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COMPARISON OF NUMERICAL SIMULATION AND EXPERIMENTAL DATA FOR STEAM-IN-PLACE STERILIZATION

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SUMMARY

A complex problem involving convective flow of a binary mixture containing a condensable vapor and noncondensable gas in a partially enclosed chamber was modelled and results compared to transient experimental values. The finite element model successfully predicted transport processes in dead-ended tubes with inside diameters of 0.4 to 1.0 cm. When buoyancy driven convective flow was dominant, temperature and mixture compositions agreed with experimental data. Data from 0.4 cm tubes indicate diffusion to be the primary air removal method in small diameter tubes and the diffusivity value in the model to be too large.

NOMENCLATURE

C_p = mixture specific heat at constant pressure, J/kg-°C
C_1 = mass fraction of air
$\mathbf{g}$ = gravitational vector, m/s^2
h = convective heat transfer coefficient, W/m^2-°C
k = thermal conductivity, W/m-°C
P = mixture total pressure, Pa
P_{sat} = saturated water vapor pressure, Pa
q = heat transfer rate, W
t = time, s
T = mixture temperature, °C
T_{amb} = ambient temperature, °C
T_{sat} = saturated steam temperature, °C
$\mathbf{u}$ = velocity vector, m/s
μ = mixture dynamic viscosity, Pa-s
u_x = x-component of velocity, m/sec
ρ = mixture density, kg/m^3
ρ_0 = mixture density at reference temperature and concentration, kg/m^3
ρ^* = mixture density used in momentum equation, kg/m^3
ρ_{sat} = saturated water vapor density, kg/m^3
α = binary mass diffusivity, m^2/s
β_C = mixture coefficient of concentration expansion
β_T = mixture coefficient of thermal expansion

Subscripts

f = fluid

t = tube

INTRODUCTION

Availability of commercial computational fluid dynamic (CFD) packages have made finite element modeling available for use as a tool in studying complex real-world fluid engineering problems. Previously most CFD work involved selection of a problem based on availability of software or study of simple geometries and very idealized problem formulations. This study describes application of a commercial CFD package to a complex problem, compromises made during model development and how the model was used in conjunction with experimental data to gain an understanding and start in quantizing the important physical parameters.

Steam-in-place (SIP) sterilization has arisen in the biotechnology and pharmaceutical industries as result of the increased need to sterilize large devices which can not be placed in an autoclave. SIP offers the additional advantage of sterilizing complete systems (i.e., filters, holding tanks and interconnecting piping) without the need for aseptic assembly of individually sterilized components. This provides greater sterility assurance, improved productivity and reduced cost over convectional sterilization in an autoclave.

SIP sterilization requires that an adequate amount of moisture at the proper temperature be delivered for a required time to all sterilization sites. The factor most often resulting in sterilization failure is air entrapment which results in inadequate temperature and moisture. Failures are most likely to occur in dead-ended geometries, deadlegs, such as safety valves, gauges/transducers and closed inlet/outlet lines. Displacement of air can result from molecular diffusion and/or buoyancy driven convective flow resulting from temperature and solutal gradients.

Published studies have focused on general SIP principles (refs. 1 and 2) or recommendations for specific pieces of equipment (ref. 3). Recently experimental data have been reported describing effect of deadleg tube orientation (ref. 4) and diameter (ref. 5) on sterilization, but no quantitative guidelines exist for design engineers and scientists. Currently, biological testing is conducted to determine if sterilization has occurred. This is time-consuming and expensive. A mathematical model is needed for the transient air/steam mixing process occurring during SIP sterilization. The model would predict temperatures and steam concentrations at locations within various deadleg geometries and could be used to determine general transport processes and critical parameters.

EXPERIMENTAL METHODS

Temperature and biological measurements were taken in tubes orientated vertically up or 5 degrees above horizontal. Inside diameters (IDs) of the tubes were 0.4, 1.0 and 1.7 cm with tube lengths ranging from 7.8 to 18.0 cm. Thermocouples or biological indicators were attached to a nylon string which ran along the tube centerline. The test fixture was pressurized with saturated steam having an average temperature of 122.4°C and pressure of 217 kPa. Details of experimental methods are given elsewhere (ref. 5).

Linear regression analysis was performed on the segment of the semi-logarithmic plots of surviving population versus time showing decreasing population. Slope of the regression line was used to determine time required to reduce the population by one log. This was termed cycle log reduction (CLR) time. Time at which the regression line intersected the initial population was termed time to start of kill (TSK). These two parameters characterized experimental kill at a location within a tube.

NUMERICAL SIMULATION

Simulation of deadleg SIP sterilization required:

1. Compressibility of air be taken into account
2. Development of expressions for physical properties of air/steam mixtures over wide ranges of composition (C_1 equal 0 to 1) and temperature (27 to 123°C).
3. Modeling of condensation from a mixture containing a noncondensable gas and condensable vapor.

Major effects associated with compressibility occur during initial pressurization with steam since the pressure is constant thereafter. This allows inclusion of these effects into the initial conditions for the problem (fig. 1b). It was assumed that all air was removed from the cross by steam flow and that the air in the tube was isentropically compressed to 217 kPa with no mixing of air and steam. This results in the top 56% of the tube initially containing all air at 100°C. The mixture was then considered to be incompressible with density ρ_0 and the Boussinesq approximation applied to the momentum equation to account for temperature and concentration induced density variations.

Actual mixture density at 217 kPa is given by :

$$\rho = \frac{217}{0.287\ T\ C_1 + \dfrac{P_{sat}\ (1-C_1)}{\rho_{sat}}} \qquad (1)$$

Density used to calculate buoyant forces in the momentum equation was determined by assuming density to be a linear function of temperature and mass fraction with coefficient of volume expansion due to concentration to be a function of temperature. Least squares analysis resulted in:

$$\rho^* = \rho_o\,(0.00407\,T + (0.01638-0.00178\,T)\,C_1) \tag{2}$$

Variations in other mixture properties were determined using published values for steam and air at 25 to 125°C and assuming properties to be proportional to mass fraction of each component. Published diffusivity values could not be found for air/steam mixtures at 217 kPa and over the appropriate temperature and mass fraction ranges. A semiemperical equation for diffusivity of low density gases which depends on total pressure, temperature and molecular properties of constituent gases was used. Diffusivity was evaluated at 217 kPa and 123°C and assumed constant (0.15 cm^2/s).

Heat transfer resulting from condensation of a condensable vapor from a binary mixture containing a noncondensable gas is complex and depends on mixture composition , as well as, fluid flow near condensation sites. Convective heat transfer coefficients can vary from 24 to 2500 W/m^2-°C as composition varies from C_1 equals 0 to 1. Heat transfer reductions of over 50% can result when only 2 to 5% mass of air is present during free convection (ref. 6 and 7). Substantially higher heat transfer occurs if forced flow is present and condensation is minimal (ref. 8 and 9).

The condensation model was kept simple since this represented a first attempt at modeling SIP sterilization and substantial computational time was required. The tube was divided into six segments and temperature of a tube wall segment was set equal to the saturated steam temperature corresponding to steam concentration in that segment when the mass fraction of air was less than 0.1 (fig. 1a). This simulates the high convective heat transfer associated with condensation and availability of an infinite steam supply. When C_1 was greater than 0.1, the mixture was treated as a simple mixture of two noncondensable gases and a constant convective heat transfer coefficient of 7 W/m^2-°C corresponding to free convection was specified on external surfaces of the tube segment. Heat transfer between the tube wall and mixture within the tube was specified as:

$$k_t\,\frac{\partial T}{\partial y}\Big|_{t@wall} = k_f\,\frac{\partial T}{\partial y}\Big|_{f@wall} \tag{3}$$

The commercially available CFD package FIDAP Version 6 (Fluid Dynamics International, Inc.) was used on a Silicon Graphics 4D35 workstation with 48 mg of memory. The governing equations were:

Continuity equation

$$\nabla \cdot u = 0 \tag{4}$$

Concentration equation

$$\frac{\partial C_1}{\partial t} + u \cdot \nabla C_1 = \alpha \nabla^2 C_1 \tag{5}$$

Momentum equation

$$\rho_0 \left[\frac{\partial u}{\partial t} + u \cdot \nabla u\right] = -\nabla P + \mu \nabla^2 u - \rho_0 g \left[\beta_T T + \beta_C C_1\right] \tag{6}$$

Energy equation

$$\rho_0 C_p \left[\frac{\partial T}{\partial t} + u \cdot \nabla T\right] = k_f \nabla^2 T \tag{7}$$

Quadrilateral finite elements were used with non-uniform meshes containing 1923 to 3464 nodes. A preliminary study was performed to evaluate the effect of two versus three dimensional modeling and to ensure grid-independence. The three-dimensional model required two weeks of CPU time to simulate 2.56 seconds of actual fluid flow, whereas the two-dimensional required 12 hours. Comparison of tube centerline temperatures showed a maximum average difference of 9.4%. Doubling the number of elements in the two dimensional model resulted in a 2.0% difference in average temperatures. Since five different tube simulations covering up to 40 minutes of real time were needed, two-dimensional models were used.

RESULTS AND DISCUSSION

Typical model velocity vector plots show steam rising at the tube centerline and the cooler air being displaced downward along the tube walls (fig. 2a). This results in a pair of counterrotating vortices. A second pair of vortices develop below the initial interface and the shear layers between the vortices become unstable (fig. 2b). The vortices pair and roll up resulting in a more homogeneous mixture in the tube (fig. 2c). The buoyant forces eventually diminish until only weak structures remain (fig. 2d).

Magnitude of convective flow is indicated by mean fluid speed (Table I). 0.4 cm ID tubes were predicted to have minimal flow. This was experimentally investigated by comparing centerline temperature profiles and biological results from vertically orientated tubes and those positioned 5 degrees above horizontal. Average temperatures were within 3.7% and neither tube showed significant biological kill above the interface after 2 hours, thereby confirming diffusion to be the primary mode of air removal.

The model predicted significant increases in convective flow with increasing tube diameter (Table I). Since mixture density depends on temperature and mixture composition (eq. (1)) and not diameter, buoyant forces are independent of diameter. Viscous or retarding forces increase with decreasing tube diameter and tend to damp out convective flow. In the case of 0.4 cm ID tubes, viscous forces quickly damp out flow and diffusion dominates. Temperature and biological data confirmed diameter to be a critical parameter for sterilization. Sterilization was achieved throughout an 18 cm long tube with 1.7 cm ID within 75 minutes whereas 185 minutes were required for a 7.8 cm long tube with 1.0 cm ID (fig. 3). Sterilization could be achieved in only the lower half of 0.4 cm ID tubes with lengths of 7.8 cm.

Model predicted and experimental temperature profiles were quantitatively compared for three different lengths of 1.7 cm ID tubes, as well as, 7.8 cm long tubes with diameters from 0.4 to 1.7 cm (fig. 4). % differences were less than 12% for all lengths of 1.7 cm ID tubes with average differences increasing with increasing tube length. % differences increased with decreasing tube diameter. Model temperatures were higher than experimental values for 0.4 and 1.0 cm tubes with % differences as high as 28%.

The model must quantitatively predict transient mixture composition, in addition to temperature, if it is to be used to predict sterilization times. Mixture composition can not be measured experimentally but can be inferred from CLR times and TSK. Young (ref. 5) has shown a correlation between CLR and saturated steam temperatures for vertical deadlegs. Therefore, experimental CLR times can be used to calculate mixture mass fraction. Model predictions of time required to reach this mass fraction can then be compared to experimental TSK. Due to % differences in temperatures for 0.4 and 1.0 cm tubes, comparisons of predicted and experimental TSK were carried out for 1.7 cm ID tubes only. % differences increased with increasing tube length. For the shortest tube, 7.8 cm, experimental TSK for the uppermost location was 2.6 minutes while the model predicted 2.2 minutes. Experimental values for 13.0 and 18.0 cm tubes were 10.2 and 17.5 minutes. Corresponding predicted values were 5.4 and 7.9

minutes.

Model and experimental data show diffusion to be the primary transport process in 0.4 cm ID tubes, but the value of diffusivity used in the model was too large. A smaller value would result in a greater time required for steam to diffuse to top of the tubes when convective flow is not present. In the case of 7.8 cm long tubes with 1.7 cm IDs, a smaller diffusivity will have minimal effect on temperatures and TSK since air is removed from the tubes within 2-3 minutes by buoyant driven convective flow. In the longer 1.7 cm tubes, convective flow dies out prior to complete removal of air. Therefore, diffusion becomes important in these 1.7 cm tubes. A smaller diffusivity would increase model predicted TSK and thereby increase agreement between model and experimental data. A similar argument can be made for 1.0 cm tubes.

An empirical value for diffusivity can be obtained by comparison of model and experimental results for 0.4 cm tubes. This value could be used in the model as a next level of refinement.

The present condensation model does not allow for accumulation of non-condensable gas at the site of condensation. This does not appear to be a problem when convective flow is present, but may present a problem in 0.4 cm tubes and larger diameter tubes when convection is minimal.

CONCLUSIONS

Although many simplifying assumptions had to be made, use of a commercial CFD package has been extremely useful in understanding transport processes and critical parameters in SIP sterilization. It correctly predicted diffusion to be the primary air removal mechanism in 0.4 cm ID tubes and diameter to have a significant effect on magnitude of buoyant driven convective flow. When convective flow is the only significant transport mechanism the model and experimental data were in good agreement. Interactive use of numerical model and experimental data has proven effective towards development of quantitative SIP guidelines.

REFERENCES

1. Berman, D., Myers, T. and Suggy, C., Factors Involved in Cycle Development of a Steam-in-Place System, *J. Parenter. Sci. Technol.*, Vol.40, No. 4, 1986, 119-121.
2. Agalloco, J., Steam Sterilization-in-Place Technology, *J. Parenter. Sci. Technol.*, Vol.44, No. 5, 1990, 253-256.
3. Myers, T. and Chrai, S., Steam-in-Place Sterilization of Cartridge Filters In-Line with a Receiving Tank. *J. Parenter. Sci. Technol.*, Vol. 36, No. 3, 1982, 108-112.
4. Young, J. H., Ferko, B. L., Temperature Profiles and Sterilization Within a Dead-ended Tube. *J. Parenter. Sci. Technol.* Vol. 46, No. 4, 1992, 100-107.

5. Young, J. H., Sterilization of Varying Diameter Dead-ended Tubes. *Biotechnology and Engineering*. Vol. 42, No. 1, 1993, 125-132.
6. Mori, Y., Hijikata, K., Free Convective Condensation Heat Transfer with Noncondensable Gas on a Vertical Surface. *Int. J. Heat Mass Transfer*. Vol. 16, 1973, 2229-2240.
7. Sparrow, E. M., Lin, S. H., Condensation Heat Transfer in the Presence of a Noncondensable Gas. *J. of Heat Transfer*. August 1964, 430-436.
8. Al-Diwany, H. K, Rose, J. W., Free Convection Film Condensation of Steam in the Presence of Non-condensing Gas. *Int. J. Heat Mass Transfer*. Vol. 16, 1973, 1359-1369.
9. Sparrow, E. M., Mincowycz, W. J. and Saddy, M., Forced Convection Condensation in the Presence of Non-Condensables and Interfacial Resistance. *Int. J. Heat Mass Transfer*. Vol. 93, No. 3, 1967, 1829-1838.

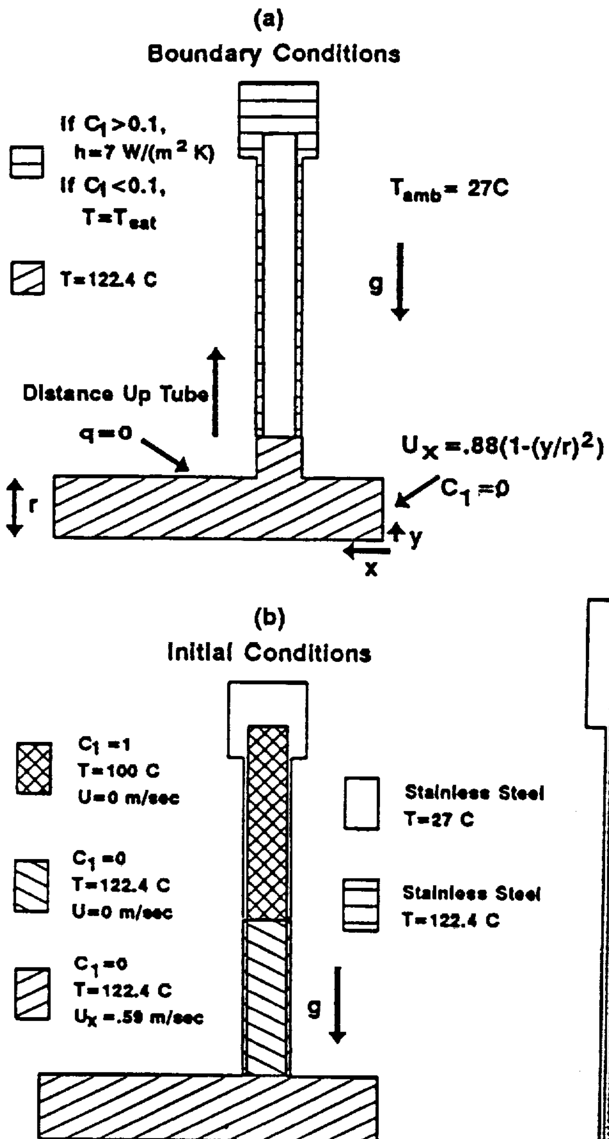

Figure 1- Schematic representation of numerical model (a) boundary conditions and (b) initial conditions.

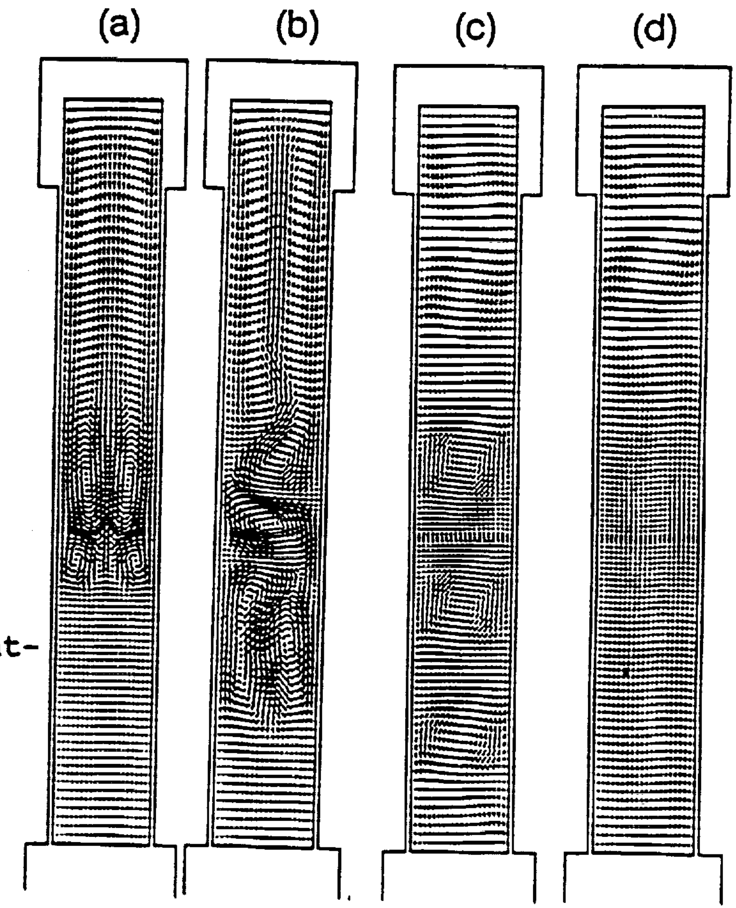

Figure 2- Velcity vector plots from 13.0 cm long tubes with 1.7 cm ID; (a) 0.48 seconds, (b) 1.75 seconds, (c) 1.02 minutes and (d) 4.75 minutes after pressurization with steam.

Time (Minutes)	Mean Speed (cm/sec)		
	0.4 cm ID Tube	1.0 cm ID Tube	1.7 cm ID Tube
0.02	0.029	1.43	3.55
0.05	0.007	0.97	3.42
0.50	0.005	1.14	2.37
1.0	0.005	0.89	2.05
3.0	0.005	0.74	0.11
5.0	0.005	0.02	0.03

Table I- Transient mean fluid speed for 7.8 cm long tubes.

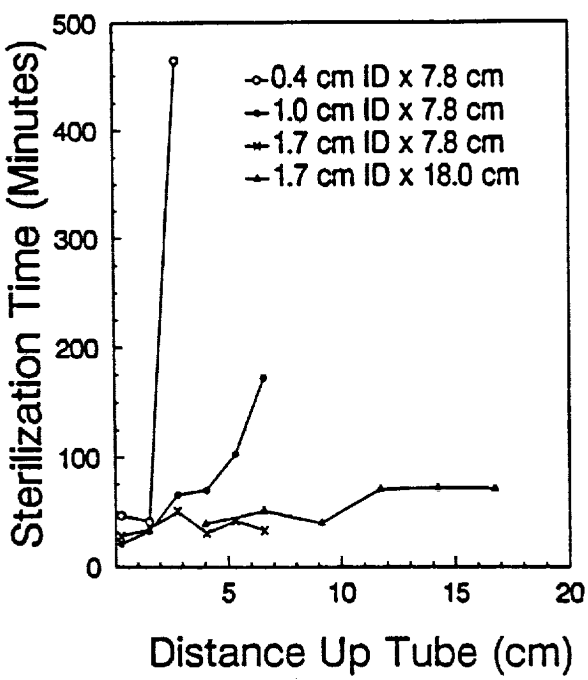

Figure 3- Time required for 12 log reduction of *B. stearothermophilus* spores

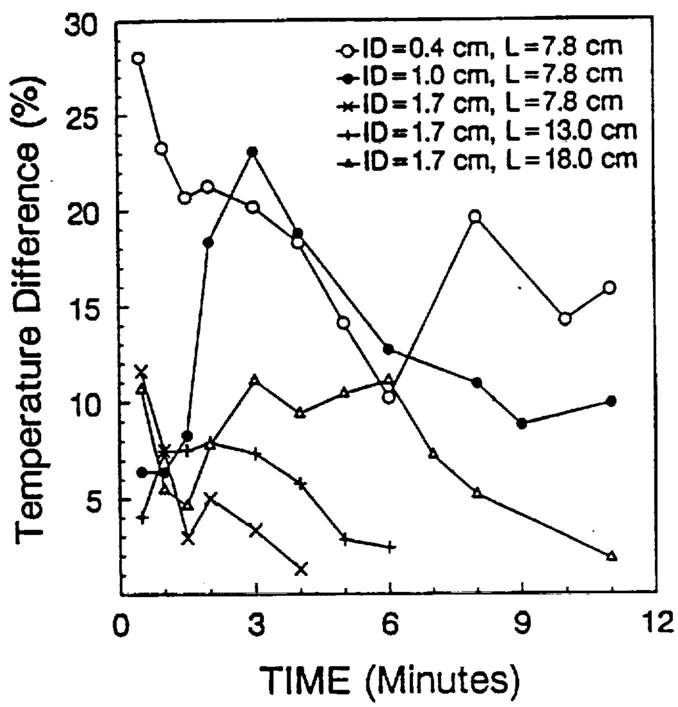

Figure 4- % difference between model and experimentally measured centerline temperatures.