ACCURACY OF THE FIREBIRD-III COMPUTER

CODE PUMP MODEL

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by

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ABSTRACT

The PUMPANC subroutine in the FIREBIRD-III thermalhydraulic computer code models pump behaviour. The accuracy of this model is examined on the basis of available pump tests.

Single phase test results on the Gentilly-2 primary heat transport pumps are compared to FIREBIRD-III predictions. As well, various two-phase pump tests are examined for the applicability of scaling small pump results to large pump results. Further areas of investigation are outlined.

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The author would like to thank Atomic Energy of Canada Limited for the opportunity to do this work. Particular thanks are given to D.F. Rennick and N.J. Spinks for their supervision.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

TABLES

1.0 INTRODUCTION

The FIREBIRD-III computer code predicts reactor behaviour during postulated loss-of-coolant accidents (LOCA). Of primary importance is the maximum fuel sheath temperature as this governs whether fuel sheath failure will occur. The fuel sheath temperature is very sensitive to the pump behaviour as the pump head governs the amount of flow and therefore the amount of heat carried away from the fuel rods. The sensitivity to the effects of the pumps is reduced by searching for the worst break size leading to the worst sheath temperature transient, but is also important for a wider range of smaller breaks.

During a LOCA the pressure in the cooling system will drop causing the heavy water to vaporize. Therefore the pump model must be able to predict pump behaviour for steam-water flows as well as single-phase flows.

Although not all pumps used in CANDU reactors are the same, the G2 primary heat transport pumps that are the main focus of this report are typical. The pump is a vertically mounted, single suction, double discharge centrifugal pump. The impeller has a diameter of approximately 2.44 feet and is driven by a 6.74 MW moter (rated) of which 4.78 MW goes to drive the impeller under normal operation. The pump will deliver 29,400 IGPM at a head of 705 feet when the impeller speed is 1800 revolutions per minute.

The PUMPANC single-phase pump model is tested by comparing a computer simulation of the rundown of the Gentilly-2 PHT pumps 1 and 2, with experimental results. Three different runs are made with each pump: one with rated head and flow set at initial experimental head and flow, and one with the impeller speed determined by the pump model; and another with the rated head and flow set equal to design rated head and flow, and with the impeller speed determined by the pump model. The last case is similar to the first except the impeller speed is fixed to follow experimental results. As well a program using the PUMPANC pump model and utilizing the experimental speed and flow determines pump head and compares it with the experimental head. This is done for all four pumps tested.

Further verification of the PUMPANC single phase model is obtained by comparing forward flow resistance of the pump at zero speed with that obtained from PUMPANC. A further analysis of locked impeller results in single and two-phase flow is done to test the assumption that fully degraded locked impeller resistance is the same as the single-phase locked impeller resistance currently in PUMPANC.

As mentioned previously, the pump model must predict behaviour in two-phase as well as single-phase flow. Full size reactor pumps are too expensive to risk two-phase testing, so tests are done on smaller pumps, then applied to the full size pumps. Scaling of single-phase results is well established, but criteria for similarity of results in two-phase flow are as yet unknown.

Two pumps of the same specific speed but different sizes were tested by EPRI[2,3]. Their homologous head and torque curves are compared for both single phase and a void of 40%. A comparison was also made in the single phase with the ANC MOD-1 Semiscale pump[4] which has a different specific speed.

Degradation curves for one of the EPRI pumps are examined to see how closely they match the curves in PUMPANC. These curves are also examined for further improvements in the pump model.

2.0 THEORY

2.1 Single Phase Operation

Scaling laws which predict the performance of large pumps based on the behaviour of small pumps are fairly well established if single-phase flow persists. By applying the principles of dimensional analysis a general function defining the relationship between important parameters can be found.

$$f\left[\frac{Q}{(gH)^{1/2}D^2}, \frac{ND}{(gH)^{1/2}}, \frac{\mu}{\rho D(gH)^{1/2}}\right] = 0 \qquad (1)$$

where,

H = head (length)
Q = capacity (volume/time)
N = speed (revolutions per minute)
D = impeller diameter
ρ = liquid density
μ = absolute liquid viscosity
g = acceleration due to gravity

Dimensional analysis states that the three groups will remain constant for similarly designed impellers and similar flow conditions irrespective of the speed or size of the impellers. If these three conditions are satisfied then the functional relationship between them remains the same. This relationship, however, can only be determined experimentally.

For a small test pump operating at a constant speed, D, ρ, N and μ are all constant and the relationship reduces to $H = f'(Q)$, the head-capacity curve.

It is common to write the functional relationship in terms of the maximum efficiency operating conditions, Q_{TO}, H_{TO} where the subscript T refers to the test pump. This gives

$$f_T \left[\frac{Q_{TO}}{(gH_{TO})^{1/2} D_T^2} , \frac{N_T D_T}{(gH_{TO})^{1/2}} , \frac{\mu T}{\rho_T D_T (gH_{TO})^{1/2}}\right] = 0 \tag{2}$$

A larger pump, such as a reactor pump, will have the same functional relationship provided the scaling laws are satisfied. Suppose the reactor pump is required to deliver a head, H_{RO}, at capacity Q_{RO} for a fluid of properties ρ_R, μ_R. The dimensional analysis requires that:

$$\frac{Q_{TO}}{(gH_{TO})^{1/2} D_{T2}} = \frac{Q_{RO}}{(gH_{RO})^{1/2} D_R^2} = \pi_1 \tag{3}$$

$$\frac{N_T D_T}{(gH_{TO})^{1/2}} = \frac{N_R D_R}{(gH_{RO})^{1/2}} = \pi_2 \tag{4}$$

$$\frac{\mu_T}{\rho_T D_T (gH_{TO})^{1/2}} = \frac{\mu_R}{\rho_R D_R (gH_{RO})^{1/2}} = \pi_3 \tag{5}$$

Squaring the second relationship and multiplying by the first yields:

$$\frac{Q_{TO} N_T^2}{(gH_{TO})^{1/2}} = \frac{Q_{RO} N_R^2}{(gH_{RO})^{1/2}} \tag{6}$$

or

$$\frac{N_T Q_{TO}^{1/2}}{(H_{TO})^{3/4}} = \frac{N_R Q_{RO}^{1/2}}{H_{RO}^{1/2}} \tag{7}$$

The expressions in equation 7 are known as the specific speeds of the pumps. Equation 7 requires that the test pump and the reactor pump must have the same specific speed in order for the head-capacity curve to be the same. This is true only for single phase.

Also, equation (4) shows that

$$H_{RO} \propto N_R^2 \; D_R^2 \tag{8}$$

and (3) and (4) combined give

$$Q_{RO} \propto N_R D_R^3 \tag{9}$$

These are the affinity laws which relate head and capacity to the size and speed of the pump.

The third scaling constant requires that

$$H_{RO} \propto \frac{\mu_R^2}{\rho_R^2} \; \frac{1}{D_R^2} \tag{10}$$

This is in direct conflict with the first affinity law which states that the head increase as the diameter increases. However, hydraulic losses are often small (on the order of 5%) so this scaling law can often be ignored.

A similar analysis shows that the torque follows the same affinity law as the head.

The affinity laws are used in analysis to normalize head-capacity curves for different impeller speeds. An example of this is given in figure 1, taken from the MOD-1 semiscale pump report[4]. This shows that the affinity relationships are followed quite well.

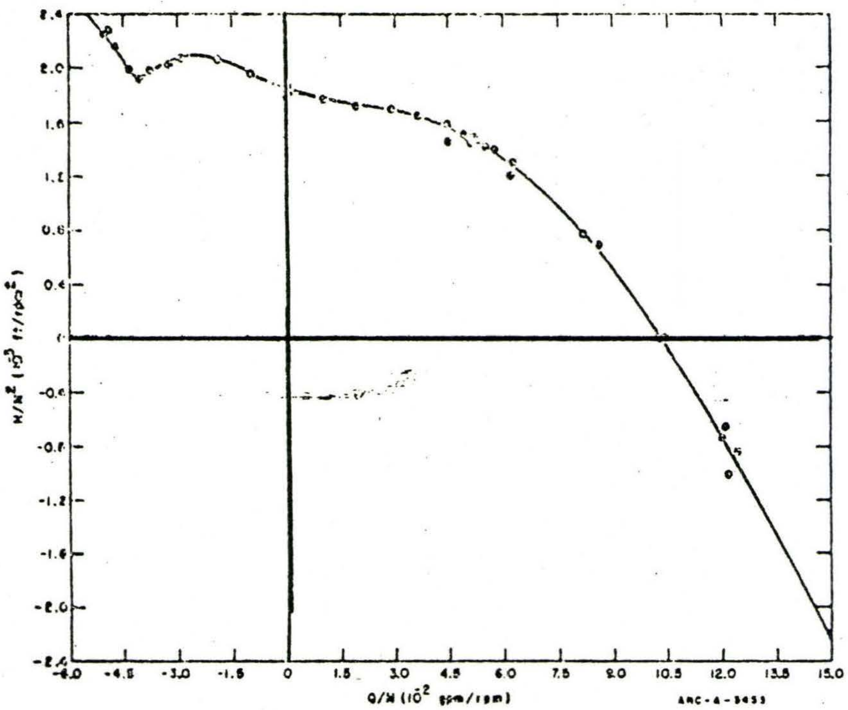


FIGURE 1: Typical Head-Capacity Curve

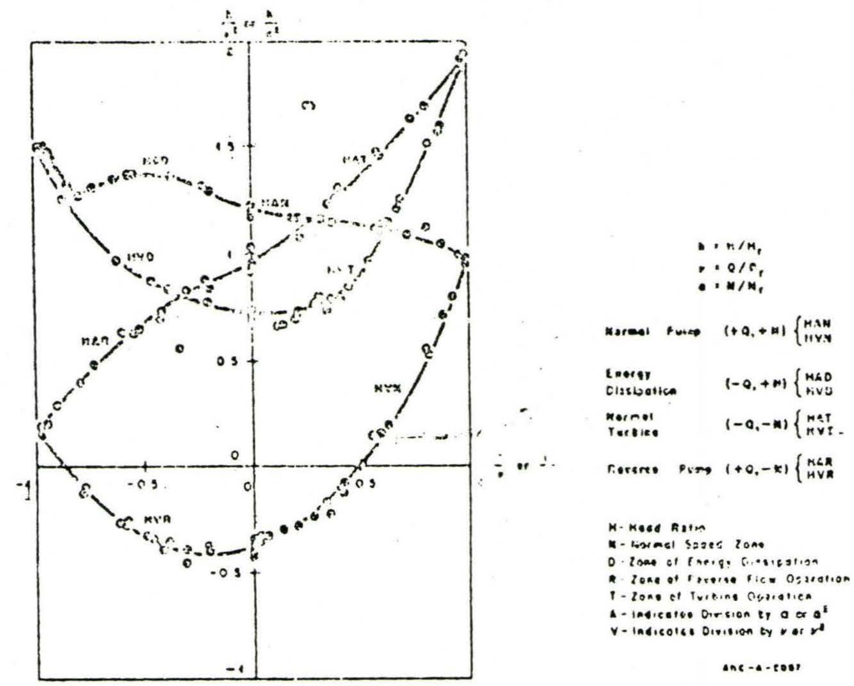


FIGURE 2: Homogous Head Curve

For computer analysis, the pump performance is characterized by homologous curves because of the ease of tabulating the input data. These curves are dimensionless extensions of the four quadrant pump characteristic curves in which the parameters are made dimensionless by dividing them by the rated (point of maximum efficiency) parameter. The homologous curves are actually two sets of four dimensionless curve segments that are plotted on the same set of coordinates. In one set, the dimensionless head ratio ($h = H/H_R$) or torque ratio ($\beta = T/T_r$) divided by the square of the dimensionless speed ($\Omega^2 = (N/N_R)^2$) is plotted as a function of dimensionless flow ($\nu = Q/Q_R$) divided by Ω, that is h/Ω^2 (or β/Ω^2) is a function of ν/Ω. In the second set h/ν^2 (or β/ν^2) is a function of Ω/ν.

An example of such a homologous head curve is shown in Figure 2 for the MOD-1 Semiscale pump[4]. A three letter acronym is used to designate the curve type. H refers to head ratio, A indicates division by Ω or Ω^2 and V indicates division by ν or ν^2. The last letter denotes the zone of pump operation. N is normal (forward flow, forward speed), R is the reverse flow zone (forward flow, reverse speed) and T is the zone of turbine operation (reverse flow, reverse speed).

2.2 Two-Phase Operation

The functional relationship must now include void giving,

$$f\left[\frac{\overline{Q}}{(gH)^{1/2}D^2}, \frac{ND}{(gH)^{1/2}}, \frac{\mu(\overline{\alpha}, \overline{Q})}{\overline{\rho}\, D(gH)^{1/2}}, \overline{\alpha}\right] = 0 \qquad (11)$$

where $\overline{Q}$, $\overline{\rho}$, and $\overline{\alpha}$ refer to the average flow conditions. This can be treated similarly to single-phase flow except that it can no longer be assumed that hydraulic losses are negligible. However, this term is commonly neglected in treating two-phase flow through pumps until its exact effect can be found. Thus the major criteria for scaling small pump results is still that of similar specific speed.

As the fluid properties (mainly density and void) will vary depending on whether they are taken at the suction or discharge, there is some ambiguity as to which void fraction should be used to correlate two-phase data. Inlet conditions (suction for forward flow and discharge for reverse flow) are commonly used. As a result there are two parameters used to get head, volumetric flow and void fraction. No account is made of the flow regime as its effect is not known.

The correlation of change in pump head with void at constant volumetric flow is called a degradation curve. A typical degradation curve is shown in Figure 3. At low voids the head remains unchanged from the single phase head. This is due to the pump being able to collapse the void so single-phase flow is passing through the pump. The head degradation increases rapidly between about 10% to 20% void after which it reaches a fairly constant value. After 90% void the head starts to return to its single-phase value.

Torque is modelled similarly to head.

2.3 Subroutine PUMPANC

The PUMPANC subroutine in FIREBIRD calculates the pump head, torque and speed for single phase and two-phase flows. It is based on a combination of ANC MOD-1[4] Semiscale and SAWFT[5] data.

The single-phase head is determined from a curve of H/Ω^2 vs. ν/Ω. A typical curve is shown is Figure 4. The curve is represented by a set of data points, with values between these points obtained by linear interpolation. At high values of ν/Ω (both positive and negative) a parabolic extrapolation is used to represent the curve as the pump is expected to approximate a simple resistance of the form $H/\Omega^2 = a + b(\nu/\Omega) + c(\nu/\Omega)^2$. Values of a, b and c and their range of use are shown in Table 1.

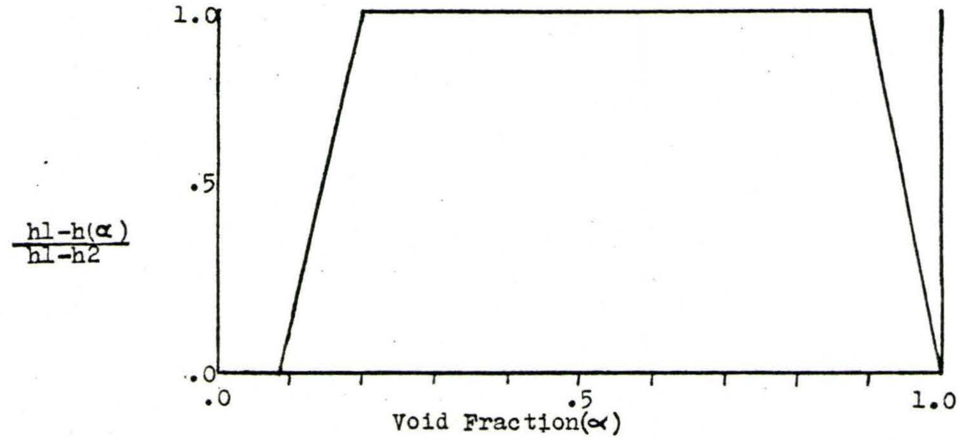


Figure 3; Typical Head Degradation Curve. h1 is single phase head, h2 is fully degraded head, and $h(\alpha)$ is the head at void fraction (α)

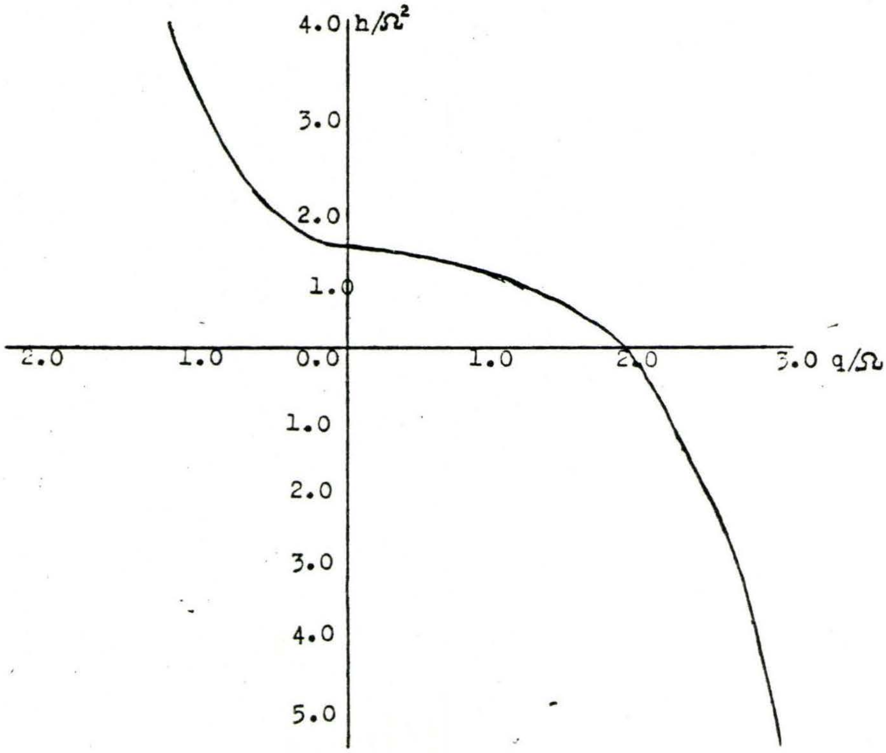


Figure 4: Example of PUMPANC Single Phase Homologous Head

TABLE 1

CONSTANTS FOR PUMP CHARACTERISTIC PARABOLIC EXTRAPOLATION

Y	ν/Ω <-5.0			ν/Ω >3.0		
	a	b	c	a	b	c
Single Phase Homologous Head	2.0	-0.1	0.7	-.68	1.07	-.55
Two-Phase Homologous Head	2.0	-0.1	0.7	-1.5	1.15	-.55

The two-phase head is found by means of the fully degraded head and the void dependent head multiplier. The fully degraded head is found similarly to the single-phase head and represents the maximum pump degradation. The head multiplier is a function of the void fraction and is used to calculate the head degradation.

The single-phase torque is determined from a curve of β/Ω^2 vs. ν/Ω similar to the single-phase head curve. This value is then multiplied by the ratio of the fluid density to the rated density. For two-phase flow this product is multiplied by a correction factor which is empirically determined. No fully degraded curve is used.

2.4 Locked Impeller Analysis

The characteristic of the pump at zero impeller speed are particularly important as in case of a pump trip, the pump may be braked to zero speed. This means that the parabolic extrapolation of head and torque must agree with the stopped impeller values at high ν/Ω. In order to do this the hydraulic resistance coefficient of the stopped pump must be found.

By definition

$$K = \frac{\Delta P}{\frac{1}{2}\rho V^2} \tag{12}$$

where K is coefficient of resistance of the stopped pump, ΔP is the pressure differential across the pump, ρ is the fluid density and V is the fluid velocity.

This gives

$$H = \frac{\Delta P}{\rho g} = \frac{-K Q^2}{2g A^2}$$

where H is the head, Q is volumetric flow, A is the area, and g is acceleration due to gravity. Dividing through by the pump rated values Q_o, H_o gives

$$\frac{h}{Q_o^2} = \frac{-K}{2gA^2} \frac{\nu^2}{H_o} \tag{14}$$

or

$$h = -K^1 \nu^2 \tag{15}$$

where

$$K^1 = \frac{K}{2gA^2} \frac{Q_o^2}{H_o} \tag{16}$$

The parabolic extrapolation used in PUMPANC can be represented by:

$$h/\Omega^2 = a + b(\nu/\Omega) + c(\nu/\Omega)^2 \tag{17}$$

as the impeller speed goes to zero.

$$\lim_{\Omega \to 0} h = \lim_{\Omega \to 0} \Omega^2 (a + b(\nu/\Omega) + c(\nu/\Omega)^2) = C\,\nu^2 \tag{18}$$

When equation (18) is compared to equation (15) it can be easily seen that $C = K^1$. Using equation (16) for the definition of K^1 and rearranging gives:

$$K = C(2gA^2) \frac{H_o}{Q_o^2} \tag{19}$$

3.0 METHOD

3.1 Coast Down Tests

A schematic of the test apparatus is shown in Figure 5. Heat to the loop was supplied by the pump itself, and the temperature was controlled by bypassing part of the water and putting it through air coolers (effect not modelled). The resistance of the reactor loop was simulated by an orifice whose resistance could be adjusted. Pressure was maintained by two plunger pumps (not shown in Figure 5). Light water at 9.51 MPa and 211°C was used to simulate heavy water at the same pressure and 266°C.

The apparatus is modelled using six nodes, with a seventh imaginary node to fix the boundary conditions. This is shown in Figure 6. Detailed dimensions were not available for the complete loop so approximate dimensions were scaled from drawings from reference (1). An example of typical input data is shown in Appendix 1.

The initial conditions are obtained by setting the pump-rated conditions equal to the experimental initial conditions and adjusting the resistance of the orifice for each case until initial conditions are achieved. No flow occurs in the pressure enthalpy boundary link at steady state. Initial conditions for the four pumps and rated conditions are shown in Table 2. Note that the characteristics for the real pumps vary significantly.

No thermal calculations for the pump or air coolers are performed as a steady state run which removed the amount of heat equivalent to the pump heat (without any corresponding thermal input from the pump) showed only a 2°F drop in 30 seconds. The maximum time for which the rundown is calculated in just over two minutes so the temperature change will have negligible effect on fluid properties. The reason for the small temperature drop is the high thermal inertia of the system, i.e., a large mass of water and pipe.

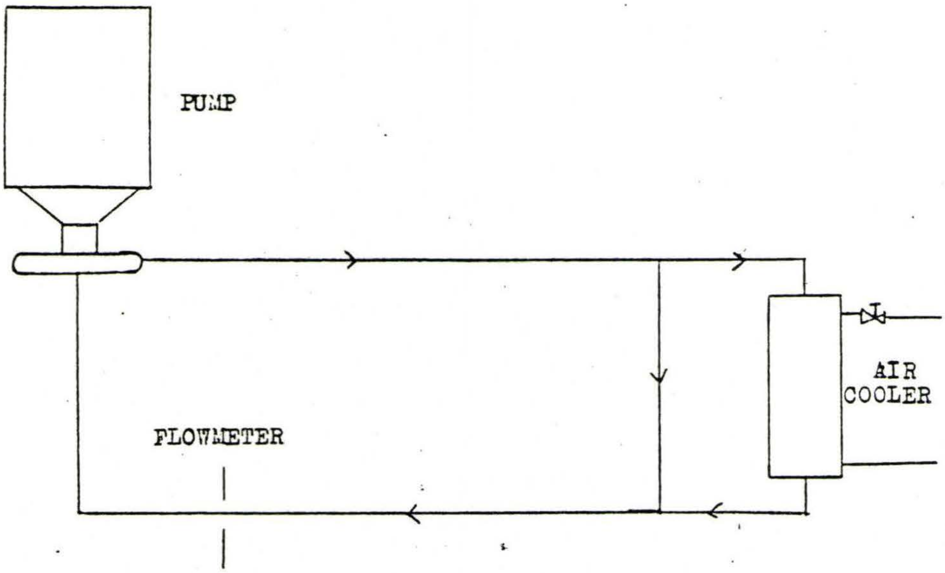


Figure 5; Schematic of Experimental Apparatus

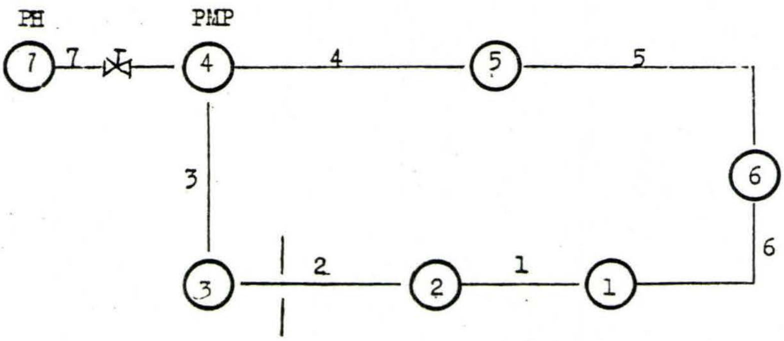


Figure 6: System of nodes and links used to model the experimental apparatus. Link 2 contains the orifice, node 4 contains the pump and node 7 sets the pressure-enthalpy boundary conditions at pump suction.

TABLE 2

COMPARISON OF RATED CONDITIONS WITH INITIAL CONDITIONS FOR G2
PUMPS 1 to 4

PARAMETER	RATED CONDITIONS	PUMP 1	PUMP 2	PUMP 3	PUMP 4
FLOW, Ft^3/s	78.7	72.3	79.5	76.5	82.2
HEAD, Ft.	705	779	617	591	709
SPEED, RPM	1790	1790	1790	1790	1790

Quantities Common to all Pumps

Suction Pressure	1363 psi
Moment of Inertia	931.5 $lb_f\text{-}ft/s^2$
Power	4.78 MW
Density of Fluid	53.59 lbm/ft^3
Specific Speed	2256 $RPM(IGPM)^{1/2}/Ft^{3/4}$

When the pump is tripped the control valve in link 7 is closed, isolating the boundary condition node, so there is no fixed suction pressure.

Three different rundowns are done with pumps 1 and 2. In the first case, the pump rated conditions are set equal to the experimental initial conditions. The pump power is set equal to the rated power of 4.78 MW and the pump is tripped at time zero. This is run until pump speed equals zero. The second case is a rerun of the first with the impeller speed forced to follow experimental results in order to eliminate errors in the torque model. The third case uses the design rated head and flow, fixing the initial flow to coincide with the experimental initial flow. The impeller speed is calculated by the code. A run was done with pump power at 4.78 MW and another done on pump 1 with pump power at 5.0 MW to see the effect of torque. A third run for case three was done with the impeller inertia changed to 900.0 from 931.5. The power was 4.78 MW for this run.

In a separate calculation, tables from the pump model for the FIREBIRD-III code are used with the experimental flow and speed as input to try to predict the head. This is done in order to remove effects due to torque or flow errors resulting from other parts of the calculation. This is done for all four pumps.

3.2 Locked Rotor Tests

Part of the reactor pump rundown tests was to brake the pump. This produced the locked impeller head flow curve shown in Figure 7. This curve is used to find the locked impeller resistance coefficient of the pump, utilizing equation (13) and this is compared to the resistance derived from equation (19).

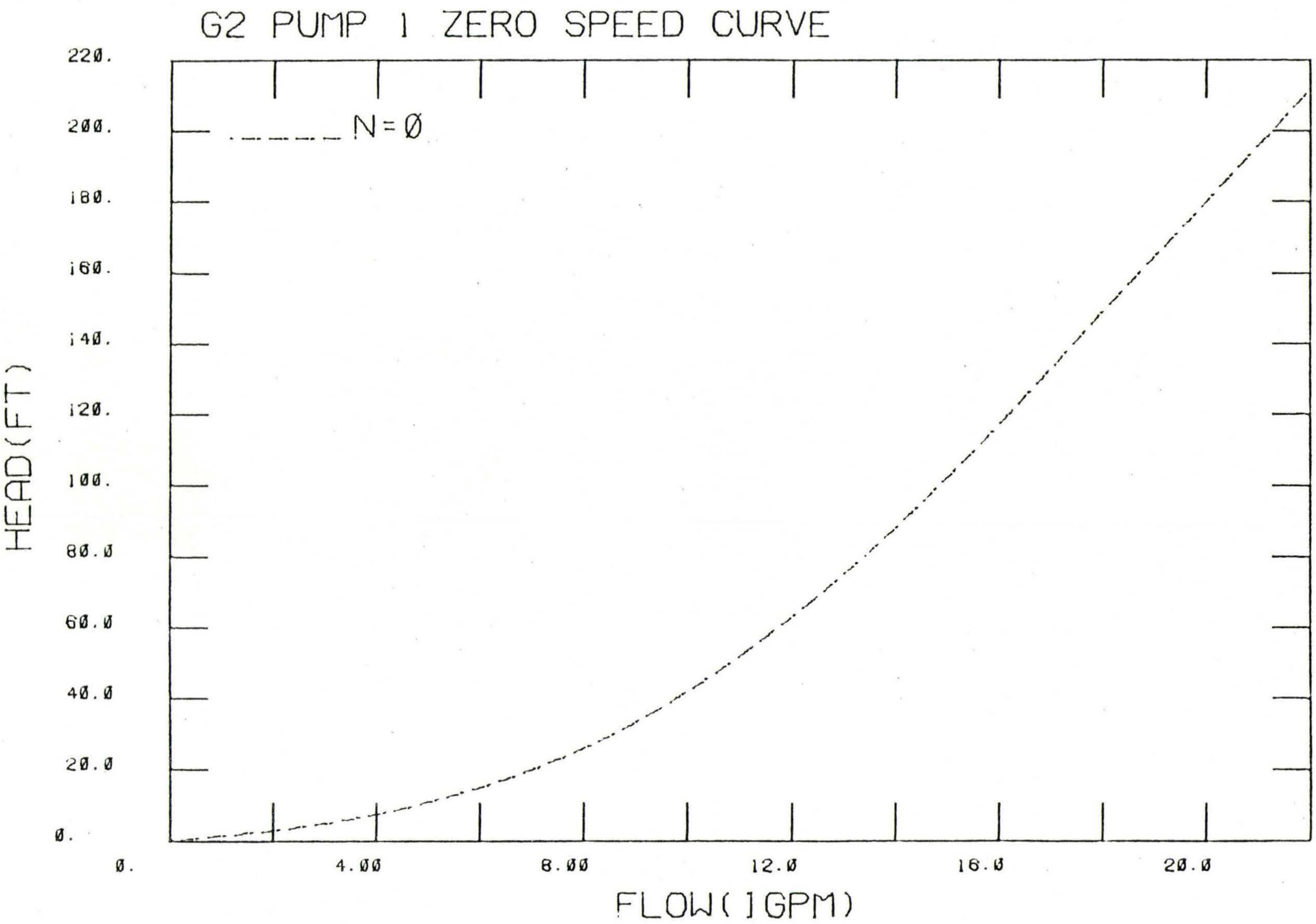

G2 PUMP 1 ZERO SPEED CURVE
N=Ø
HEAD(FT)
220.
200.
180.
160.
140.
120.
100.
80.Ø
60.Ø
40.Ø
20.Ø
Ø.
0.
4.ØØ
8.ØØ
12.Ø
16.Ø
20.Ø
FLOW(IGPM)

Figure 7

The second part of the locked impeller analysis is to compare the resistances of pumps in both single-phase and two-phase flow. This has a dual purpose. One is to check the current assumption that the single-phase and two-phase locked impeller resistance is the same and the second is to give information on scaling.

The pumps used are the ANC MOD-1 Semiscale[4], the C-E/EPRI 1/5[2] scale, the EPRI/CREARE 1/20[3] scale and the MODEL[5] pump. Their characteristics are given in Table 3. Data from the ANC pump tests are available[7] so the average of the locked impeller results is used. For two-phase flow only inlet voids between 30% and 90% are used so as to ensure fully degraded flow. The data are given in Table 4.

Only homologous curves are available for the two EPRI pumps. These are shown in Figures 8 - 11. The forward flow resistance is taken from where the HVN curve crosses the y-axis, and the reverse flow resistance is taken from where the HVT curve crosses the y-axis. At these points the speed equals zero so the value can be represented by $h/\nu^2 = C$, the C corresponding to that in the parabolic extrapolation used in PUMPANC. These values can be compared directly as they are dimensionless.

The MODEL pump has no locked impeller values available, and the available curves are of the same form as the PUMPANC curves. Parabolics are fitted to three points at high flows where the pump is expected to approximate a resistance. The points used are shown in Figures 12 - 14.

TABLE 3

PROPERTIES OF TEST PUMP

PUMP	ANC	EPRI/CREARE	C-E/EPRI	MODEL
Head (Ft)	192	252	252	15.7
Rated Flow (IGPM)	150	182	2914	209
Rated Speed (RPM)	3560	18000	4500	1180
Specific Speed $(RPM(IGPM)^{1/2}/Ft^{3/4})$	845	3840	3840	2163

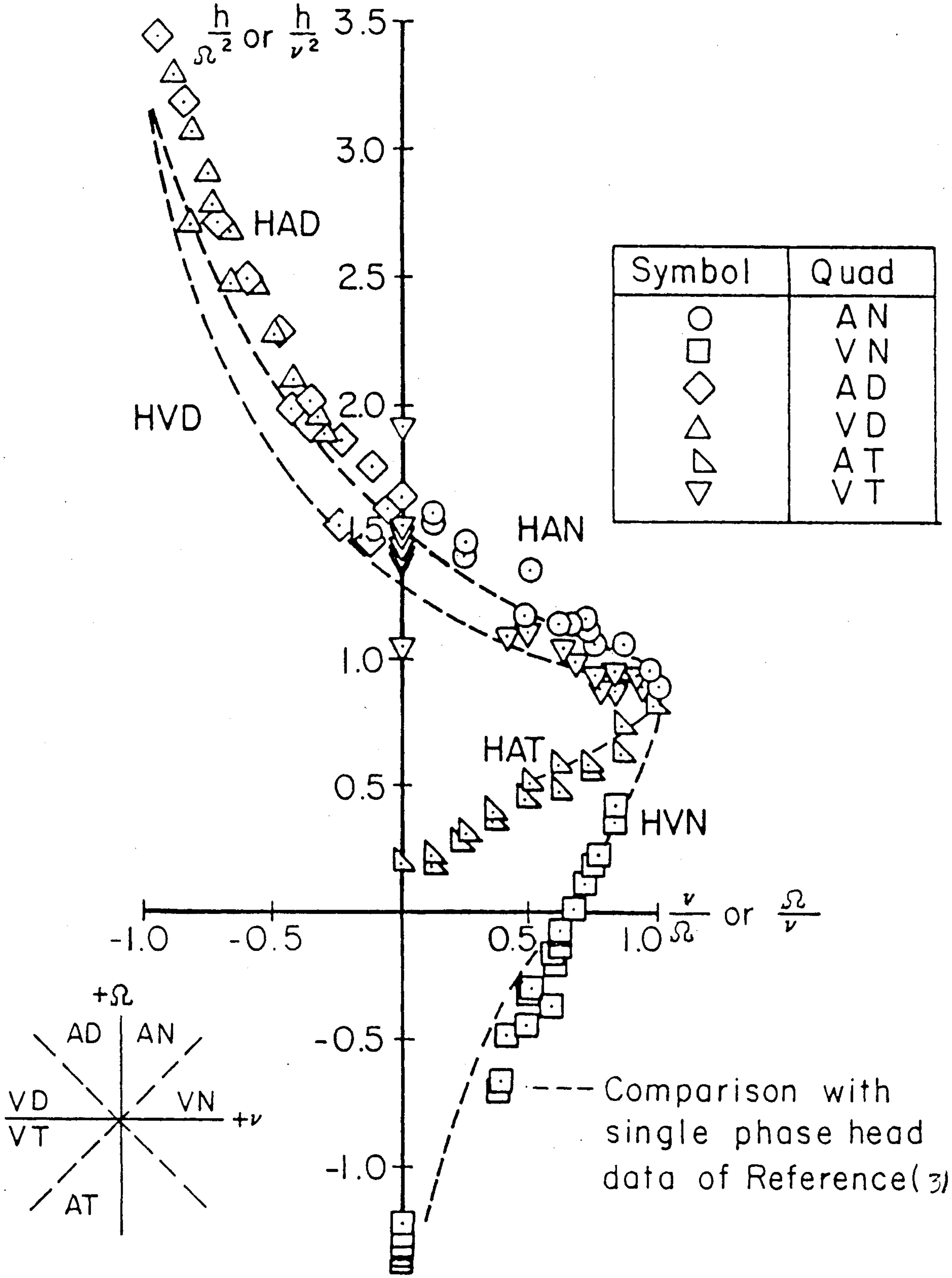
TABLE 4

ANC ZERO SPEED DATA

	FLOW (USGPM)	HEAD (Ft)	h/ν^2	TEST NO.
1) Forward Flow				
a) Single Phase				
	101	-21.2	-.35	001
	164	-55.7	-.35	002
	228	-104.9	-.34	003
	341	-282.5	-.41	004
	276	-159.9	-.35	005
	67	-10.2	-.38	006
	149	-43.7	-.33	136
Average = -.36				
b) Fully Degraded ($.3 < \alpha < .9$)				
	207.1	-92.1	-.35	003
	228.1	-105.4	-.34	004
	289.0	-284.2	-.57	012
	256.9	-222.1	-.57	013
	246.9	-192.9	-.53	014
	209.6	-154.8	-.59	015
	65.3	-9.6	-.38	028
	56.8	-15.2	-.80	034
	58.8	-20.6	-1.01	036
	121.7	-56.4	-.64	046
	113.8	-40.7	-.53	047
	303.1	-319.2	-.59	055
	122.8	-43.8	-.49	059
	103.1	-13.8	-.22	063
	343.5	-346.3	-.50	070
	121.4	-52.6	-.60	074
Average = -.55				

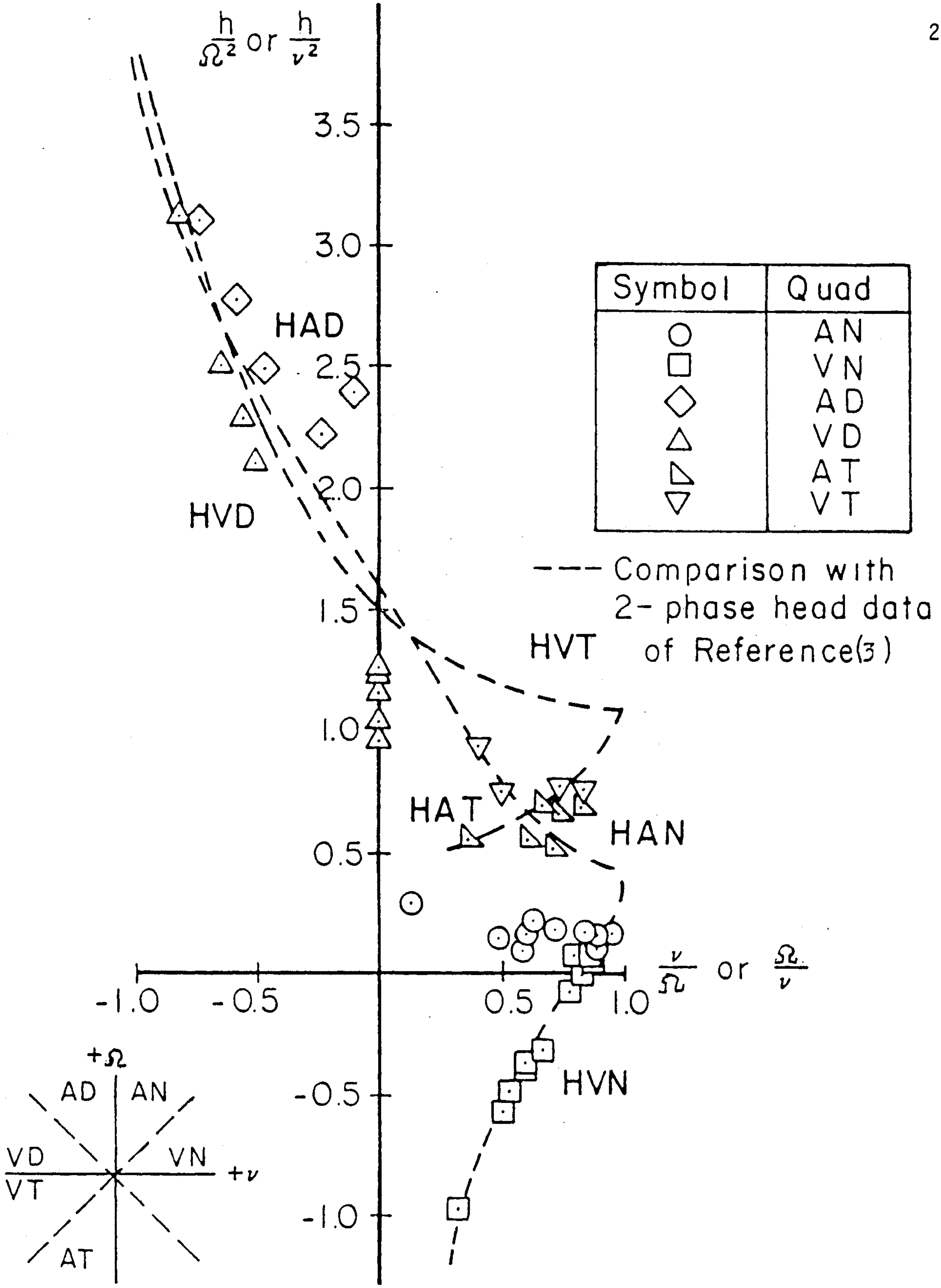
TABLE 4 (Cont'd)

	FLOW (USPGM)	HEAD (Ft)	h/ν^2	TEST NO.
2) Reverse Flow				
a) Single Phase				
	-68	19.4	.71	008
	-106	49.5	.74	009
	-154	101.8	.72	010
	-190	154.6	.72	011
	-255	280.6	.73	012
	-255	280.3	.73	013
	-64	16.7	.69	080
Average = .72				
b) Fully Degraded ($.3 < \alpha < .9$)				
	-89.5	40.2	.85	149
	-112.9	76.2	1.01	164
	-250.7	251.0	.67	201
Average = .84				



SINGLE-PHASE HEAD DATA

Figure 8



TWO-PHASE HEAD DATA, $\alpha_F = 40\%$

Figure 9

FIGURE 10

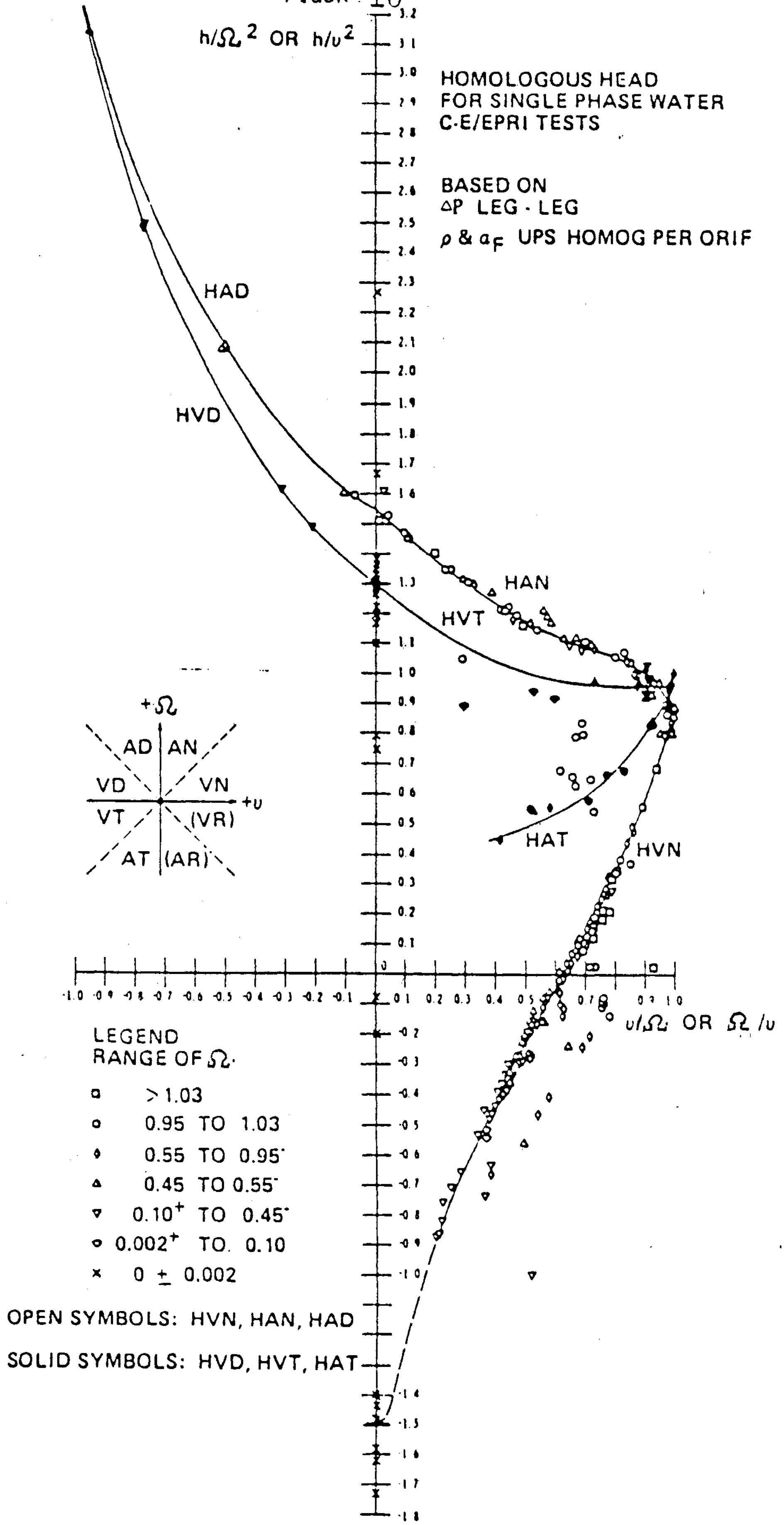

h/Ω² OR h/υ²
HOMOLOGOUS HEAD
FOR SINGLE PHASE WATER
C-E/EPRI TESTS
BASED ON
ΔP LEG - LEG
ρ & a_F UPS HOMOG PER ORIF
HAD
HVD
HAN
HVT
HAT
HVN
+Ω
AD
AN
VD
VN
+υ
VT
(VR)
AT
(AR)
υ/Ω OR Ω/υ
LEGEND
RANGE OF Ω
> 1.03
0.95 TO 1.03
0.55 TO 0.95⁻
0.45 TO 0.55⁻
0.10⁺ TO 0.45⁻
0.002⁺ TO 0.10
0 ± 0.002
OPEN SYMBOLS: HVN, HAN, HAD
SOLID SYMBOLS: HVD, HVT, HAT

Figure 11

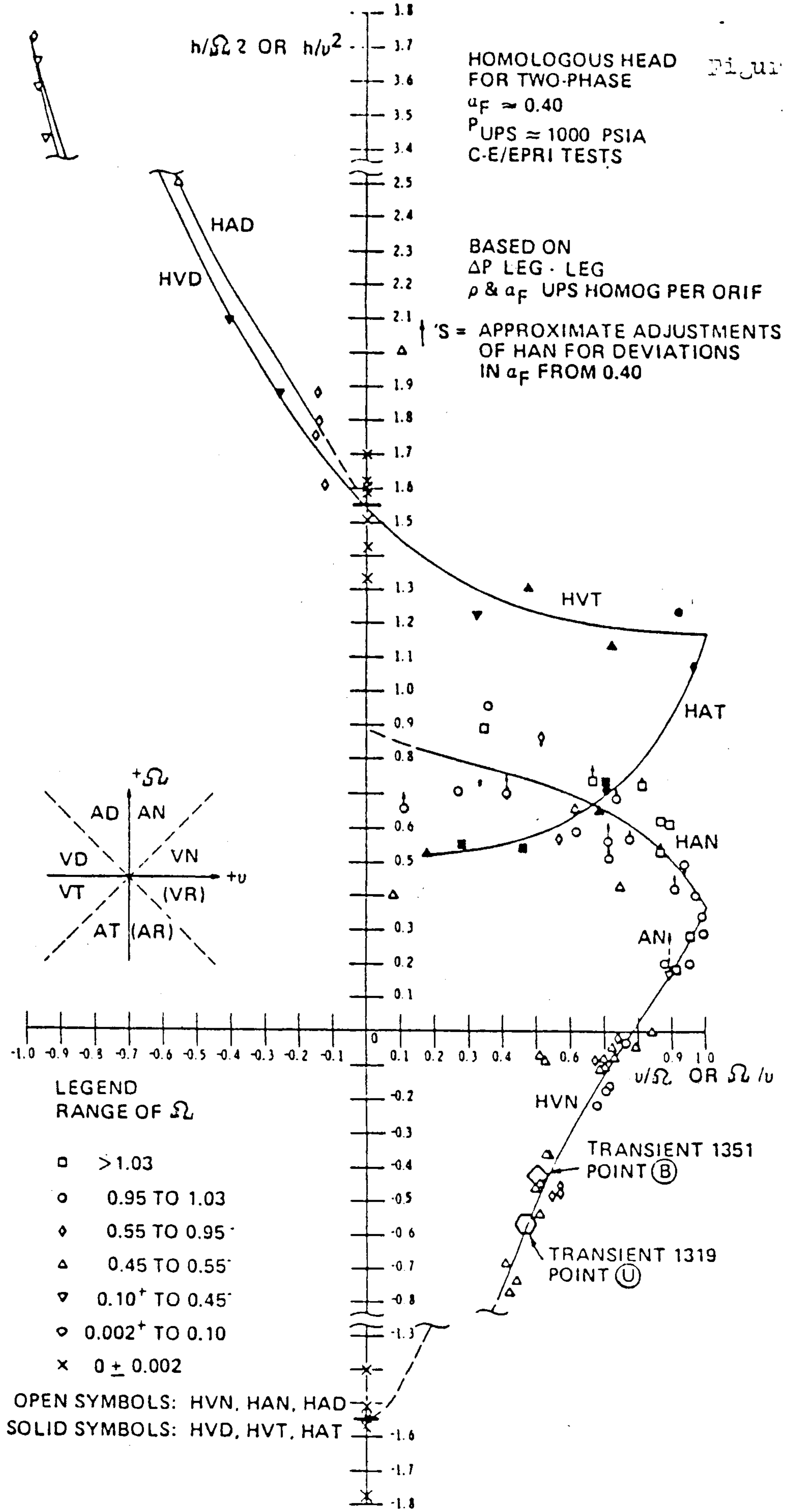
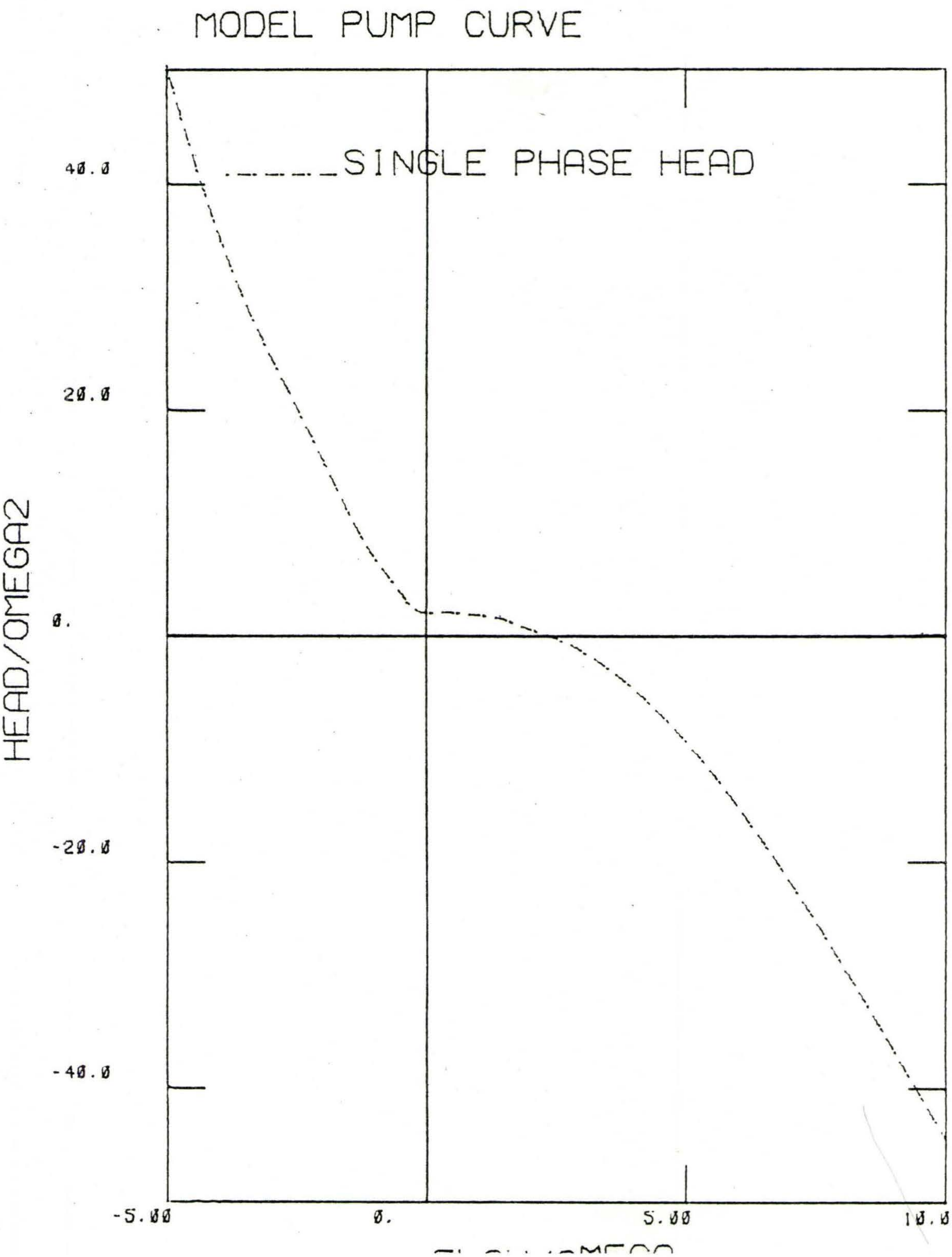

HOMOLOGOUS HEAD
FOR TWO-PHASE
$a_F \approx 0.40$
$P_{UPS} \approx 1000$ PSIA
C-E/EPRI TESTS
BASED ON
ΔP LEG - LEG
ρ & a_F UPS HOMOG PER ORIF
'S = APPROXIMATE ADJUSTMENTS
OF HAN FOR DEVIATIONS
IN a_F FROM 0.40
h/Ω2 OR h/υ2
υ/Ω OR Ω/υ
HAD
HVD
HVT
HAT
HAN
AN
HVN
TRANSIENT 1351
POINT (B)
TRANSIENT 1319
POINT (U)
+Ω
+υ
AD
AN
VD
VN
VT
(VR)
AT
(AR)
LEGEND
RANGE OF Ω
>1.03
0.95 TO 1.03
0.55 TO 0.95
0.45 TO 0.55
0.10+ TO 0.45
0.002+ TO 0.10
0 ± 0.002
OPEN SYMBOLS: HVN, HAN, HAD
SOLID SYMBOLS: HVD, HVT, HAT

Figure 12


MODEL PUMP CURVE
SINGLE PHASE HEAD
40.0
20.0
0.
-20.0
-40.0
HEAD/OMEGA2
-5.00
0.
5.00
10.0

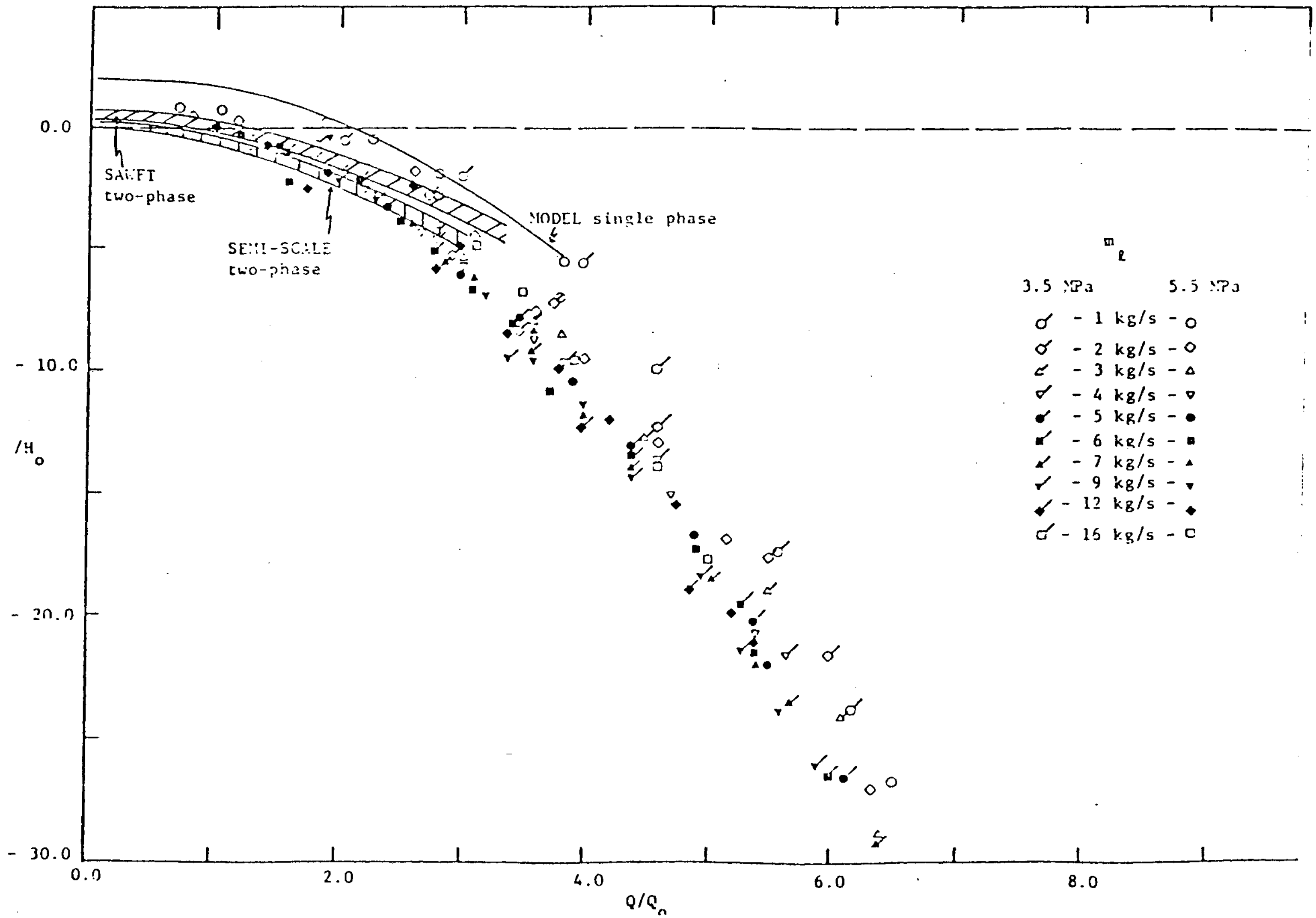


FIG. 13 NORMALIZED HEAD VS VOLUMETRIC FLOW FOR HALF-RATED SPEED OPERATION

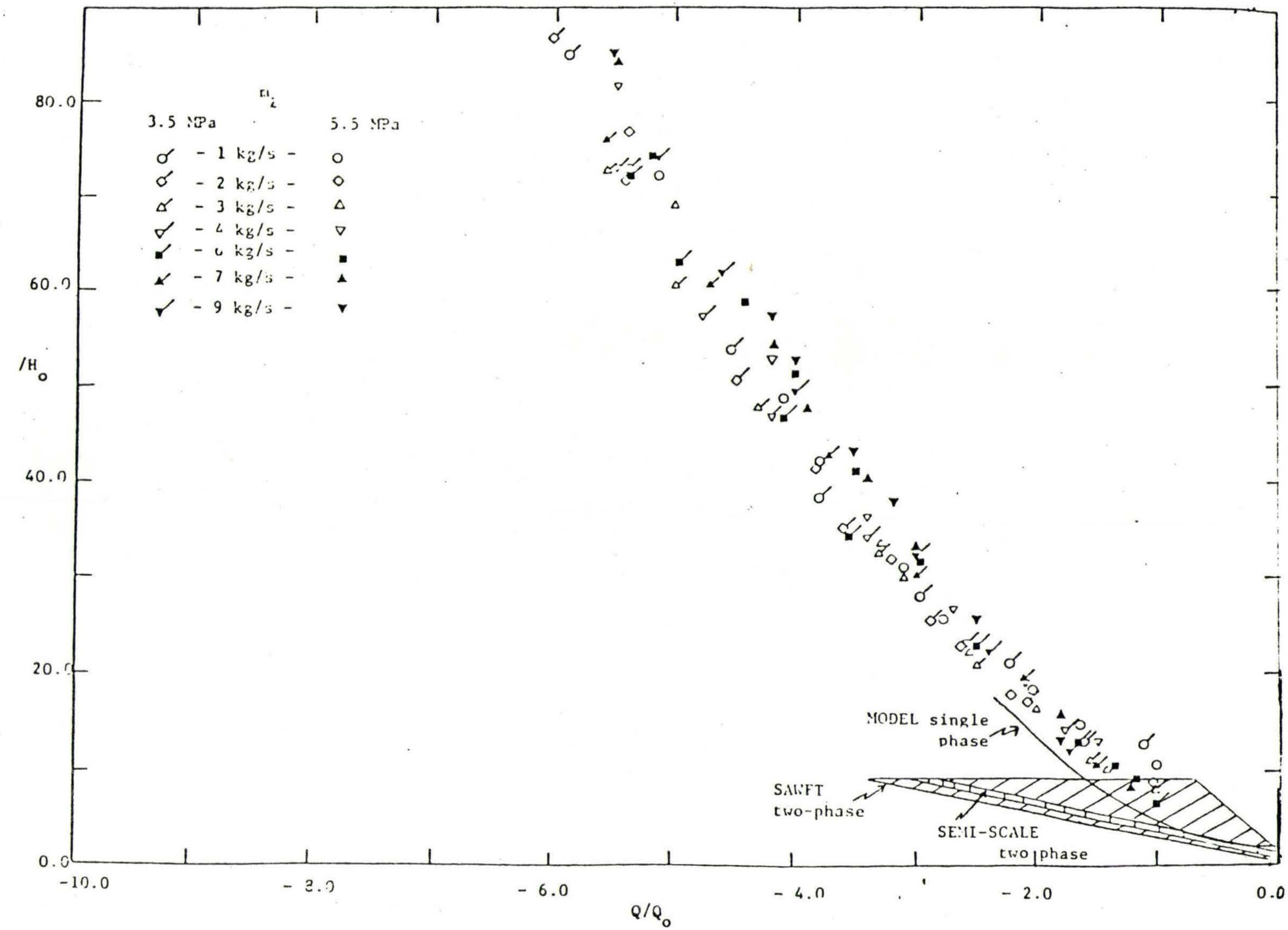
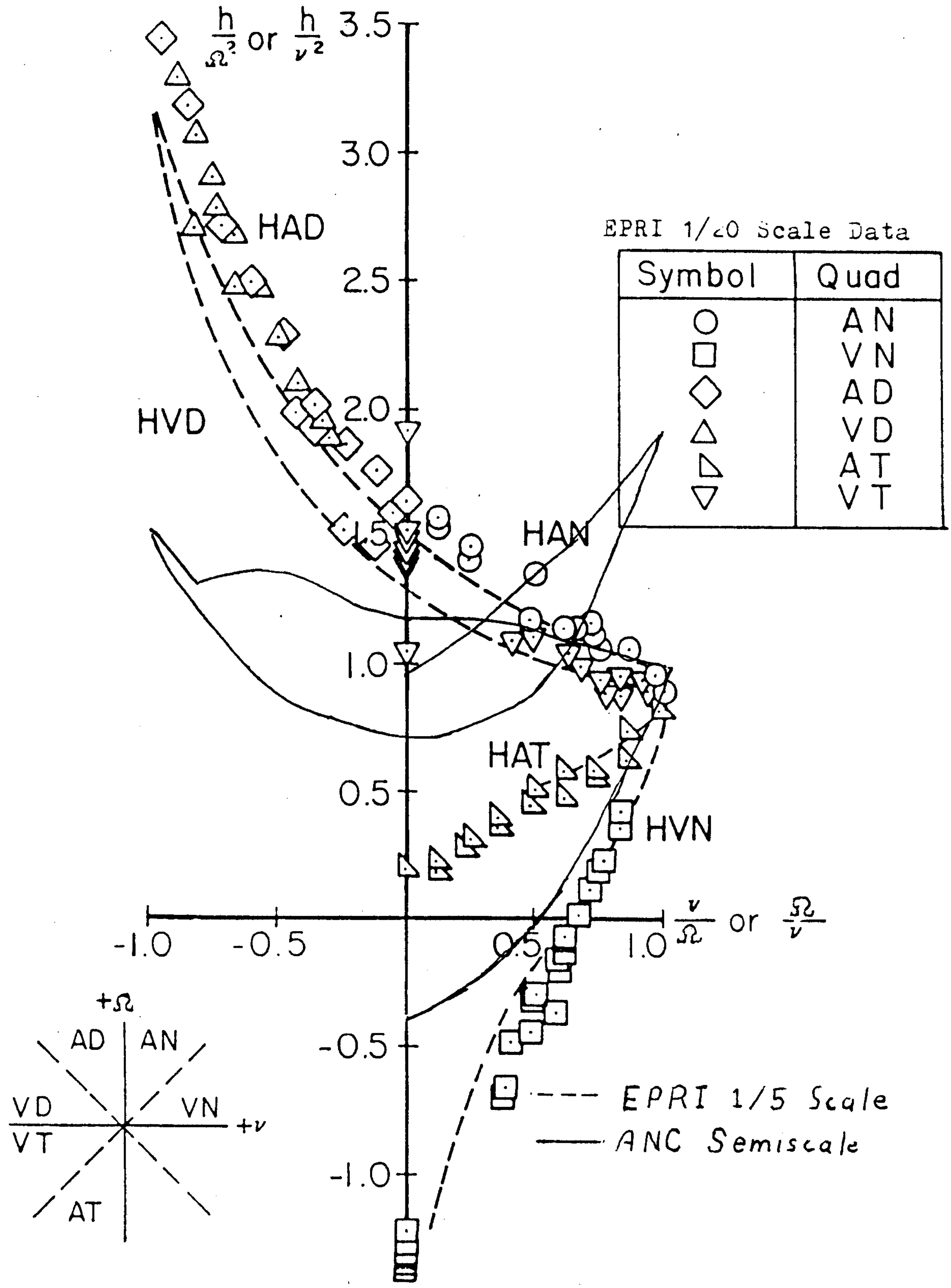


FIG. 14 NORMALIZED HEAD VS VOLUMETRIC FLOW FOR HALF-RATED SPEED OPERATION

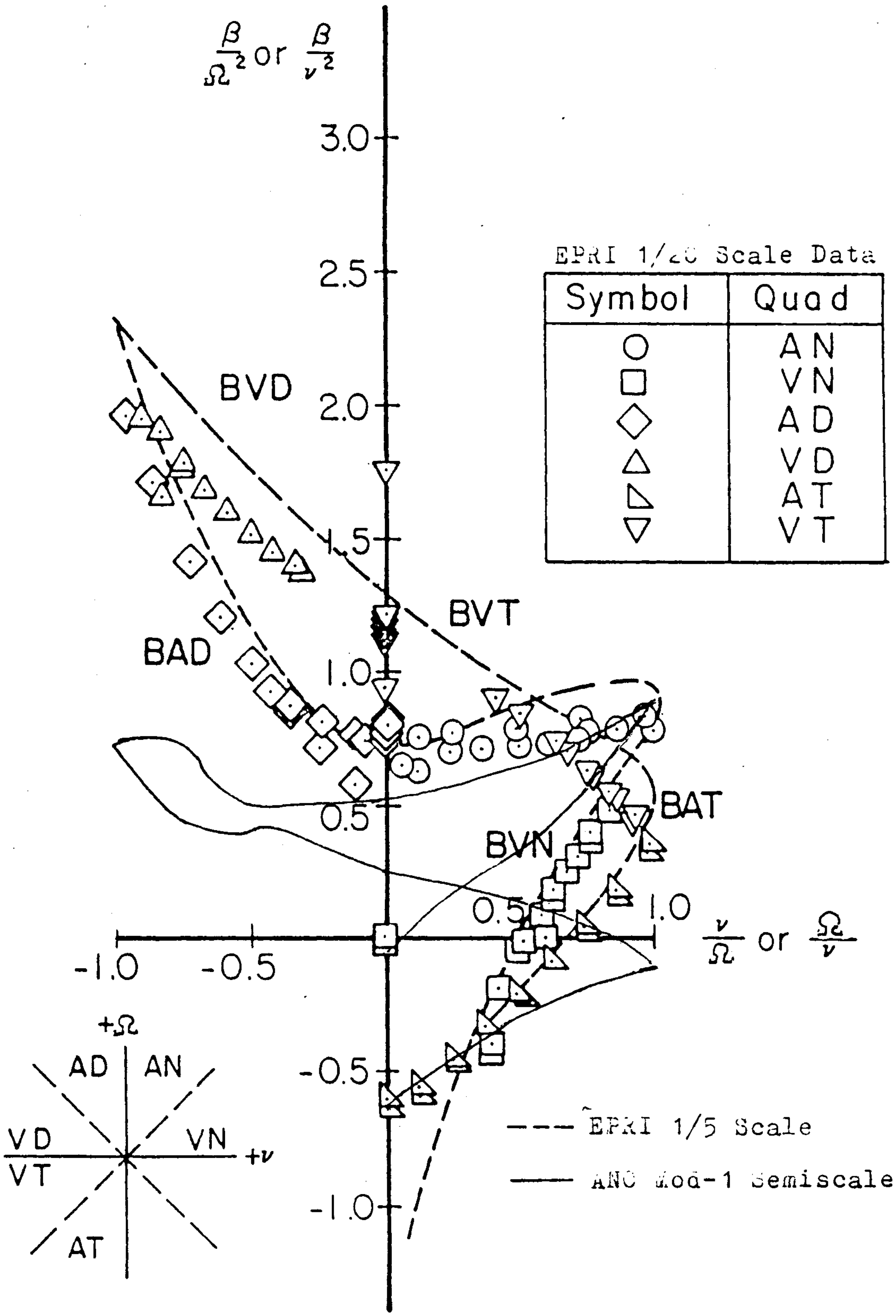
3.3 Pump Scaling Analysis

The homologous head curves for the two EPRI pumps and the ANC pump are compared in Figures 15 - 18. Head and torque degradation curves are shown in Figures 19 - 22 for the C-E/EPRI 1/5 scale pump. The ANC head degradation curves are shown in Figures 23, 24 and 25.



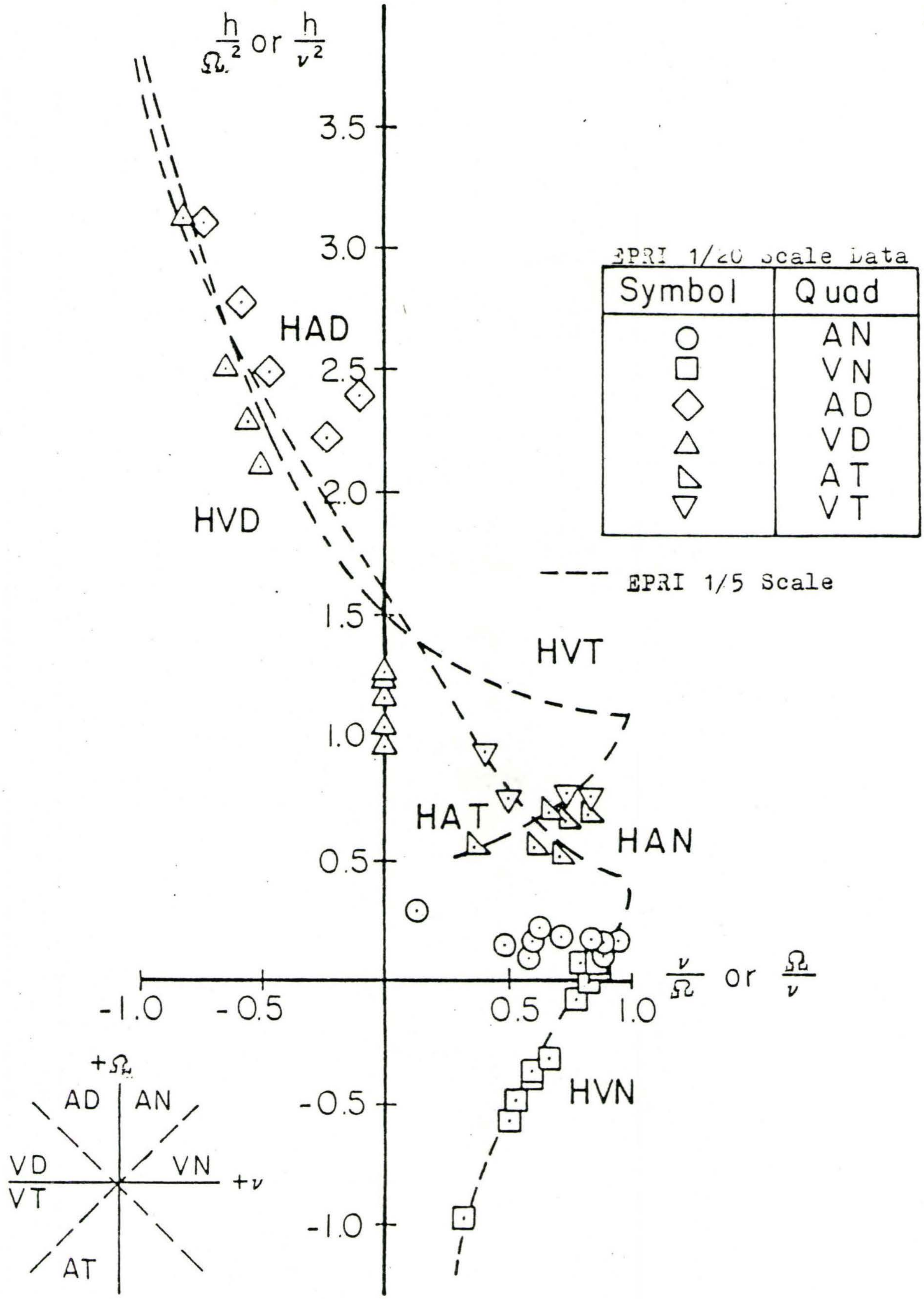
SINGLE-PHASE HEAD DATA

Figure 15



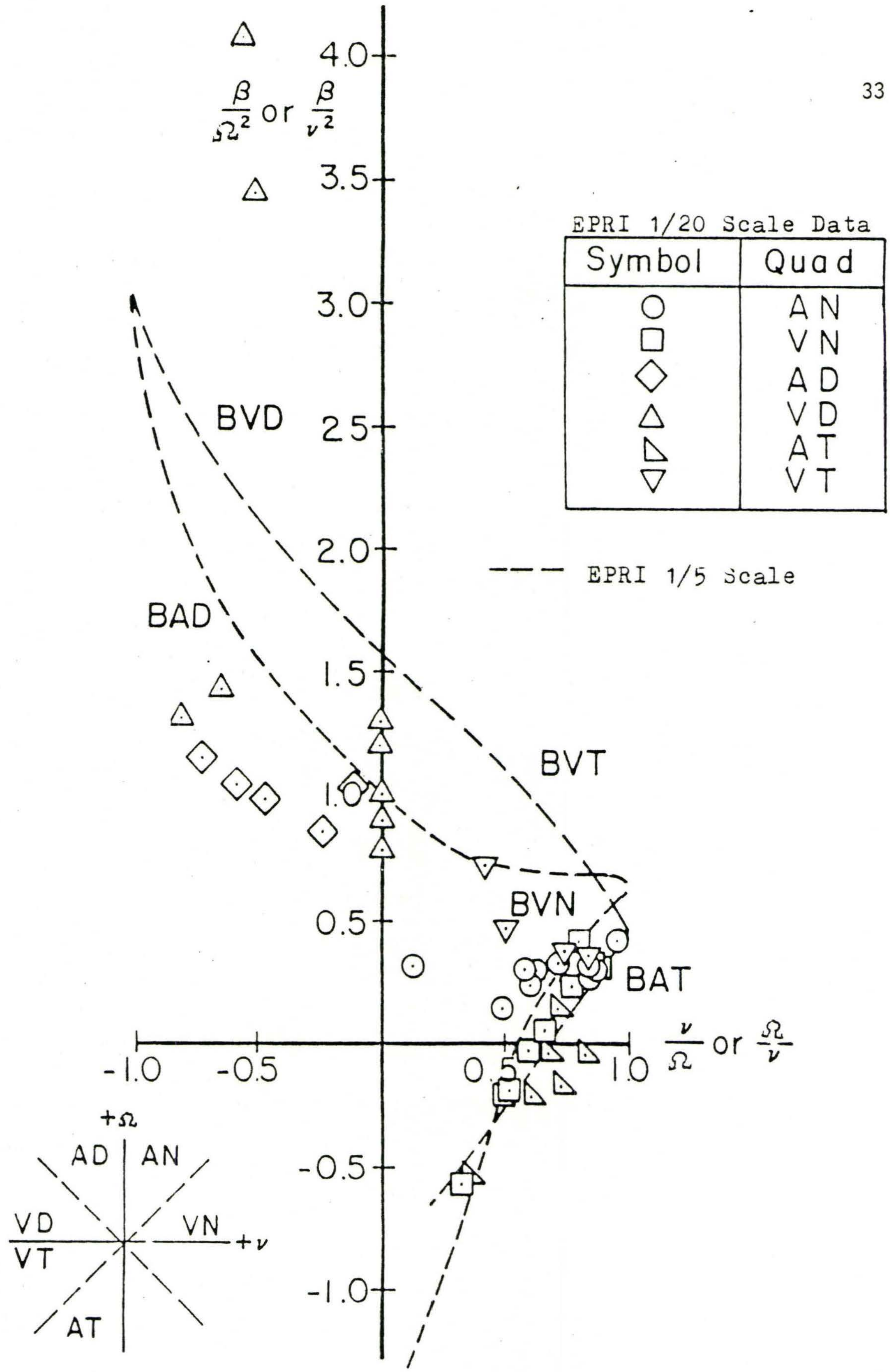
SINGLE-PHASE TORQUE DATA

Figure 16



TWO-PHASE HEAD DATA, $\alpha_F = 40\%$

Figure 17



TWO-PHASE TORQUE DATA, $\alpha_F = 40\%$

Figure 18

EFFECT OF VOID FRACTION ON HOMOLOGOUS HEAD RATIO
FOR RATED SPEED AND NEAR RATED FLOW (EPRI 1/5)
$\Rightarrow \upsilon/\Omega \approx 1.0$

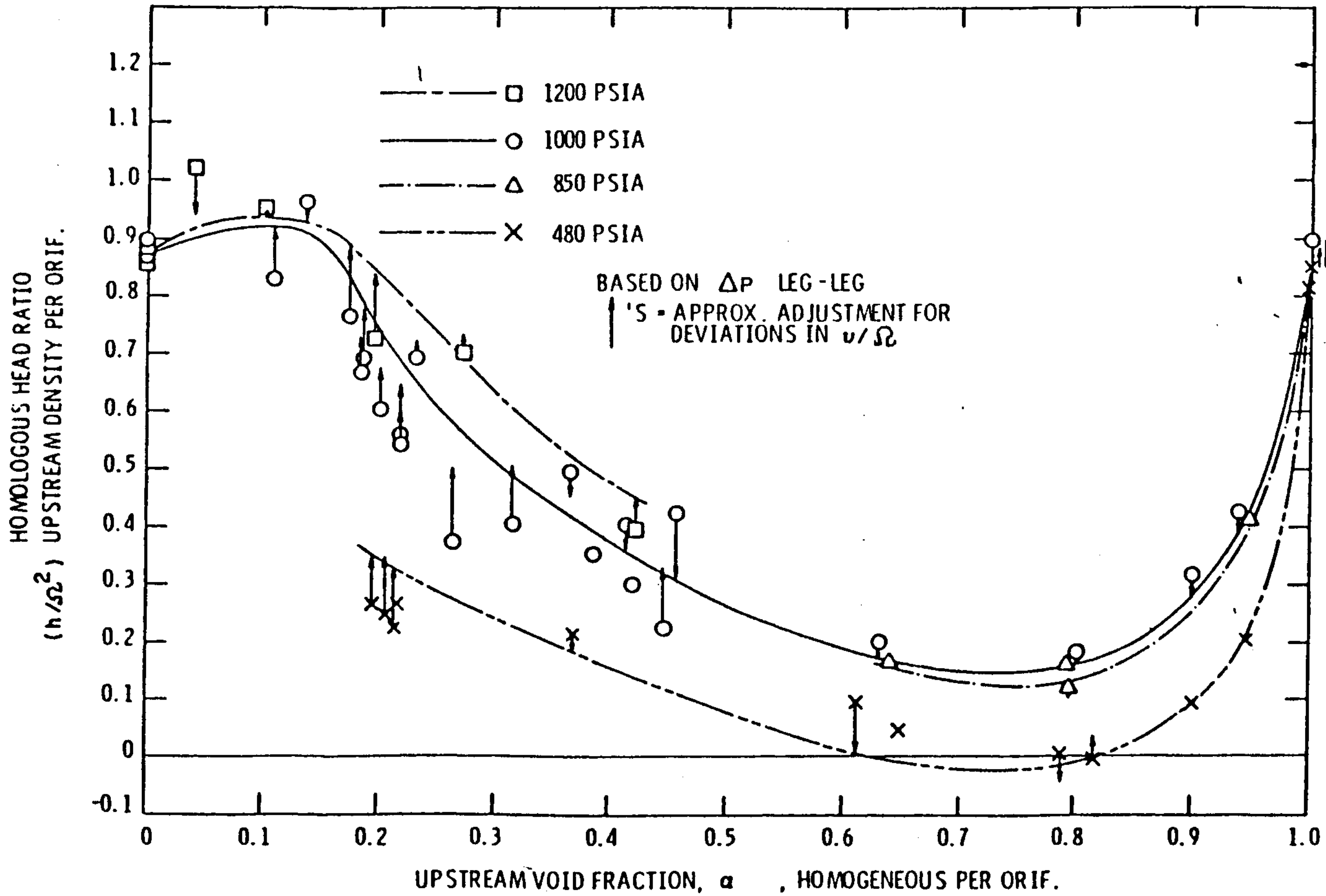
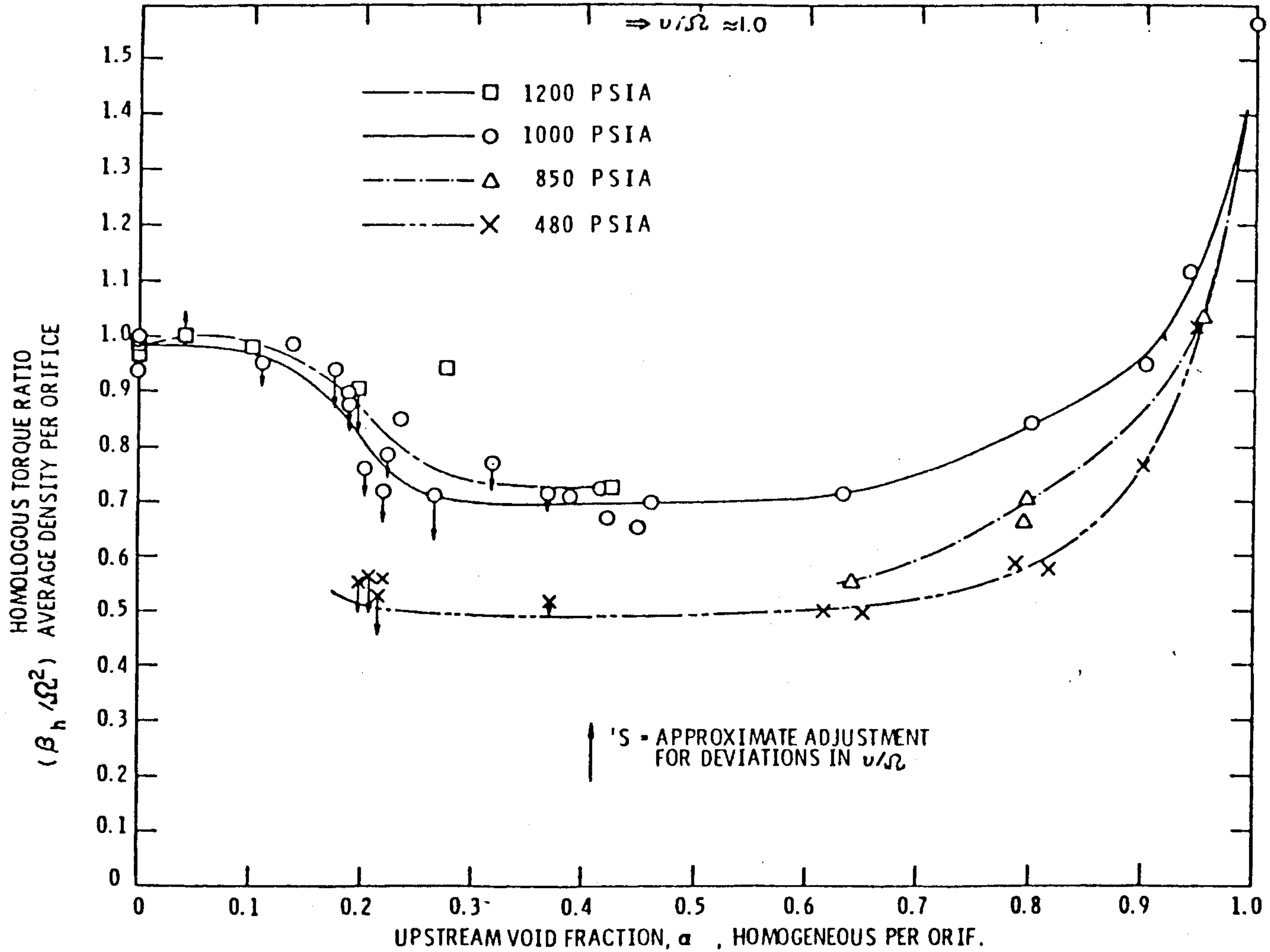


Figure 19

FIGURE 20

EFFECT OF VOID FRACTION ON HOMOLOGOUS TORQUE RATIO
FOR RATED SPEED AND NEAR RATED FLOW (EPRI 1/5)



EFFECT OF VOID FRACTION ON HOMOLOGOUS HEAD RATIO
FOR 1/2 RATED SPEED AND NEAR RATED FLOW (EPRI 1/5)
$\Rightarrow v/\Omega \approx 2.0$

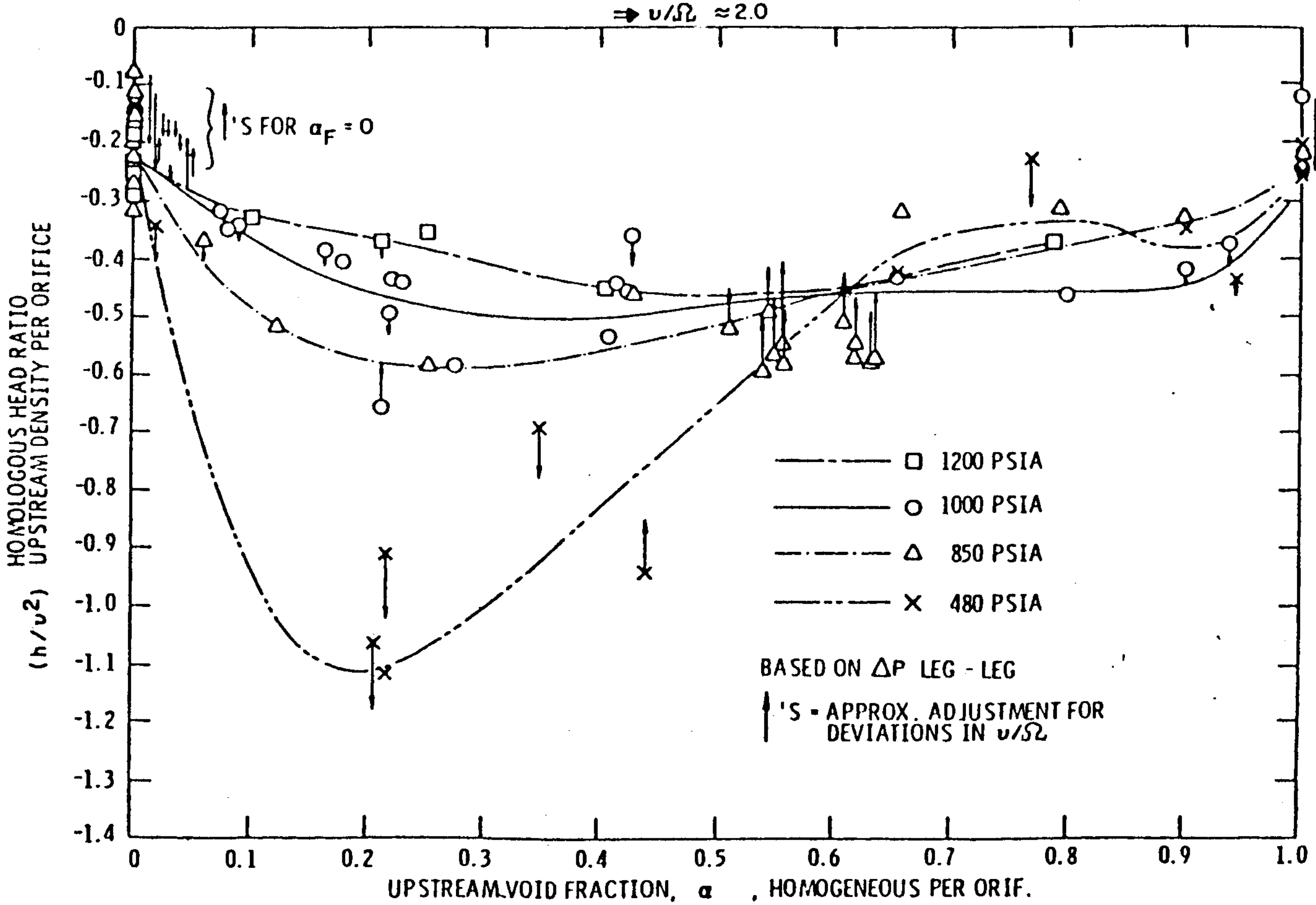


Figure 21

EFFECT OF VOID FRACTION ON HOMOLOGOUS TORQUE RATIO
FOR 1/2 RATED SPEED AND NEAR RATED FLOW(EPRI 1/5)
$\Rightarrow \nu/\Omega \approx 2.0$

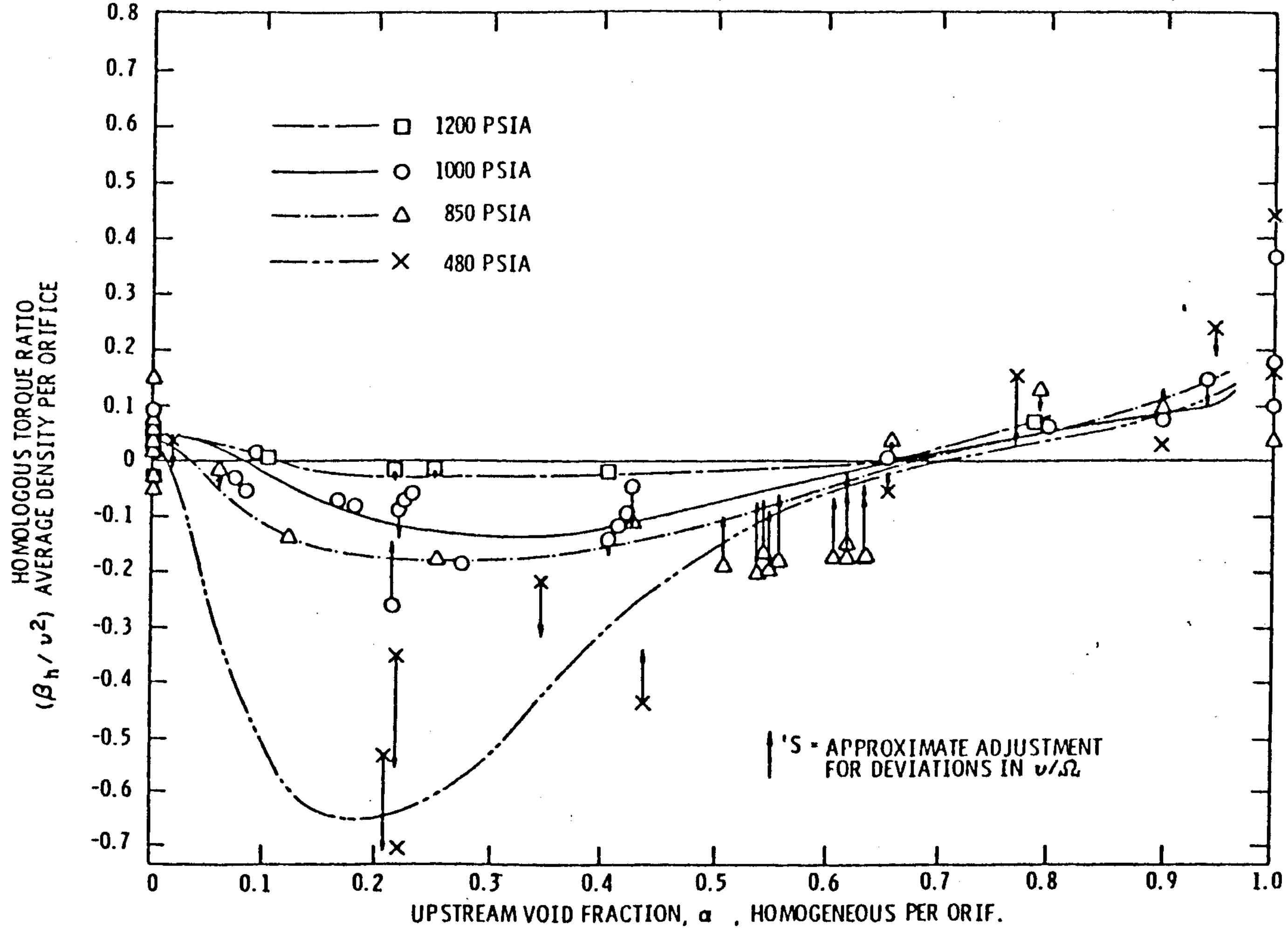


Figure 22

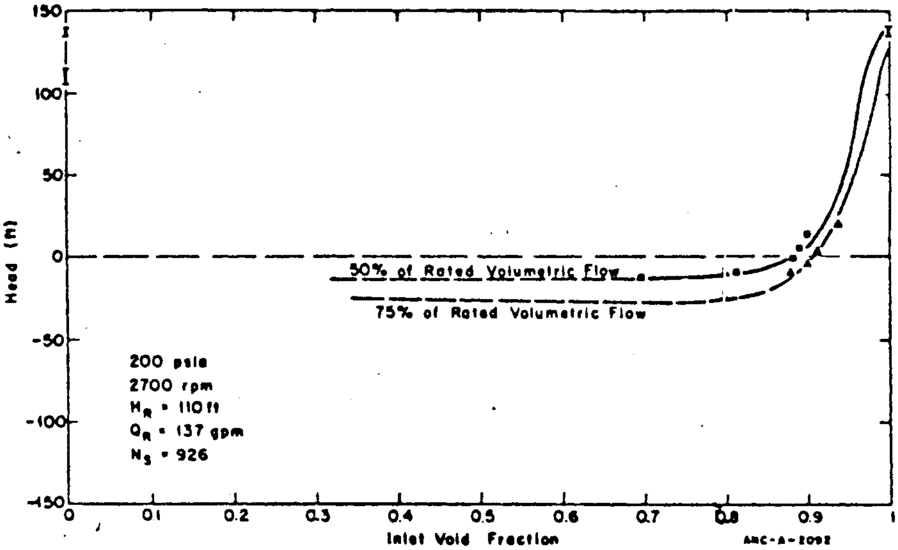


Fig.23 Semiscale Pump Head Versus Void Fraction at 200 psia Inlet Conditions.

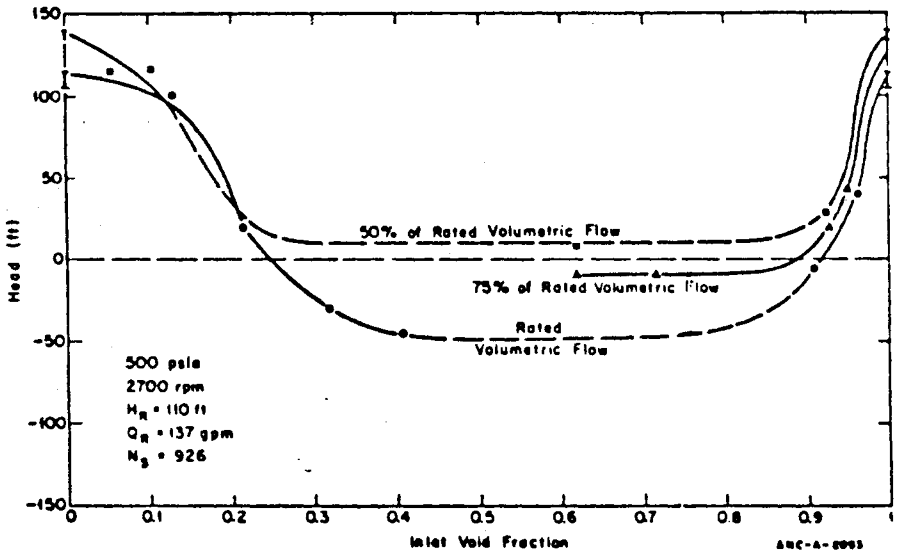


Fig. 24 Semiscale Pump Head Versus Void Fraction at 500 psia Inlet Conditions.

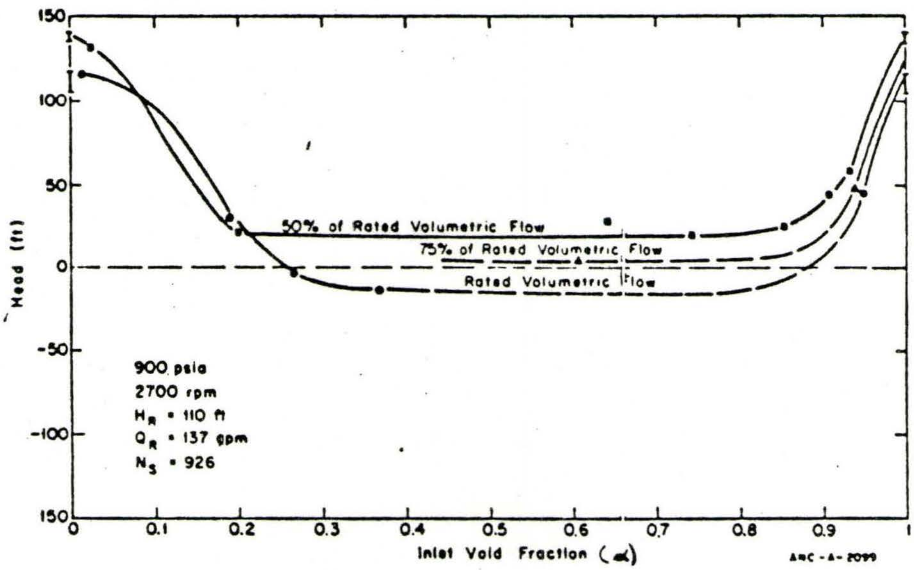


Fig.25 Semiscale Pump Head Versus Void Fraction at 900 psia Inlet Conditions -- Steady-State Tests

4.0 RESULTS

4.1 Pump Rundown Results

I) Pump rated conditions set equal to initial conditions.
Pump 1 (initial conditions, head = 779 ft., flow = 72.3 ft^3/s, orifice resistance = 501.8)

A) Rundown Calculated by the Model

i) Speed
The speed is underpredicted at first, but nears experimental values ater sixty seconds (Figure 6). Experimental data for times beyond sixty seconds are not presently available.

ii) Head
The head is always underpredicted (Figure 27). Considered as a percentage of the actual head, the error increases as time increases.

iii) Flow
There is good agreement between predicted flow and actual flow (Figure 8).

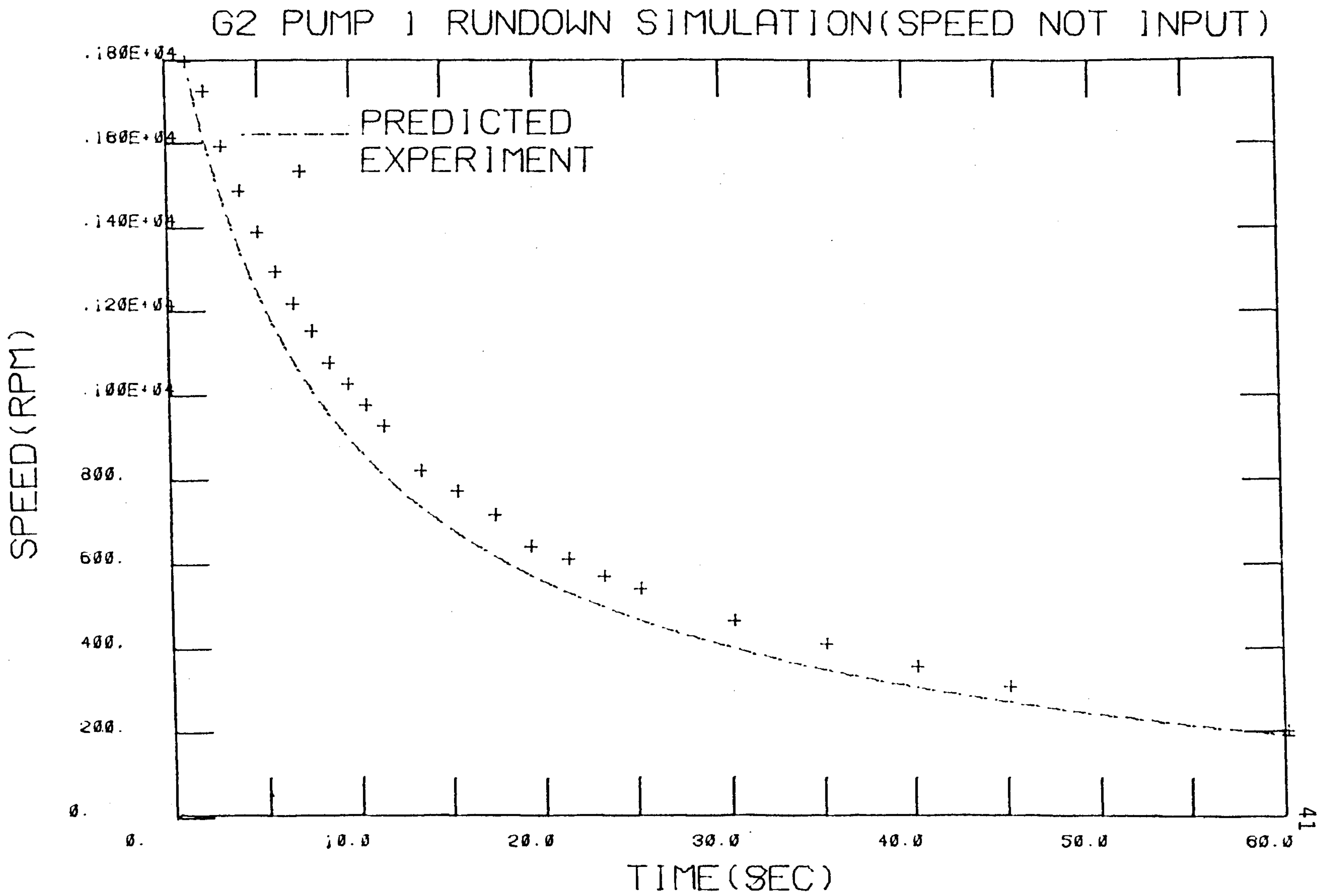
iv) Rundown time to full stop

Predicted	-	137 seconds
Actual	-	290 seconds

B) Speed Input as a Function of Time

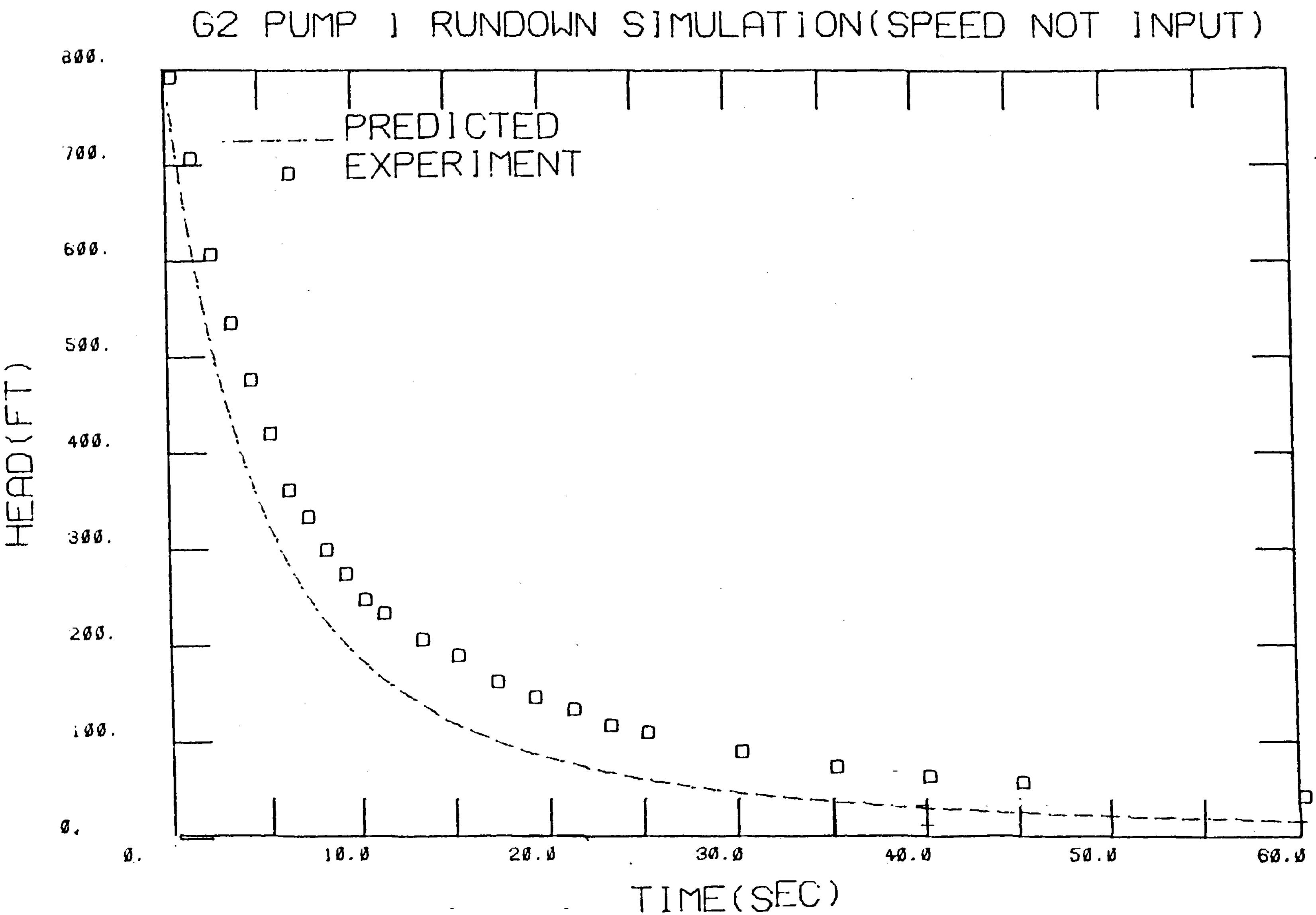
i) Head
There is good agreement for the first ten seconds. The head is then underpredicted, the error increasing with time (Figure 29).

FIGURE 26



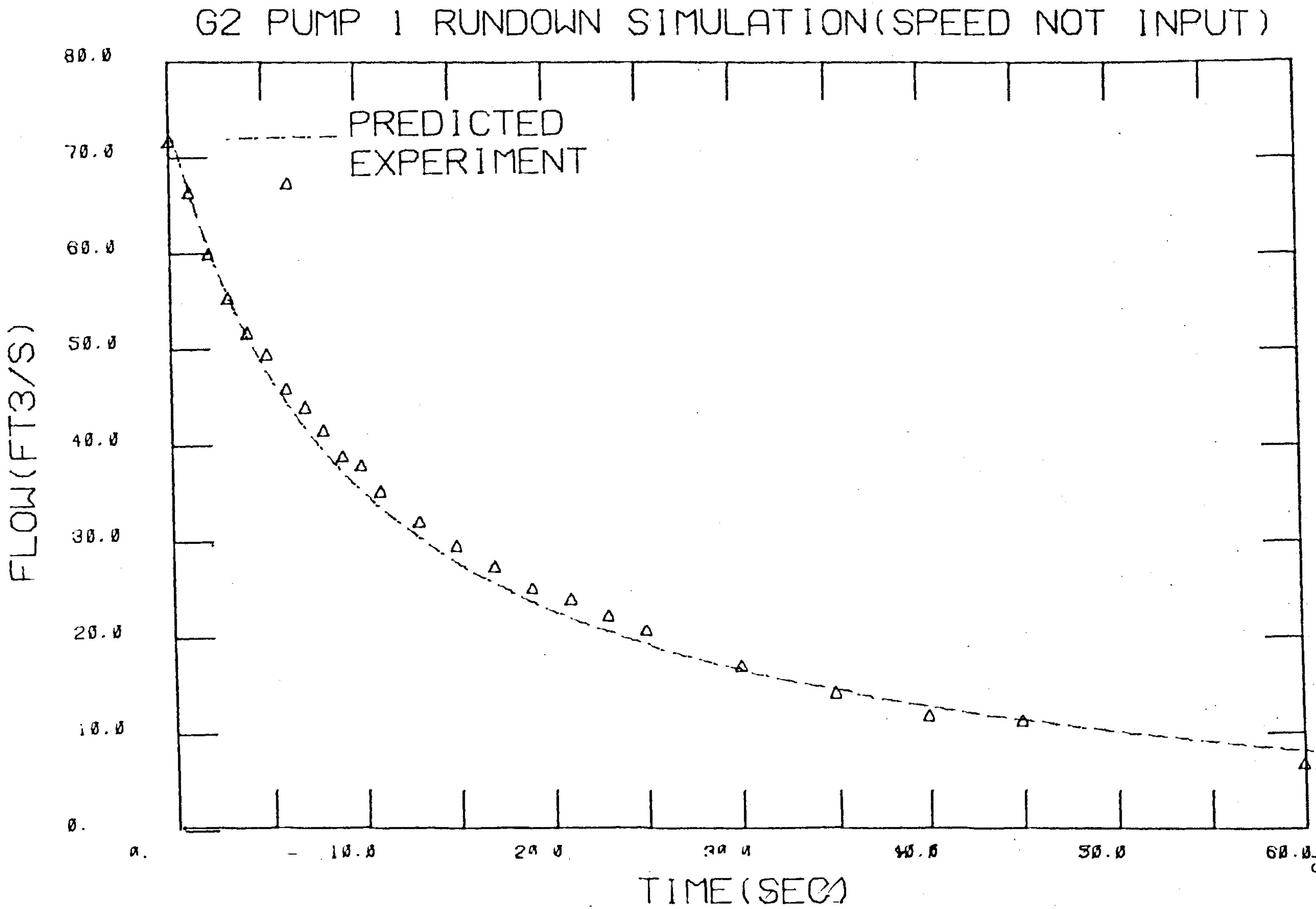
G2 PUMP 1 RUNDOWN SIMULATION(SPEED NOT INPUT)
PREDICTED
EXPERIMENT
SPEED(RPM)
.180E+04
.160E+04
.140E+04
.120E+04
.100E+04
800.
600.
400.
200.
0.
0.
10.0
20.0
30.0
40.0
50.0
60.0
TIME(SEC)

FIGURE 27



G2 PUMP 1 RUNDOWN SIMULATION(SPEED NOT INPUT)
PREDICTED
EXPERIMENT
HEAD(FT)
800.
700.
600.
500.
400.
300.
200.
100.
0.
0.
10.0
20.0
30.0
40.0
50.0
60.0
TIME(SEC)

FIGURE 28



G2 PUMP 1 RUNDOWN SIMULATION(SPEED NOT INPUT)
PREDICTED
EXPERIMENT
FLOW(FT3/S)
TIME(SEC)
80.0
70.0
60.0
50.0
40.0
30.0
20.0
10.0
0.
0.
10.0
20.0
30.0
40.0
50.0
60.0

ii) Flow

The flow is slightly overpredicted at first, but agrees well with experiment after ten seconds (Figure 30).

Pump 2 (initial conditions, head = 617 ft, flow = 79.5 ft^3/s, orifice resistance = 304.2)

A) Rundown Calculated by the Model

i) Speed

There is good agreement with experiment until twenty seconds (Figure 31). The speed is then overpredicted, the error increasing with time.

ii) Head

There is good agreement between prediction and experiment (Figure 32).

iii) Flow

The flow is consistently underpredicted, the error increasing with time (Figure 33).

iv) Rundown time to full stop

Predicted	-	137 seconds
Actual	-	270 seconds

B) Speed Input as a Function of Time

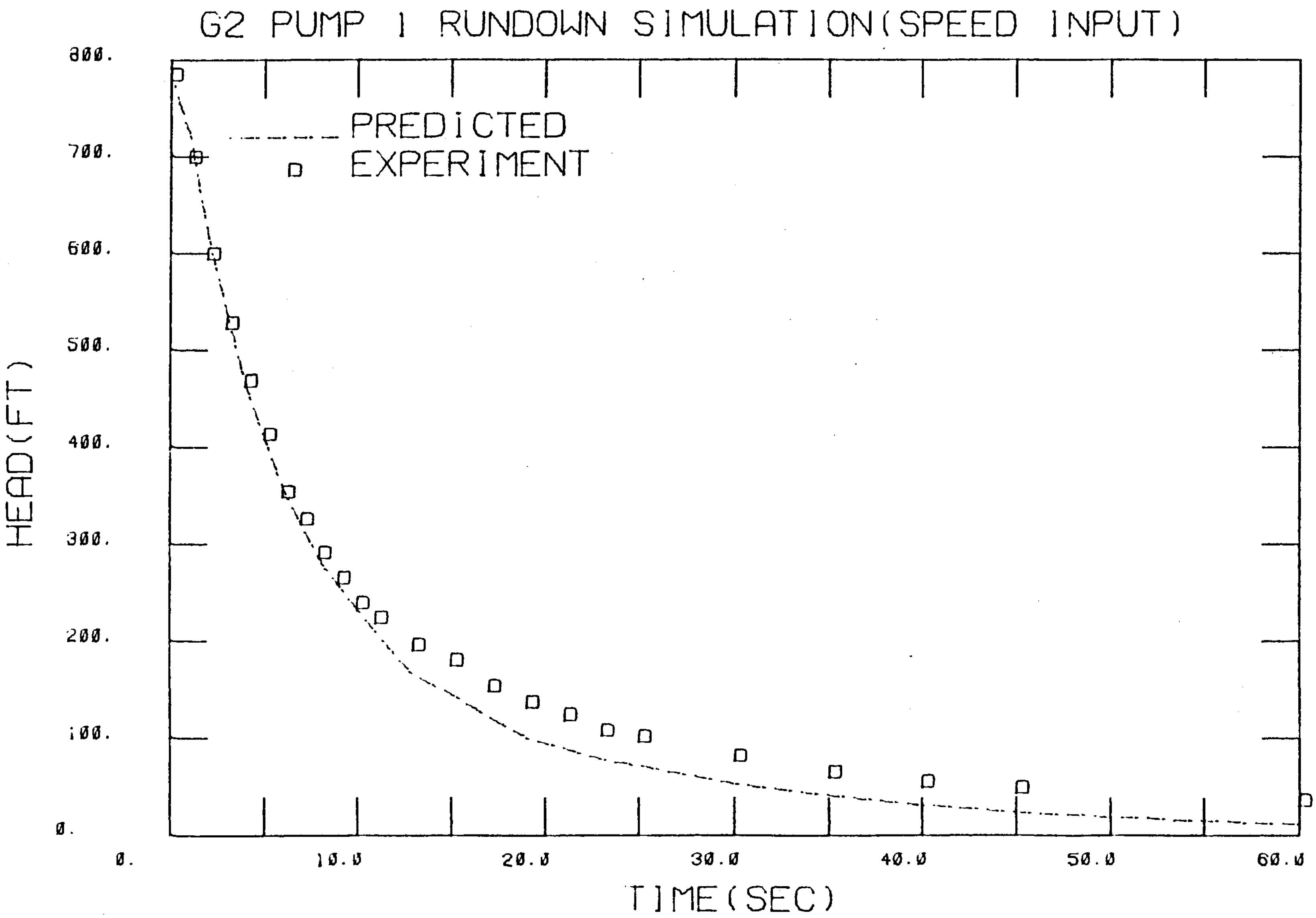
i) Head

There is good agreement up to ten seconds after which the head is increasingly underpredicted (Figure 34).

ii) Flow

There is good agreement up to ten seconds after

FIGURE 29



G2 PUMP 1 RUNDOWN SIMULATION(SPEED INPUT)
PREDICTED
EXPERIMENT
HEAD(FT)
TIME(SEC)
800.
700.
600.
500.
400.
300.
200.
100.
0.
0.
10.0
20.0
30.0
40.0
50.0
60.0

FIGURE 30

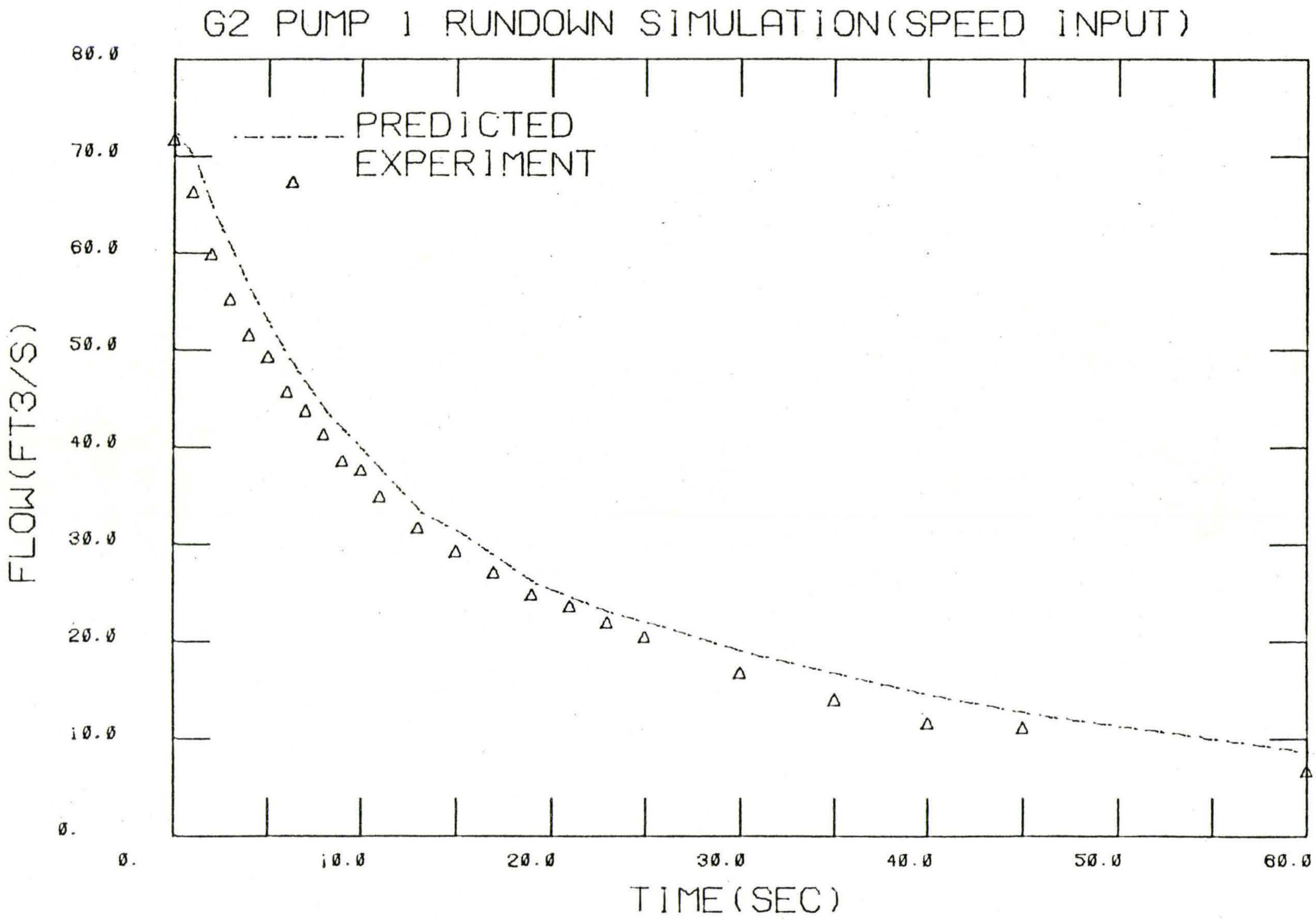

G2 PUMP 1 RUNDOWN SIMULATION(SPEED INPUT)
PREDICTED
EXPERIMENT
FLOW(FT3/S)
TIME(SEC)
0.
10.0
20.0
30.0
40.0
50.0
60.0
70.0
80.0

FIGURE 31

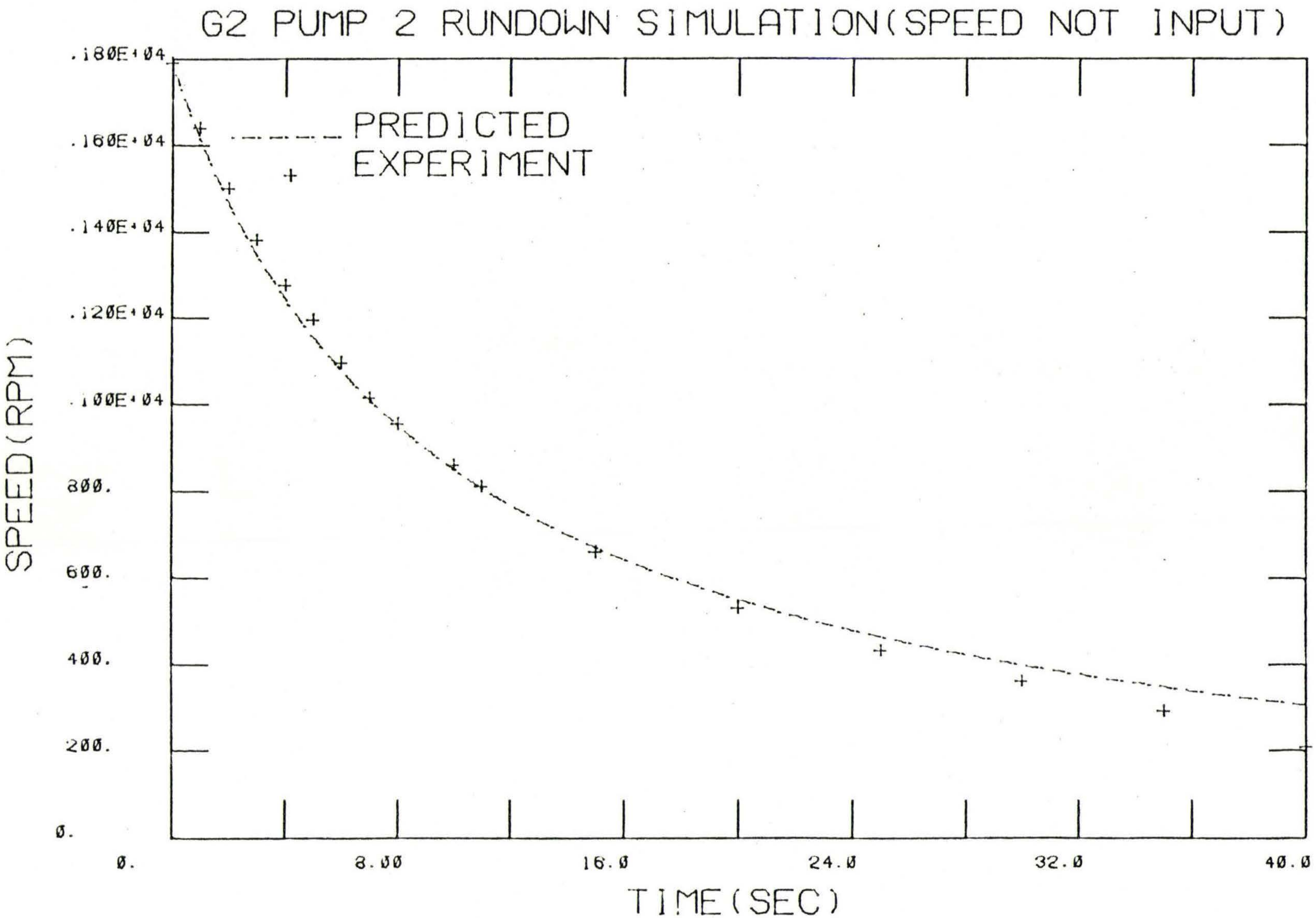
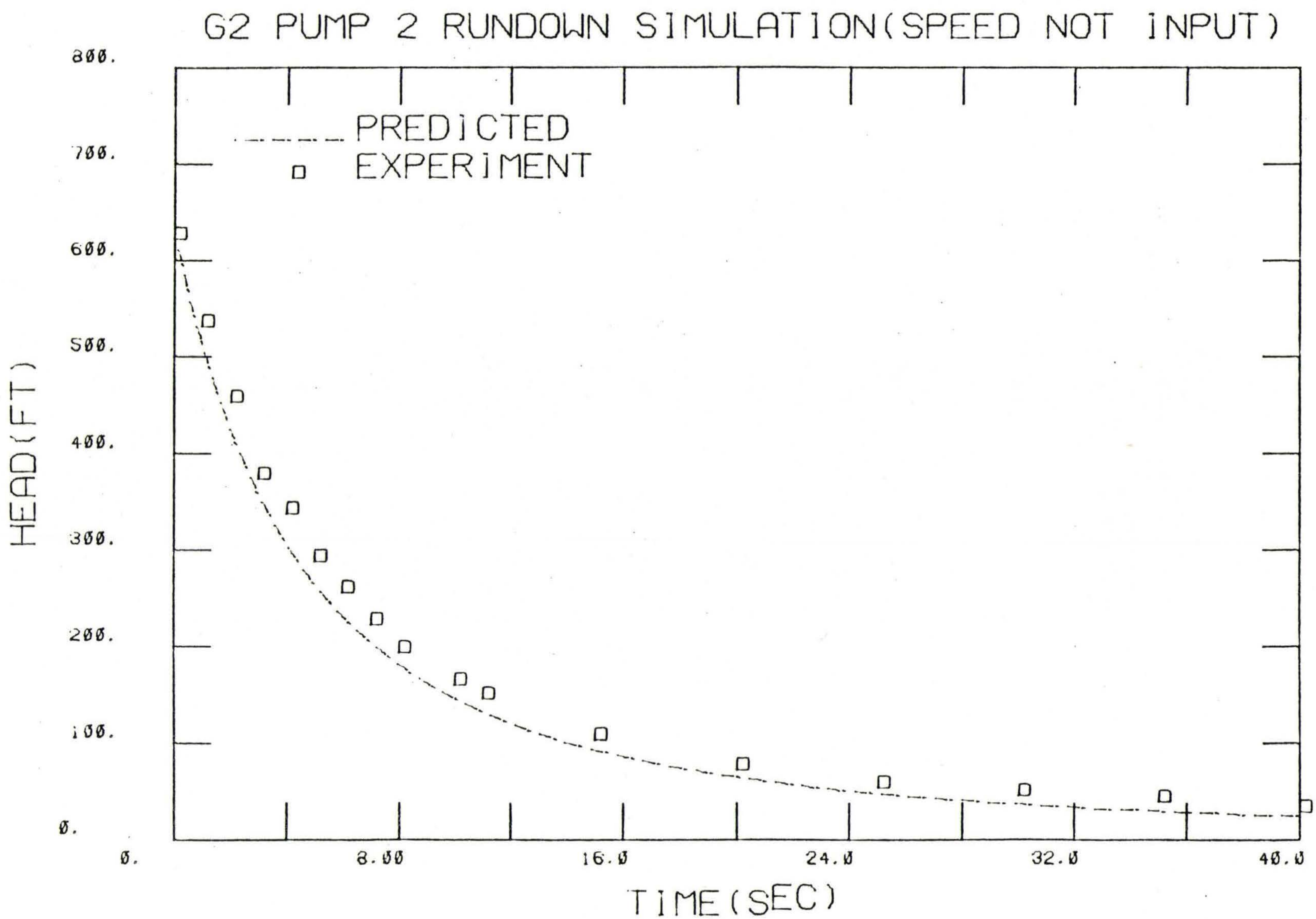
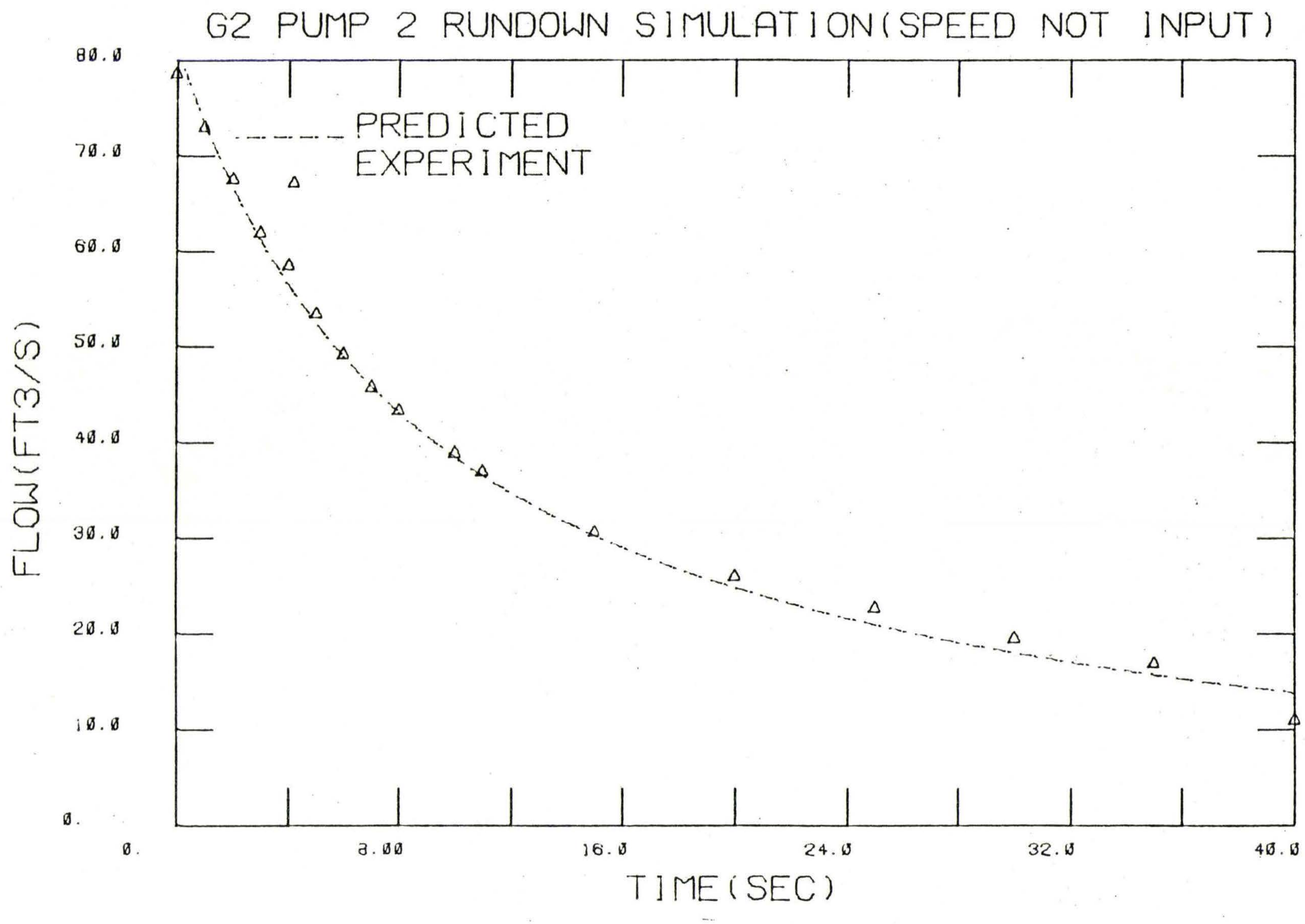

G2 PUMP 2 RUNDOWN SIMULATION(SPEED NOT INPUT)
PREDICTED
EXPERIMENT
SPEED(RPM)
TIME(SEC)
.180E+04
.160E+04
.140E+04
.120E+04
.100E+04
800.
600.
400.
200.
0.
0.
8.00
16.0
24.0
32.0
40.0

FIGURE 32



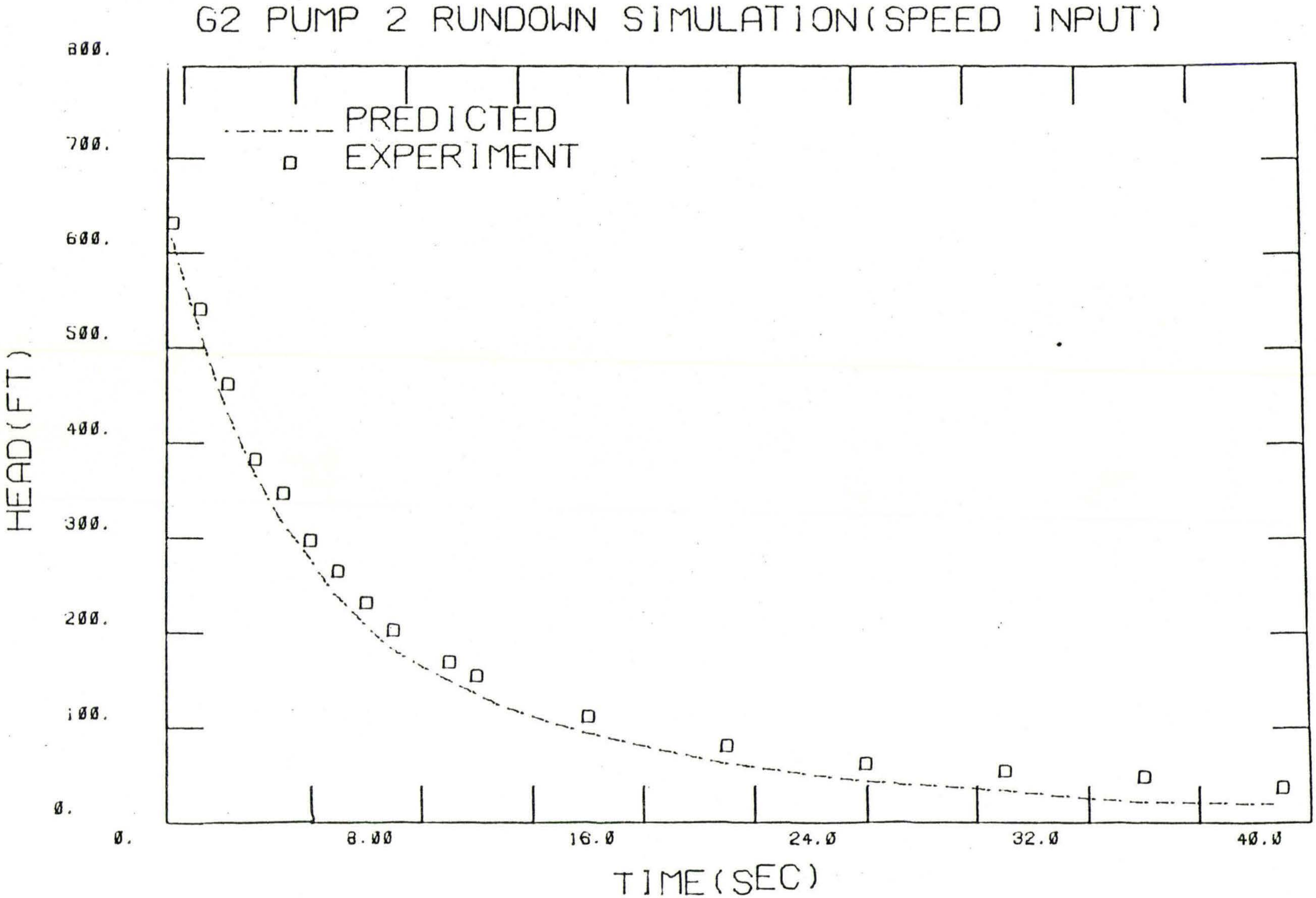
G2 PUMP 2 RUNDOWN SIMULATION(SPEED NOT INPUT)
PREDICTED
EXPERIMENT
HEAD(FT)
800.
700.
600.
500.
400.
300.
200.
100.
0.
0.
8.00
16.0
24.0
32.0
40.0
TIME(SEC)

FIGURE 33



G2 PUMP 2 RUNDOWN SIMULATION(SPEED NOT INPUT)
PREDICTED
EXPERIMENT
FLOW(FT3/S)
80.0
70.0
60.0
50.0
40.0
30.0
20.0
10.0
0.
0.
8.00
16.0
24.0
32.0
40.0
TIME(SEC)

FIGURE 34


G2 PUMP 2 RUNDOWN SIMULATION(SPEED INPUT)
PREDICTED
EXPERIMENT
HEAD(FT)
800.
700.
600.
500.
400.
300.
200.
100.
0.
0.
8.00
16.0
24.0
32.0
40.0
TIME(SEC)

which the flow is increasingly underpredicted (Figure 35).

II) Pump Rated Conditions Set Equal to Design Rated Conditions

A) Pump 1 (initial conditions, head = 734 ft., flow = 72.11 ft^3/s orifice resistance = 474.02)

i) Head

The head originally is lower than the previous prediction, however, it follows the previous prediction exactly after fourteen seconds (Figure 36).

ii) Speed

The original prediction is followed for the first ten seconds after which the speed is slightly higher (Figure 37).

iii) Flow

There is no difference from the previous prediction (Figure 38).

B) Pump 2 (initial conditions, head = 698 ft., flow = 79.9 ft^3/s, orifice resistance = 304.22)

i) Head

The head is initially well above the previous prediction and gradually approaches it (Figure 39).

ii) Speed

There is no difference from the previous prediction (Figure 40).

iii) Flow

There is no difference from the previous prediction (Figure 41).

FIGURE 35

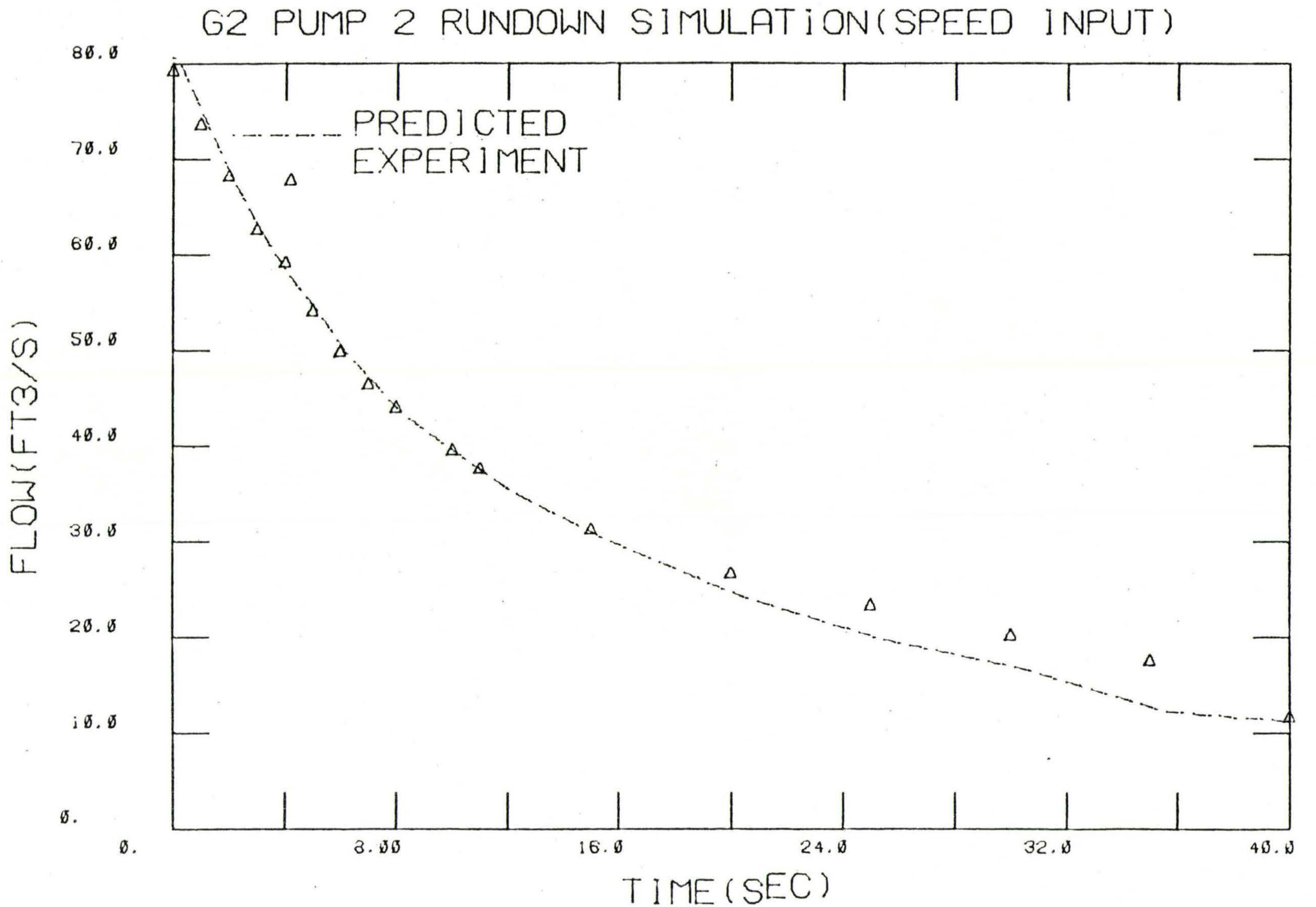

G2 PUMP 2 RUNDOWN SIMULATION(SPEED INPUT)
PREDICTED
EXPERIMENT
FLOW(FT3/S)
TIME(SEC)
80.0
70.0
60.0
50.0
40.0
30.0
20.0
10.0
0.
0.
8.00
16.0
24.0
32.0
40.0

FIGURE 36

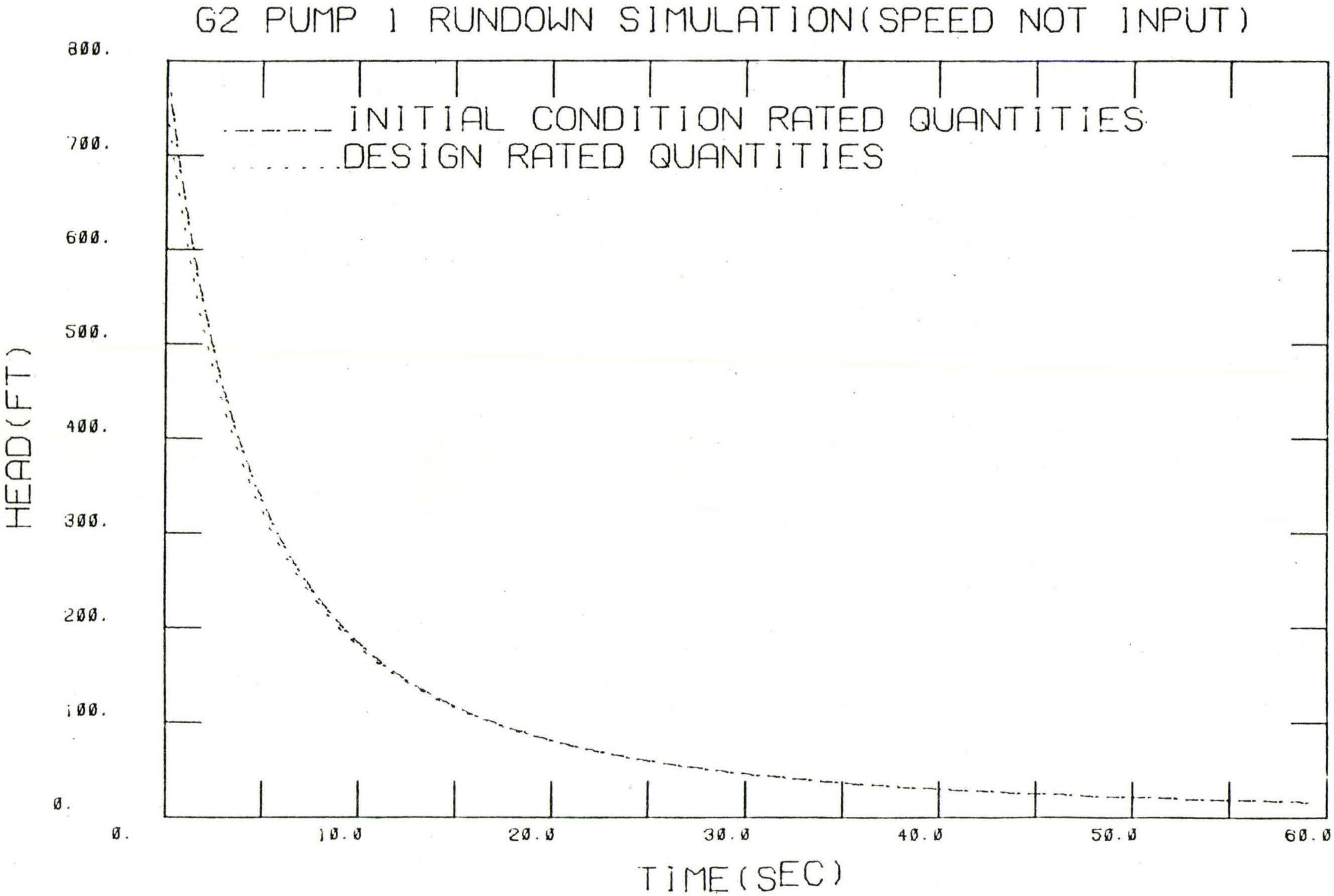
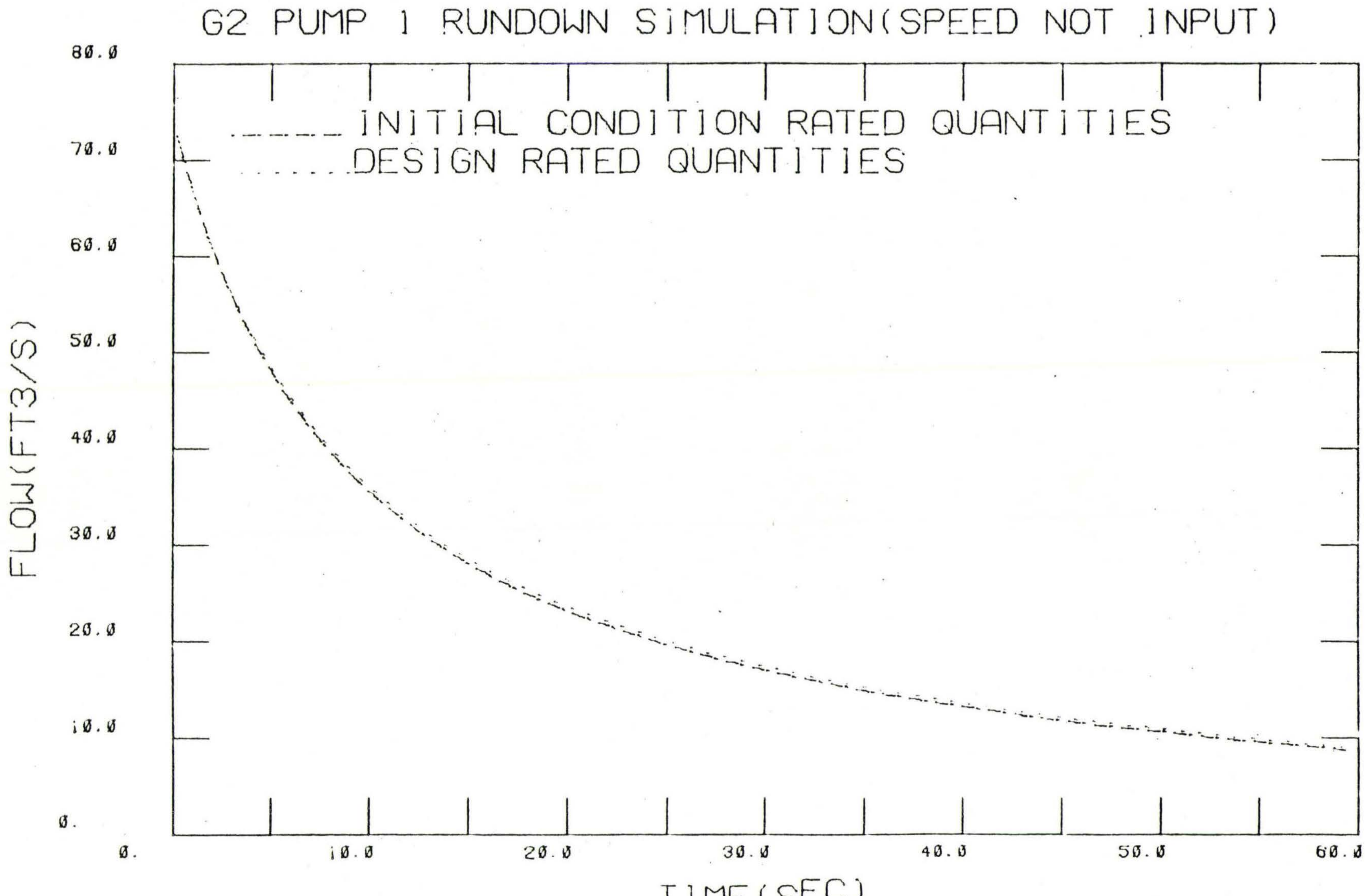
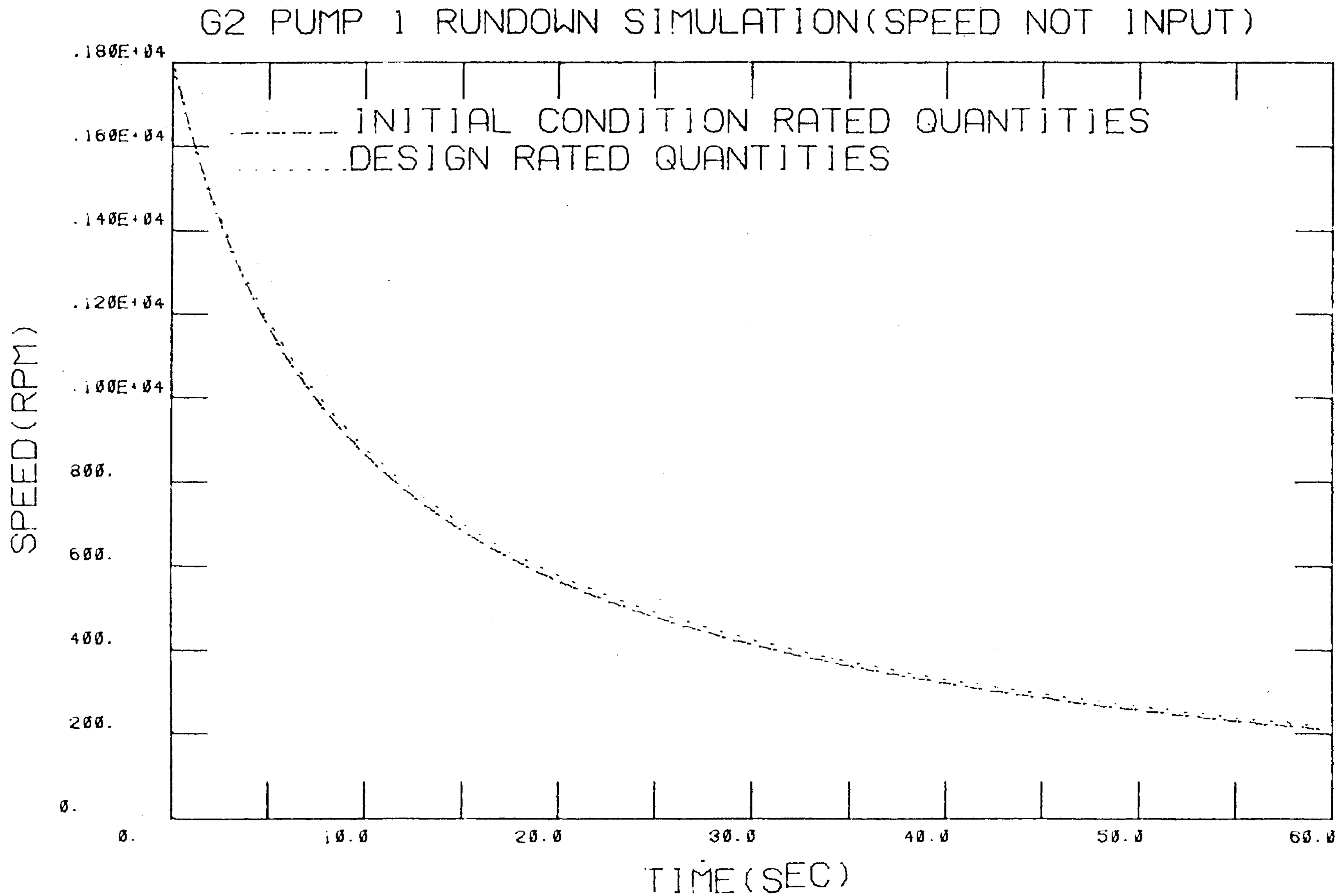

G2 PUMP 1 RUNDOWN SIMULATION(SPEED NOT INPUT)
INITIAL CONDITION RATED QUANTITIES
DESIGN RATED QUANTITIES
HEAD(FT)
TIME(SEC)
800.
700.
600.
500.
400.
300.
200.
100.
0.
0.
10.0
20.0
30.0
40.0
50.0
60.0

FIGURE 37



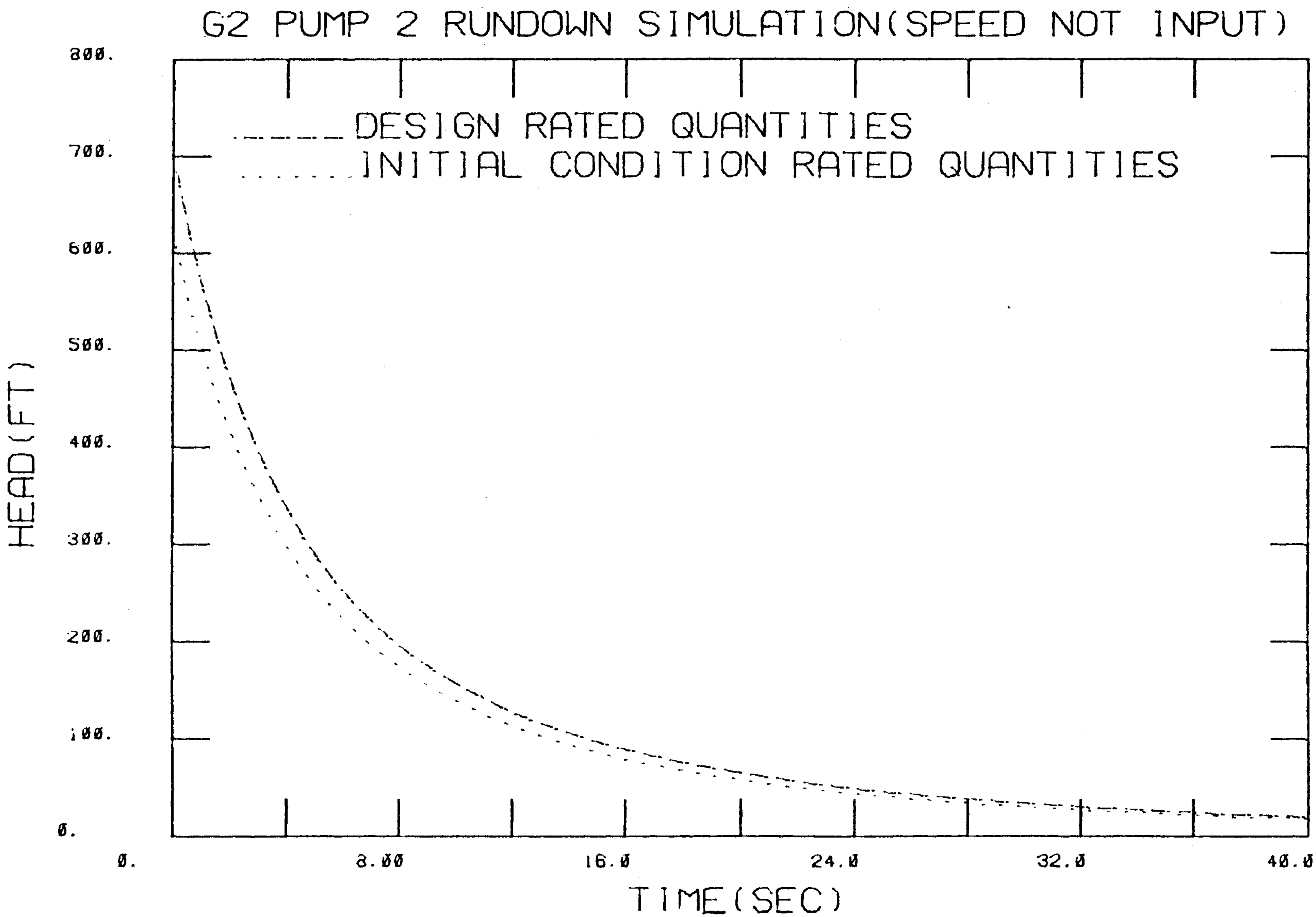
G2 PUMP 1 RUNDOWN SIMULATION(SPEED NOT INPUT)
INITIAL CONDITION RATED QUANTITIES
DESIGN RATED QUANTITIES
FLOW(FT3/S)
80.0
70.0
60.0
50.0
40.0
30.0
20.0
10.0
0.
0.
10.0
20.0
30.0
40.0
50.0
60.0
TIME(SEC)

FIGURE 38



G2 PUMP 1 RUNDOWN SIMULATION(SPEED NOT INPUT)
INITIAL CONDITION RATED QUANTITIES
DESIGN RATED QUANTITIES
.180E+04
.160E+04
.140E+04
.120E+04
.100E+04
800.
600.
400.
200.
0.
SPEED(RPM)
0.
10.0
20.0
30.0
40.0
50.0
60.0
TIME(SEC)

FIGURE 39



G2 PUMP 2 RUNDOWN SIMULATION(SPEED NOT INPUT)
DESIGN RATED QUANTITIES
INITIAL CONDITION RATED QUANTITIES
HEAD(FT)
800.
700.
600.
500.
400.
300.
200.
100.
0.
TIME(SEC)
0.
8.00
16.0
24.0
32.0
40.0

FIGURE 40

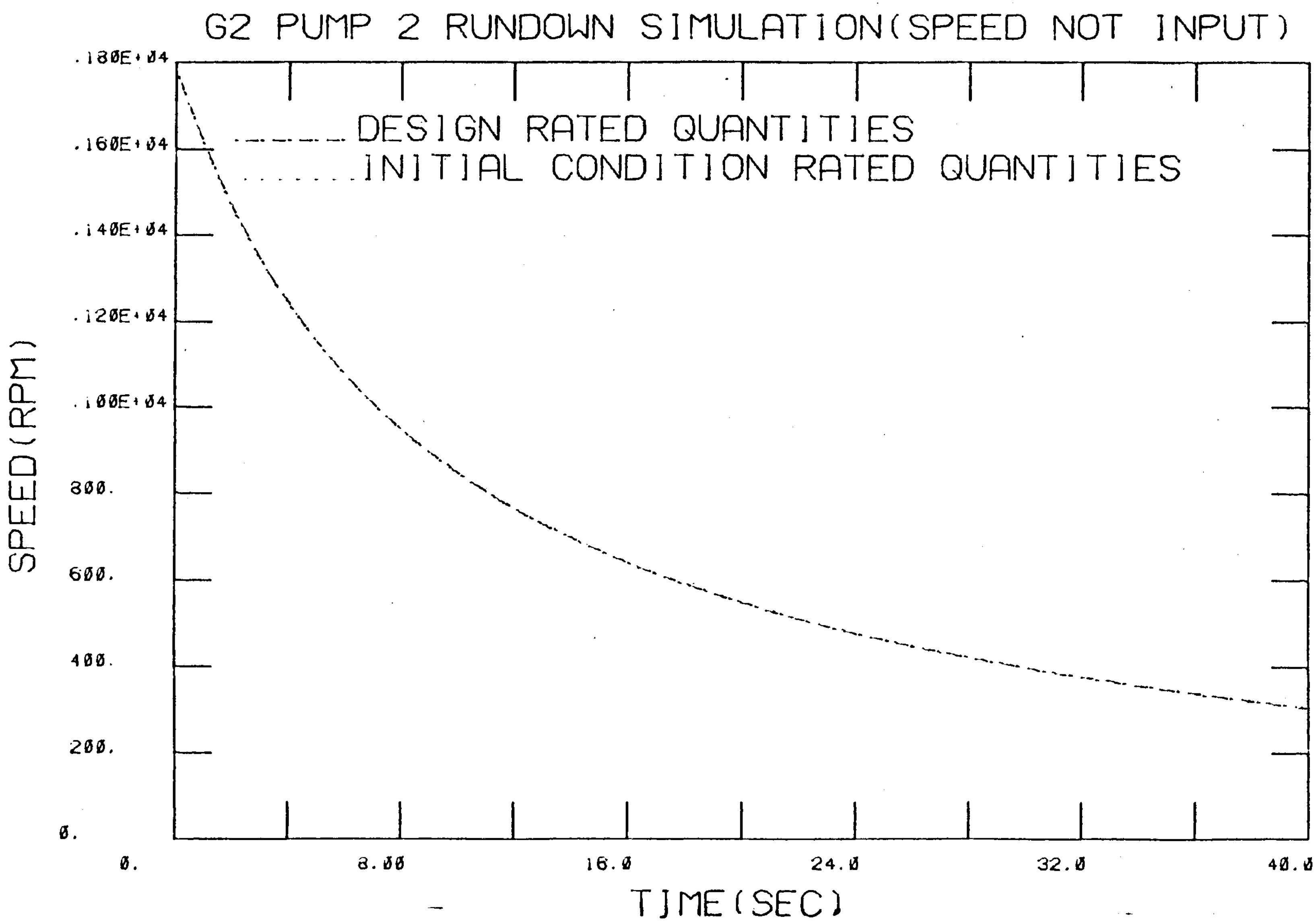
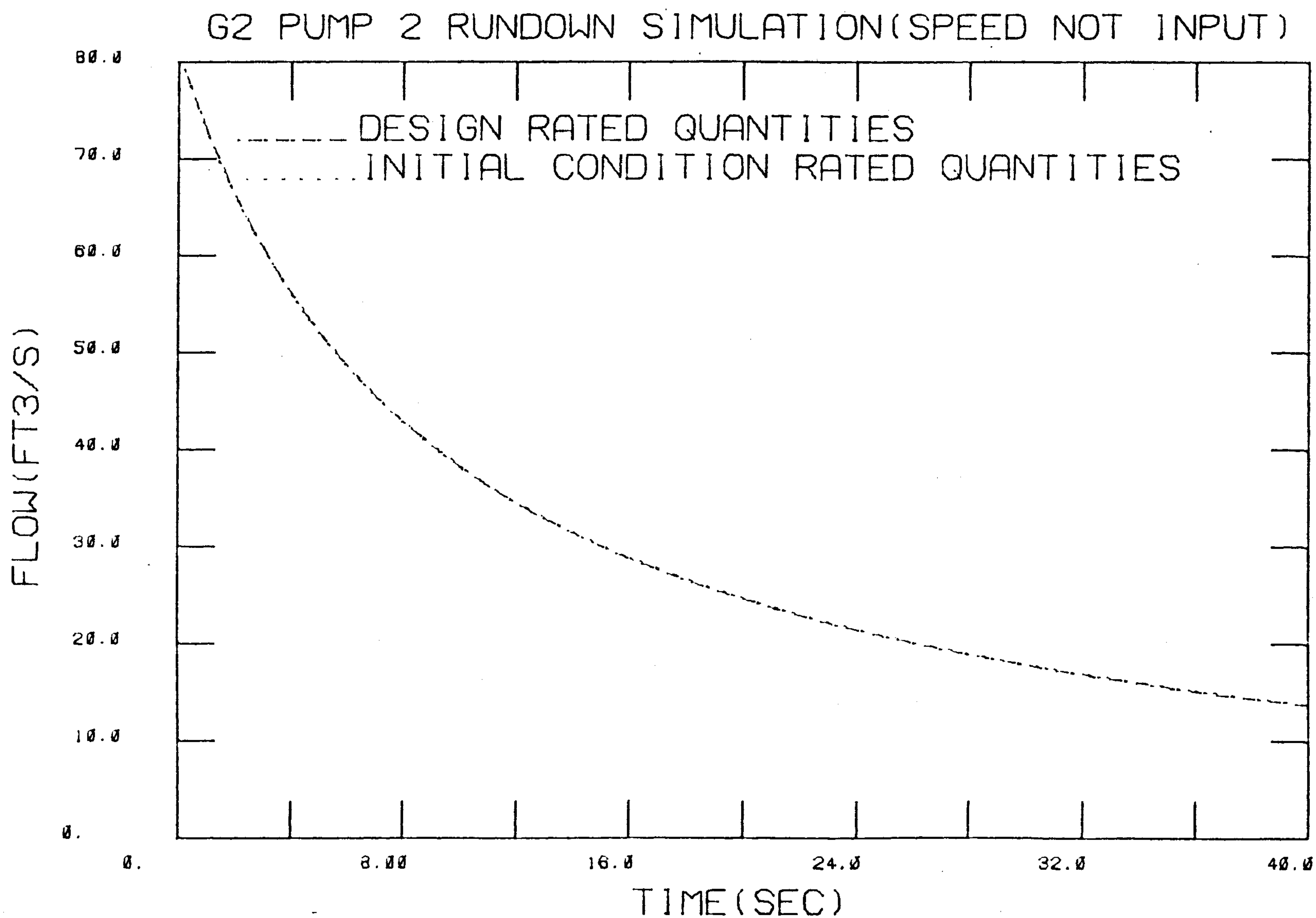

G2 PUMP 2 RUNDOWN SIMULATION(SPEED NOT INPUT)
DESIGN RATED QUANTITIES
INITIAL CONDITION RATED QUANTITIES
.180E+04
.160E+04
.140E+04
.120E+04
.100E+04
800.
600.
400.
200.
0.
SPEED(RPM)
0.
8.00
16.0
24.0
32.0
40.0
TIME(SEC)

FIGURE 41


G2 PUMP 2 RUNDOWN SIMULATION(SPEED NOT INPUT)
DESIGN RATED QUANTITIES
INITIAL CONDITION RATED QUANTITIES
FLOW(FT3/S)
80.0
70.0
60.0
50.0
40.0
30.0
20.0
10.0
0.
0.
8.00
16.0
24.0
32.0
40.0
TIME(SEC)

III) Pump Rated Torque Changed to 5.0 MW (design rated head and flow)

This is only done for pump 1 (initial conditions, head = 734 ft., flow = 72.1 ft^3/s, orifice resistance = 474.02).

i) Head

The head is slightly less than the design rated head and flow case from two to thirty-six seconds. Otherwise the head agrees fairly well (Figure 42).

ii) Speed

The speed is slightly less than the previous case. A plot is shown in Figure 43. The flow shows a similar discrepancy.

iii) Flow

The flow is slightly less than the previous case (Figure 44).

IV) Impeller Inertia Changed to 900.0 lb-ft-s^2

This case is similar to the above case, although the effect is slightly less. An example is down in Figure 45.

V) Pump Model Predictions of Head Using Experimental Flow and Speed as Input

The numerical results are shown in Table 5. Only a summary of these are given here.

Pump 1

Overpredicted in the first 8 seconds, then underpredicted.

Pump 2

Underpredicted, the error increasing with time.

FIGURE 42

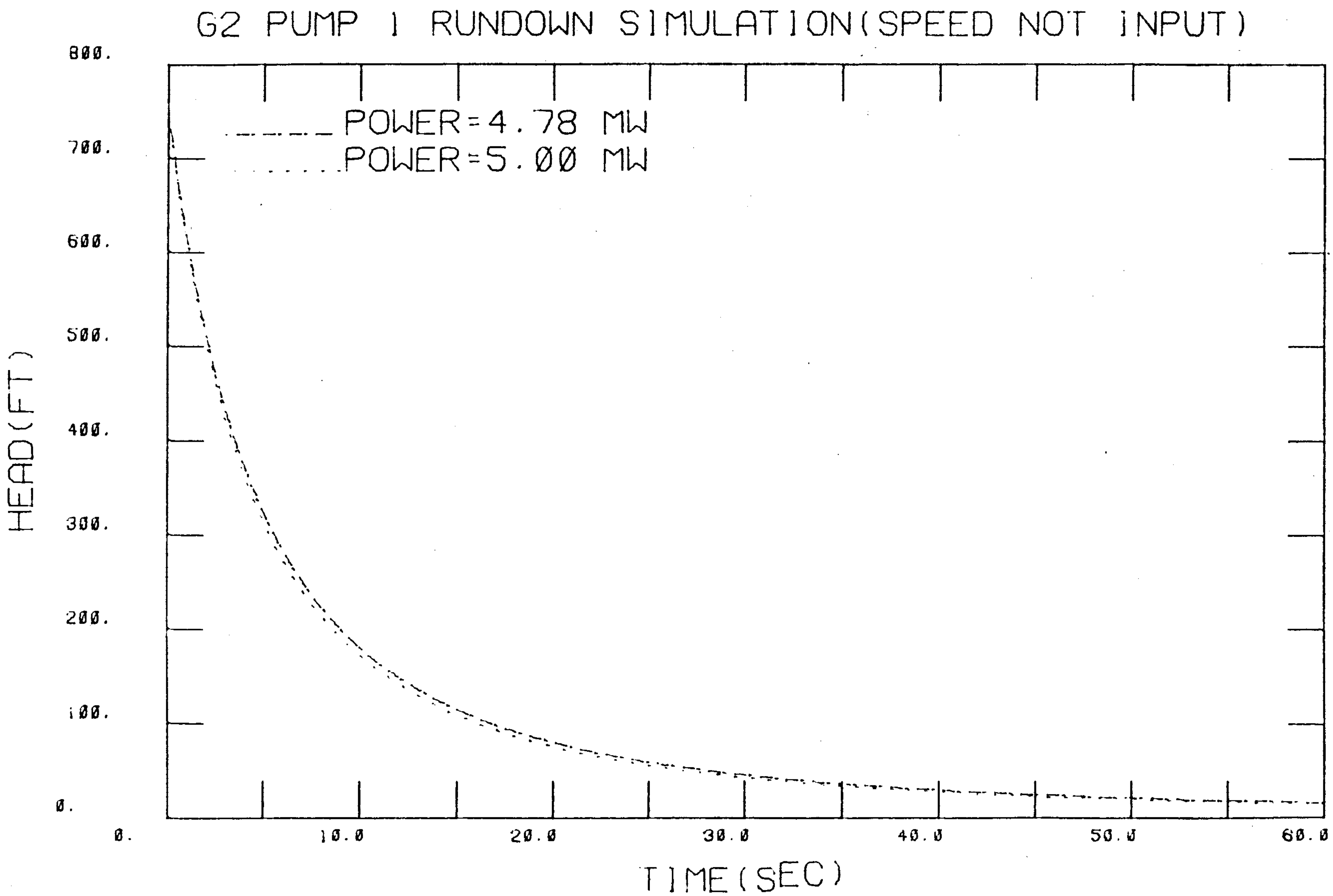

G2 PUMP 1 RUNDOWN SIMULATION(SPEED NOT INPUT)
POWER=4.78 MW
POWER=5.00 MW
HEAD(FT)
800.
700.
600.
500.
400.
300.
200.
100.
0.
0.
10.0
20.0
30.0
40.0
50.0
60.0
TIME(SEC)

FIGURE 43

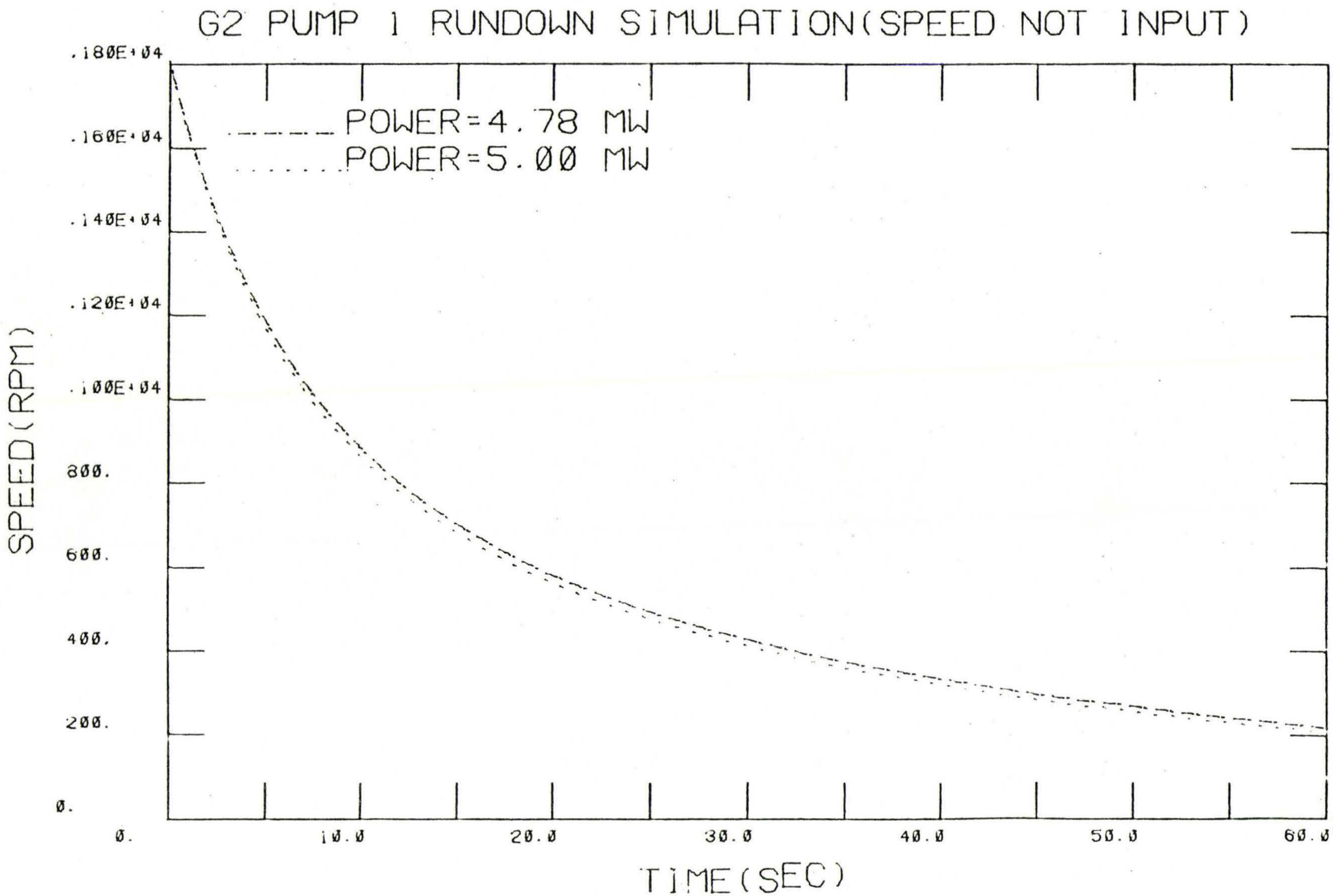

G2 PUMP 1 RUNDOWN SIMULATION(SPEED NOT INPUT)
POWER=4.78 MW
POWER=5.00 MW
SPEED(RPM)
.180E+04
.160E+04
.140E+04
.120E+04
.100E+04
800.
600.
400.
200.
0.
0.
10.0
20.0
30.0
40.0
50.0
60.0
TIME(SEC)

FIGURE 44

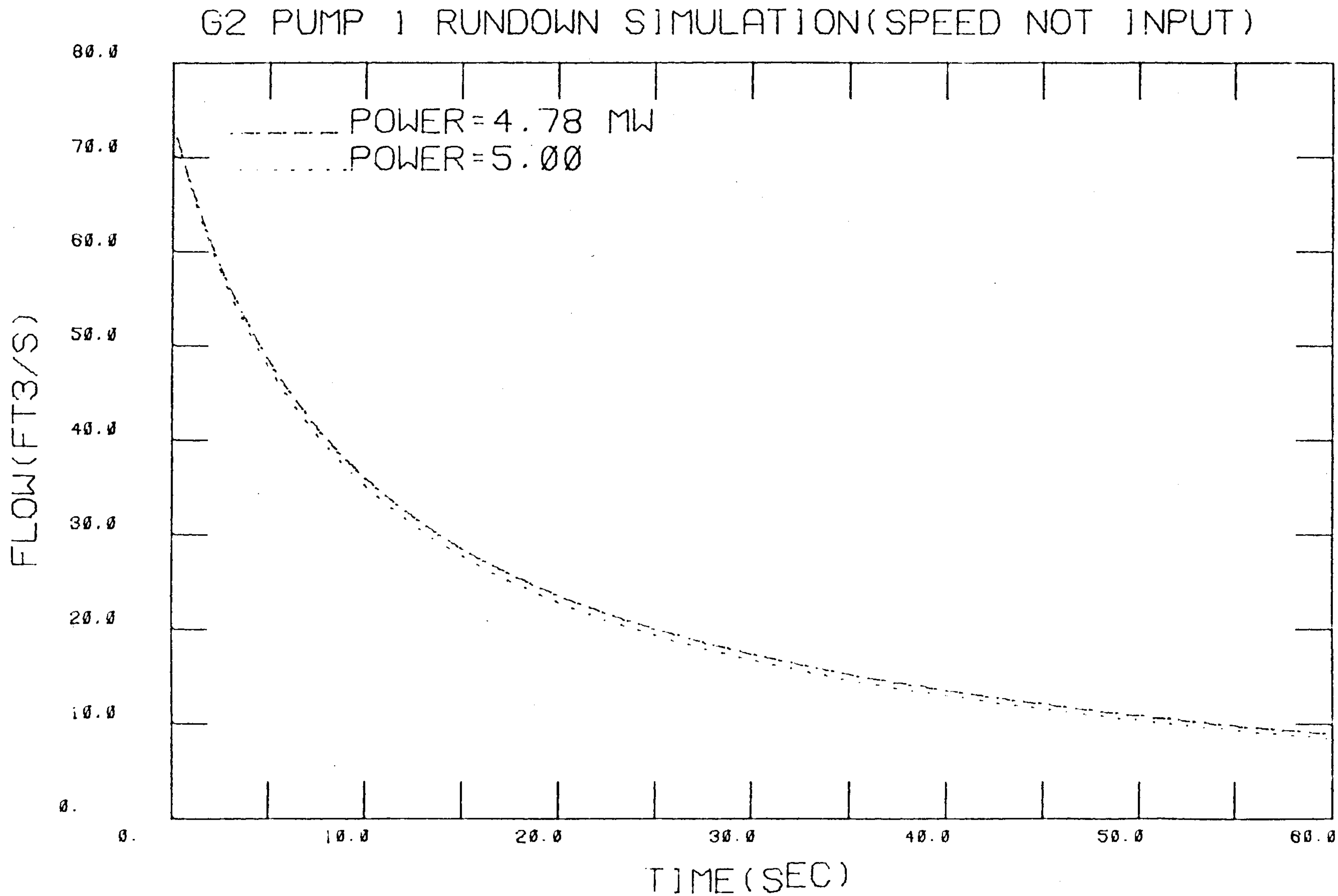

G2 PUMP 1 RUNDOWN SIMULATION(SPEED NOT INPUT)
POWER=4.78 MW
POWER=5.00
FLOW(FT3/S)
TIME(SEC)
0.
10.0
20.0
30.0
40.0
50.0
60.0
70.0
80.0

FIGURE 45

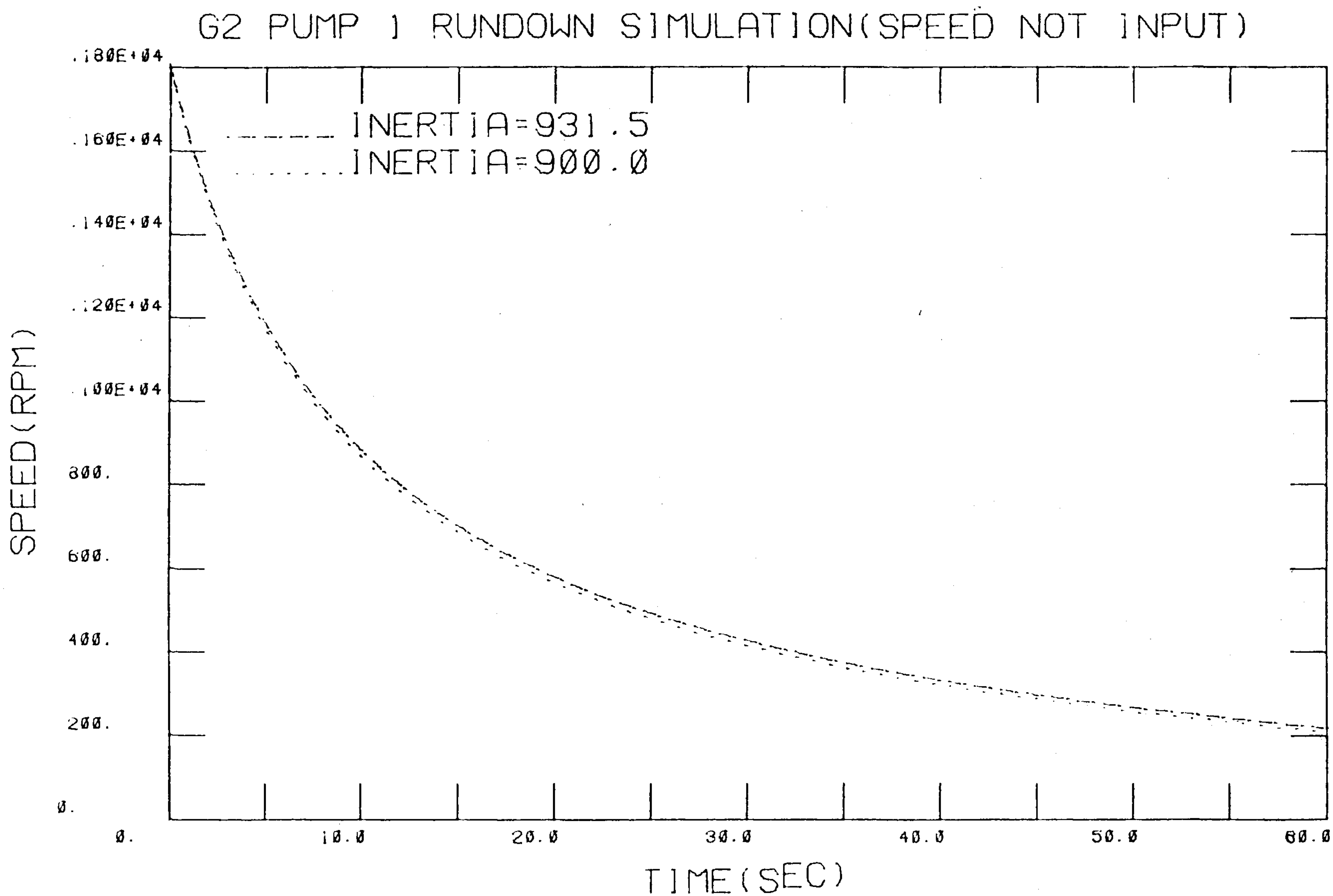

G2 PUMP 1 RUNDOWN SIMULATION(SPEED NOT INPUT)
INERTIA=931.5
INERTIA=900.0
.180E+04
.160E+04
.140E+04
.120E+04
.100E+04
800.
600.
400.
200.
0.
SPEED(RPM)
0.
10.0
20.0
30.0
40.0
50.0
60.0
TIME(SEC)

TABLE 5

COMPARISON OF CALCULATED HEAD WITH EXPERIMENTAL HEAD USING EXPERIMENTAL SPEED AND FLOW AS INPUT

TIME (s)	ν/Ω	h/Ω^2 (calculated)	h/Ω^2 (experimental)	HEAD(M) (calculated)	HEAD(M) (experimental)
Pump 1					
0.0	1.000	1.000	1.000	237.5	237.5
1.0	.963	1.018	.965	223.3	211.5
2.0	.943	1.029	.966	192.8	181.0
3.0	.932	1.034	.973	169.0	159.0
4.0	.937	1.032	.999	145.6	141.0
5.0	.945	1.027	1.005	126.7	124.0
6.0	.943	1.029	.981	111.6	106.0
7.0	.955	1.022	.995	100.2	97.5
8.0	.965	1.017	1.016	87.1	87.0
9.0	.947	1.026	1.014	79.9	79.0
10.0	.965	1.018	1.008	71.7	71.0
11.0	.951	1.024	1.048	65.0	66.5
13.0	.971	1.015	1.150	51.2	58.0
15.0	.961	1.019	1.206	44.8	53.0
17.0	.959	1.021	1.188	38.7	45.0
19.0	.986	1.007	1.318	30.6	40.0
21.0	.985	1.008	1.305	27.8	36.0
23.0	.988	1.006	1.310	23.8	31.0
25.0	.966	1.017	1.342	22.0	29.0
30.0	.926	1.037	1.439	16.6	23.0
35.0	.888	1.056	1.445	13.2	18.0
40.0	.843	1.079	1.562	10.4	15.0
45.0	.975	1.013	1.885	7.0	13.0
60.0	.896	1.052	2.754	3.4	9.0

TABLE 5 (cont'd)

TIME (s)	ν/Ω	h/Ω^2 (calculated)	h/Ω^2 (experimental)	HEAD (M) (calculated)	HEAD (M) (experimental)
Pump 2					
0.0	1.000	1.000	1.000	188.0	188.0
1.0	1.011	.993	1.008	157.6	160.0
2.0	1.028	.981	1.030	129.6	136.0
3.0	1.025	.983	1.002	109.2	112.0
4.0	1.066	.956	1.059	91.2	101.0
5.0	1.022	.985	1.026	82.8	86.0
6.0	1.025	.983	1.070	69.8	76.0
7.0	1.034	.977	1.093	59.1	66.0
8.0	.999	1.000	.981	58.1	57.0
10.0	1.041	.973	1.083	42.2	47.0
11.0	1.043	.971	1.104	37.4	42.5
15.0	1.071	.952	1.155	24.3	29.5
20.0	1.147	.902	1.214	14.9	20.0
25.0	1.272	.817	1.336	8.9	14.5
30.0	1.289	.806	1.578	6.1	12.0
35.0	1.372	.750	2.026	3.7	10.0
Pump 3					
0.0	1.000	1.000	1.000	180.0	180.0
1.0	1.038	.975	1.018	141.8	148.0
2.0	1.054	.964	.981	124.8	127.0
3.0	.971	1.014	.974	114.6	110.0
4.0	.980	1.010	.985	98.5	96.0
5.0	.990	1.005	.939	83.6	82.0
6.0	.996	1.002	1.016	74.0	75.0
7.0	1.002	.998	.953	64.9	62.0
8.0	1.004	.997	.968	56.7	55.0
9.0	.979	1.011	.946	51.8	48.5
10.0	.995	1.002	.967	46.1	44.5
15.0	.919	1.040	1.019	30.6	30.0
20.0	.899	1.050	1.011	20.8	20.0
25.0	.947	1.026	.951	14.6	13.5

TABLE 5 (Cont'd)

TIME (s)	ν/Ω	h/Ω^2 (calculated)	h/Ω^2 (experimental)	HEAD (M) (calculated)	HEAD (M) (experimental)
Pump 3 (Cont'd)					
30.0	1.098	.934	1.045	8.5	9.5
35.0	.750	1.125	.862	7.9	6.0
Pump 4					
0.0	1.000	1.000	1.001	216.0	216.0
1.0	1.010	.993	1.003	182.3	184.0
2.0	.999	1.000	.985	159.9	157.5
3.0	.990	1.005	.993	137.6	136.0
4.0	1.028	.981	1.029	113.5	119.0
5.0	1.021	.986	1.068	99.8	108.0
6.0	1.018	.988	1.088	88.1	97.0
7.0	1.000	1.000	1.096	79.4	87.0
8.0	1.001	1.000	1.084	70.1	76.0
9.0	.986	1.007	1.014	64.5	65.0
10.0	.982	1.009	1.052	57.6	60.0
15.0	1.017	.988	1.058	35.5	38.0
20.0	1.161	.892	1.191	20.2	27.0
25.0	1.221	.851	1.297	13.8	21.0
30.0	1.296	.801	1.112	8.6	12.0

Pump 3

Very good agreement.

Pump 4

Good agreement for the first three seconds, then the head is underpredicted.

4.2 Locked Impeller Results

I) Comparison with Experiment

Data is taken from Figure 7.

$$K = 2gA^2 \frac{H^2}{Q^2}$$

$Q\ (ft^3/s)$	$H\ (ft)$	K
21.40	-28	-14.07
26.76	-43	-13.82
40.13	-104	-14.86
53.52	-180	-14.46
	Average =	-14.30

From Table 1 for forward flows:

$$C = -.55$$

$$K = -.55(2 \times 32.2 \times 1.89^2)\ \frac{705}{78.68^2} = -14.41$$

II) Comparison of Single Phase and Fully Degraded Resistance

Pump	Flow Direction	Phase	$h/\nu^2 = C$
EPRI/CREARE	Forward	Single phase	-1.30
		Fully degraded	<-1.30
	Reverse	Single phase	1.4
		Fully degraded	1.1

Pump	Flow Direction	Phase	$h/\nu^2 = C$
C-E/EPRI	Forward	Single phase	-1.50
		Fully degraded	-1.54
	Reverse	Single phase	1.28
		Fully degraded	1.55
ANC	Forward	Single phase	-.36
		Fully degraded	-.55
	Reverse	Single phase	.72
		Fully degraded	.84
MODEL	Forward	Single phase	-.7
		Fully degraded	-.56
	Reverse	Single phase	1.96
		Fully degraded	2.00

The MODEL pump values are derived from the following points.

Forward Flows

a) Single Phase

ν/Ω	h/Ω^2
2.5	-1.1
3.0	-2.15
3.5	-3.65

$$h/\Omega^2 = -.5 + 1.55\ (\nu/\Omega) - 0.7\ (\nu/\Omega)^2$$

b) Fully Degraded

ν/Ω	h/Ω^2
3.0	-5.4
4.7	-15.0
6.0	-25.0

$$h/\Omega^2 = 4.11 - 1.49\ (\nu/\Omega) - 0.56\ (\nu/\Omega)^2$$

<u>Reverse Flows</u>

a) <u>Single Phase</u>

ν/Ω	h/Ω^2
-2.5	19.3
-4.1	35.8
-5.0	49.5

$$h/\Omega^2 = 13.6 + 2.62\ (\nu/\Omega) + 19.6\ (\nu/\Omega)^2$$

b) <u>Fully Degraded</u>

ν/Ω	h/Ω^2
-4.0	46.5
-5.0	67.0
-6.0	91.5

$$h/\Omega^2 = 4.5 - 2.5\ (\nu/\Omega) + 2.0\ (\nu/\Omega)^2$$

5.0 DISCUSSION

Agreement between the code prediction and experiment for the rundown simulation is generally good for the first few seconds, then an overall tendency to underestimate the experimental values is shown. Those few cases where good agreement did occur could be merely due to a fortuitous cancellation of errors. This only happens with one quantity at the most for each case.

The tremendous discrepancy between experimental and predicted rundown times to full stop indicates that quantities at low flows are continually underestimated. This would have a cumulative effect resulting in the large rundown time difference.

Another contributing effect may be the mechanical pump losses. Although a fractional value of .01 was used, it is not known how valid this is but this arbitrary value is normally used in FIREBIRD. This affects the friction torque on the pump.

The above points indicate that perhaps the pumps did not have the same characteristic curve as the pump used to make the FIREBIRD-III pump model, so the available data points were used to construct a homologous head-capacity curve for pumps 1 and 2 (Figures 46 and 47). Unfortunately almost all the data is clustered around the point (1.0,1.0) so that an idea of the shape of the curve is impossible to obtain. However, it was found that the data was fairly near the theoretical curve at high flows, but was well above the curve at low flows. The discrepancy increased as the flow decreased, indicating that the scaling laws are not followed at low flows.

Problems with the interpretation of results occur from lack of information about the test loop. This is due to experimental data being obtained from pump commissioning tests as this is the only data available for full-size PHT pumps. For example, the orifice resistance required to achieve initial conditions were 501.8 and 304.2 for pumps 1 and 2 respectively. This could indicate the effect of temperature/

FIGURE 46

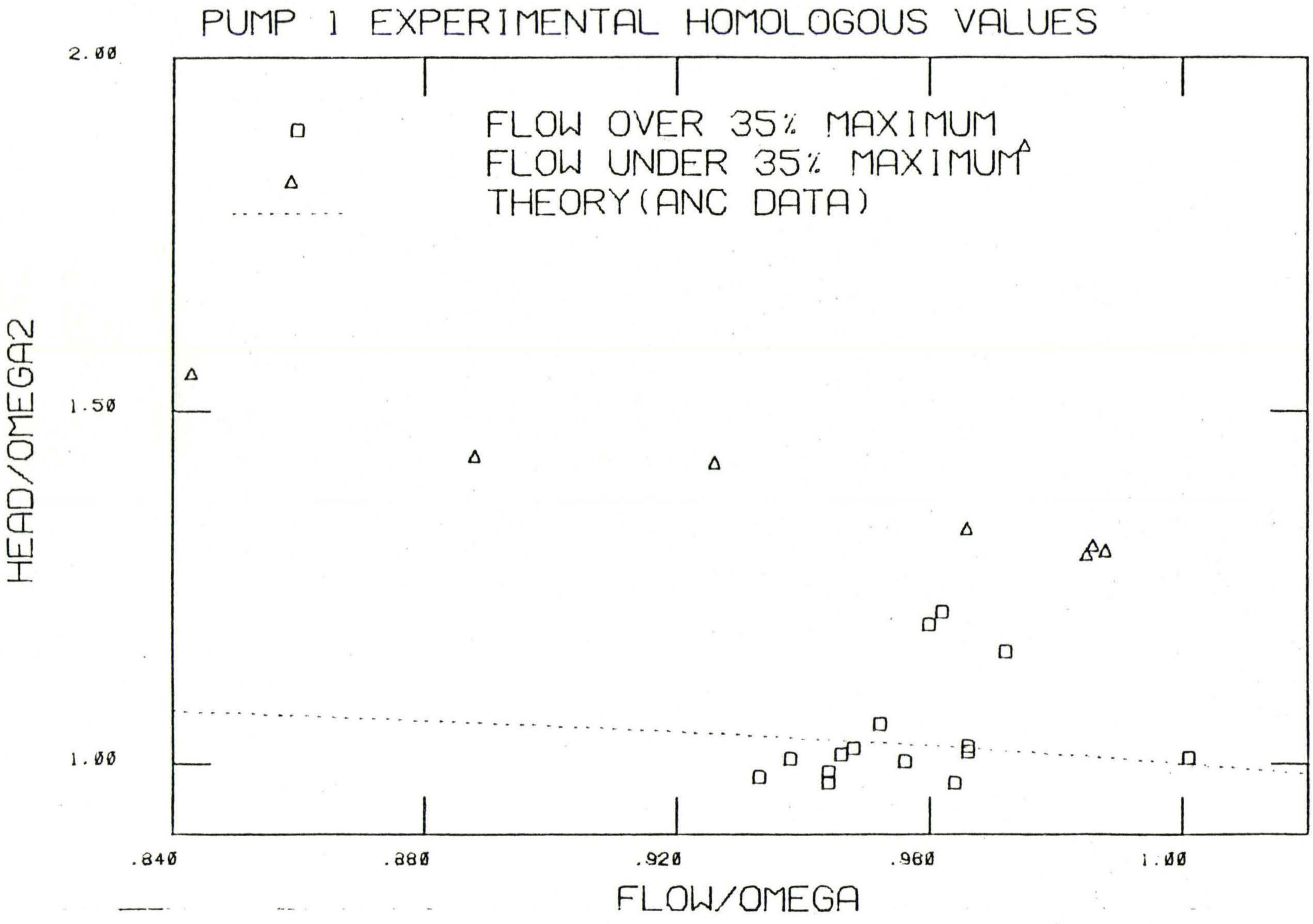
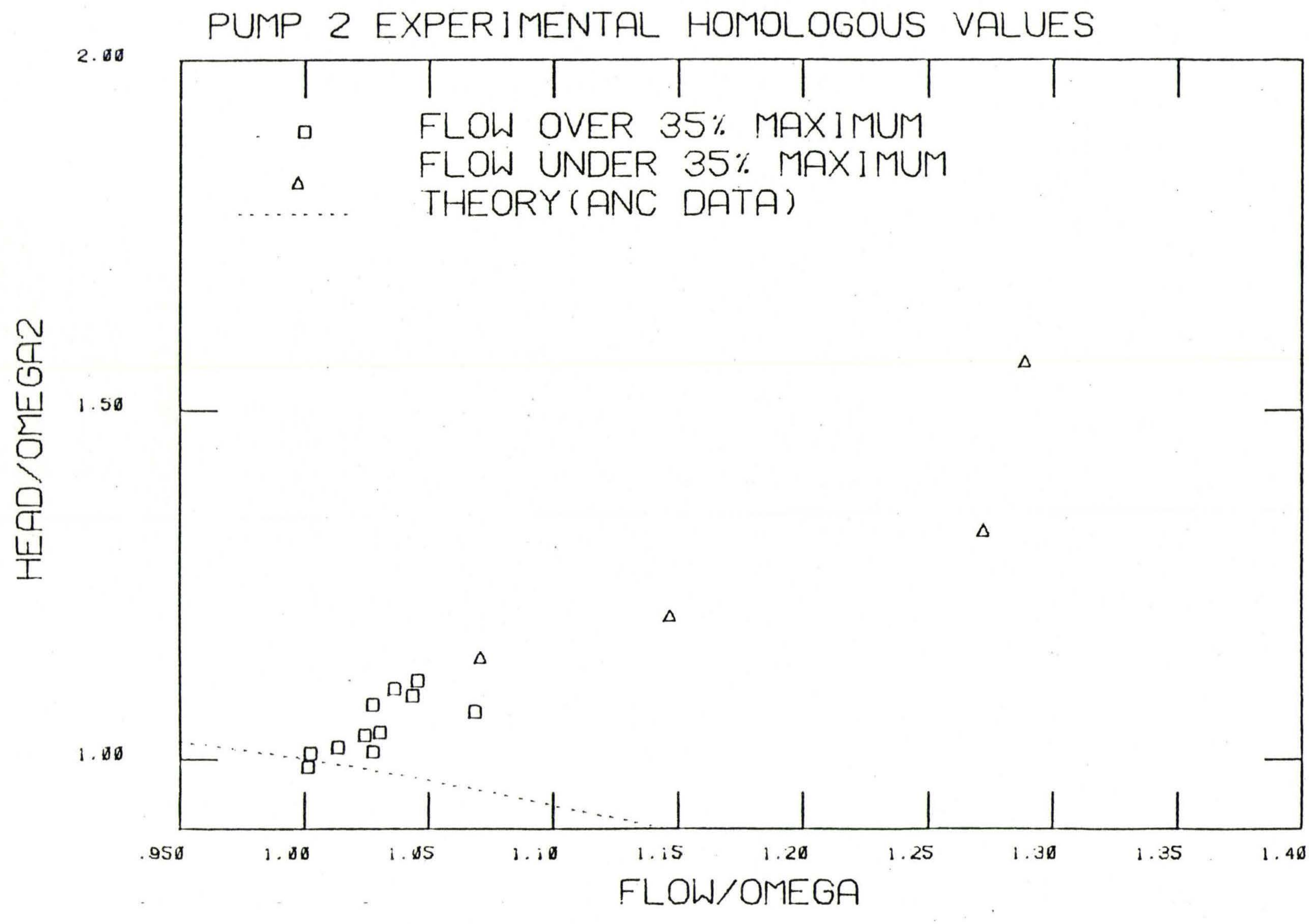

PUMP 1 EXPERIMENTAL HOMOLOGOUS VALUES
FLOW OVER 35% MAXIMUM
FLOW UNDER 35% MAXIMUM
THEORY(ANC DATA)
HEAD/OMEGA2
2.00
1.50
1.00
.840
.880
.920
.960
1.00
FLOW/OMEGA

FIGURE 47


PUMP 2 EXPERIMENTAL HOMOLOGOUS VALUES
FLOW OVER 35% MAXIMUM
FLOW UNDER 35% MAXIMUM
THEORY(ANC DATA)
2.00
1.50
1.00
HEAD/OMEGA2
.950
1.00
1.05
1.10
1.15
1.20
1.25
1.30
1.35
1.40
FLOW/OMEGA

pressure control or a markedly different characteristic curve.

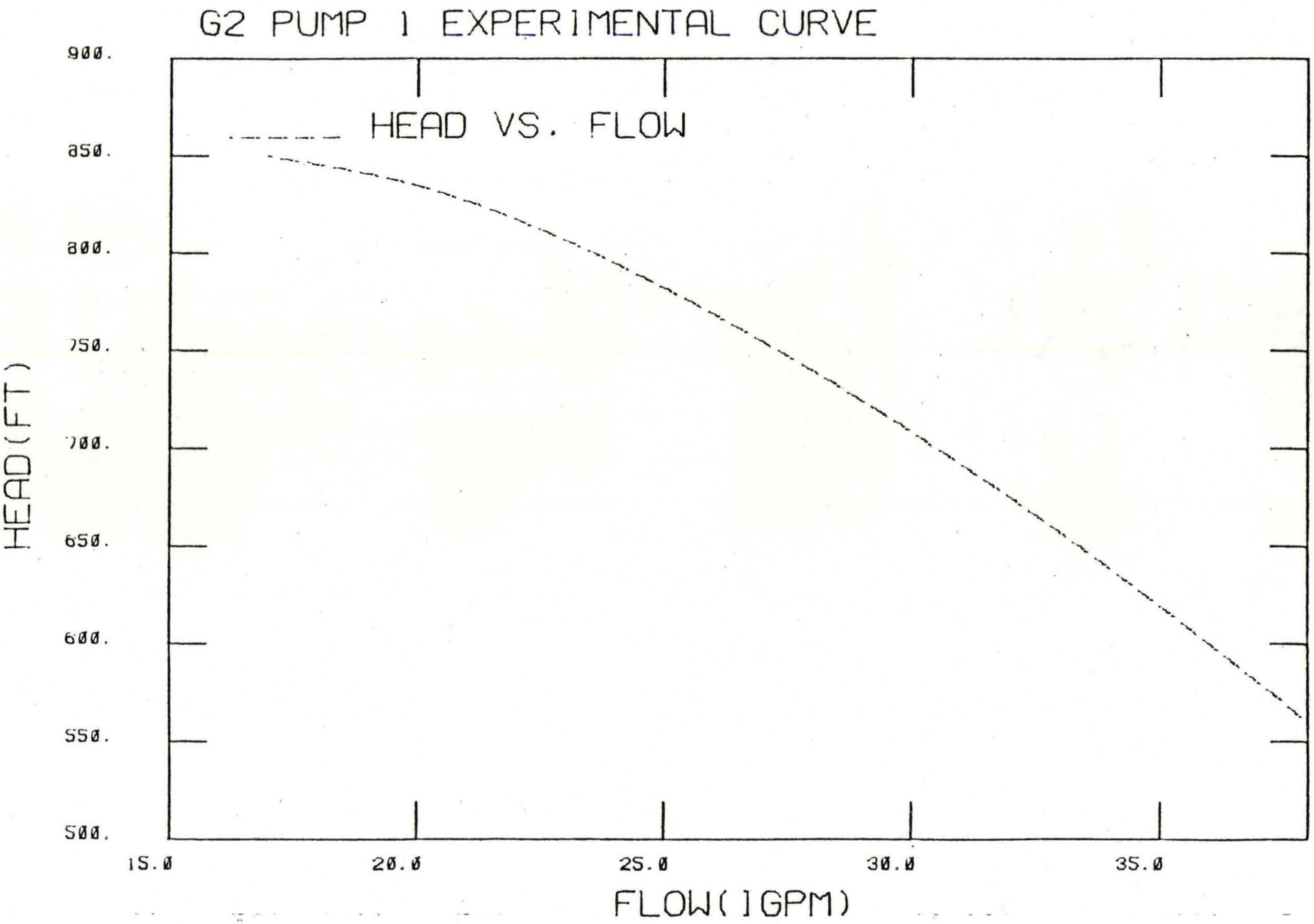
It is not certain whether the initial conditions are the rated conditions for that pump (the rated condition is the point of maximum efficiency). The head-capacity curve for pump 1 (Figure 48) showed that the pump would operate at nominal rated conditions, and that the D_2O simulation test from which point the pumps were run down was run near design rated conditions. However, the conditions at time zero not only have a large difference from rated conditions, they also do not lie on the head-capacity curve for pump 1.

No data on the suction temperature and pressure during the run is available so it is not known if the suction pressure is maintained or if the estimate of a very small temperature change during the run is valid because of the interaction of the air coolers. The suction varies considerably during a typical run, increasing to a value of above 1600 psig from its original value of 1363 psig. However, a rundown was made on pump 2 with the control valve left open so the suction pressure would remain constant. This run was in excellent agreement with the previous run so the suction pressure has little effect on the prediction.

To test the effect of various other changes on the predictions changes were made in various rated quantities. The rated head and flow was changed from the initial conditions to the design conditions as described in the method. This had an extremely small effect on the predictions as did similar runs with the rated torque and the moment of inertia slightly change. These results suggest that the pump model has a minor effect on the rundown and that its primary influence is the test loop and the large fluid inertia.

The program will give a maximum 20% error in head prediction for the range of values used here if both the flow and speed are in error by 5%. The major source of error is the speed, the flow

FIGURE 48


G2 PUMP 1 EXPERIMENTAL CURVE
HEAD VS. FLOW
HEAD(FT)
900.
850.
800.
750.
700.
650.
600.
550.
500.
15.0
20.0
25.0
30.0
35.0
FLOW(IGPM)

contributing only a 5% error. However, this is only a factor when both the flow and speed are calculated by the code so the head-flow curves for the speed input should have been closer if the model was correct. No information is available on the accuracy of the experimental data.

The zero speed experimental results show excellent agreement with the value now used as the coefficient of $(\nu/\Omega)^2$ for forward flow.

Comparison of fully degraded flow locked impeller had loss with single-phase head loss shows that the fully degraded head loss is consistently greater than the single-phase head loss. This is to be expected as some extra loss occurs due to separation of flow.

The two EPRI pumps with the same specific speed (3,840) show very good agreement between the homologous curves for both head and torque (Figures 15 and 16). This is to be expected from the discussion on single-phase scaling in the theory section. The ANC pump with a much lower specific speed has a completely different set of homologous curves, even in single phase.

Two-phase homologous curves were available only for the two EPRI pumps (Figures 17 and 18). They were both done at a void fraction of 40%, but at different pressures. The C-E/EPRI 1/5 scale pump data was obtained at 1000 psia while the EPRI/CREARE 1/20 scale pump data was obtained at under 500 psia.

The agreement here is not as good as it was for the single-phase curve. The 1/20 scale results are consistently lower than the 1/5 scale results, although the curves are of the same shape. An explanation for this can be found from the degradation curves of the 1/5 scale pump (Figures 19 - 22) and those of the ANC pump (Figures 23 - 25). These clearly show a dependence of head degradation on the absolute pressure at suction - the degradation decreasing as pressure increases. This decreased degradation at high pressures accounts for the difference in the two curves.

This implies that the absolute pressure as well as the specific speed is needed to scale pumps in two-phase flow. As well as the different shape of the degradation curve at $\nu/\Omega = 2.0$ implies that different degradation curves are needed for each quadrant. The curves in the first quadrant (see the ANC curves) have fairly constant shape throughout.

6.0 CONCLUSIONS

The single-phase head curve is reasonably accurate for high flows, but underestimate the head at low flows and low speed. The reason for this appears to be that the pump doesn't follow the scaling laws at low flows. The difference between the predicted and actual rundown times indicate either that the torque curve may be in error or that the mechanical losses are overestimated. Another contributing factor to this discrepancy is the underprediction of the head at low flows. This would stop the pump sooner than normal as the flow would be decreased.

As demonstrated by changing the various rated quantities, the prediction is not very sensitive to rated quantities. This means that the rundown is mainly governed by the loop. However, the head underprediction at low flows would have a major effect.

The coefficient of $(\nu/\Omega)^2$ for the forward flow parabolic extrapolation should be kept the same as it agrees extremely well with experimental results.

There is no justification for the assumption in the present PUMPANC subroutine that the fully degraded parabolic extrapolation should approach the single phase at low speeds. Unfortunately, no data exists as yet to enable more accurate values to be calculated.

In order to scale head and torque characteristics accurately between pumps of different sizes, it is necessary to have agreement between both the specific speeds and the head of the two-phase flow. This is due to the compression of the void by the head.

Absolute pressure has a significant effect on the head and torque degradation, expecially at low pressures. Therefore a pressure correlation should be incorporated into the PUMPANC subroutine. As well, degradation curves for each quadrant should be used.

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