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ENERGY AND HEAT FLOW IN AN AIRBAG SYSTEM

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Paper was presented at the 23rd International Workshop on Human Subjects for Biomechanical Research. This paper has not been screened for accuracy nor refereed by any body of scientific peers and should not be referenced in the open literature.

ENERGY AND HEAT FLOW IN AN AIRBAG SYSTEM

The objective of this study was to address: 1) the use of an energy model to simulate the temperature and pressure variations in the air bag system, and 2) a heat energy study to quantify the influence of the inflator porous media on the response characteristics of the air-bag system.

TIME-VARYING THERMAL ENERGY MODEL IN 2-D COMPRESSIBLE GAS

The purpose of this part of the investigation was to provide a means to numerically study the thermal energy distribution in the airbag during the deployment of the airbag system.

Assumptions

The time varying thermal energy model includes the unsteady conservation laws of mass, momentum (in 2-directions) and energy. The following fundamental concepts are used in the development of this model.

(1) The model is developed for application to a general two-dimensional domain. It can be used to simulate one-dimensional problems by proper specification of boundary conditions.

(2) The compressible gas is considered non-viscous. The process is highly transient with high pressure, temperature and velocity changes, therefore, the viscous dissipation of momentum and energy is relatively small with respect to other energy components, and may be neglected.

(3) A heat source or sink may or may not exist inside the domain. The heat exchange could occur between gas and solid boundary, e.g., between gas and bag wall or between gas and canister wall, a local energy equilibrium condition is applied here to supply a boundary condition.

(4) The model is developed to treat flow velocities up to sonic conditions. For velocities in excess of Mach number 1.0 shock waves develop in the physical system. More detailed equations than coded herein are required to handle the details of abrupt pressure, density, velocity, and temperature changes at shock fronts. Thus, conceptually the code should not be used to attempt to simulate even slightly supersonic flow fields. However, for low velocity supersonic jet flows ($M \leq$ approximately 1.3) only weak shocks occur physically, so the numerical procedure provides reasonable results.

Basic Equations

The basic equations for this model are listed:

$$\frac{\partial \rho u}{\partial x}+\frac{\partial \rho v}{\partial y}+\frac{\partial \rho}{\partial t}=0 \quad (1)$$

$$\frac{\partial p}{\partial x}+\frac{\partial \rho u^2}{\partial x}+\frac{\partial \rho uv}{\partial y}+\frac{\partial \rho u}{\partial t}=0 \quad (2)$$

$$\frac{\partial p}{\partial y}+\frac{\partial \rho uv}{\partial x}+\frac{\partial \rho v^2}{\partial y}+\frac{\partial \rho v}{\partial t}=0 \quad (3)$$

$$\frac{\partial[(\rho E+p)u]}{\partial x}+\frac{\partial[(\rho E+p)v]}{\partial y}+\frac{\partial \rho E}{\partial t}=\rho Q \quad (4)$$

$$E=e+\frac{1}{2}(u^2+v^2) \quad (5)$$

$$e=C_v T_g=C_v\frac{p}{R\rho}=\frac{1}{k-1}\frac{p}{\rho} \quad (6)$$

in which p, u, v are the pressure and velocity components in x- and y- directions, respectively; t = time; ρ = the density of the gas; Q is the amount of heat transferred per unit mass and unit time, which is applied only inside the domain if an internal heat exchange mechanism exists; E = internal energy; e = intrinsic energy; T_g = the temperature of the gas; C_v = the specific heat of the gas under constant volume; R = gas constant; and κ = the ratio of specific heat. The set of equations is first transformed into the equations with the pressure p, velocities u, v and density ρ as the four dependent variables. The characteristics-like method is then used to place the equations in numerical form for computer solution. With appropriate boundary treatments, a numerical energy model is set up for simulating the compressible gas flow with transient mass and heat transfer.

The mass flow rate measurement

The mass flow rate was obtained through the use of a large receiving tank (100 liters) that contains the inflator. Measurements are the inflator pressure history, the tank pressure history and final temperature. Conceptually it should be relatively direct to compute the time history of the inflator mass flow rate from the pressure and temperature sensor readings. Such is not the case due to the hostile environment, and due to considerable variability between inflators. Robust temperature probes with sufficiently fast response times are not available to measure the temperature inside the inflator. Sensors to provide pressure histories for the inflator also are not always accurate for a number of reasons -- one being that generate or particulate matter may block the pressure measurements. Inflators used in the tests may not be identical, and the inflators used in the tank tests could not be the ones used in the air bag deployment test.

The procedure adopted to produce the mass flow history, needed for experimental evaluation and for numerical simulation, is first discussed, followed by detailed equations. A sampling (statistical) procedure was used in order to reduce variability and to provide consistent mass flow rate data. From a single lot of inflators 15 were randomly chosen. Of the 15 inflators, ten were used in the tank tests and five were set aside for the deployment test. For the tank tests the inflator weight was measured before and after to provide a measure of the total mass issued into the tank, and during the test the inflator pressure history, the receiving tank pressure history, and the final temperature were recorded. If the maximum tank pressure varied between tests by more than 5% then the lot was rejected. The inflator pressure history and the total mass are the most sensitive parameters. Since the inflator pressure measurement may contain inaccuracies that lead to large deviation from the true mass flow rate, a minimum of 8 inflator pressure measurements without experimental contamination was required. Contamination was determined by making a comparison of the total mass obtained by integration of the inflator mass flow rate and the measured total mass. If the two differed by more than 5% then that inflator's results were not used. If the above criteria were met, the five remaining modules were then used in the deployment test. In the deployment tests, the pressure inside the inflator was measured and used to calculate the mass flow rate. If this measured pressure was outside the range of those that were obtained in the tank pretest, that deployment test was rejected. Otherwise the average of the mass flow rate obtained from the tank test was used for the numerical model calculations. Although this procedure did not completely remove the experimental error, it helped to reduce test to test variation, and it addressed the problems of not being able to measure the mass flow rate and temperature of the tested system.

Calculation of mass flow rate of inflator

By assuming critical flow (sonic flow) at the minimum open area of an orifice (throat), the mass flow rate is:

$$\mathrm{m} = \rho^* A^* V^* \tag{7}$$

where the asterisks indicate conditions in the throat at sonic flow, m = mass flow rate, ρ = density of gas, A = the cross-sectional area, V = the velocity. By assuming that the gas flow is isentropic [Streeter and Wylie, 1985] conditions in the throat can be related to conditions inside the inflator

$$\frac{T*}{T_0}=\frac{2}{k+1}; \quad \frac{p*}{p_0}=\left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}; \quad \frac{\rho*}{\rho_0}=\left(\frac{2}{k+1}\right)^{\frac{1}{k-1}} \tag{8}$$

where T^*, p^*, and ρ^* are the temperature, pressure and density at the throat, T_0, p_0, and ρ_0 are the temperature, pressure and density inside the inflator, κ = the specific heat ratio. Substitution of Eq. (8) in Eq. (1) leads to

$$m = \frac{A^* p_0}{\sqrt{T_0}} \sqrt{\frac{k}{R} \left(\frac{2}{k+1}\right)^{\frac{1}{k-1}}} \tag{9}$$

where R = gas constant. Since 99% of the gas from an inflator is nitrogen then k = 1.4 and R = 297m. N/kg. ° K, and

$$m = \frac{0.686 A^* p_0}{\sqrt{R}\sqrt{T_0}} \tag{10}$$

For a sharp edged orifice, a discharge coefficient is needed which includes a contraction coefficient and a velocity coefficient. Although the inflator pressure history may be measured with some reliability, the temperature history can not. The time history of temperature is estimated from fuel (used in the inflator) burning tests.

Two methods were used to check the consistency of the total mass of gas from each inflator: one used the final temperature and pressure from the tank test, and the second used the weight of the inflator before and after discharge. In the first method the mass was calculated from the perfect gas law

$$m_f = \frac{p_f V}{R T_f} \tag{11}$$

where T_f, p_f *and* m_f indicate the final values of temperature, pressure and total mass, and V is the tank volume. If the difference between this calculated total mass m_f and the total mass from weighing the inflator before and after the test is less than 5%, then this m_f is used for verifying and correcting the calculated mass flow rate from Eq. (4). The mass flow rate calculated with Eq. (10) was adjusted proportionately so the total mass, obtained by integrating m over time, equaled the value obtained with Eq. (11). Since the mass into the tank is known from Eq. (10), the time history of the temperature can be estimated from Eq. (10) and the temperature of the gas exiting the orifice.

It is difficult to physically measure the temperature pattern inside the airbag during deployment due to the speed of the process, and to its aggressive dynamic nature. The published experimental data of the temperature in an airbag ranges by a factor of 3 or more. Thus the need for a reliable simulation.

Two dimensional airbag model

This two-dimensional dynamic fluid flow model is unable to account for fabric unfolding and expansion, therefore, the airbag inflation process is not simulated. The semiaxes dimensions of the ellipse-shaped computational domain were 1.77 and 1.175 ft yielding a volume of the domain equal to the volume of the inflated airbag, 6.54 ft^3. The computational mesh, which represents half of the domain, is shown in Figure 1. The spatial step was Δx = 0.01041 ft. An actual vent hole of 2 inches in diameter was represented by two slots, 12 inches long and 0.01057

Assumptions

The model adopted here is an adaptation of the three-equation model described above. The equations of the conservation of mass, energy, and momentum in two-directions have been adapted for flow in porous media. There are several assumptions made for this model.

(1) A linear viscous loss and a nonlinear inertial loss term have been added to the momentum equations. The coefficients involved in those two terms are estimated with the relations recommended in Sozen and Vafai, 1993.

(2) The heat exchange between the metal screen and the gas follows the linear law, i.e., the heat transferred is directly proportional to the temperature difference between the metal screen and the gas.

(3) Heat conduction of the gas is neglected in the energy equation, since a short transient process (order of 10^{-2} second) is considered in present case.

(4) The metal screen temperature is calculated by the first law heat energy balance in the metal screen only. Heat conduction in the metal screen is neglected due to the same reason as stated in (3).

(5) The frame of the metal screen is fixed, no deformation occurs during the fuel burning and discharging process.

(6) There is no chemical reaction taking place between metal screen and the gas.

Basic Equations

The whole set of equations for two-dimensional unsteady compressible gas flow through a porous media are summarized as follows:

$$\frac{\partial \rho u}{\partial x}+\frac{\partial \rho v}{\partial y}+\frac{\partial \rho}{\partial t}=0 \tag{12}$$

$$\frac{\partial p}{\partial y}+\frac{\partial \rho uv}{\partial x}+\frac{\partial \rho v^2}{\partial y}+\frac{\partial \rho v}{\partial t}+\frac{\rho g}{K}\beta v+\rho R_{ms}\beta^2 v\sqrt{u^2+v^2}=0 \tag{13}$$

$$\frac{\partial p}{\partial x}+\frac{\partial \rho u^2}{\partial x}+\frac{\partial \rho uv}{\partial y}+\frac{\partial \rho u}{\partial t}+\frac{\rho g}{K}\beta u+\rho R_{ms}\beta^2 u\sqrt{u^2+v^2}=0 \tag{14}$$

$$\frac{\partial[(\rho E+\rho)u]}{\partial x}+\frac{\partial[(\rho E+\rho)v]}{\partial y}+\frac{\partial pE}{\partial t}=-\frac{1}{\eta}K_{ex}a_{sv}(T_g-T_{ms}) \tag{15}$$

$$\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}-(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x})=0 \tag{16}$$

$$E=e+\frac{1}{2}(u^2+v^2) \tag{17}$$

$$e=C_vT_g=C_v\frac{p}{R\rho}=\frac{1}{k-1}\frac{p}{\rho} \tag{18}$$

$$(1-\eta)\rho_{ms}C_{pms}\frac{\partial T_{ms}}{\partial t}-K_{ex}a_{sv}(T_g-T_{ms})=0 \tag{19}$$

$$\beta=1-\sqrt{1-\eta} \tag{20}$$

where p is the pore-volume averaged pressure, u,v are the area-averaged pore velocities (in x-direction and in y-direction), K and R_{ms} are the hydraulic conductivity and non-linear resistance coefficient of the metal screen, respectively. T_g and T_{ms} are the gas and metal screen temperatures, respectively. ρ and ρ_{smc}, C_v and C_{pms} are density and specific heat of the gas and metal screen, respectively. η is the porosity of the porous medium. β is the ratio of pore cross-sectional area. k_{ex} is the convective heat transfer coefficient between the gas porous medium. a_{sv} is the specific surface area of the porous media particles. Equation (17) is an identity used for the numerical transformation only.

The set of equations is transformed by the characteristics-like method, and placed in finite difference form for numerical solution. The input boundary conditions are mass flow rate and temperature at the fuel interface. The boundary condition at the inflator wall is a no-flow condition, except for the gas-flow holes. An orifice formulae with the 'choke' condition employed is used on the solid boundary. Extensive computations were carried out to calibrate the heat transfer coefficients, k_{ex} k_w by using the data.

The computational domain, which represents one quarter of a section through the inflator metal screen is shown in Figure 8. Graphic displays of the pressure contour, temperature contour, ft. wide each. The total area of the slots was equal to the total area of the vent hole. The thermal conductivity of the gas was 0.002966 ft-lb/ft-s-°R, the specific heat was 4281.0 ft-lb/slug-°R, and heat exchange coefficient k_w at the canister and inflator walls was 0.0065 ft-lb/ft^2-s-°R.

The input boundary conditions are mass flow rate obtained from the above procedure and temperature, which were obtained from the inflator porous media computation, described below. The boundary condition on the inflator wall and canister wall was a no-flow condition, and a local heat energy equilibrium condition was used, except for the gas-flow holes. An orifice formula, with a 'choke' condition employed, was used at the vent hole.

Figure 1 is the computational mesh for the airbag, and canister system. Figure 2 contains the input mass flow rate used to inflate the bag as well as the temperature of that gas. Figure 3 is the temperature profile with heat exchange, and Figure 4 is the temperature profile in the bag without heat exchange. Figure 5 is the average temperature time history at the center of the bag and at the exhaust hole. Figure 6 is the energy time history for different air-bag components. Several preliminary points may be drawn from the results obtained at the present stage of this study.

(1) The pressure and temperature are almost uniformly distributed in the airbag at any instant after deployment. A big drop in pressure and temperature occurs in a very small region next to the vent hole due to the high discharge velocity.

(2) The maximum temperature is about 700°F in the bag and 640°F at the venthole in the heat exchange case. The amount of heat absorbed by the canister and inflator wall from the hot gas during deployment is about 2.5 Btu, which may raise the temperature of the canister wall about 5°F.

(3) The obvious temperature gradient along the canister wall, shown in the temperature contour plot, in the case of heat exchange, compared with the temperature contour in the case of heat exchange, indicates the effect of the heat transfer between gas and canister and inflator wall.

Model for compressible gas flow through porous media

A simplified model showed that the residual heat in the porous media (inflator screens and wall) and in the fuel ash at the end of deployment must be responsible for the temperature response obtained over the next 200 seconds on the inflator outer wall (see measured response, Figure 7). This conclusion encouraged the development of a more complete model of gas flow and heat transport and transfer in the porous media during the first 60-80 ms of deployment. The objective was to understand the transient flow pattern, the pressure and temperature distributions in the domain of the porous media, and to provide a thermal energy balance over the first 60-80 ms. This information, together with a thermal energy distribution in the porous metal region at the end of this early time analysis, would provide a deeper insight to the process of the airbag deployment.

and flow pattern at 23 ms are shown in Figures 9-11. The pressure and velocity distributions along sections A - A and B - B are shown Figures 12 and 13. The comparison of input mass flow rate with the computed mass flow rate through the discharging holes, conservation of total mass (input and computed outgoing through the discharging holes), the time histories of pressure at the input and the discharging holes, the time histories of the temperature of the gas and metal screen at the input and at the discharging holes, and lastly the time histories of total input thermal energy, the energy stored in metal screen, the energy going out with gas and the energy stored in the gas contained in metal screen, are shown in Figures 14 to 18. It can be seen that the conservation of mass is well satisfied, the conservation of energy is also reasonably well satisfied, along with the temperature response obtained over the next 200 seconds on the inflator wall (Figure 19). The energy brought out by the outgoing gas is also closely equal to what is needed for heating up the gas to the final temperature of around 650°F in the tank test (Figure 20).

CONCLUSIONS

Models have been established for investigating the airbag deployment process, including thermal energy. The results from these models provide dynamic and energy patterns inside the inflator and in the domain between the inflator discharge openings and the air bag. The models should be useful for further studies.

(1) The agreement between the measured and the computed pressure and temperature time histories for the tank test implies that the time-varying thermal energy model provides reliable results for this case. The inclusion of heat transfer at the tank wall has some effect on the thermal analysis. With the experience gained in simulating the tank test, the thermal energy model may be used with some confidence for further energy studies of the unsteady compressible gas flow field between the inflator discharge openings, the canister and the air bag.

(2) The results from the model of compressible flow through the porous media show a complete picture of the dynamics of compressible gas flow and thermal energy distribution in the domain of the metal screen during deployment. Overall mass and energy balances are achieved. The model has not been evaluated extensively against experimental data. However, it has the potential to provide results which may be compared with data from other sources.

REFERENCES

Hoffman, E. J., "Heat Transfer Rate Analysis," PennWell Books, 1980.

Moody, F. J., "Introduction to Unsteady Thermofluid Mechanics," John Wiley & Sons, 1990, pp. 89 - 97.

Streeter, V. L., and Wylie, E. B., "Fluid Mechanics," McGraw-Hill, 1985, 250.

Suo, L. S., and Wylie, E. B., "Two-Dimensional Numerical Model of Airbag System," Report to Chrysler, #UMCEE 90-20, 1990, 69 pp.

Wylie, E. B., and Streeter, V. L., "Fluid Transients in Systems," Prentice Hall, New Jersey, 1993, pp. 410 - 413.

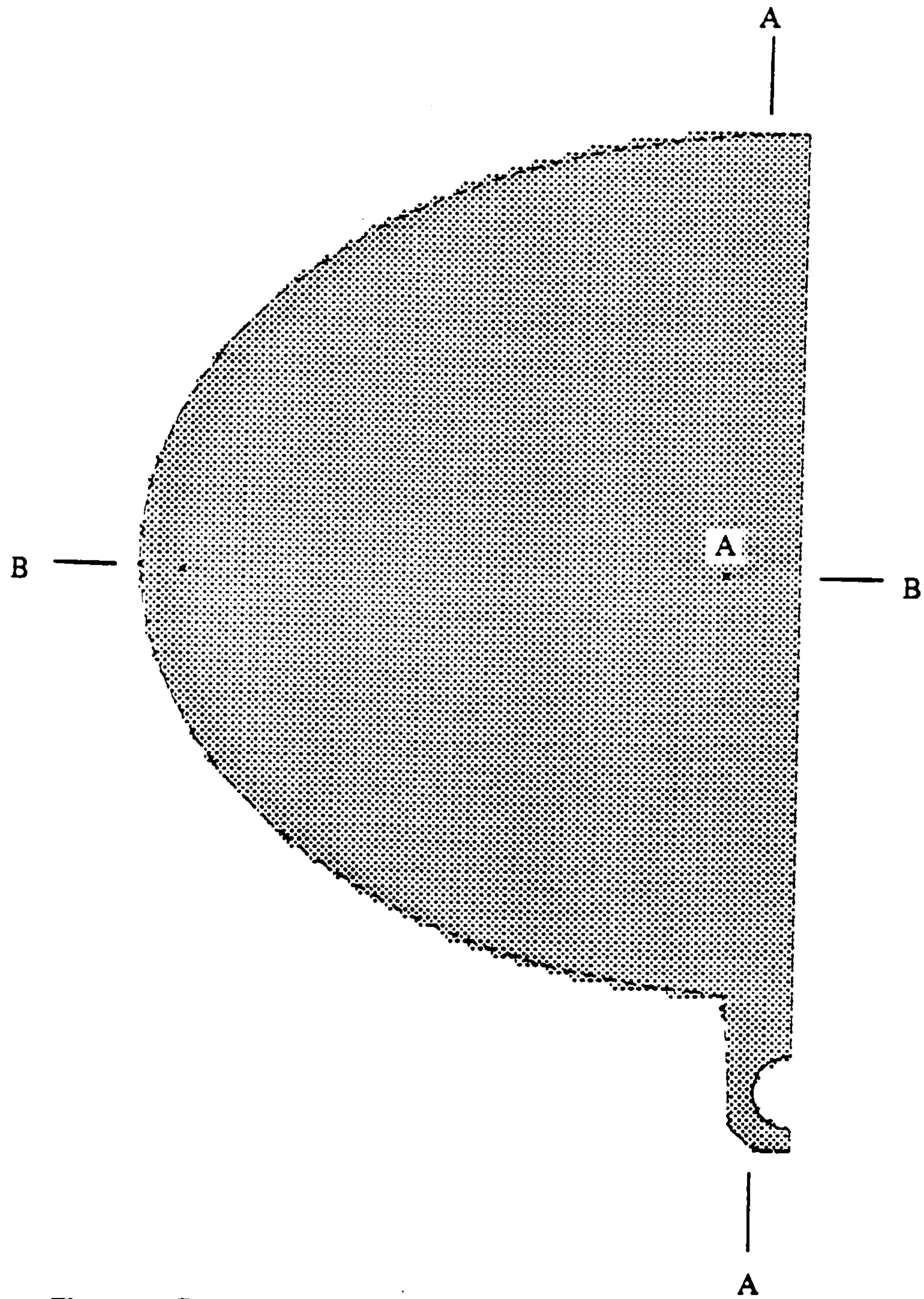
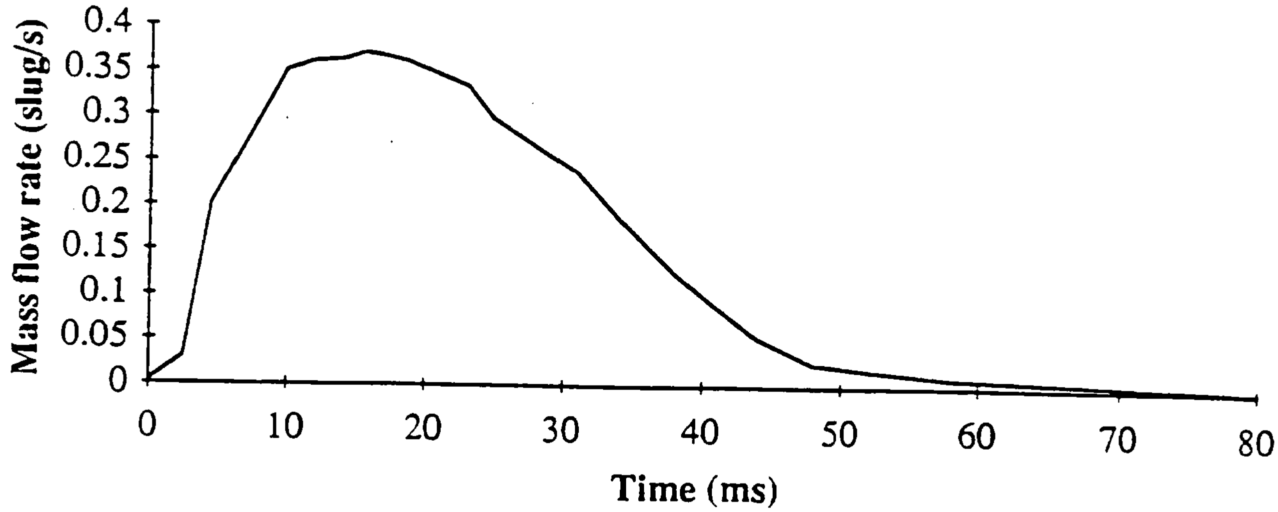


Fig. 1 Computational mesh for airbag inflator-canister system



Input mass flow rate for airbag simulation

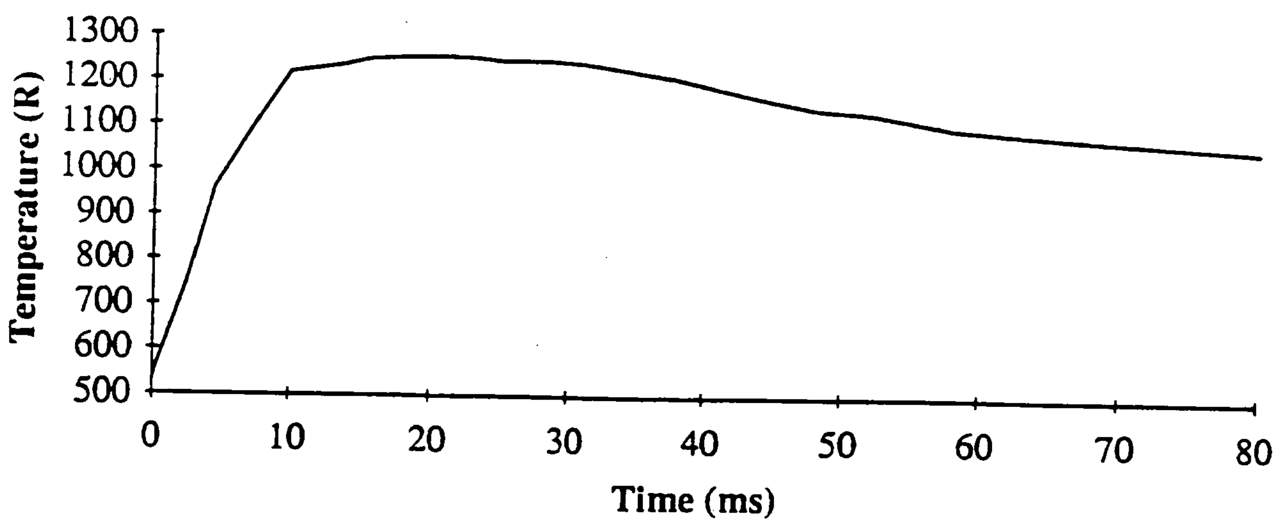
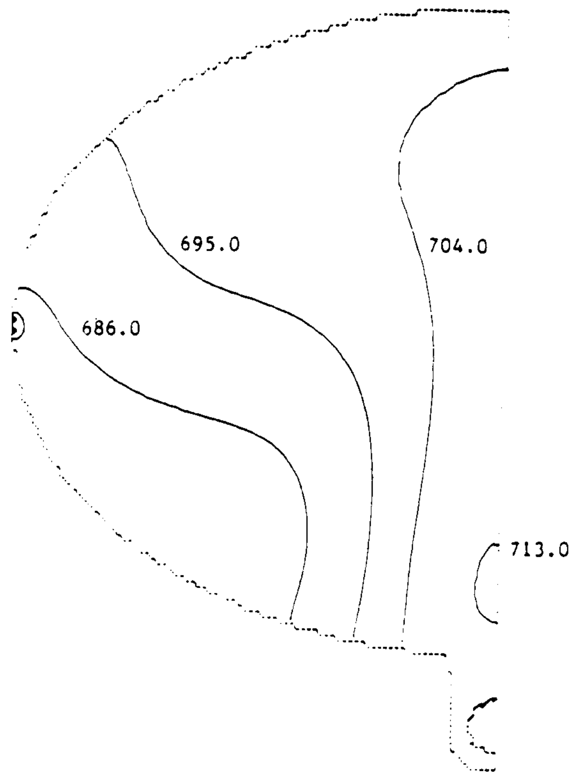


Fig. 2   Input temperature for airbag simulation



Temperature contours at t=44 ms, $\Delta T = 9°F$ (with heat exchange)

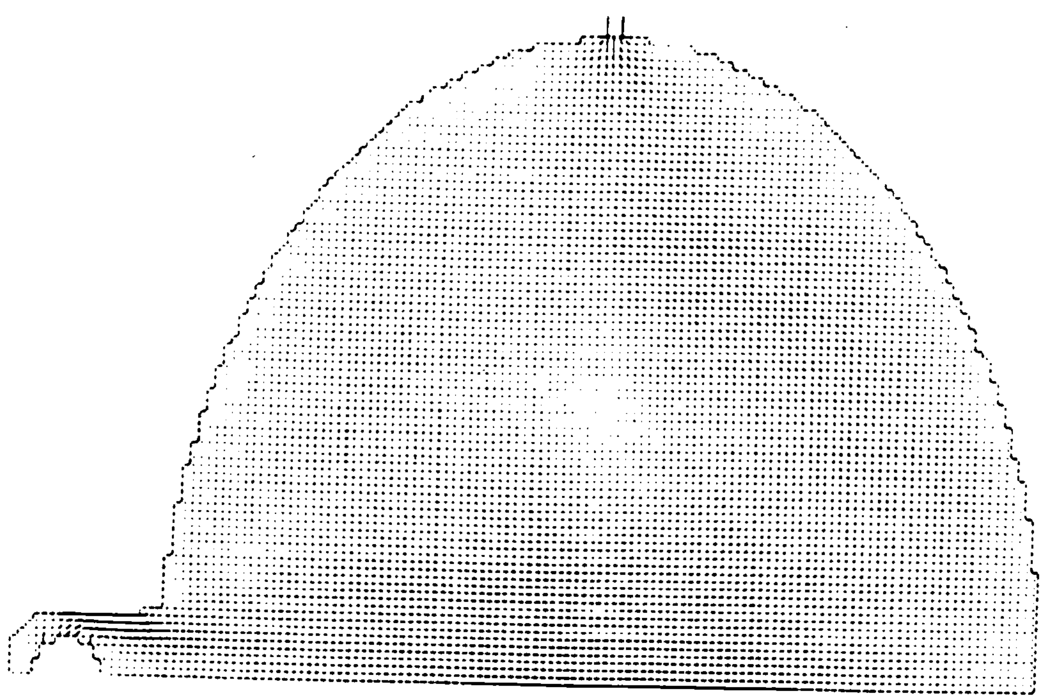
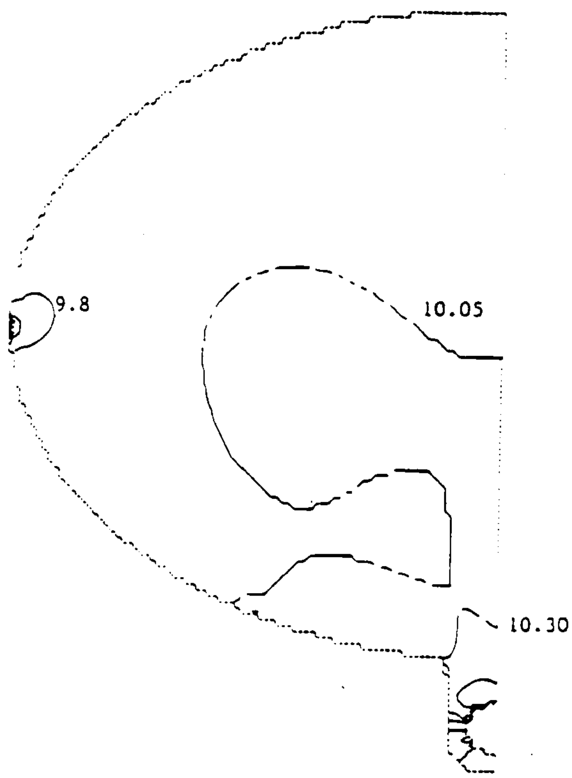


Fig. 3 Velocity vectors at t=44 ms (with heat exchange)



Pressure contours at t=43 ms, $\Delta p = 0.25$ psi (no heat exchange)

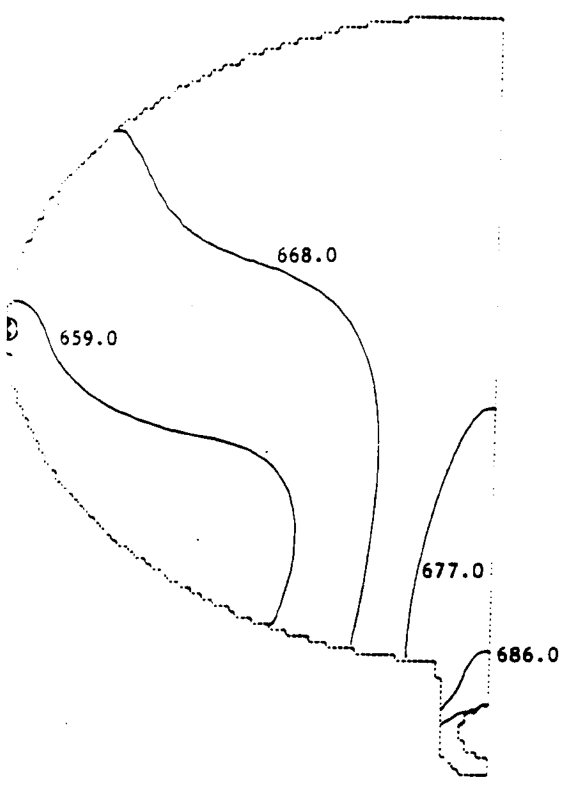


Fig. 4 Temperature contours at t=43 ms, $\Delta T = 9°F$ (no heat exchange)

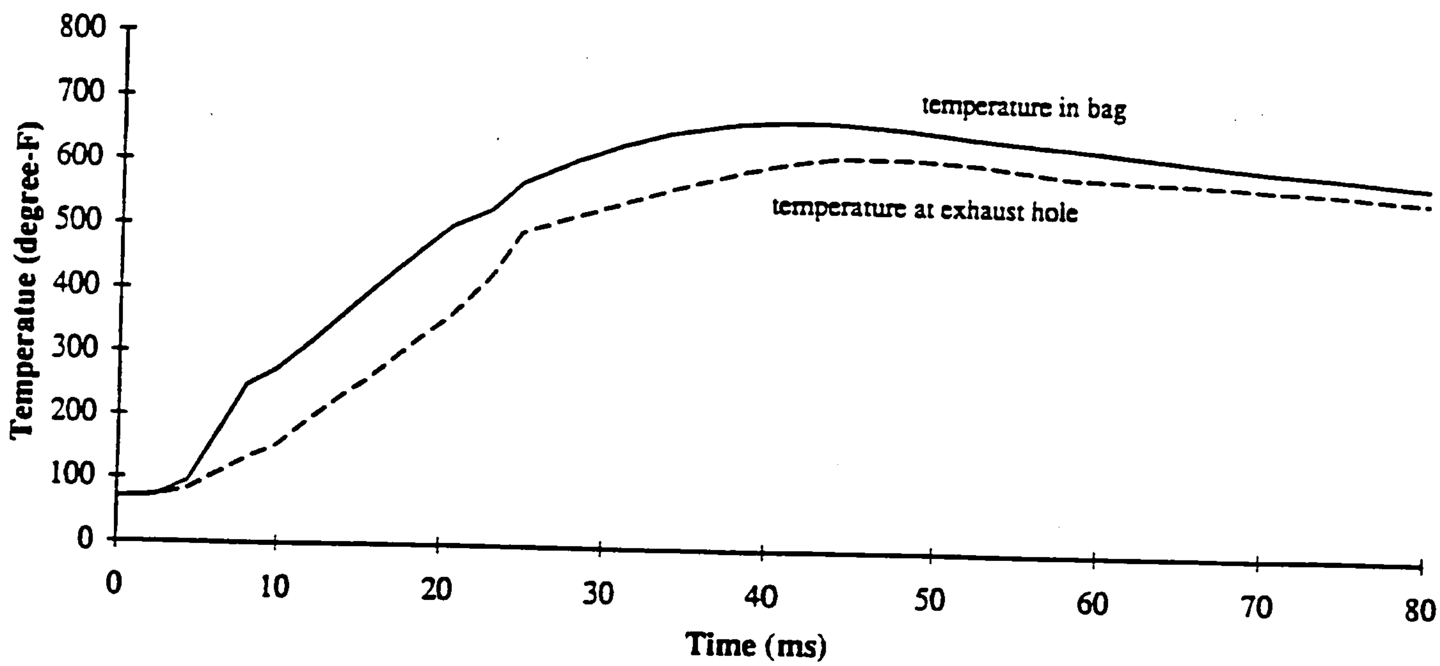
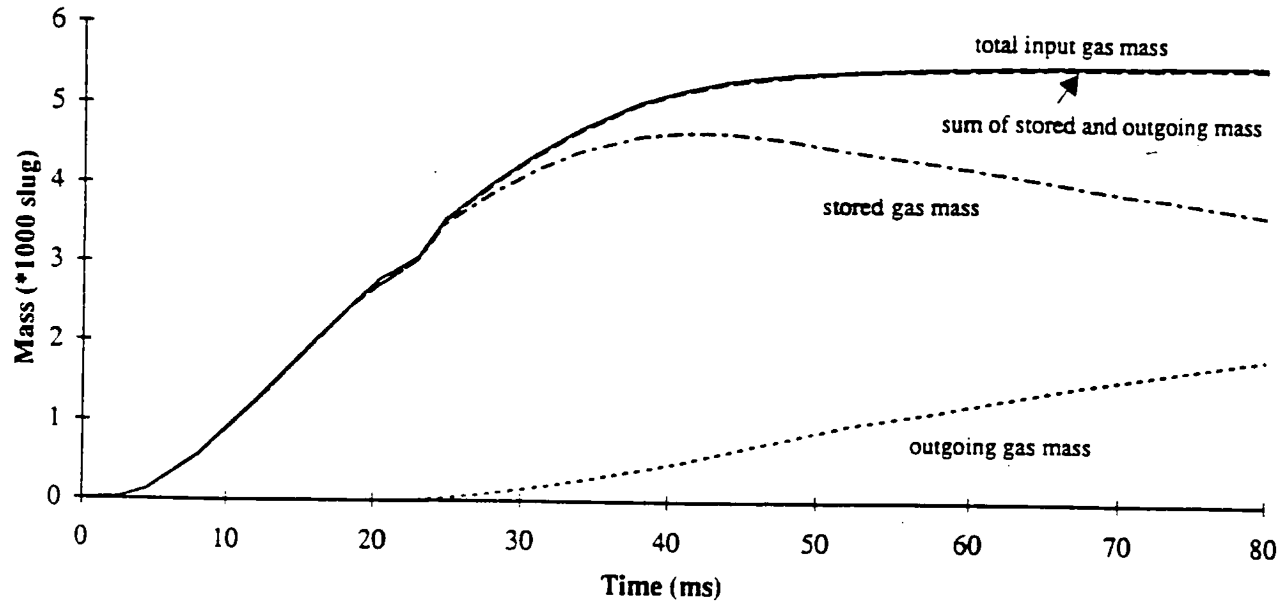


Fig. 5 Computed temperature time history at node A in bag and at vent hole (with heat exchange)



Temporal mass variations (with heat exchange)

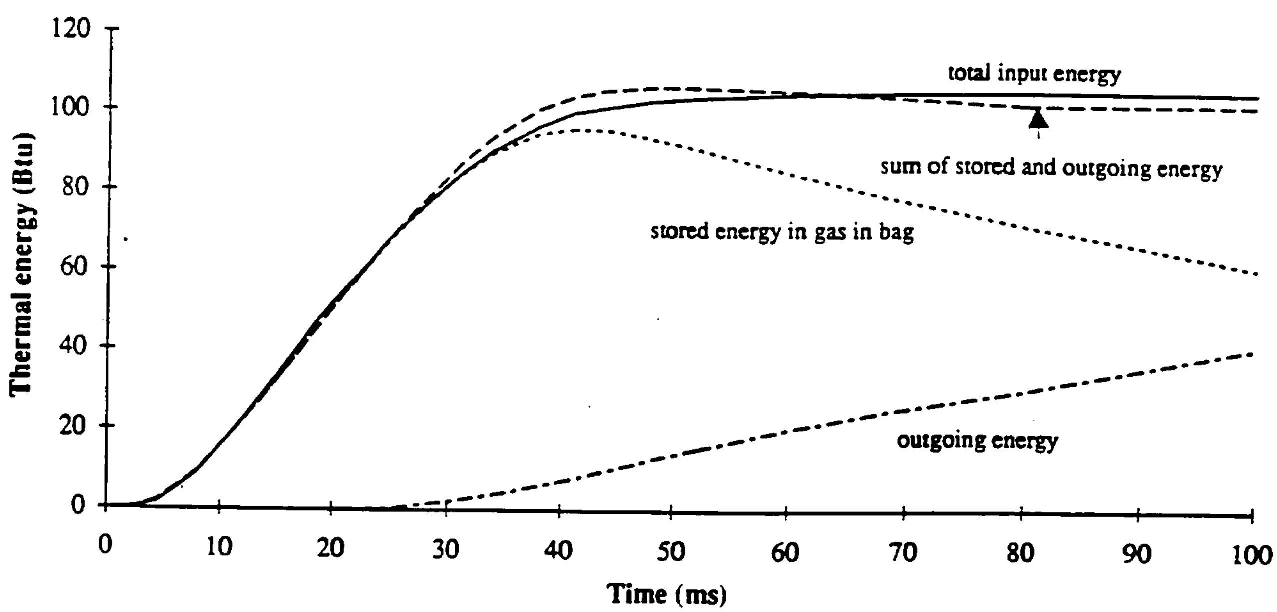


Fig. 6 Temporal energy variations (with heat exchange)

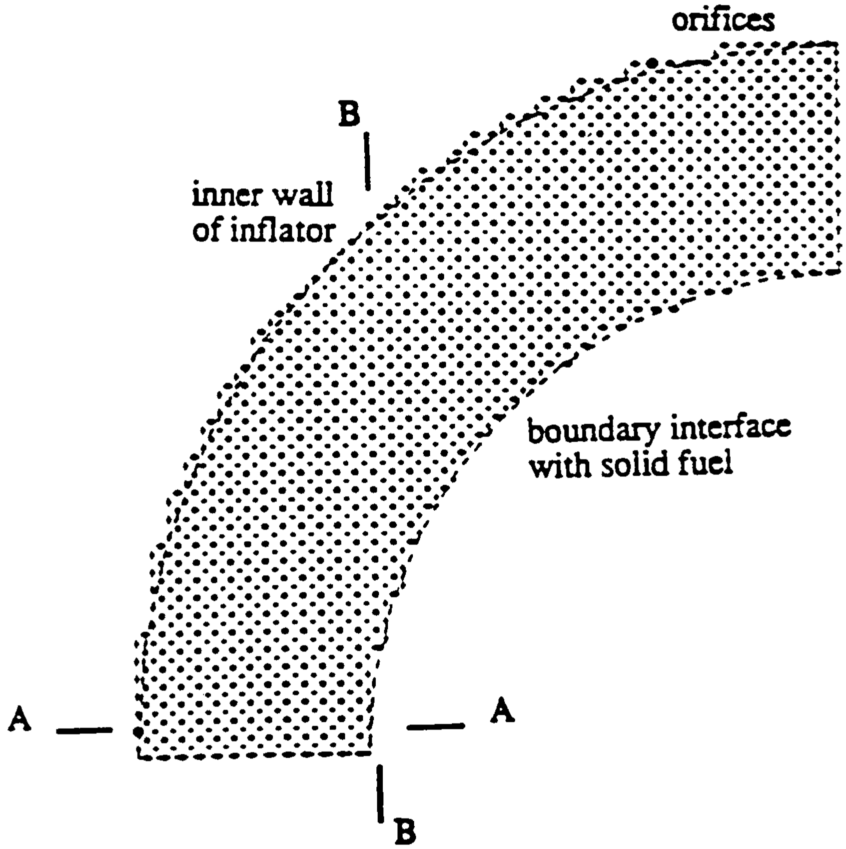


Fig. 8 Computational mesh in the porous metal screen

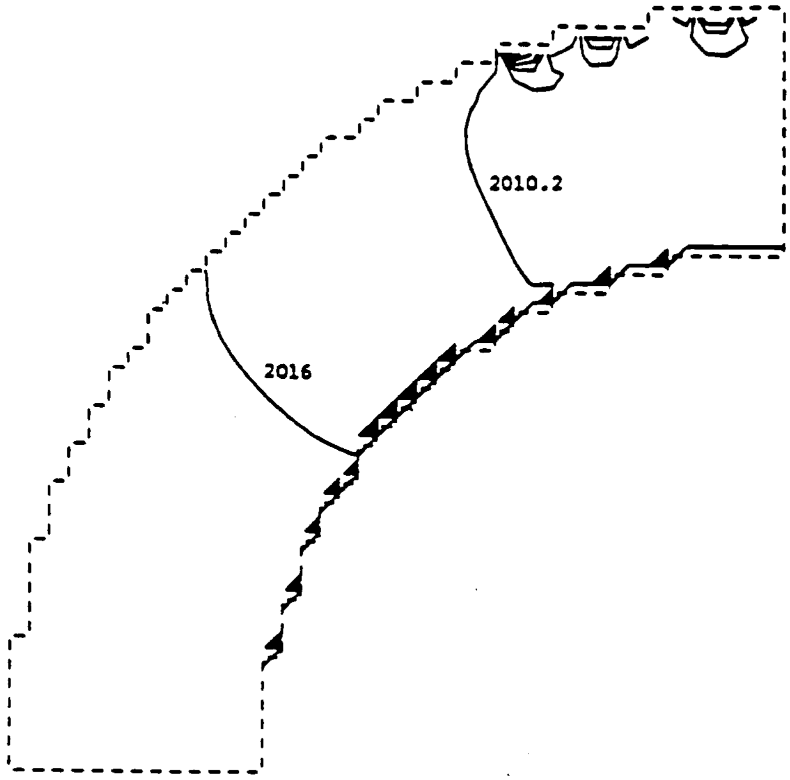


Fig. 9 Pressure contours in metal screen at t=23 ms, $\Delta p = 5.8$ psi

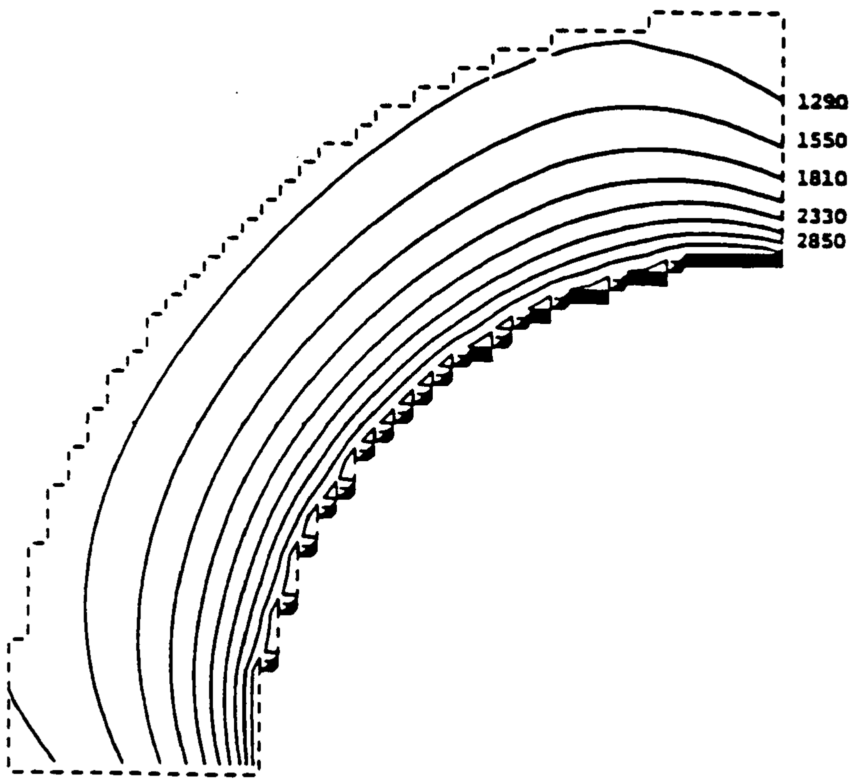


Fig. 10 Temperature contours in metal screen at t=23 ms, $\Delta T = 260°F$

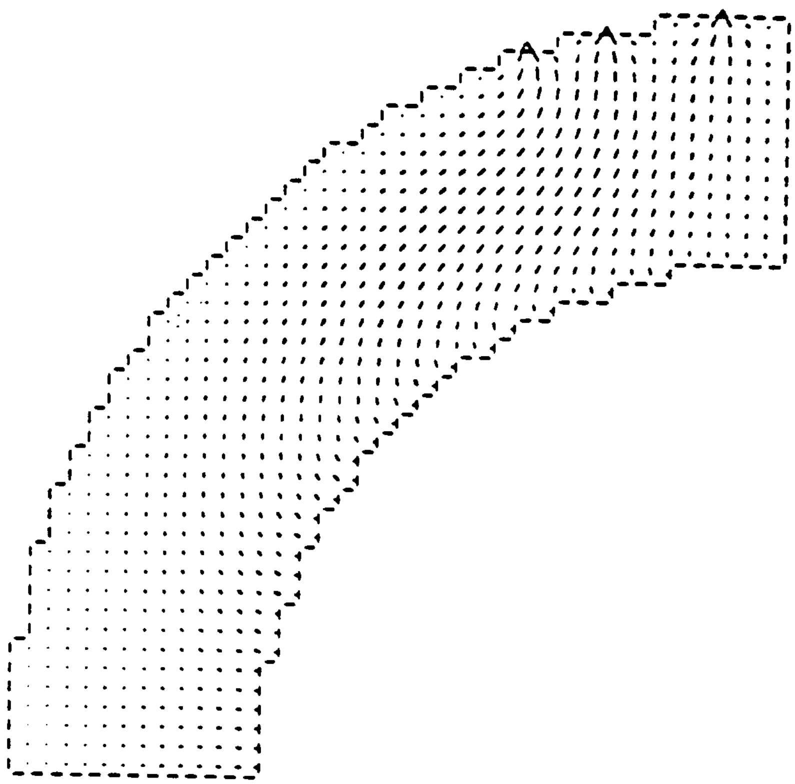


Fig. 11 Velocity vectors in metal screen at t=23 ms

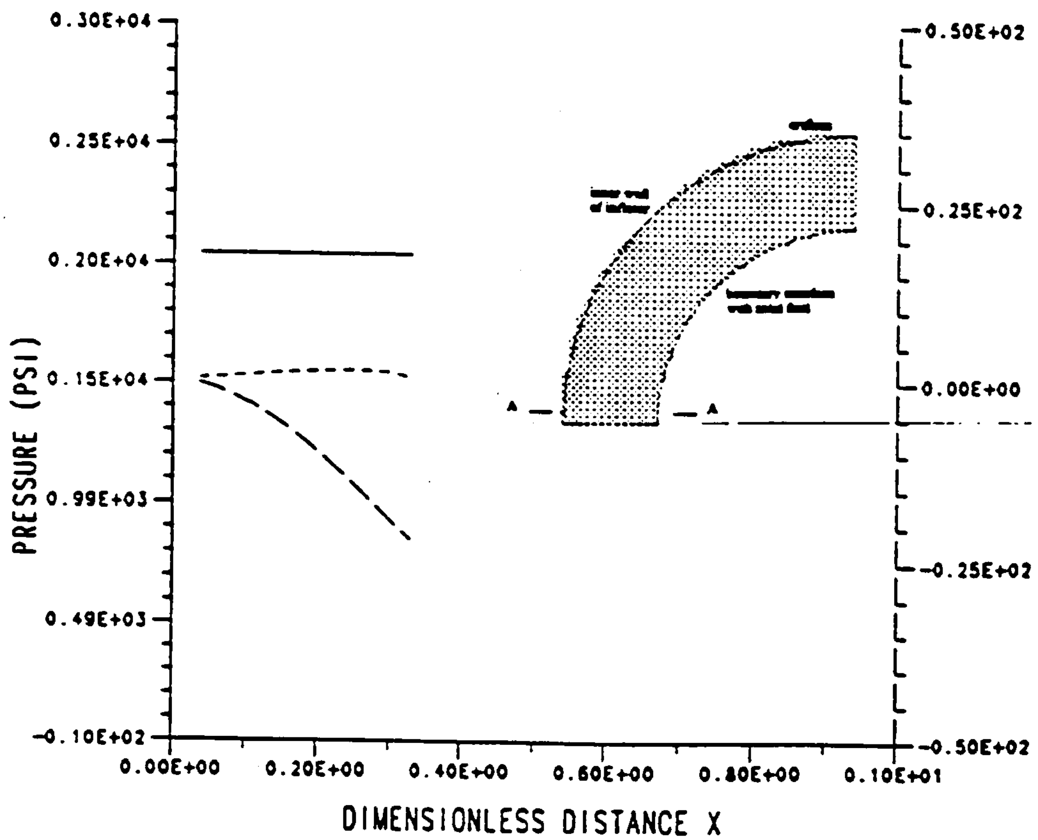


Fig. 12 Pressure and velocity along section A – A in metal screen at t=23 ms

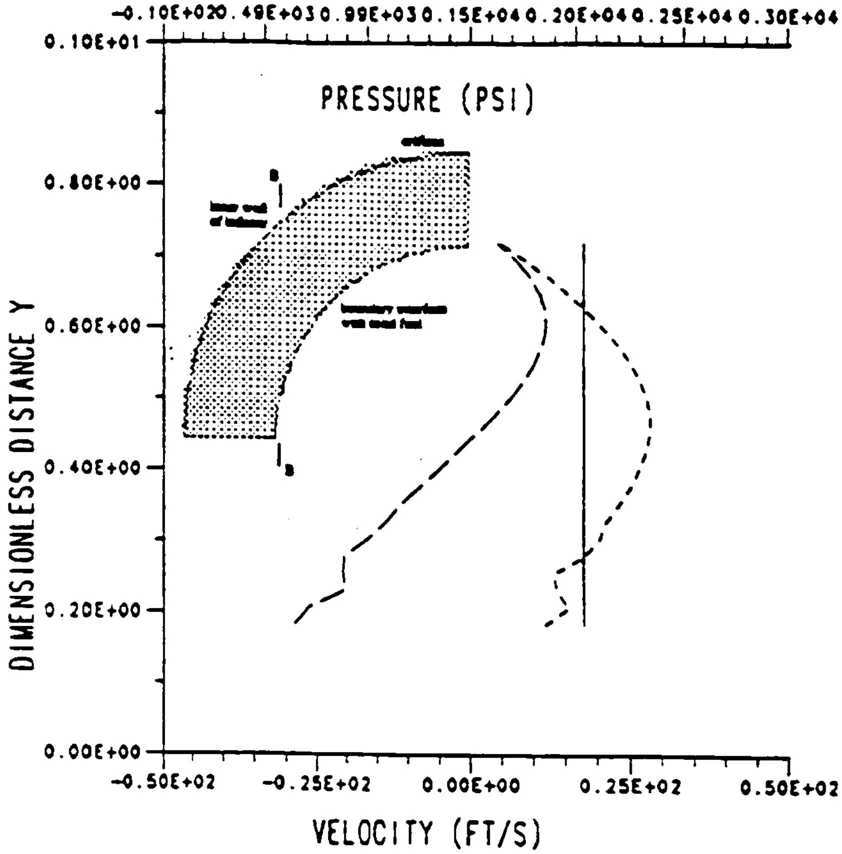


Fig. 13 Pressure and velocity along section B – B in metal screen at t=23 ms

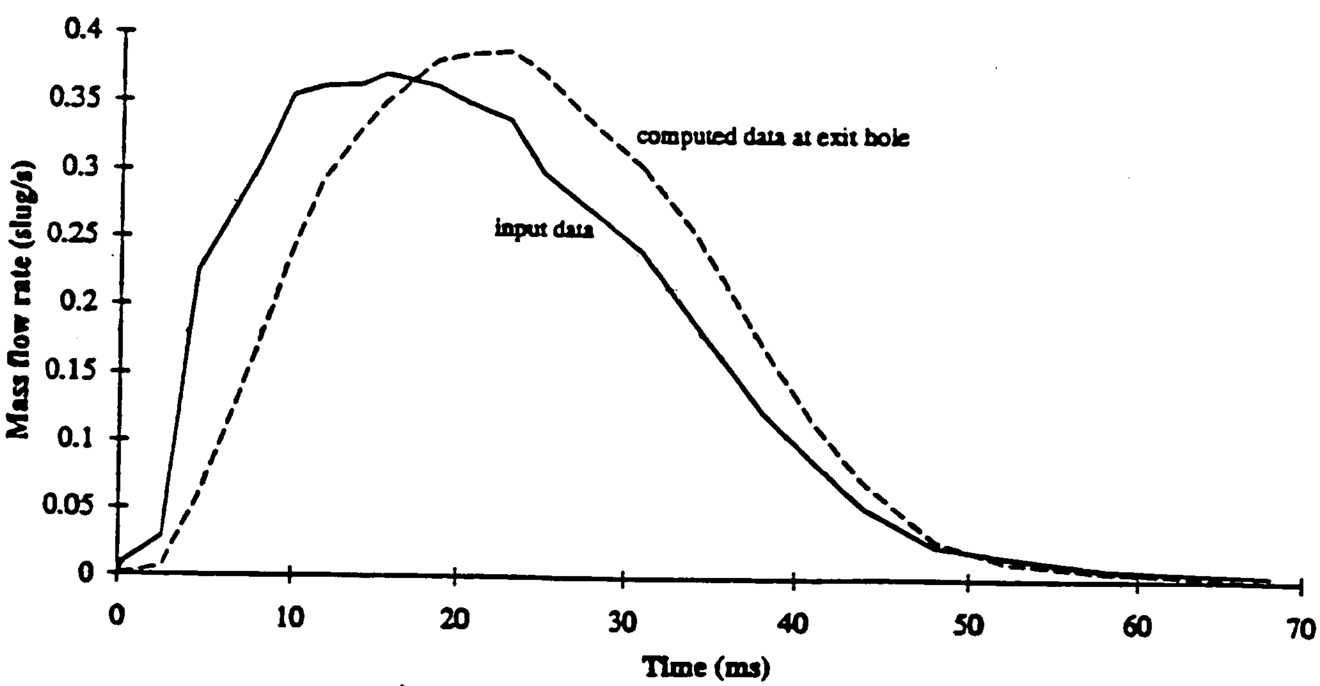


Fig. 14 Input mass flow rate and computed mass flow rate at exit hole

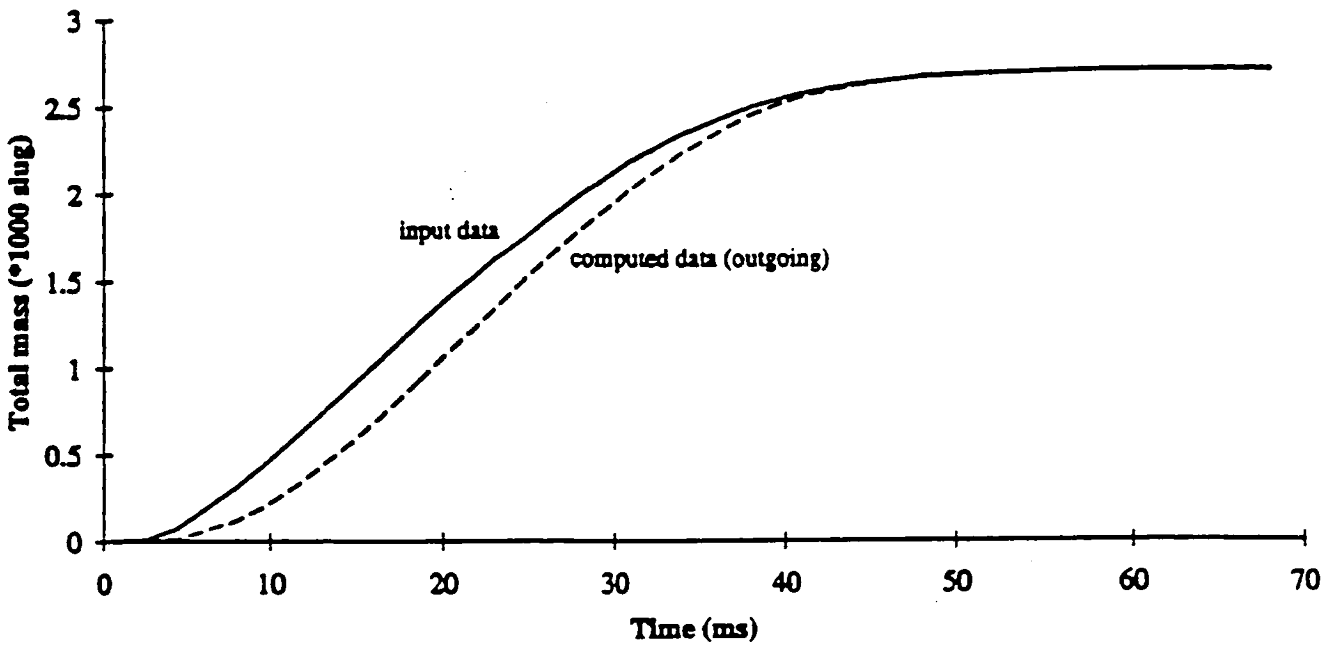


Fig. 15 Input total mass and computed total mass out of exit hole

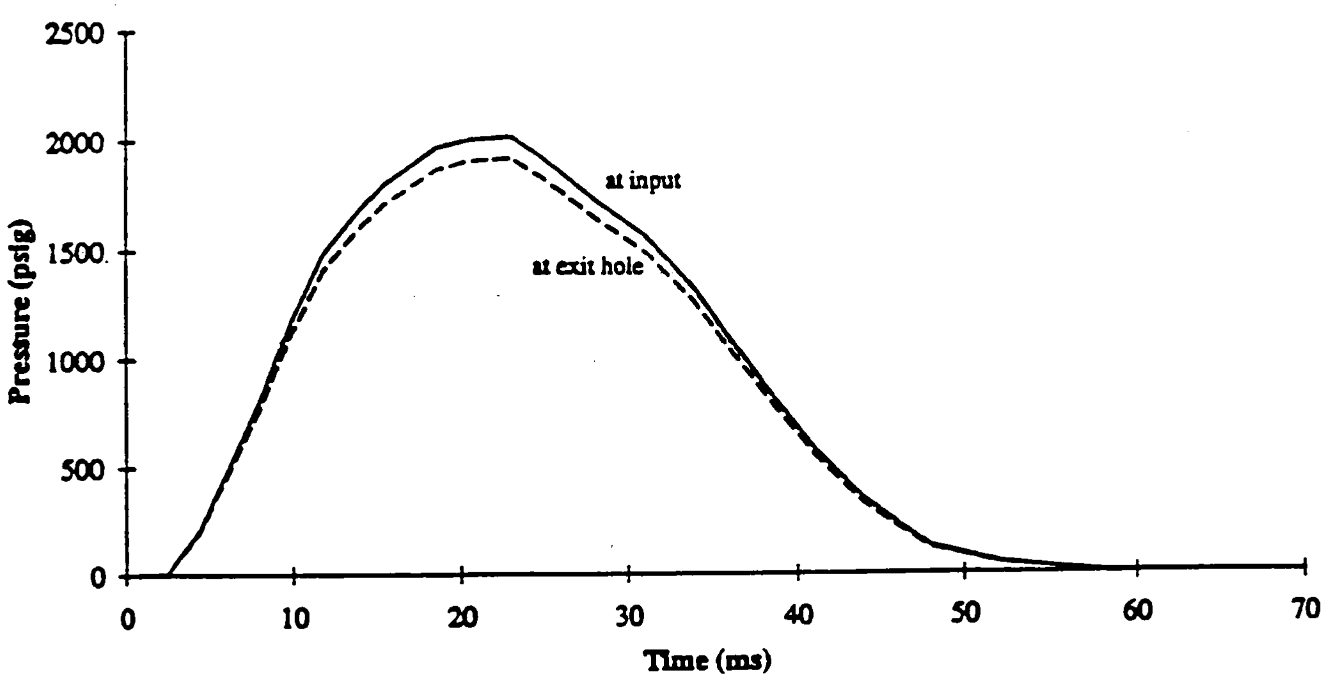


Fig. 16 Pressure time histories at input and at exit hole

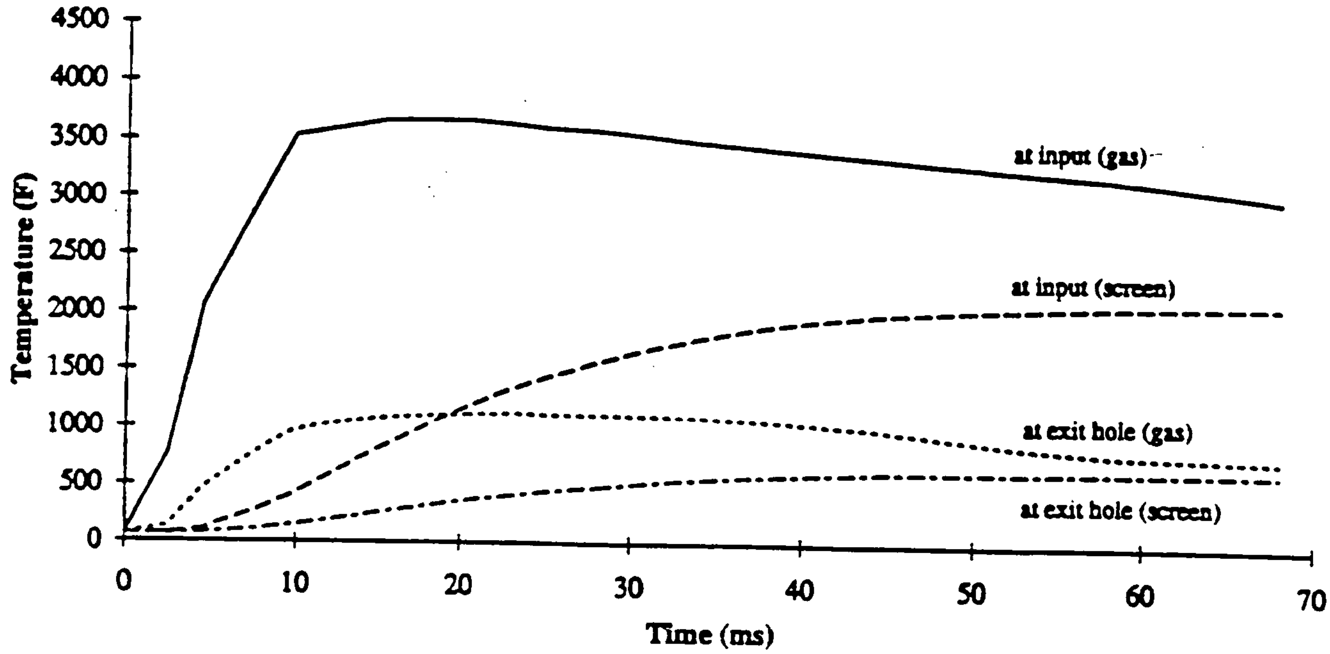


Fig. 17 Temperature time histories

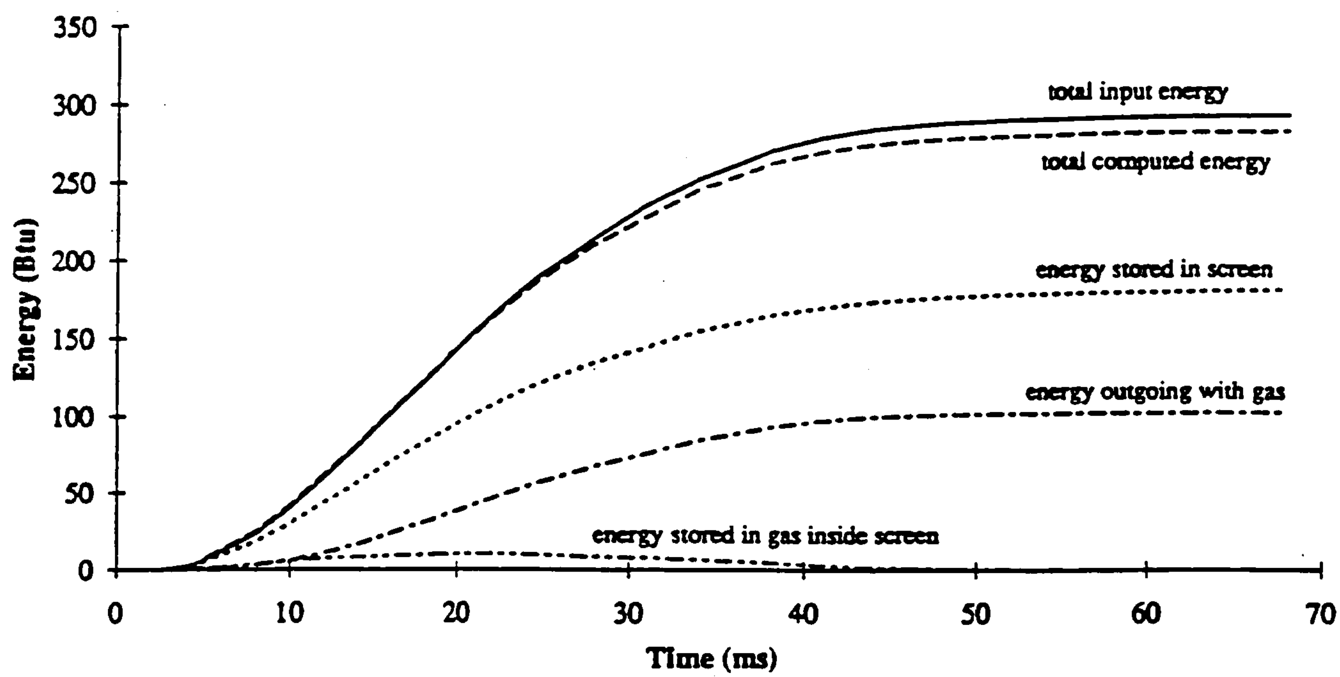


Fig. 18 Energy time histories

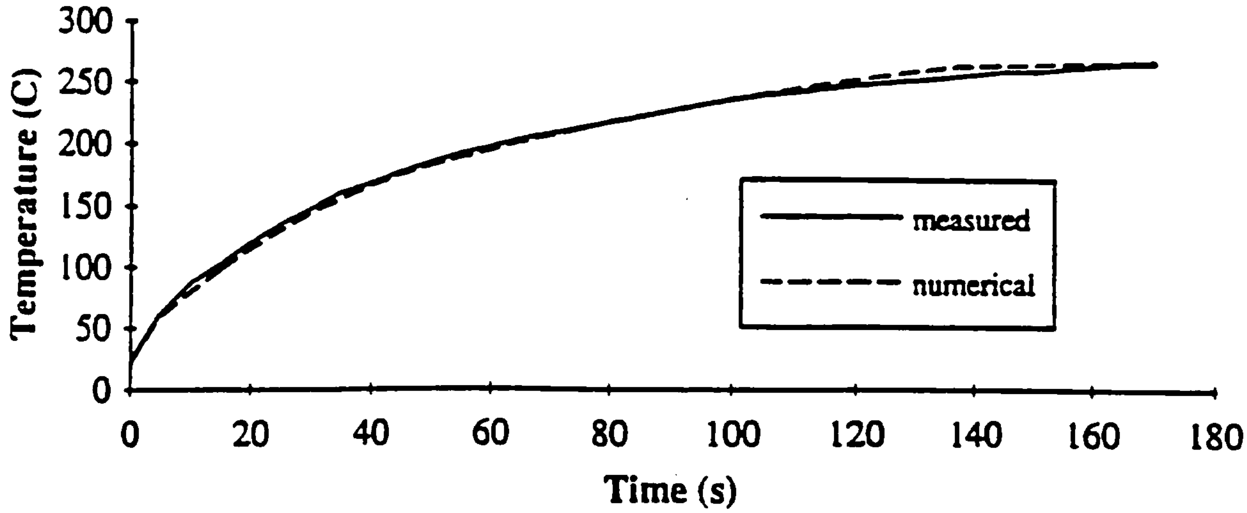
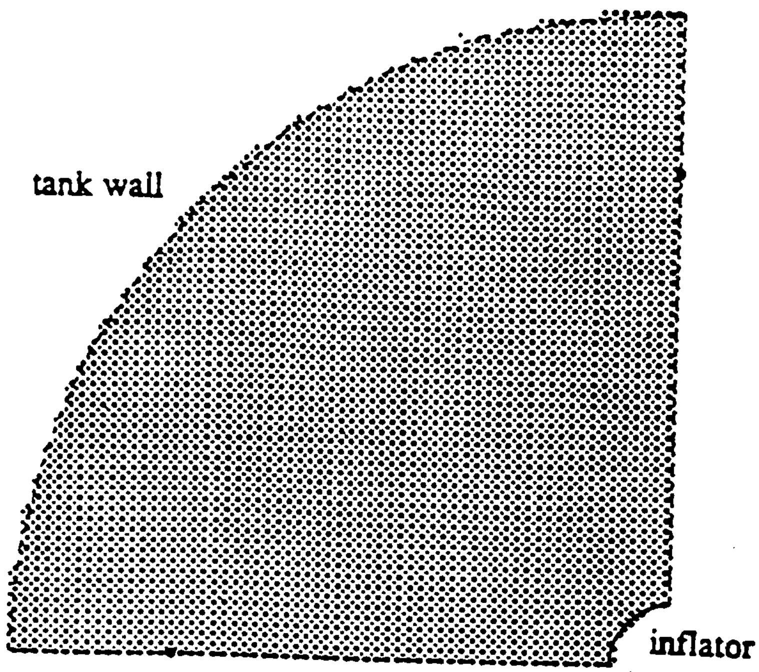


Fig. 19 Computed vs measured temperature on outer wall of inflator: one-D heat diffusion model



Computational mesh for tank test

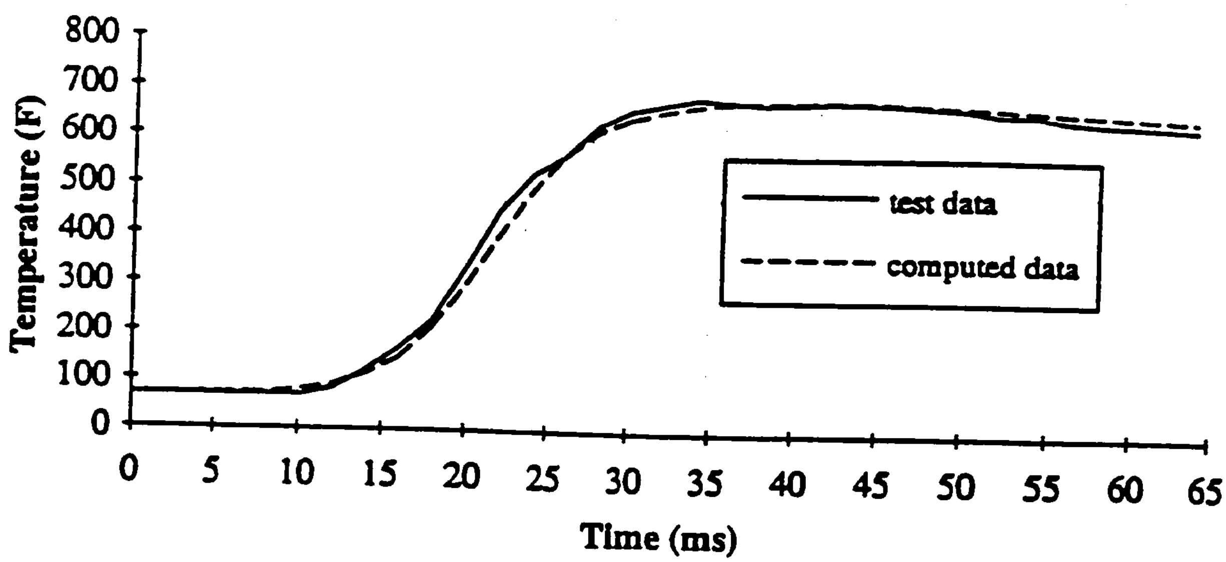


Fig. 20 Computed vs measured temperature in tank test

REFERENCES

Hoffman, E. J., "Heat Transfer Rate Analysis," PennWell Books, 1980.

Moody, F. J., "Introduction to Unsteady Thermofluid Mechanics," John Wiley & Sons, 1990, pp. 89 - 97.

Streeter, V. L., and Wylie, E. B., "Fluid Mechanics," McGraw-Hill, 1985, 250.

Suo, L. S., and Wylie, E. B., "Two-Dimensional Numerical Model of Airbag System," Report to Chrysler, #UMCEE 90-20, 1990, 69 pp.

Wylie, E. B., and Streeter, V. L., "Fluid Transients in Systems," Prentice Hall, New Jersey, 1993, pp. 410 - 413.

Wylie, E. B., and Wang, D. G., "Time-Varying Thermal Energy Model," Report to Chrysler, #UMCEE 95-3, 1995, 108 pp.

DISCUSSION

PAPER: **Energy Consideration In Passenger Air Bag Deployment**

PRESENTER: Guy Nusholtz, Chrysler Corporation

QUESTION: Tony Sances, Medical College of Wisconsin

You did a very nice job, Guy. I remember when I was in physics. I wrote the book "What Entropy Is Not." We are always interested in that. Could you perhaps expand a little bit upon the amount of force, the reaction force in the bag, between that and the occupant in the position and out of position situations?

ANSWER: That is actually an extremely complicated problem. We've done some work where we've cut the bag off and we've got fairly good estimates that the gas jet can put anywhere between 500 and 1,000 lbs. This is for a limited set of passenger inflators that we've tested.

I don't know what the full spectrum is but we started with a set of experimental passenger inflators. We've adjusted them a little bit and we're getting anywhere between 500 and 1,000 lbs. just from the gas jet without the bag. Now how much force you really put on the occupant on a typical deploying air bag, we haven't fully estimated but it's going to be somewhat above there. It may be as much as three times but it may only be twice that.

Q: And that's in quite a short time I presume, in the first twenty, thirty milliseconds.

A: It's all during the early deployment which is generally not much more than thirty milliseconds.

Q: Thanks.

Q: Frank Pintar, Medical College of Wisconsin

I have to ask Guy a question because he is always so nicely to ask everybody else questions. So, I come from a laboratory that does cadaver testing and we do air bag testing with cadavers. We don't have things like tank tests and we buy commercially available air bags and inflators. Do we have to worry about these fine differences and results in terms of tank tests?

A: Depends on what you're doing. If you are running experiments to get a perception with regard to the way the occupant is going to respond to the bag, you probably don't have to worry a lot about it. If you are trying to model precisely what's going on, you are going to need all that information.

Unless you know what the mass input and what the temperature is going into the bag and what the temperature relations are, it is hard to get a detailed estimate of what's happening. You can get some fairly good estimates by some simple empirical models.

Q: When you buy commercially available air bags for laboratory testing, are those tank test results? Do they have a significant deviation or are they pretty standardized?

A: That's also a very big question. Sometimes there are significant deviations in the air bags, particularly, when you purchase the experimental. What we've discovered is, we get a lot of maybe a hundred air bags and we end up having to burn about fifty of them to find out what the deviation is. If the deviation is tight enough, then we take it for granted that the other fifty are reasonably tight. If there is a large deviation, sometimes we can't run the experiments that we are trying to do.

Q: OK. Thank you.

Q: N. Yoganandan, Medical College of Wisconsin
Just to follow what Frank was asking, what would be your suggestion for human cadaver laboratory type of research, to use the kind of air bag? And where would you get the kind of information for the tank test because I have checked around and I cannot find it. Obviously, I'm not going to call your company up but where can I get this kind of information so that I can think of doing some empirical models.

A: Ask Harley. He'll tell you.

Q: OK.

A: It depends on which company made your air bag. Let's say, you went to TRW
Which is where Harley is from? Then you would contact them and try and get that information. That's normally the source that I go to or we run our own tank tests.

Q: Do you know of any other laboratories which do tank tests other than them?

A: I'm sure all the OEM's do tank tests and all of the air bag suppliers will be doing tank tests. You mean, do I know any commercial laboratories where you can take your inflators to and test them that way?

Q: Yes.

A: And the answer is no.

Q: OK. Thanks.