - \bar{d})^2 \right] \quad 2.3.6$$

where $\bar{d}$ is a mean diameter value and S is the standard deviation from this mean.

For drop size distribution in sprays, Mugele and Evans (1951) proposes the following function:

$$\frac{dQ}{dy} = \frac{\delta}{\sqrt{\pi}} \exp \left[-\frac{\delta^2 y^2}{\pi} \right] \quad 2.3.7$$

where y is given by:

$$y = \ln \frac{ad}{d_m - d} \quad 2.3.8$$

Figure 2.3.3 Spectrum of drop sizes

Figure 2.3.4 Typical drop size histogram

δ is related to the standard deviation of y and hence, of d , and a is a dimensionless constant. Here, d_m denotes the maximum drop size and as y goes from $-\infty$ to $+\infty$, the diameter d goes from its minimal value d_0 to d_m .

The Sauter Mean Diameter (SMD) is the diameter of the drops whose ratio of volume (or mass) to surface area is the same as that of the entire spray:

$$SMD = \frac{\sum N_i d_i^3}{\sum N_i d_i^2} \quad 2.3.9$$

or

$$SMD = \frac{\int_{d_0}^{d_m} d^3 (dN/dd) dd}{\int_{d_0}^{d_m} d^2 (dN/dd) dd} \quad 2.3.10$$

This mean value is proposed by Mugele and Evans for use in mass transfer processes. These authors have used the distribution function presented above (Eq. 2.3.7) and have reached a mathematical expression for the SMD; however, this expression is difficult to calculate. Furthermore, the physical processes in atomizers are not yet fully understood and the prediction of mean diameter based on the equations derived from basic principles is not satisfactory yet. As an outcome of these shortages, correlations based on experimental data generalizations are used.

One of the proposals of Lefebvre for pressure swirl atomizers is the following expression:

$$SMD = 2.25 \left(\frac{\sigma \mu_L m_L}{\rho_a} \right)^{0.25} \cdot \Delta P_L^{-0.5} \quad 2.3.11$$

which is considered to be one of the best available for engineering calculations of mean drop size.

Flow in Atomizers

The discharge coefficient is related to the nozzle flow rate by the equation:

$$\dot{m}_L = C_D A_o (2\rho_L \Delta P_L)^{0.5} \quad 2.3.12$$

A detailed schematic of swirl pressure atomizer is shown in Fig. 2.3.5, and the following are its dimensions:

L_p	-	inlet port length
D_p	-	inlet port diameter
L_s	-	swirl chamber height
D_s	-	swirl chamber diameter
l_o	-	orifice length
D_o	-	exit nozzle diameter
2θ	-	cone angle of discharge flow

The air core exists along the swirl chamber and exit nozzle centers are essential to atomizer performance. From theoretical considerations it can be shown that the discharge coefficient, C_D , relates to atomizer geometry as follows:

$$C_D = \left[\frac{(1-x)^3}{1+x} \right]^{0.5} \quad 2.3.13$$

where

$$x = \frac{A_a}{A_o} \quad 2.3.14$$

A_a is the air core area at nozzle exit plane, and A_o is the nozzle area. Equation 2.3.13 was found to give C_D values that were too low compared to experimental data (Giffen and Muraszew), and correction factors were introduced.

Other investigations show that C_D could be related to atomizer geometry without considering air-core area, as the latter is not easily be measured, and an expression of the form:

$$C_D = f\left(\frac{A_p}{D_s D_o}\right) \left(\frac{D_s}{D_o}\right) \quad 2.3.15$$

Figure 2.3.5 A detailed schematic of a swirl pressure atomizer

is suggested by different authors (Taylor, 1950; Eisenklam, 1961; Dombrovinsky and Hassan, 1969). Rizk and Lefebvre (1985) came out with the coefficient:

$$C_D = 0.35 \left(\frac{A_p}{D_s D_o} \right)^{0.5} \left(\frac{D_s}{D_o} \right)^{0.25} \quad 2.3.16$$

Jones (1982) included in his correlation the effect of liquid velocity on C_D as well, and his correlation is:

$$C_D = 0.45 (Re_{\text{nozzle}})^{-0.02} \left(\frac{l_o}{D_o} \right)^{-0.03} \left(\frac{L_s}{D_s} \right)^{0.05} \left(\frac{A_p}{D_s D_o} \right)^{0.52} \left(\frac{D_s}{D_o} \right)^{0.23} \quad 2.3.17$$

Note that the exponents in Eq. 2.3.16 are close to those presented by Jones for the same nondimensional groups.

The correlations presented above (Eq. 2.3.16 and 2.3.17) were in good fit to experimental data achieved by using atomizers of $\frac{D_s}{D_o}$ at the range 1.41-8.13.

The geometrical parameters and fluid properties included in Jones experimental study are listed in Table 2.3.1.

Table 2.3.1

Parameters and properties included in Jones experimental study

$$\frac{D_s}{D_o} = 1.41 - 8.13$$

$$\frac{l_o}{D_o} = 0.1 - 0.9$$

$$\frac{L_s}{D_s} = 0.31 - 1.26$$

$$\frac{L_p}{D_p} = 0.79 - 3.02$$

$$\frac{A_p}{D_o D_s} = 0.19 - 1.21$$

$$\frac{\mu_L}{\mu_a} = 279 - 2235$$

$$\frac{\rho_L}{\rho_a} = 694 - 964$$

$$Re_{\text{nozzle}} = 11.5 \cdot 10^3 - 3.55 \cdot 10^5$$

$$We_{L, \text{nozzle}} = 1.913 \cdot 10^3 - 21.14 \cdot 10^3$$

2.4 Separators

Separators for gas liquid mixture were investigated during the last few years at the Center for MHD Studies, especially for application in liquid metal-gas systems. An effective separation process is the one which considers low or negligible carry under of bubbles into the liquid stream, and carry over of droplets into the gas stream. High efficiency is achieved when the energy losses are minimal. The conclusion of an early study (Farchi, 1990) is that a cylindrical cyclone separator with tangential two phase inlet and liquid exit ports is preferred.

The separation process is governed by high centrifugal forces, and gravity effect is negligible. Good performance is achieved with volute type spiral housing of the inlet and exit ports, which cause a uniform radial velocity distribution. The latter is important in order to prevent instabilities which may bring to carry under and carry over. In this geometry the kinetic inlet energy is recovered. In addition such a separator, if designed properly for given flow conditions, is compact and minimizes liquid metal inventory.

3. THERMODYNAMIC ANALYSIS OF MIXING STREAM CONDENSER

3.1 Basic Thermodynamic Model

Figure 3.1.1 MSC Control Volume

In order to evaluate the performance of the MSC, a thermodynamic analysis was carried out. A schematic diagram of the condenser control volume is shown in Fig. 3.1.1. At the inlet section - i, the conditions are defined as follows: The inert gas mass flow rate, $\dot{m}_{Gi}$ enters with pressure (partial) of P_{Gi} , and its temperature is T_{Gi} . The vapor mass flow rate $\dot{m}_{Vi}$ enters with pressure (partial) of P_{Vi} and its temperature is T_{Vi} . The subcooled liquid flow rate, $\dot{m}_{Li}$, enters with pressure P_{Li} and its temperature is T_{Li} . At the exit, the gas, vapor and liquid flow rates, partial pressures and temperatures are $\dot{m}_{Go}$, $\dot{m}_{Vo}$, $\dot{m}_{Lo}$, P_{Go} , P_{Vo} , P_{Lo} , T_{Go} , T_{Vo} and T_{Lo} , respectively.

The assumptions for the calculations are:

- 1) Gas and vapor partial pressures obey Dalton's rule

$$P_{Gi} + P_{Vi} = P_i \quad 3.1.1$$

$$P_{Go} + P_{Vo} = P_o \quad 3.1.2$$

- 2) Frictional and accelerational pressure drops along the MSC are neglected. This assumption was found valid after calculated flow rates values were considered.
- 3) The process along the MSC is adiabatic. The condenser is isolated from its ambient.

- 4) Change of kinetic energy along the MSC is negligible.
- 5) Change in potential energy does not occur (horizontal condenser or non-gravity space)
- 6) Gas solubility is neglected.

The first law of thermodynamics yields:

$$\sum_{j=1}^3 \dot{m}_j C_{p_j} T_j \text{ (at inlet)} = \sum_{j=1}^3 \dot{m}_j C_{p_j} T_j \text{ (at outlet)} \quad 3.1.3$$

In our process, the gas and vapor enter the MSC at the same temperature

$$T_{Vi} = T_{Gi} = T_i \quad 3.1.4$$

and their partial pressures relate to each other:

$$\frac{P_v}{P_G} = (\dot{m}_v / \dot{m}_G) (M_G / M_v) \quad 3.1.5$$

where M is the molecular weight.

Furthermore, the condensation process ends at the point where the 3 component — vapor, gas, and liquid — are at the same temperature, which is the saturation temperature, and the relative humidity is equal to unity. Thus at the exit,

$$T_{Vo} = T_{Go} = T_{Lo} = T_o \quad 3.1.6$$

The extracted vapor mass flow rate, $\Delta \dot{m}$, is defined as:

$$\dot{m}_{Vi} - \dot{m}_{Vo} = \Delta \dot{m} \quad 3.1.7$$

and

$$\dot{m}_{Lo} - \dot{m}_{Li} = \Delta \dot{m} \quad 3.1.8$$

The subcooled liquid is heated from its inlet temperature T_{Li} to the saturation temperature $T_o = T_{sat}$ (assuming that the process is completed). This heat is absorbed from the vapor and the gas as follows: The inert gas is cooled from its inlet temperature T_{Gi} to the exit temperature T_o , and the vapor is first cooled to the saturation temperature (assuming only

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homogeneous condensation is possible); then the extracted vapor flowrate undergoes condensation. According to this scheme, Eq. 3.1.3 reduces to:

$$(\dot{m}_G C_{pG} + \dot{m}_{v_i} C_{pV})(T_i - T_{sat}) + \Delta \dot{m} h_{fg} = \dot{m}_L (T_{sat} - T_{Li}) \quad 3.1.9$$

Dividing Eq. 3.1.9 by $\dot{m}_G$ yields:

$$(C_{pG} + \frac{\dot{m}_{v_i}}{\dot{m}_G} C_{pV})(T_i - T_{sat}) + \frac{\Delta \dot{m}}{\dot{m}_G} h_{fg} = \frac{\dot{m}_L}{\dot{m}_G} (T_{sat} - T_{Li}) \quad 3.1.10$$

At a given set of inlet conditions e.g., pressures, temperatures and flow rates, the solution of Eq. 3.1.10 yields the liquid flow rate required for a given extraction factor: $(1 - \Delta \dot{m} / \dot{m}_{v_i})$. For this solution, Eq. 3.1.5 is used for calculation of exit vapor partial pressure and is combined with an appropriate saturation-temperature curve fit equation (For example those of Vargaftik, 1975). Thermodynamic properties are assumed to be constant along the process and values are taken at an average temperature.

3.2 Cesium-Hydrogen Cycle

Referring to the statement of work (SOW), the conditions for the process are tabulated in Table 3.2.1

Table 3.2.1 - Process Conditions for MSC According to SOW

Gas:	hydrogen - H ₂
$\dot{m}_G$:	4 kg/s
subcooled liquid and vapor substance:	Cesium - Cs
inlet gas temperature T _i :	1000°K
inlet vapor to gas mass flow rate ratio:	1:100
extraction factor: $1 - \frac{\Delta \dot{m}}{\dot{m}_{v_i}}$	1:100
total inlet pressure:	0.02 MPa

The result of this analysis is that for an inlet subcooled liquid temperature of 305°K, the recovery of 99% of the vapor seeds requires about 2300 kg/s of subcooled liquid. In this process at given extraction factor and concentration, the vapor flow rate at the exit defines its partial pressure and hence the saturation temperature. Thus the decrease of gas temperature and the increase of liquid temperature along the MSC are determined. In addition heat capacity of the H₂, compared to that of the cesium is very high ($C_{pH_2}/C_{pCs} \cong 14.2/0.25$). These two arguments result in the high liquid flow rate value required for the process. Note that most of the heat transfer takes place between the H₂ and the subcooled liquid.

This analysis has been further expanded to consider flow conditions at the neighborhood of those listed above and the results are shown in Fig. 3.2.1. as the extraction factor vs. subcooled liquid flow rate. The ratio of inlet vapor to gas mass fraction is selected as a parameter, while at all runs, the H₂ flow rate and inlet temperature are kept constant: $\dot{m}_G = 4$ kg/s, $T_{Gi} = 1000^\circ\text{K}$ and inlet subcooled liquid temperature is 305°K. As seen from the figure, an increase in the extraction factor is associated with a decrease of liquid flow rate, and at a given extraction factor the increase of inlet vapor to gas fraction also decreases liquid flow rate. The higher the vapor flow rate is, the higher are the partial pressure, saturation temperature and liquid temperature gradient. The relative influence of increasing extraction factor or initial concentration is more significant at lower vapor concentration.

The problems which could arise from the high values of subcooled liquid flow rates derived from this analysis has suggested some alternatives that will reduce the subcooled liquid flow rate.

First, an adiabatic turbine allocated downstream the gas-vapor path enables a reduction of the inlet temperature (and pressure) of the mixture. Note that the major part of the subcooled liquid heating is required for the cooling of the H₂ stream, as mentioned above. The same effect of reducing the heat absorption from the H₂ stream by reducing its temperature could be achieved by a second alternative, which is to inject liquid H₂ downstream the mixture path. This will compensate the highly specific heat influence. A third alternative is to inject solid C_s particles instead of subcooled liquid, absorbing heat during the melting process. Alternative 3 could be adapted either together with alternatives 1 or 2. These alternatives are shown in Figs. 3.2.2. and 3.2.3. The influence of the selection of the different alternatives on liquid flow rate is shown in Fig. 3.2.4. Resulted

C_s flow rates are plotted vs. the extraction-factor for gas to vapor flow rate ratio of 0.01. The inlet conditions are those detailed in Table 3.2.1. Comparing Fig. 3.2.1 to Fig. 3.2.4. shows a significant reduction in C_s flow rate, which influences C_s inventory, capital cost (pumps, power etc.), weight and other important parameters of the system.

Figure 3.2.1

**Subcooled liquid flow rate vs. extraction factor for Cs-H₂ MSC.
Inlet vapor concentration is the parameter.**

Figure 3.2.2 Schematic of MSC with a turbine at the inlet

Figure 3.2.3 Schematic of MSC with liquid hydrogen injection

Figure 3.2.4

**Subcooled liquid flowrate vs. extraction factor for Cs-H₂ MSC
combined with different heat sinks.**

- a) Turbine installed and subcooled Cs injection**
- b) Turbine installed and solid Cs particle injection**
- c) Injection of liquid H₂ and subcooled Cs**
- d) Injection of liquid H₂ and solid Cs particles**

3.3 Other Fluid Pairs

The thermodynamic parametric study was also run for other fluid pairs, in order to estimate flow conditions needed for the design of simulation experiments: namely, mercury as liquid metal and helium or nitrogen representing the inert gas. For the Hg-He pair, the gas to liquid Cp ratio is about 40, while for the pair Hg-N₂ this ratio is only about 5.2.

Another important parameter is the gas to liquid metal molecular weight ratio, which influences the vapor partial pressure at the exit, and hence the saturation temperature. This value is about 65 for Cs-H₂, 50 for Hg-He and 7.16 for Hg-N₂. The initial conditions for the parametric study of the Hg-He and Hg-N₂ loops are listed in Tables 3.3.1. and 3.3.2 respectively. The results of this parametric study are shown in Figs. 3.3.1 to 3.3.7. In Figs. 3.3.1 to 3.3.6 Hg-He, loop performance is presented. In Fig. 3.3.7 the results of the Hg-N₂ loop parametric study is shown. The vapor to gas mass flow rate was selected as a parameter. The tendencies shown above are clearly recognized in these plots. Out of these results one can design the capacities of experimental facility; however a hydrodynamic model is also required in order to be sure that heat transfer rate will enable the conduction of the process at reasonable channel dimensions. This is shown in the next chapter.

Table 3.3.1

Process conditions for MSC with mercury helium mixture

Gas	helium-He
$\dot{m}_G$	0.017 kg/s
inlet gas temperature range	220-370°C
subcooled liquid and vapor substance	mercury-Hg
inlet vapor to gas mass flowrate ratio range:	0.01 - 4
inlet subcooled liquid temperature:	20°C
extraction factor angle:	0.05 - 0.4
total inlet pressure	0.111 MPa

Table 3.3.2

Process conditions for MSC with mercury nitrogen mixture

Gas	nitrogen - N ₂
$\dot{m}_G$	0.084 kg/s
inlet gas temperature range	300°C
subcooled liquid and vapor substance	mercury-Hg
inlet vapor to gas mass flowrate ratio range:	0.01 - 0.3
inlet subcooled liquid temperature:	30°C
extraction factor angle:	0.05 - 0.4
total inlet pressure	0.111 MPa

Figure 3.3.1
 Subcooled liquid flow rate vs. extraction factor for Hg-He MSC.
 Inlet vapor concentration is the parameter.
 Inlet gas temperature $T_{GI} = 220^{\circ}\text{C}$.

Figure 3.3.2
 Subcooled liquid flow rate vs. extraction factor for Hg-He MSC.
 Inlet vapor concentration is the parameter.
 Inlet gas temperature $T_{GI} = 270^{\circ}\text{C}$.

Figure 3.3.3
Subcooled liquid flow rate vs. extraction factor for Hg-He MSC.
Inlet vapor concentration is the parameter.
Inlet gas temperature $T_{GI} = 300^{\circ}\text{C}$.

Figure 3.3.4
Subcooled liquid flow rate vs. extraction factor for Hg-He MSC.
Inlet vapor concentration is the parameter.
Inlet gas temperature $T_{GI} = 320^{\circ}\text{C}$.

Figure 3.3.5

Subcooled liquid flow rate vs. extraction factor for Hg-He MSC.

Inlet vapor concentration is the parameter.

Inlet gas temperature $T_{GI} = 340^{\circ}\text{C}$.

Figure 3.3.6

Subcooled liquid flow rate vs. extraction factor for Hg-He MSC.

Inlet vapor concentration is the parameter.

Inlet gas temperature $T_{GI} = 370^{\circ}\text{C}$.

Figure 3.3.7

Subcooled liquid flow rate vs. extraction factor for Hg-N₂ MSC.
Inlet vapor concentration is the parameter.
Inlet gas temperature $T_{GI} = 300^{\circ}\text{C}$.

4. Parametric Study of the Mixing Stream Condenser

4.1 Introduction

In this chapter the parametric study of MSC is presented. It was conducted in the following stages:

- Modifying a single droplet direct-contact condensation model.
- Extending the single droplet model to a multi-droplet model for Homogeneous
- Mist Flow Condensation (HMFC).
- Conducting a parametric study by using the above models.

The single droplet model, which was previously developed for the analysis of the evaporation of a single fuel droplet in an air stream, is employed for our study. Such models are widely used for spray-combustion simulation. They are based on the assumption that heat and mass transfer occur through uniform thickness transport layers. With this assumption, the analysis is one-dimensional, which is much simpler than a two-dimensional analysis. However, the assumption of the uniform transport layers for condensation processes should be proved experimentally.

The Homogeneous Mist Flow Condensation (HMFC) model, is an extension of the single droplet model to a multi-droplet homogeneous stream ($U_G = U_{\text{droplet}}$), which is based on the assumption that the distance between the droplets is large enough so that no interactions occur between them. This model evaluates the upper limit for the condensation length, as will be discussed later in more detail.

From the runs of the single droplet condensation model discussed in the following section, and of the HMFC model it appears that the mist flow over at least half the length of the MSC is far from being homogeneous. In other words, at least half of the MSC length is an acceleration zone, for which the homogeneous assumption is conservative in the sense of heat and mass transfer. This subject is discussed later on. As a result of this conclusion, a further modification of the HMFC model is considered.

The following sections, 4.2 and 4.3, present the single droplet model and the HMFC model. Section 4.4 presents and discusses the parametric study results.

4.2 Single-Droplet Condensation Model

The single-droplet condensation model presented here evaluates the dynamics as well as the heat and mass transfer of a single droplet moving in a gas stream. This model was originally developed for a fuel spray vaporization analysis (Abramzon and Sirignano 1987,1989). From the physical point of view, the vaporization and condensation processes on a single moving droplet in the presence of noncondensing gases are similar. The only difference is the direction of the diffusional flow of vapor, and can be taken into account by appropriate selection of the sign in the expression for the diffusion flux. The droplet dynamics, heat-up and radius variation are governed by the following equations and initial conditions.

$$\frac{dZ_L}{dt} = U_L \quad 4.2.1$$

$$\frac{dU_L}{dt} = \frac{3C_D}{2r_s} \left(\frac{\rho_G^\infty}{\rho_L} \right) |U_G - U_L| (U_G - U_L) \quad 4.2.2$$

$$\frac{dr}{dt} = - \frac{\dot{m}}{4 \pi r_s^3 \rho_L r_s^2} \quad 4.2.3$$

$$\frac{dT_L}{dt} = \frac{3Q_L}{4 \pi r_s^3 \rho_L C_{pL}} \quad 4.2.4$$

Initial conditions:

at $t = 0$

$$Z_L = 0 ; U_L = U_L^i ; T_L = T_L^i ; \quad 4.2.5$$

$$r = r_i$$

The drag coefficient is usually expressed as a function of the Reynolds number, $C_D = C_D(Re)$, where $Re = 2 \rho_G^\infty |U_G - U_L| r_s / \bar{\mu}_G$, and $\bar{\mu}_G$ is the gas viscosity evaluated at "reference" temperature and concentration:

$$\bar{T} = T_s + (T_G^\infty - T_s)/3 ; \quad \bar{Y}_v = Y_{v,s} + (Y_v^\infty - Y_{v,s})/3 \quad 4.2.6$$

For dense spray region, where the distance between the droplets compared with the droplets diameter is relatively small, the drag coefficient should also depend on a local droplet concentration in the gas flow. Some studies predict that the drag coefficient of the spherical particle, surrounded by other particles, may be considerably lower than that for a

single sphere in the unbounded flow (Yuen and Chen, 1976). However, no practical correlations including this effect have been suggested in the literature. In the present study, we used the following correlation for the drag coefficient (Faeth, 1977):

$$C_D = \frac{24}{Re} \left[1 + \frac{Re^{2/3}}{6} \right] \quad 4.2.7$$

which approximates the "standard drag curve" of a sphere in an unbounded flow.

To calculate the rate of condensation on a moving droplet, we employ the "extended film model", which includes the effects of non-unitary Lewis number and variable physical properties in the gas film near the droplet surface, and the effect of the Stefan flow on heat and mass transfer between the droplet and gas. The extended film model yields the following expression for the instantaneous condensation rate on the droplet:

$$\dot{m} = 2\pi \bar{\rho}_G \bar{D}_G r_s Sh^* \ln(1+B_M) \quad 4.2.8$$

and

$$\dot{m} = 2\pi (\bar{K}_G / \bar{C}_{p_v}) r_s Nu^* \ln(1+B_T) \quad 4.2.9$$

where $\bar{\rho}_G$, $\bar{D}_G$ and $\bar{K}_G$ are the average density, binary diffusion coefficient and thermal conductivity of the gas mixture in the film, respectively; $\bar{C}_{p_v}$ is the average vapor specific heat in the film; and Sh^* and Nu^* are the nondimensional parameters, which can be termed the "modified" Sherwood and Nusselt numbers, and are expressed as:

$$Sh^* = 2 + (Sh_o - 2) / F_M \quad 4.2.10$$

$$Nu^* = 2 + (Nu_o - 2) / F_T \quad 4.2.11$$

Here, Sh_o and Nu_o are the Sherwood and Nusselt number for the spherical particle in the case of very low rates of condensation or vaporization on the surface when the effect of the Stefan flow is negligibly small. F_M and F_T are the correction factors, which account for the effect of the Stefan flow on the thickness of the diffusional and temperature film, respectively.

The values B_M and B_T in Eqs. 4.2.8 and 4.2.9 are referred to as the Spalding mass and heat transfer numbers, respectively, and defined as:

$$B_M = \frac{Y_{v_s} - Y_{v_\infty}}{1 - Y_{v_s}} \quad 4.2.12$$

$$B_T = \frac{\bar{C}_{p_v} (T_\infty - T_s)}{h_{fg}(T_s) + Q_L/\dot{m}} \quad 4.2.13$$

Here Y_{v_s} , Y_{v_∞} are the mass fraction of vapor near the droplet surface and in the bulk flow, respectively. T_s and T_∞ are the temperatures of the gas mixture at the droplet surface and far from the droplet (bulk flow), respectively. $h_{fg}(T_s)$ is the latent heat of vaporization/ condensation at the temperature T_s .

It was found that the correction factors F_M and F_T can be approximated as

$$F_M = F(B_M), \quad F_T = F(B_T) \quad 4.2.14$$

where $F(B)$ is the universal function:

$$F(B) = (1+B)^{0.7} \frac{\ln(1+B)}{B} \quad 4.2.15$$

Note that for the condensation process, $\dot{m} < 0$, $-1 < B_M \leq 0$, and $F(B_M) \leq 1$. The Stefan flow direction is toward the surface of the droplet. In this situation, the Stefan flow acts similar to the "suction" effect, which decreases the thicknesses of the hydrodynamic, thermal and diffusional boundary layers, and, thus, causes the intensification of the heat and mass transfer, and to a certain extent, the growth of the drag experienced by the particle moving in the gas environment.

For the non-dense sprays, where the distance between droplets $l > 10d$ (or gas void fraction less than 10^{-3}), the Nusselt and Sherwood numbers can be calculated using the correlations for a single droplet in an unbounded flow (Clief et al., 1978):

$$Nu_0 = 1 + (1 + Re \cdot Pr)^{1/3} \cdot f(Re) \quad 4.2.16$$

$$Sh_0 = 1 + (1 + Re \cdot Sc)^{1/3} \cdot f(Re) \quad 4.2.17$$

where $f(Re) = 1$ at $Re \leq 1$ and $f(Re) = Re^{0.077}$ at $Re \leq 400$.

In dense sprays, there are several additional factors affecting the heat and mass transfer between droplets and the gas flow. The first factor is the thermal and diffusional interactions between the neighboring droplets of the spray, which, generally, tend to reduce the Nusselt and Sherwood number of an individual droplet. This effect manifests itself at

low Reynolds numbers, where the thickness of the thermal or diffusional boundary layer is comparable with the distance between the droplets. For example, a droplet which is located in the wake behind another droplet "feels" the reduced driving force for the heat or mass transfer more than an isolated droplet in an infinite flow. The second factor is the flow disturbances produced by neighboring particles. These disturbances result in a better mixing of the flow, a larger level of turbulence, and, finally, a tendency to increase the heat or mass transfer rate.

Unfortunately, there are no reliable quantitative data on heat and mass transfer in the multi-droplet system. Therefore, Eqs. 4.2.16 and 4.2.17 are used in the present study as a first approximation.

Equating of the RHS of Eqs. (4.2.8), (4.2.9) yields:

$$B_T = (1 + B_M)^\varphi - 1, \quad 4.2.18$$

where

$$\varphi = (\bar{C}_{p_v} / \bar{C}_{p_g}) \cdot (Sh^* / Nu^*) \cdot (1 / \bar{Le}), \quad 4.2.19$$

where $\bar{C}_{p_g}$ is the average specific heat of the gas mixture in the film and $\bar{Le} = \bar{K}_G / (\rho_G D C_{p_g})$ is the average Lewis number. Note that the classical theory of the droplet vaporization/condensation assumes that $Le = 1$ and $\varphi = 1$, which in turn yields $B_M = B_T$. In our case, however, vapor molecules (cesium) are much heavier than the inert gas molecules (H_2 , He), $C_{p_v} \ll C_{p_g}$ and the Lewis number appears to be of the order of 5. Therefore, the classical model, which neglects the differences between the physical properties of the components of the mixture, is not applicable in the present situation.

The heat penetrating into the droplet interior is calculated from Eq. 4.2.13 as

$$Q_L = \dot{m} \left[\frac{\bar{C}_{p_v} (T_\infty - T_s)}{B_T} - h_{fg}(T_s) \right] \quad (4.2.20)$$

This value is then used in Eq. 4.2.4 to find the variation of the droplet average temperature, T_L . As an additional assumption, the present model postulates that due to the high thermal conductivity of the liquid metal droplet, the temperature within the droplet is spatially uniform, i.e. $T_L = T_s$.

The system of ordinary differential equations 4.2.1 to 4.2.4 along with the nonlinear algebraic equations 4.2.6 to 4.2.20, is solved by numerical methods. A detailed description of the numerical algorithm is presented in the above mentioned paper by Abramzon and Sirignano (1987, 1989).

4.3. Homogeneous Mist Flow Condensation (HMFC) Model

The HMFC model is an extension of the single droplet model for the case of homogeneous flow ($U_G = U_{\text{droplet}}$). It is based on the assumption of no interactions between the droplets. This assumption implies that the condensation rate on each droplet ($\dot{m}$) can be calculated by using the single droplet model.

As a result of the above, the gas-temperature along the MSC is reduced according to the equation:

$$\dot{m}_G C_{pG} \frac{dT_G}{dt} = -\dot{N} Q_L \quad 4.3.1$$

$$\dot{m}_G \frac{dY_v}{dt} = \dot{m} \dot{N} \quad 4.3.2$$

where $\dot{N}$, the number of droplets flow rate, is:

$$\dot{N} = \dot{m}_L / m_o \quad [s]^{-1} \quad 4.3.3$$

Substituting it into Eqs. 4.3.1 and 4.3.2 yields:

$$\frac{dT_G}{dt} = - \frac{\dot{m}_L}{\dot{m}_G} \frac{Q_L}{C_{pG} m_o} \quad 4.3.4$$

$$\frac{dY_v}{dt} = \frac{\dot{m}_L}{\dot{m}_G} \frac{\dot{m}}{m_o} \quad 4.3.5$$

These equations (4.3.4 and 4.3.5) are integrated in the single droplet model and are solved with the initial conditions:

$$\text{at } Z = 0 \quad T_G = T_G^i ; \quad Y_v = Y_v^i \quad 4.3.6$$

This integrated model is designated as the HMFC model.

Note that the present simplified homogeneous flow model is believed to be conservative and overestimates the required condenser length. Due to the assumption of no slip between the droplet and gas flow, the Nusselt and Sherwood numbers for heat and mass transfer to a single droplet appear to be close to their minimum values $Nu = Sh = 2$. Thus, the calculated heat and mass transfer rate is significantly underestimated. To correct this model, we assumed that for the interphase heat and mass transfer calculations, the relative velocity between the droplets and the gas is approximately equal to the turbulent fluctuation velocity, which, in turn, is estimated as $U' = 0.07 U_G$.

Furthermore, the actual MSC has an acceleration zone that might occupy a considerable channel portion, where the droplet "residence time" is much longer than the one predicted using an assumption of homogeneous flow for the same length. Thus, the actual duration of direct contact between the droplets and the gas is longer than that predicted by a homogeneous model. Combining this fact with the minimum predicted heat and mass transfer coefficients yields that the HMFC model predicts an upper limit of the MSC length.

The above discussion shows that the final validation of the model should be verified and/or corrected by direct comparison with experimental results.

4.4. Parametric Study

The above models were used for conducting the parametric study presented and discussed below.

The single-droplet model was used to evaluate the longitudinal droplet velocity distribution. This distribution is supposed to be approximately the same as in the case of a multi-droplet stream with negligible interactions between the droplets. At no interactions, which is the case of relatively large distance between the droplets, the acceleration of the single droplet is equal to that in the multi-droplet system.

Calculations for a single cesium droplet moving in a hydrogen stream ($H_2 - Cs$) and in a helium stream ($He - Cs$) were carried out. Initial conditions for the runs with hydrogen-cesium mixture are listed in Table 4.4.1., and the respective values for helium-cesium in Table 4.4.2.

Table 4.4.1

**Initial Conditions for Calculations of Velocity of Cesium
Droplet in Hydrogen-Cesium Mixture**

Gas velocity	160 m/s
Gas inlet temperature	400°K
Inlet subcooled cesium droplet temperature	300°K
Cesium droplet diameters	60, 100 and 160 microns
Process pressure	0.2 bar
Vapor initial concentration	0.01 (as defined in the SOW of this project)

Table 4.4.2

**Initial Conditions for Calculations of Velocity of Cesium
Droplet in Helium-Cesium Mixture**

Gas velocity	160 m/s
Gas inlet temperature	400°K
Inlet subcooled cesium droplet temperature	300°K
Cesium droplet diameters	60, 100 and 160 microns
Process pressure	1.0 bar
Vapor initial concentration	0.10

The low temperature difference between the gas and the droplet was chosen to avoid evaporation that causes a reduction in the droplet diameter. In the MSC in question, the droplet diameter growth is not supposed to be significant. Additional calculations for H₂-Cs were made at higher gas temperature and higher vapor concentration, in order to estimate the influence of the properties change. This influence was found to be negligible.

The calculations with the HMFC model were carried out for droplet diameters: 40, 80, 120 and 160 microns, flowing along the MSC. For hydrogen-cesium (H₂-Cs) the run conditions are listed in Table 4.4.3, and the respective conditions for helium-cesium in Table 4.4.4.

Table 4.4.3

**Boundary Condition for Calculation of MSC Performance with
Hydrogen-Cesium Mixture**

Inlet gas temperature	1000°K
Inlet vapor concentration	0.0100
Exit vapor concentration	0.0001
Process pressure	0.2 bar
Inlet subcooled cesium droplet temperature	300°K
Inlet gas velocity	160 m/s

Table 4.4.4

**Boundary Condition for Calculation of MSC Performance with
Helium-Cesium Mixture**

Inlet gas temperature	750°K
Inlet vapor concentration	0.10
Exit vapor concentration	0.01
Process pressure	1.0 bar
Inlet subcooled cesium droplet temperature	300°K
Inlet gas velocity	160 m/s

The results of the numerical simulations are given in Figs. 4.4.1 through 4.4.5. In Figs. 4.4.1 a, b, and c, the acceleration process of a single droplet in a carrier gas flow is illustrated. We can approximately define the acceleration distance as the distance required for a droplet to reach 80% of the gas flow velocity. The acceleration distance depends substantially on the droplet diameter. As the droplet diameter changes from 60 to 100 μm , the acceleration distance increases from ≈ 1 m to 3 m in the case of the helium flow, and much larger in the case of hydrogen flow.

Figure 4.4.1 Single cesium droplet velocity vs. longitudinal location in hydrogen and helium streams.

Figures 4.4.2 and 4.4.4 show the volume condensation rate J [kg/(s·m³)] as a function of the longitudinal coordinate Z . The volume condensation rate represents the vapor mass condensing per unit time per unit volume of the condenser:

$$J = n \dot{m} \quad 4.4.1$$

where n is the number of droplets per unit volume at a given location, [1/m³], and $\dot{m}$ is the average condensation mass rate on a single droplet [kg/s]. J also represents the total condensate flow rate, per unit channel volume.

As follows from these figures, the volume condensation rate, $J = J(Z)$, increases at small distances near the atomizer, passes through some maximum value and then decreases as a function of Z . An initial growth of $J(Z)$ is associated with both the droplet heating and the corresponding increase of the binary diffusion coefficient in the boundary layer near the droplet surface (see Eq. 4.2.8). However, at higher distances, the driving force for mass transfer (namely, the mass transfer number, B_M) decreases rapidly due to both the reduction in the vapor concentration in the bulk flow and the growth of the vapor pressure near the droplet surface. As a result, the volume condensation rate decreases with Z . Figures 4.4.3 and 4.4.5 show the vapor mass fraction Y_v in the bulk flow as a function of the longitudinal coordinate Z . The condensation length can be defined from these data as the length of the condenser required to reduce the vapor fraction in the bulk flow below the prescribed level (say, 1-3% of the inlet vapor fraction).

As the average droplet diameter increases, the condensation length grows approximately as d^2 . Note that the condensation length for both helium-cesium and hydrogen-cesium flows is considerably shorter than the acceleration distance for the corresponding conditions. This result shows that the homogeneous mist flow model does not provide a good representation of the spray condenser simulation for the parameters defined for this project. However, this model is useful in the sense that it provides conservative estimates for the required condenser length.

Figure 4.4.2 Condensation rate vs. longitudinal location for hydrogen-cesium homogeneous mist flow condensation process.

Figure 4.4.3 Vapor mass fraction in the gas bulk stream vs. longitudinal location for hydrogen-cesium homogeneous mist flow condensation process.

Figure 4.4.4 Condensation rate vs. longitudinal location for helium-cesium homogeneous mist flow condensation process.

Figure 4.4.5 Vapor mass fraction in the gas bulk stream vs. longitudinal location for helium-cesium homogeneous mist flow condensation process.

In Figs. 4.4.6 and 4.4.7 the results of the parametric study for mercury representing the subcooled liquid and different gases is shown. The selected parameters are listed in Table 4.4.5.

Table 4.4.5

Initial Conditions for Calculations of MSC Performance with Mercury Vapor Mixed With Different Gases

Gases substance	H ₂ , He, N ₂
Gas inlet velocity	70 m/s
Gas inlet temperature	600°K
Inlet subcooled liquid droplet temperature	300°K
Droplet diameter	100 μm
Process pressure	1.2 bar
Vapor inlet concentration	0.15

The longitudinal coordinate of the MSC is chosen for the independent variable. In Fig. 4.4.6 the condensation rate is shown, while in Fig. 4.4.7 the bulk flow vapor concentration is shown. For all gases it is seen that both the concentration and condensation rate decrease with the length coordinate. When the selected gas is H₂, the condensation process stops after a relatively short distance, and at a relatively high concentration. The model predicts a zero condensation rate at about 0.23 m, then evaporation starts and the concentration increases again. This is a result of the high heat capacity of H₂ with respect to the other two gases under study. For He and N₂ the process ends as soon as the concentration difference between the droplet surface (held at saturation) and the gas bulk flow is diminished. As already discussed, the differences in properties of the selected gases influence the performance of the MSC.

Figure 4.4.6 Condensation rate vs. longitudinal location for mercury and different inert gases, in a homogeneous mist flow condensation process

Figure 4.4.7 Vapor mass fraction in the gas bulk stream vs. longitudinal location for mercury and different inert gases, in a homogeneous mist flow condensation process.

5. EXPERIMENTAL PHASE

5.1 Introduction

As already mentioned, the model used in our study was originally developed for vaporization processes, and therefore it was found necessary to validate its applicability for condensation processes. Although the model is based on well known physical considerations, an experimental proof is necessary.

In order to construct the experimental simulation of the HMFC the working fluids had to be selected. From a pure scientific point of view each pair of subcooled liquid (and its vapor) and a noncondensable gas is a potential candidate. On the other hand, from an engineering point of view the alternatives are limited.

Thus Cs-H₂ MSC performance could be simulated either by the pair Hg + gas or by the pair H₂O + gas, providing that the process is feasible from a thermodynamic point of view. Taking into account the parameters governing the process it appears that the gas to liquid heat capacity ratio of Cs-H₂ mixture is in the same order of magnitude as the corresponding value of Hg-H₂ and H₂O-N₂. The binary diffusivity of the 3 above mentioned pairs are also in the same order of magnitude, as well as other non-dimensional parameters.

Based on the above, it was decided to include H₂O-N₂ mixture in the first stage of the experimental phase of the project, in order to validate the HMFC model. Since the properties of air are very close to that of N₂, it is assumed that the results of the model for both gases will not deviate significantly from each other.

Nevertheless, considering the importance of gathering experience in running of test loops including liquid metals, the work with mercury, representing cesium, was found necessary.

Thus, in the second stage of the experimental phase, atomization and spraying of mercury in air was tested, as described in Section 5.5 and 5.6.

5.2 MSC Experimental Simulation Model

The experimental rig

The experimental air-steam water facility is constructed of a horizontal test section 4m length and a 2" diameter galvanized steel pipe. The schematic of the facility is shown in Fig. 5.2.1 and its details in Figs. 5.2.2 - 5.2.6. At the exit from the test section a cylindrical separator (8" diameter) is installed. It is shown in Fig. 5.2.2, prior to its isolation. Referring to the figure, two-phase entrance is located at the separator left hand side, through a volute type spiral housing designed in order to maintain a constant radial velocity of the mixture. The liquid water, after separation, flows to the downflow pipe through the volute type spiral housing exit located on the lower right hand side, and the liquid level control valve. Another valve is installed at the base of the separator for drainage. Exhaust gases are ventilated at the opening of the separator top.

The selected atomizers for this facility are of type H530x93 manufactured by Monarch, USA. Its characteristics are: spray angle: 80°, droplet mean diameter: 250-500µm, and a solid cone spray geometry. This geometry is reached due to 2 inlet ports — one tangential to the swirl chamber and the other one directed toward the chamber center. This configuration was already addressed in Section 2.3. This type of atomizers was selected from a variety of alternatives which were checked in the facility in order to fit the desired flow rates.

Two water atomizers are installed across the other end of the test section, with sprays directed downflow, (see Fig. 5.2.3). Subcooled water are fed to the atomizer high pressure, flexible inlet tubes (see Fig. 5.2.4) through a manifold and a 10 bar centrifugal pump, shown in Fig. 5.2.5. The T-connection at the pump discharge port is the pump bypass loop entrance. At the pipe left side inlet, a filter is mounted to prevent atomizers plugging.

Upstream the spraying zone air and steam are mixed together in a T-connection mixer, shown in Fig. 5.2.6. Air from the compressor flows up in the right side pipe, turns and enters the mixer from above. Steam inlet is at the bottom of the mixer.

Figure 5.2.1

Schematic of MSC experimental simulation

Figure 5.2.2

View of exit zone and separator of the experimental facility

Figure 5.2.3

View of Inlet zone of the test section

Figure 5.2.4

View of Inlet subcooled liquid manifold

Figure 5.2.5

View of high pressure centrifugal pump

Figure 5.2.6

View of facility Inlet mixer and flow rate regulators

Accessories

Aside from the facility previously described, the following components are used.

a) **Steam boiler**

The Etgar-3 plant steam boiler is used, manufactured by H.V.E. Thermoduct, Ltd, UK. with a capacity of 15 kg dry steam at 9.5 bar, ($T \cong 185^{\circ}\text{C}$).

b) **Air compressor**

The screw air compressor of the Matar facility, manufactured by Oholiav, Israel, is used. It has a capacity of 4200 liter per minute free air, at about 7 bar, and is equipped with a 40 HP electrical motor.

In series to the compressor a 470 liter pressure vessel is connected, which compensates pressure fluctuations, and oil and particle filters.

c) **Heat exchanger and cooling system**

In order to feed the sprayers with low temperature subcooled water, a tube and shell heat exchanger was installed. At the shell envelope, chilled water from the university net is fed. A centrifugal pump circulates the water from a 1000 liter vessel to the heat exchanger tubes, and from there to the high pressure centrifugal pump inlet. The water, after leaving the MSC, is fed back through the downflow pipe to the cooled water vessel.

Instrumentation

The facility is equipped with the following instruments.

A. **Temperature measurement**

3 thermo-resistors, type PT100 RTD, with an accuracy of 0.5% manufactured by "Madei Taas," Israel, are installed in the following locations: one at the cold water manifold, downstream the high pressure centrifugal pump (see Fig. 5.2.4). This one measures the atomizers inlet temperature. The length of the tube from the measuring point up till the spraying nozzles is about 1.5m, and the hosts are made

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of an insulated material. It was found that, the accuracy of the measurement is high enough for our purposes. The other 2 are located upstream the airflow meter and the steam flowmeter. Measured values of the latter two are necessary for mass flow rates calculations.

A semiconductor element is installed in the humidity meter probe tube, (see Fig. 5.2.3, with a measuring range of 0~100°C and accuracy of 0.1°C, manufactured by Steinecker Elektronik GMBH, Germany. It is installed downstream the air steam mixer and measures the MSC inlet gases temperature.

The wet bulb and dry bulb exhaust air from the test section, as well as the exit water temperatures are measured by mercury in glass tube thermometers with a least count of 0.2°C and accuracy of 0.1°C. The ranges of the thermometers were selected in accordance with the experimental temperature range. In addition, the inlet and outlet temperatures of the heat exchanger were measured by industrial bi-metal dial thermometers.

B. Pressure measurements

Bourdon tube type pressure gauges with an accuracy of 1%, manufactured by MegoAfek, Israel, were used for pressure measurement of atomizer water inlet and the condenser air-steam inlet (see Figs. 5.2.4 and 5.2.3).

Druck U.K. pressure transducers were used to measure the flow meter inlet pressure. The one that measures the air flow pressure is seen at the middle of the right hand side pipe shown in Fig. 5.2.6.

C. Humidity measurements

A combined relative-humidity and temperature instrument, manufactured by Steinecker, Electronic GMBH, Germany was used (see Fig. 5.2.3). Its humidity sensor is PMO1, and its range is 0-100% relative humidity. Accuracy is less than 1% up to 95% relative humidity. The sensor is wind protected by a sintered stainless steel cup, (which lengthens its response time. Special care has to be taken to prevent its sensor from getting wet.

D. Flow rates

The volumetric flow rate is measured upstream the air steam mixer by a 1 1/4" turbine flow meter, manufactured by Hoffer, Flow Controls, Inc., USA, which has a range of 50 cfm and accuracy of 1% of reading. It is shown upstream the pressure transducer in Fig. 5.2.6.

Subcooled liquid flow rate is calculated either from the pressure vs. flowrate calibration chart, using the measured pressure, or at high flow rate by using the Foxboro, USA, magnetic flow meter, both seen in Fig. 5.2.4. The latter has a 40 mm flow tube size, and its maximal upper flowrate range is 680 L/M.

Steam flow rate is measured by a pitot tube flow meter, installed upstream the air steam mixer.

Control and Data Acquisition

The process is manually controlled. A pressure regulator controls the inlet air flow rate, and a suitable valve controls the steam inlet. Both are shown on the right-hand side of Fig. 5.2.6. The steam valve is in the lower front of the figure, while the air regulator is located behind it. The atomizer subcooled water flowrate is controlled by the valves shown in Fig. 5.2.4. The atomizers are controlled independently from each other. The high-pressure centrifugal pump connected to the atomizer feed lines in parallel to a bypass loop, its lower part is shown in Fig. 5.2.5. The bypass valve is used for the control of the atomizer manifold inlet flowrates. The temperature of the inlet gases stream is controlled not only by the relative air and steam flow rates, but also by the steam temperature, and the latter is controlled by the boiler automatic pressure regulator setup, which can be varied at the range of 2-9.5 bar.

The separator downflow pipe is equipped with a valve for the control of separator liquid level. This level is significant for high-quality separation.

All electronic output instruments are equipped with a 2 wire 4-20 mA transmitter. The output signals are fed into the BMC computer for the on-line calculation of mass flow rate, with the option for automatic computerized data analysis. Glass thermometers and dial pressure gauge readouts are manually recorded.

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Test Plan Implementation

After the boiler and compressed air capacitor have reached stable conditions, the air regulator and steam valve were operated to give required flow rates. After heat exchanger circulation pump was operated, the high pressure cold water pump was turned on and by setting the atomizers and by pass valves the desired subcooled liquid flow rate was reached.

After stabilization of the read out of all instruments was attained, the data was recorded several times in order to have sufficient information for averaging. The stabilization time was never less than 15 min, in order to bring the test section to a stable temperature. After one set of data was recorded, the dry air mass flow rate was increased while the steam and subcooled liquid flow rates were held constant. This procedure was continued until maximum air flow rate was reached. Thereafter the procedure was repeated at another value of the subcooled liquid flow rate.

5.3 Data Reduction

The mathematical model presented in Chapter 4, is reduced to a nonlinear system of five time dependent ordinary differential equations for the following dependent variables:

- U_k - droplet velocity
- r_k - droplet radius
- T_k - droplet temperature
- T_{GF1} - stream temperature
- Y_{VF1} - stream vapor concentration

The numerical scheme to work out the solution of the system was chosen to be a Runge-Kutta of the fourth order.

In the final run the value of the dimensionless time step, dt , was set to 0.05, which corresponds to $5 \cdot 10^{-5}$ sec under the current input set of parameters. A careful examination of the output result file which was created with $dt = 0.1$ reveals a maximum relative

difference of the order of $5 \cdot 10^{-4}$ between the two cases. This maximum error is detected in the first step result when calculating VCR, the rate of vapor condensation.

During the computer run the following values are computed:

- Q_{LIQ} - the amount of heat involves convection heat transfer modes only.
 Q_{LIQ} is due to the temperature difference between the stream and droplet.
- Q_{PHASE} - the added or subtracted heat to the droplet due to condensation or the evaporation process, respectively.
- Q_{HV} - heat that is required to change the condensate temperature from that of the stream to that of the droplet, keeping in mind that Q_{PHASE} occurs during a process that involves no temperature change, and takes place on the droplet surface.

Based on the first law of thermodynamics and the other assumptions that were made in the above mathematical model, the following assumptions (regarding the above three heat flux quantities) are applied to the general thermal balance:

Process	Heat added to:	$T_k \geq T_{stream}$	$T_k < T_{stream}$
Evaporation	Droplet	$-Q_{PHASE} - Q_{LIQ}$	$-Q_{PHASE} - Q_{LIQ}$
	Stream	$+Q_{VH} + Q_{LIQ}$	$-Q_{VH} - Q_{LIQ}$
Condensation	Droplet	physically impossible	$Q_{PHASE} + Q_{VH} + Q_{LIQ}$
	Stream	physically impossible	$-Q_{LIQ}$

The initial conditions are calculated from the results of the different measurements that were performed on the experimental system. The termination condition for each computer run occurs when

$$|Y_{VF} - Y_{VS}| \leq 4 \cdot 10^{-5}$$

Careful examination of the computation results reveals that only the condensation case occurs under the current experimental conditions.

Dry air mass flow rate was calculated from the volumetric measured values using the equations of state, assuming idle gas. Its vapor content was measured prior to its mixing with the steam, and that vapor mass flow rate was added to the inlet steam flow rate. The steam volumetric flow rate measurements were found not stable, thus its temperature and pressure measurement served for the calculation of steam flow rate. The mixer control volume enthalpy balance was carried out for this calculation.

In some runs condensation occurs at the mixer. In those cases the mixer relative humidity was set as 100% and the resulting steam flow rate was calculated. The condensate flow rate was added to the atomizers flow rate. The humidity at the gas mixture was calculated by means of a computerized steam chart.

5.4 MSC Simulation Results

All the individual runs of the experimental facility were gathered into 12 average data points. The results of the MSC simulation are presented in the Appendix and are shown in Figs. 5.4.1 to 5.4.3. In Fig. 5.4.1 the condensate flow rate is plotted vs. the inlet vapor to liquid mass fraction. As expected, an increase of inlet vapor flow rate is associated with an increase of the condensate flow rate. Note that the experimental data is divided into two ranges: one is the lower vapor concentration, and in the figure, the respective points are overlapping each other at the lower left point. The other group of points spread along the figure relates to higher steam flow rate values.

In Fig. 5.4.2 the MSC effectiveness is plotted vs. the inlet vapor mass fraction. The MSC effectiveness is defined by:

$$\text{Effectiveness} = (\dot{m}_{V_i} - \dot{m}_{V_o}) / \dot{m}_{V_i} \quad 5.4.1$$

From the figure it is shown that the condenser is more effective at higher values of inlet vapor mass fraction.

In Fig. 5.4.3 the measured extraction factor is plotted vs. the inlet vapor mass fraction. The lower the inlet vapor concentration the higher the extraction factor.

The HMFC model was used in order to calculate the predicted MSC length required for the completion of the process. In Figs. 5.4.4 to 5.4.7 the results of the model are

Figure 5.4.1

Measured condensate flow rate vs. inlet vapor to gas mass fraction

Figure 5.4.2

Measured MSC effectiveness vs. inlet vapor to gas mass fraction

Figure 5.4.3

Measured extraction factor vs. inlet vapor to gas mass fraction

shown. In each one of the figures the right side frames show the bulk to droplet surface vapor mass fraction as a function of MSC longitudinal coordinate.

Two curves are shown in each one of the frames. The right one relates to 500 μm droplets diameter, while the curve to the left represents the model results when droplet mean diameter is assumed to be 250 μm . These two values were selected, in accordance with the manufacturer data. In each one of the plots the difference between MSC lengths for the different diameter is clearly seen. Cases 22 to 25 were run without steam injection. Hence, the condensation rate is significantly smaller compared to the other cases. However, for a simple comparison also cases Nos. 22 to 25 are shown at the same scale as the rest of the data.

The left side frames of each one of the Figs. 5.4.4 to 5.4.7 represents the temperature longitudinal distribution for both: subcooled liquid and vapor seeded gas for the two droplet diameters. The upper 2 curves show the decrease of gas temperature, while the lower 2 curves represent liquid warming up during the process. Note that as soon as the concentration curve, at a given droplet diameter, reaches a constant value, the temperature curves of the liquid and gas coincide and have a constant value. From the figures one can conclude that for both diameters, the condensation process, according to the model, is ended at a length of not more than half the length of the test section.

While comparing the measured exit temperature of the gases (dry bulb temperature of the data Table of the Appendix) to the values predicted by the model, a difference is noted. In the experiments, during the flow along the test section, the droplets were gathered at the lower part of the test section and from there were flowing as a stratified two-phase flow pattern. Thus on the one hand the gravity effects together with droplet impingement have stopped the dropwise condensation, and on the other hand condensation continues, employing another mode of condensation which is not the case of the model. As an outcome, the experimental results show that the condensate flow rate was higher than the predicted values, and this is shown in Fig. 5.4.8. Note that in cases where condensation occurs in the mixer, stratification starts prior to the atomizer cross-section.

The comparison between the experimental data and calculated values is also shown in Fig. 5.4.9. Here, the effectiveness of the MSC simulation are compared to those predicted by the model. The deviation is larger at lower vapor flow rates. The comparison of measured and calculated extraction factor values is shown in Fig. 5.4.10 and one can see

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Figure 5.4.4 Longitudinal distribution of bulk stream to droplet surface concentration ratio (right side) and gas and liquid temperature (left side) for 250 and 500 μm droplet diameter - Case 14, 15, 16

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Figure 5.4.5 Longitudinal distribution of bulk stream to droplet surface concentration ratio (right side) and gas and liquid temperature (left side) for 250 and 500 μm droplet diameter - Case 17, 18, 19

Figure 5.4.6 Longitudinal distribution of bulk stream to droplet surface concentration ratio (right side) and gas and liquid temperature (left side) for 250 and 500 μm droplet diameter - Case 20, 21, 22

Figure 5.4.7 Longitudinal distribution of bulk stream to droplet surface concentration ratio (right side) and gas and liquid temperature (left side) for 250 and 500 μm droplet diameter - Case 23, 24, 25

Figure 5.4.8

Measured condensate flow rate vs. calculated values by the model

Figure 3.4.9

Measured MSC effectiveness vs. calculated values by the model

Figure 5.4.10

Experimental extraction factor vs. calculated values by the model

that the predictions are closer to the measured values at higher extraction factor values. The general conclusion is that the models prediction is experimentally validated, and additional effects, besides the spray type condensation mode are influencing the performance of the MSC.

5.5 Experiments with Liquid Metal

As shown above, droplet diameter effects MSC performance significantly. The characteristics of pressure swirl atomizer with tangential inlet ports has been examined by various investigators, and is described in Section 2.3 of this report. However, to the best of the author's knowledge the correlations presented in the literature were not based on information gathered from experiments with liquid metal. For example: atomizer performance could be presented by:

$$U = K \left(\frac{2\Delta p}{\rho_L} \right)^{1/2} \quad 5.5.1$$

where K relates to the energy losses in the atomizer. An idle atomizer is characterized by K equal to unity. For practical calculations Butterworth (Hetsroni, 1982) recommends: $K = 0.8$. In our experiments with mercury spraying K has a much lower value, as it is shown below. Thus it appears necessary to include experimental data with liquid metal in order to properly design MSC.

The experimental facility which was constructed and tested for this purpose is shown in Fig. 5.2.1. It is constructed using a 1.5 m long steel tube, 1 1/4" in diameter, that is vertically mounted. At its top a high pressure nitrogen cylinder was connected via a copper tube, while at the bottom the tested atomizer was connected in series to a full port ball valve. The column was filled with liquid and pressurized by nitrogen from the high pressure cylinder via a pressure regulator, at different constant pressures. The liquid flow rate through the tested atomizer was measured, by a stop watch and a calibrated container. At each pressure value the measurement was repeated 3-6 times in order to take average values. Two different atomizers were tested in this facility. One was the Monarch 631 x 69 giving a hollow cone spray angle of 80° at 40 psi. Its dimensions are listed in Table 5.5.1. The other one was the H-530 x 93 made of the same manufacturer, described previously (Section 5.2). Two different liquids were tested: mercury and water,

Figure 5.5.1

View of mercury sprayer experimental device

allowing calculation of pressure vs. flow rate characteristics. The measured value of nitrogen pressure at the top of the column was added to the hydrostatic pressure of the liquid column above the atomizer exit. The results of the measurements are listed in the table of the Appendix and discussed in the next section.

Table 5.5.1
Dimensions of Atomizer 631

L_p - inlet port length	7 mm
D_p - inlet port diameter	0.67 mm
L_s - swirl chamber height	4.1 mm
D_s - swirl chamber diameter	11.5 mm
l_o - orifice length	4.4 mm
D_o - exit nozzle diameter	0.67 mm
2θ - cone angle of discharge flow	80°

5.6 Results of Tests with Liquid Metal

The experimental results of the atomizers tests are shown in Figs. 5.6.1 and 5.6.2. Figure 5.6.1 shows the pressure vs. volumetric flow rate of atomizer 631 for both liquids. The respective results for atomizer H-530 are presented in Fig. 5.6.2. While plotting these results as pressure drop vs. Q^2 a straight line fit was found with a very high correlational coefficient for all data. These plots are presented in Fig. 5.6.3 for atomizer 631 and in Fig. 5.6.4 for the H-530 atomizer. The manufacturers data for water flow was compared to our relevant experimental data, and a good fit was found.

According to the recommendations made by Butterworth (Eq. 5.5.1), the pressure drop was calculated for the experimental flow rate values and compared with the experimental results for the 631 atomizer. The comparison is shown in Fig. 5.6.5.

The resulting experimental average value of K is 0.415, however, one can see from the figure that each liquid could be characterized by its specific K value. Thus, Eq. 5.5.1 could not be regarded as a general description.

The discharge coefficient, C_D (Eq. 2.3.12) was found to correlate the experimental results much better, and it is shown in Fig. 5.6.6 as a function of Re number, for the 631 atomizer.

The experimental data is generalized with a good correlation by the equation

$$C_D = 3.18 \text{ Re}^{-0.21} \quad 5.6.1$$

where

$$\text{Re} = \frac{4\dot{m}_L}{\mu_L \pi D_o} \quad 5.6.2$$

C_D values as calculated by Eq. 2.3.16 are also shown in Fig. 5.6.6, and they are much lower than the experimental values. Note that Eq. 2.3.16 does not consider the influence of the velocity on the discharge coefficient.

Using the form suggested by Jones for C_D (Eq. 2.3.17) yields:

$$C_D = 9.162 \text{ Re}^{-0.21} \left(\frac{L_o}{D_o}\right)^{-0.03} \left(\frac{L_s}{D_s D_o}\right)^{0.52} \left(\frac{D_s}{D_o}\right)^{0.23} \quad 5.6.3$$

The multiplier 9.162 and the power -0.21 in Eq. 5.6.3 differ significantly from the corresponding figures of Eq. 2.3.17, however the Re number of the latter correlation is based on the discharge velocity, and for given flow rates it results in much higher Re number values than those calculated by Eq. 5.6.3. Note also that either Jones or Rizk and Lefebvre experiments were conducted with larger atomizers than the type 631 used in our experiments (see Table 2.3.1 and Table 5.5.1). For a given pressure drop the atomizer flow rate is determined by Eq. 5.6.1 together with Eq. 2.3.12. The respective average droplet diameter, SMD, is calculated by substituting Eqs. 5.6.1 and 2.3.12 into Eq. 2.3.11.

Note that in case of two components gas mixture (vapor and inert gas) an equivalent gas density is to be applied for the calculation of SMD from Eq. 2.3.11. This equivalent density is:

$$\rho_{eq} = \frac{\rho_v \rho_g}{\rho_v \dot{m}_g / (\dot{m}_g + \dot{m}_v) + \rho_g \dot{m}_v / (\dot{m}_g + \dot{m}_v)} \quad 5.6.4$$

Due to the complexity of the flow in the H530 atomizer, the experimental results are less correlated by an expression of the form Eq. 5.6.1, as shown in Fig. 5.6.7. However, for pressure drops approximation, the generalizing curve:

$$C_D = 2.45 \text{ Re}^{-0.124} \quad 5.6.3$$

could be used.

Unfortunately, the correlations for mean drop size, appearing in the literature, for pressure swirl atomizers, could not be employed for this atomizer configuration. This H530 atomizer could be considered as a superposition of a plane orifice and a pressure swirl atomizer. For both configurations the general tendency is that an increase of surface tension as well as liquid viscosity results in an increase of the droplet mean diameter SMD.

Figure 5.6.1

Pressure drop vs. volumetric flowrate in atomizer 631.

Figure 5.6.2

Pressure drop vs. volumetric flowrate in atomizer H-530

Pressure drop vs. volumetric flow rate squared in atomizer 631.

Figure 5.6.3

Figure 5.6.4

Pressure drop vs. volumetric flow rate squared in atomizer H-530.

Figure 5.6.5

**Comparison between experimental and calculated (Eq. 5.5.1)
pressure drop values for atomizer 631.**

Figure 5.6.6
 C_D vs Re number for atomizer 631.

Figure 5.6.7
 C_D vs Re number for atomizer H530

6. Conclusions and Recommendation

The results of the experimental simulation of the performance of an MSC lead to the conclusion that the homogeneous mist flow condensation model could be applied to the process under study. It is assumed that at non-gravity condition, the droplets flow along the MSC will not develop into a stratified two-phase flow. Thus the condensation process is as described by the model. At given inlet conditions the required lengths for the condenser is as presented in Chapter 4. Special care must be taken in order to prevent the droplets from wetting the walls.

A swirl pressure atomizer is preferred for liquid cesium atomization. This atomizer gives a hollow cone type dispersion of the atomized spray, which demonstrates good mixing of the droplets and the gas. By extrapolating the experimental correlations to the conditions specified in the SOW, it is predicted that each 631 type atomizer will supply 0.011 kg/s of cesium, and the droplets mean diameter is supposed to be about 70 μm . The condensation process requires an MSC length of about 0.3 m. If alternatively the H530 atomizer type will be used, it is predicted that for each atomizer, Cs flow rate will be 0.019 kg/s however no accurate prediction for droplet size is available. Based on general tendencies, the mean drop size will be less than that of water, due to lower values of both: surface tension and viscosity of Cs at the given temperature.

Still, the alternative of injecting the Cs as particles or powder is to be considered since it is associated with a significant reduction of Cs flow rate compared to liquid spraying. However, neither the scientific aspects nor the engineering for this type of direct contact condensation was studied during the course of this project.

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APPENDIX

MSC simulation measured values

MSC simulation calculated experimental values

MSC performance data

Atomizer tests' experimental data

AIR FLOWMETER				STEAM FLOWMETER			MIXER EXIT			ϕ %	ATOMIZERS		MSC EXIT TEMPERATURE [°C]		
CASE	Q [L/S]	P [BARg]	T [°C]	m [g/s]	P [BARg]	T [°C]	P [BARg]	T [°C]	Q [L/S]		T [°C]	DRY BULB	WET BULB	WATER	
14	10.74	0.03	21.4	11	9.3	184.4	0	78.5		0.209	14.2	34.4	33.9	32.8	
15	12.98	0.05	21.6	4.4	9.34	183	0	75.5		0.209	15.4	35.6	35.4	34.4	
16	11.586	0.14	21.8	4.5	9.69	183	0	69		0.209	15.8	36.2	36.2	34.6	
17	10.854	0.23	21.8	5.2	9.5	183	0	64.5		0.209	15.8	34.2	34.2	32.6	
18	10.17	0.4	22	4.4	9.48	182	0.15	60		0.209	16	34.4	34.4	33.3	
19	10.193	0.39	22.2	0.9	9.26	181	0.15	60		0.228	16.8	34.1	33.9	33	
20	10.807	0.24	22.2	3.6	9.09	181	0	64.5		0.228	16	35.2	34.9	33.2	
21	11.586	0.14	23.8	3.8	9.38	182	0	68.8		0.228	14.8	34.2	34.2	32.6	
22	5.52	0	24.6	no steam injection			0	28	42.6	0.223	8.2	13.2	12.2	10	
23	4.22	0	24				0	21.3	68.3	0.223	7.6	14.4	12.4	10	
24	11.85	0.035	25.2				0	28.2	39.7	0.018	9.6	14	13	10.6	
25	11.85	0.035	25.2				0	27.8	41.1	0.018	9.6	14	12.8	10.4	

MSC simulation measured values

case	mixer exit			msc exit	
	air mass flow rate [g/s]	vapor mas flow rate [g/s]	relative humidity %	relative humidity %	
14	12.92	5.21	88.39	96.66	
15	15.91	6.01	96.05	98.68	
16	15.41	4.07	100	100	
17	15.58	3.13	100	100	
18	16.6	2.16	100	100	
19	16.5	2.15	100	98	
20	15.61	3.14	100	98	
21	15.3	4	100	100	
22	6.35	0.065	measured	89.4	
23	4.86	0.053		79.9	
24	14.4	0.135		89.65	
25	14.1	0.136		87.6	

MSC simulation calculated experimental values

CASE No.	inlet vapor concentration	condensate flow rate [g/s]		msc effectiveness [%]		extraction factor [%]	
		experiment	model	experiment	model	experiment	model
14	0.404	4.77	3.33	91.5	93.4	8.5	6.6
15	0.378	5.41	3.79	90	91.4	10	8.6
16	0.264	3.46	2.83	85	91	15	9
17	0.201	2.58	2.28	82.8	89.9	17.2	10.1
18	0.13	1.57	1.65	74	88.3	26	11.7
19	0.13	1.58	1.64	74.8	88.1	25.2	11.9
20	0.201	3.1	2.29	82.2	90.2	17.8	9.8
21	0.261	3.45	2.83	86.6	91.1	13.4	8.1
22	0.0102	0.01	0.0184	16	30.2	84	59.8
23	0.0109	0.013	0.0183	24.5	37.6	75.5	62.4
24	0.0095	0.0077	0.0224	5.7	17.1	94.3	82.9
25	0.0093	0.012	0.0246	8.9	18.5	91.1	81.5

MSC performance data

		PRESSURE DROP [atm]	VOLUMETRIC FLOW RATE [L/S]	MASS FLOW RATE [Kg/s]
H 530	mercury	4.31	0.018	0.25
		5.85	0.024	0.321
		7.69	0.0273	0.371
		9.5	0.031	0.415
	water	1.88	0.061	0.0608
		4.12	0.077	0.077
		6.32	0.097	0.097
		8.34	0.104	0.104
631	mercury	3.4	0.001	0.01367
		5.15	0.00122	0.01653
		6.87	0.00132	0.01793
		8.9	0.00149	0.02029
	water	2.12	0.003386	0.003374
		4.6	0.004711	0.004694
		6.3	0.00558	0.005563
		8.6	0.006277	0.006254

Atomizer tests' experimental data

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