INTERIM REPORT

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INTERIM REPORT

Transient Critical Heat Flux

1. Future Tests in Critical Heat Flux (J. C. Leung and R.E.Henry)

Installation of Test Section IV has been completed. Test Section IV has a skew heat-flux profile, with ID and heated length identical to the previous test sections. At present it was mounted with the peaked heat flux near the bottom so as to simulate the beginning-of-life (BOL) of reactor fuel rod. Steady-state critical heat flux tests are continuing and blowdown tests are scheduled afterward. This section will be positioned with the peaking near the top, thus simulating the end-of-life (EOL) fuel, and similar tests will be conducted.

2. Analytic Support for Transient CHF (J.C.Leung and K. Gallivan)

During the transient, some knowledge of the thermal-hydraulic behavior of the system would be helpful in the interpretation of the experimental data. A simple one-dimensional transient coolant dynamics model which bears some similarity to SAS[1],[2] and BACTRAC[3] codes, is proposed. The model considers the single-phase liquid to be incompressible such that both the pressure and velocity disturbances are propagated at an infinite velocity. In the two-phase region, a homogeneous equilibrium mixture is assumed. The conservation equations of mass, momentum, and energy for a constant area round duct can be written as:[4]

$$\frac{\partial \rho}{\partial t} + \frac{\partial G}{\partial z} = 0 \tag{1}$$

$$\frac{\partial G}{\partial t} + \frac{\partial}{\partial z}\left(\frac{G^2}{\rho}\right) = -\frac{\partial P}{\partial z} - \frac{2f}{D}\,\frac{G|G|}{\rho} - \rho g \tag{2}$$

$$\frac{\partial E}{\partial t} + \frac{\partial}{\partial z}\left[\frac{G}{\rho}(E + P)\right] = \frac{4\phi}{D} \tag{3}$$

where ρ = density

G = mass velocity

f = friction factor

E = energy per unit volume,

ρe_o; e_o stagnation internal energy

ϕ = wall heat flux

D = diameter of channel

P = pressure

The four unknowns are ρ, G, E, and P; hence, one more equation is needed to close the system. This comes from the equation of state which can generally be represented by

$$P = P(\rho, E) \tag{4}$$

The initial steady-state conditions can easily be established by setting the time derivative terms to zero in Eqs. 1-3 and solving for $\rho(z,0)$, $G(z,0)$, $E(z,0)$, and $P(z,0)$.

The most rigorous solution to Eqs. 1-4 would be given by the "sectionalized compressible model" as described by Meyer.[4] However, a numerical solution using this method requires a knowledge of the acoustic velocity in both single- and two-phase fluid. The use of the equation of state in the indicated form, Eq. 4, causes accuracy problems in the compressed liquid region as experienced in RELAP3.[5] The difficulty was illustrated in Ref. 6 by noting that at a temperature of 340°F, the density of water at 800 psia is 56.13 lbm/ft^3 and at 900 psia it is 56.16 lbm/ft^3. Thus, large oscillations can be set up in the pressure with a small error in the density. Furthermore any explicit scheme of solving the conservation equations would be limited by very small time step in the liquid region. Hence, an incompressible treatment of the liquid slug was sought. Meyer suggested the "momentum integral" form which was adopted in SAS code.[1,2] Basically the momentum equation was integrated over the whole liquid slug region which extends from z_1 to z_2,

$$\int_{z_1}^{z_2} \frac{\partial G}{\partial t}\, dz + \left(\frac{G_2^2}{\rho_2}\right) - \left(\frac{G_1^2}{\rho_1}\right) = P_1 - P_2 - \int_{z_1}^{z_2} \frac{2f}{D} \frac{G|G|}{\rho}\, dz$$

$$- \int_{z_1}^{z_2} \rho g dz \qquad (5)$$

The pressures P_1 and P_2 are determined either by the lower and upper plenum pressures or by the interface pr ssures in the two-phase mixture below and above the liquid. Further computational simplification can be obtained if the effect of fluid expansion is neglected. That is, if $\partial\rho/\partial t$ is small, then by Eq. 1, $\partial G/\partial z \sim 0$; a uniform mass velocity can be assumed in the liquid slug. This approximation is not equivalent to the assumption of constant liquid density as in BACTRAC code since variations of density with axial location are included. This assumption implies that the mass velocity for the liquid slug can be determined from the solution of one equation

$$\frac{dG}{dt} = \frac{1}{L} \left(\Delta P - F\right) \qquad (6)$$

where

$$L = z_2 - z_1$$

$$\Delta P = P_1 - P_2$$

and F has been used to represent the total resistance to fluid flow

$$F = G^2 \left(\frac{1}{\rho_2} - \frac{1}{\rho_1}\right) + \frac{2f}{D} G|G| \int_{z_1}^{z_2} \frac{dz}{\rho} + g \int_{z_1}^{z_2} \rho dz \qquad (7)$$

A pointwise difference approximation of the energy equation is employed to obtain local changes in internal energy, e, and the local density can be found from the simplified equation of state

$$\rho = \rho(e) \tag{8}$$

In the two-phase region, compressibility effect is deemed important and the well-known two-step Lax-Wendroff[7] scheme is used for the solution procedure. For the purpose of formulating the numerical scheme, the governing differential equations are written in the following matrix form:[8]

$$\frac{\partial \vec{U}}{\partial t} + \frac{\partial}{\partial z} \vec{F}(\vec{U}) = \vec{S}(\vec{U}) \tag{9}$$

where

$$\vec{U} = \begin{bmatrix} \rho \\ G \\ E \end{bmatrix}$$

$$\vec{F} = \begin{bmatrix} G \\ (G^2/\rho) + P \\ (E + P)G/\rho \end{bmatrix}$$

$$\vec{S} = \begin{bmatrix} 0 \\ -\rho g - \frac{2f}{D}\frac{G|G|}{\rho} \\ 4\phi/D \end{bmatrix}$$

First, provisional values are calculated at th. centers of the rectangular meshes of the net in the Z,t - plane:

$$U_{j+1/2}^{n+1/2} = \frac{1}{2}\left(U_{j+1}^{n} + U_{j}^{n}\right) - \frac{\Delta t}{2\Delta z}\left(F_{j+1}^{n} - F_{j}^{n}\right) + \frac{\Delta t}{4}\left(S_{j+1}^{n} + S_{j}^{n}\right) \tag{10}$$

Then the flux vector, $\vec{F}$, and the source vector, $\vec{S}$, are determined at these half-step points with the use of equation of state. The final values are then obtained from the equation

$$U_j^{n+1} = U_j^n - \frac{\Delta t}{\Delta z}\left(F_{j+1/2}^{n+1/2} - F_{j-1/2}^{n+1/2}\right) + \frac{\Delta t}{2}\left(S_{j+1/2}^{n+1/2} + S_{j-1/2}^{n+1/2}\right) \quad (11)$$

This scheme is dictated in the numerical stability by the Courant-Friedricks-Lewy criterion,

$$\Delta t \leq \frac{\Delta z}{C + |u_{max}|} \quad (12)$$

where C and u_{max} are the acoustic velocity and the maximum fluid velocity.

The local fluid conditions computations will be coupled with a wall heat-transfer model by means of heat transfer coefficient relationship. At each axial node, a one-dimensional transient heat conduction in the heated wall is related to the fluid bulk temperature by the following boundary conditions at the inside surface. Prior to CHF a single-phase convection boundary condition will be used at those surfaces which are below system saturation temperature during initial steady-state operation. For those surfaces which are above the saturation temperature initially (i.e., either in subcooled or saturated nucleate boiling), a surface temperature superheat similar to Jens-Lottes equation will be prescribed. Beyond CHF, a post-CHF heat transfer coefficient will be used, e.g., one can use Dittus-Boelter coefficient based on saturated vapor properties. The CISE Freon CHF correlation[9] is used to predict CHF condition in the channel. It has been tested against both uniform and nonuniform CHF data taken in the present Freon apparatus and yielded very good agreement over a wide range of pressures.

Due to the thin-wall test section, a one-dimensional heat conduction equation in Cartesian coordinate is deemed adequate:

$$\frac{\partial^2 T}{\partial x^2} + \frac{q'''}{k} = \frac{1}{\alpha} \frac{\partial T}{\partial t} \tag{13}$$

Where q''', k, and α are the volumetric heat generation rate, thermal conductivity, and thermal diffusivity of the wall material. The Crank-Nicolson[10] implicit solution scheme has been employed and results have been tested for various boundary conditions against analytical solutions with good agreement. The use of implicit scheme is to avoid complication of the choice of time step for the overall scheme further by restrictions in the heat conduction model.

Currently, effort is concentrated on the liquid-slug flow reversal problem with pressure-forced calculation.

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Reflood Tests (Y. S. Cha, R. E. Henry, P. A. Lottes)

The reflood test facility has been modified. A schematic of the new reflood test facility is shown in Fig. 1. The arrangement shown in Fig. 1 differs from previous arrangement in the following two respects: (1) Two ball valves driven by a variable speed motor are used to control the flow to the test section. This is different from previous arrangement where a three-way solenoidal valve and an oscillator were used. (2) An orifice plate is placed between the test section and the ball valve system so that both the forward and the reverse flow to the test section are measured by the same orifice plate. This is different from the previous arrangement where two orifice plates were used to measure the forward and the reverse flow separately.

Preliminary tests were performed to see how the ball valve system behaves under oscillatory conditions. These tests were performed under cold (i.e., without heating up the test section) conditions. Figures 2 to 4 show the flow profiles at three different frequencies of oscillation. The dashed lines shown in these figures correspond to zero flow condition which has an output of 5 volts from the differential pressure transducer.

The orifice plate was calibrated under steady state conditions in both the forward and the reverse flow directions. It is important to verify that the steady state calibration data can be applied to the oscillatory (transient) conditions in the range of interest of the present reflood tests. To demonstrate this, cold tests were performed by closing the throttle valve in the reverse flow line so that there was only flow in the forward direction during the entire test. Water leaving the test section was collected in the recirculation tank. The increase in the amount of water in the recirculation tank divided by the total number of cycles of oscillation during the test gives the amount of flow to the test section per cycle. This number can be used to check the results obtained

by graphically integrating the flow profile using the steady state calibration data. The results are shown in Table I, where f is the frequency of oscillation, V_m is the measured increase in the amount of water in the recirculation tank per

TABLE I - Comparison between V_m and V_c at three different frequencies of oscillation

F (Hz)	V_m (cm^3/cycle)	V_c (cm^3/cycle)	% difference	$\frac{1}{\Delta t \cdot f}$
2.857	1.149	1.201	4.54	28
0.990	3.423	3.317	3.22	20
0.294	11.152	11.238	0.77	34

cycle, V_c is the calculated volume flow into the test section per cycle obtained by integrating the flow profile and using the steady state calibration data, and Δt is the time interval employed in performing the integration. The last column in Table I indicates the total number of intervals employed in performing the integration during one cycle. As shown in Table I, the maximum error introduced by using the steady state calibration data is less than 5 per cent. This error can be reduced by increasing the number of intervals employed in the integration of the flow profile. The results shown in Table I also indicate there is a slight increase in percentage error as the frequency of oscillation is increased. However, in the range of interest of the present reflood tests (f = 0.3 to 3.0 Hz), the errors introduced by using the steady state calibration data are not considered to be significant.

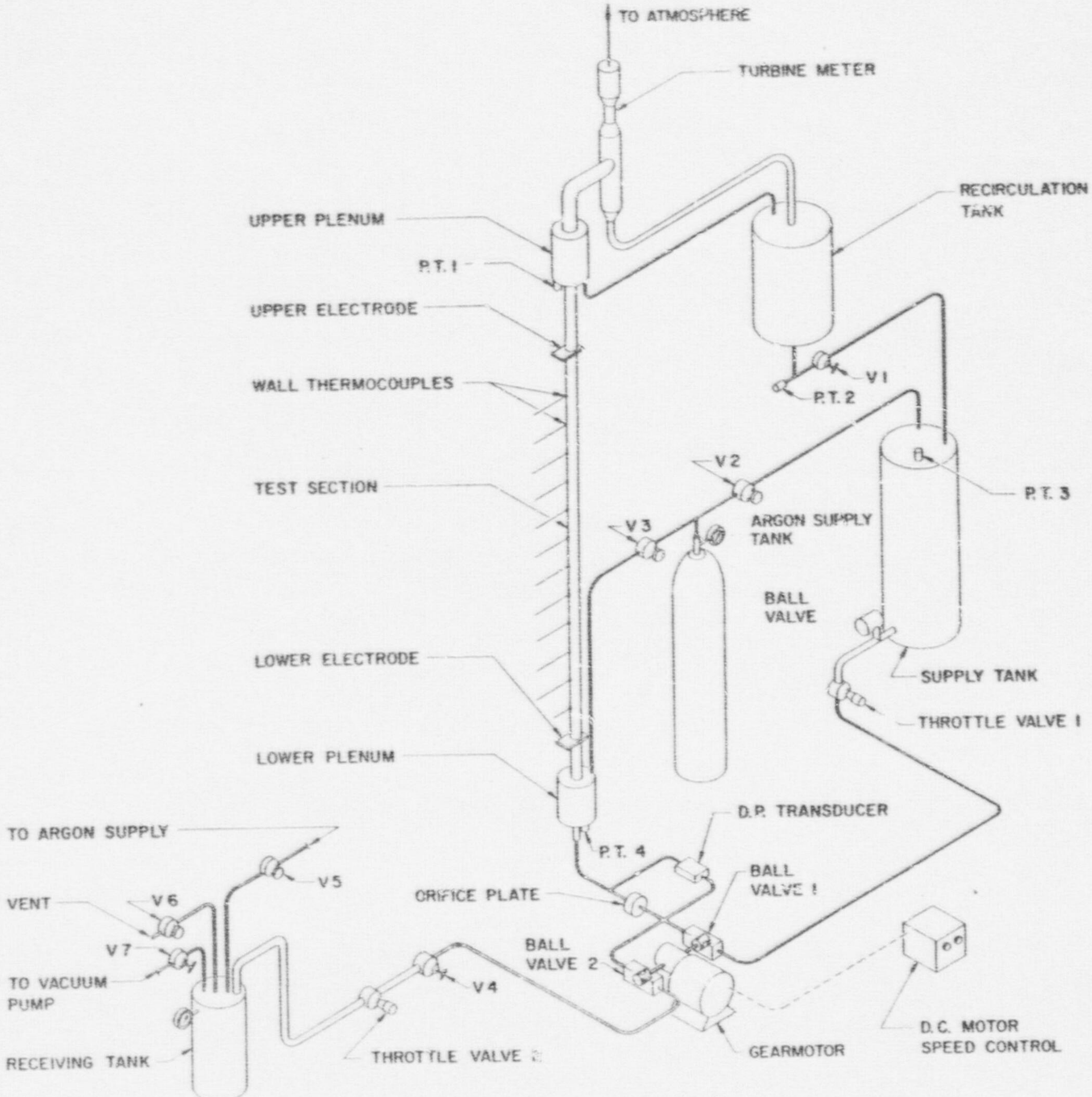

FIG. 1 Schematic of the modified reflood test facility

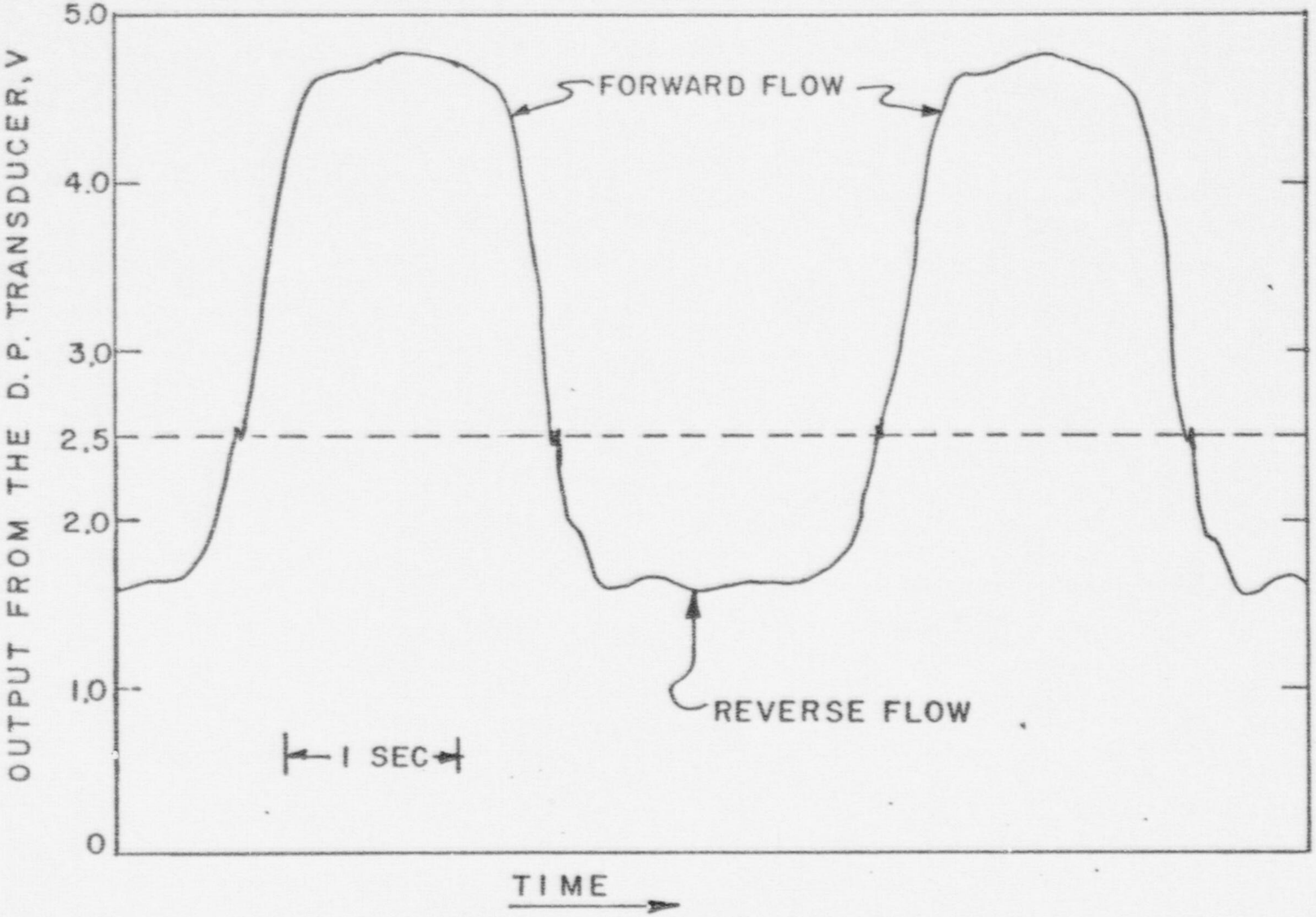

FIG. 2 Output from the differential pressure transducer versus time for test with f = 0.274 Hz.

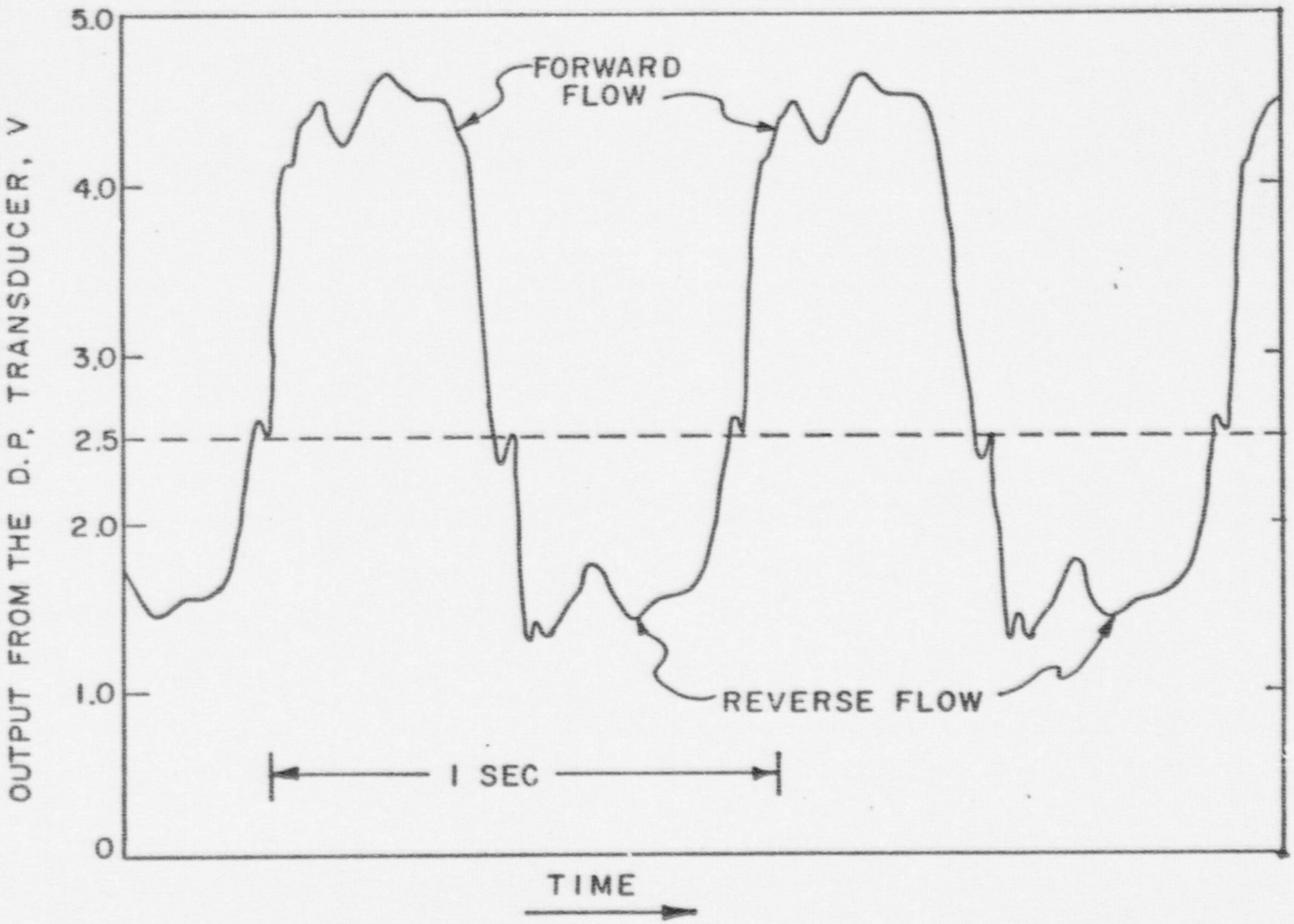

FIG. 3 Output from the differential pressure transducer versus time for test with f = 1.034 Hz.

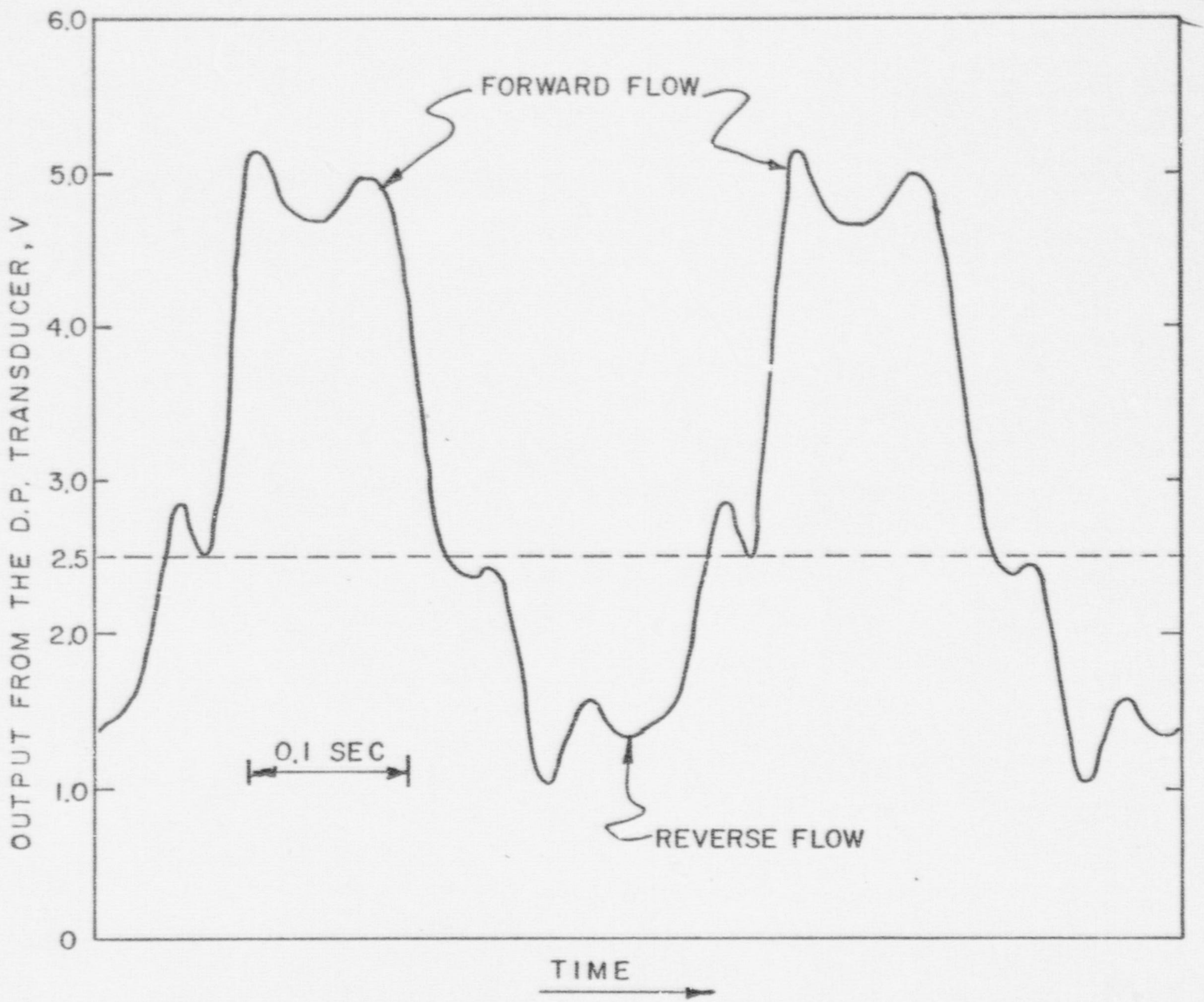

FIG. 4 Output from the differential pressure transducer versus time for test with f = 2.857 Hz.