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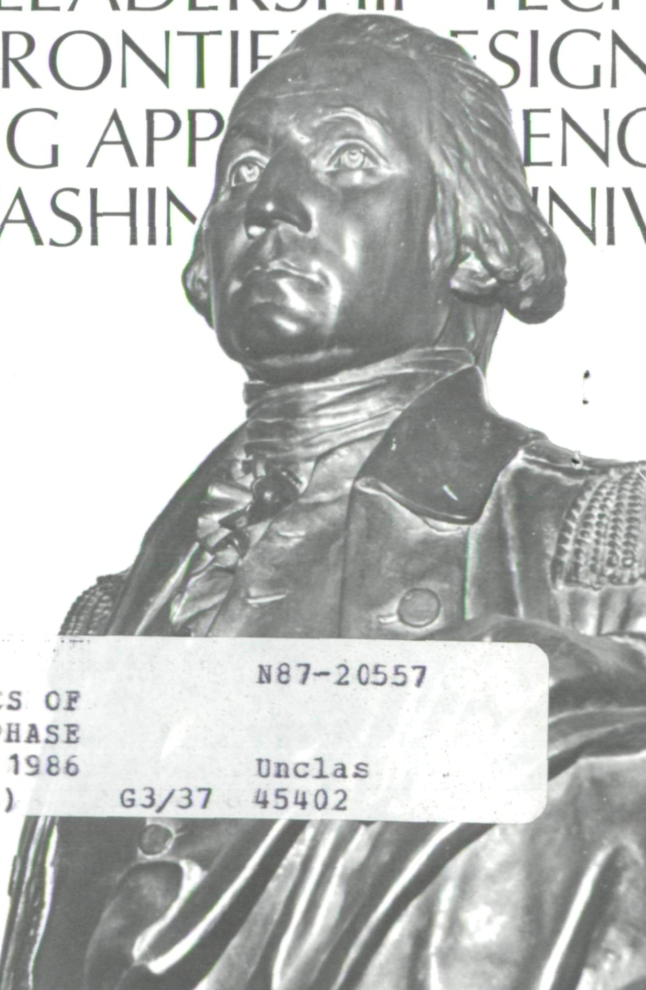

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INVESTIGATION OF THERMAL-FLUID MECHANICAL CHARACTERISTICS OF THE CAPILLARY PUMP AND THE PUMPED TWO-PHASE LOOP

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INTRODUCTION

This first semi-annual report summarizes progress made on NASA Goddard Space Flight Center (GSFC) Grant NAG 5-834 during the period September 1, 1986 to February 28, 1987. The overall goal of the project is to gain a better understanding of the transient behavior of the Capillary Pump Loop (CPL) developed and tested by the GSFC. The investigation is directed toward development of analytical models to represent the transient thermal-fluid mechanic processes occurring in different parts of the CPL engineering model. Evaluation of the available test data has been the starting point for the investigation. Based on results of this evaluation, supplementary tests will be conducted by using a CPL test system already operational in Heat Transfer laboratory of the university. Of particular interest is the oscillatory behavior of the CPL engineering model exhibited during some of the earlier test runs conducted at NASA-GSFC and Johnson Space Center (JSC).

PROGRAM WORK STATEMENT

This section of the present report summarizes project objectives and expected significance of the research work being conducted.

NASA-GSFC's involvement in the space station thermal management technology program includes development and testing of a Capillary Pump Loop (CPL) engineering model. The main objective of the CPL test program has been to gain better understanding of the system performance under each type of operation and broaden the data base for the future CPL design and development. Tests conducted under 1-g conditions demonstrated all of the basic operating principles of the CPL. To confirm zero-g performance, test data were gathered during the shuttle flight experiments.

During critical analysis and assessment of the collected CPL test data, some of the observed transient data trends could not be explained on the basis of available mathematical models used to simulate the CPL system operation. It has been concluded that such peculiar data trends, if not explained, could affect adversely reliability of the CPL system operation.

It is known that [1] heat pipe dependent operating parameters, such as vapor pressure, vapor temperature, working fluid flow rate, change smoothly in response to changes in independent parameters, such as the power input, when conditions within the heat pipe are normal, i.e. when the capillary structure is fully wetted. For such operation, the existing CPL computer modeler [2] can be used to predict the steady state and the time-averaged transient performances. During transient operation under adverse conditions, however, such as vapor nucleation inside the porous heat pipe wick, account must be taken of internal fluid dynamic characteristics of the heat pipe operation. The mentioned CPL computer modeler cannot handle such cases.

As an example, reference [3] noted that boiling in a porous wick structure begins at much lower values of specific heat flux than does boiling in a large volume. The complexity and spatial branching of the porous material produce unwetted zones, particularly at places where elements of the body touch each other and these are potential vapor generation centers; even without presence of the non-condensible gas bubbles inside the porous wick. Transition from the evaporation regime to the boiling regime in the heat pipe wick may cause oscillatory behavior and raise the question of stability of the heat pipe operation. In transient operation, similar adverse conditions can also occur caused by flow excursions, propagation of disturbances under two-phase flow conditions, variation in inlet subcooling, etc.

In view of the foregoing considerations the program work statement includes the following tasks.

I. Assessment of the available test data and identification of transient data trends.

II. Based on this assessment, tentative explanations of possible thermal-fluid mechanic phenomena causing the observed transient data trends.

III. Experimental verification or modification of conclusions reached in task II activities.

IV. Design of mathematical models which can be incorporated into the existing CPL computer modeler. Such a modified computer modeler should be able to predict the oscillatory as well as the time-averaged transient performance of the CPL system.

V. Study of the stability conditions.

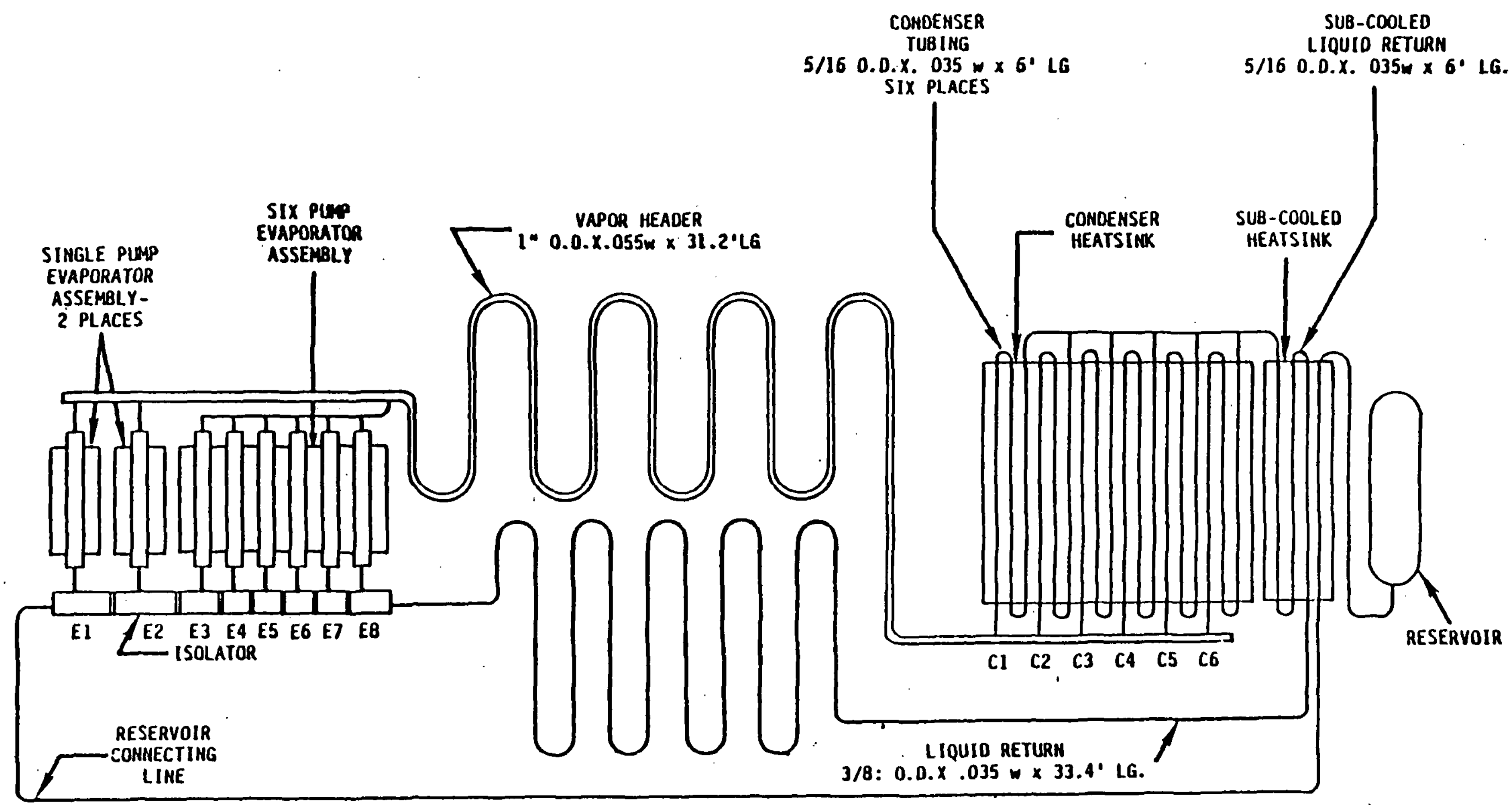
During the past six months first two tasks were completed and the CPL test model and the data acquisition system have been readied for Task III activities.

I. ASSESSMENT OF THE EARLIER TEST DATA

A considerable amount of CPL engineering model test data were gathered during test runs conducted both at the OAO Corporation and the NASA-GSFC laboratories. The CPL-1 Engineering Model, built and tested by OAO for NASA-GSFC, is shown in Fig.1. The CPL-2 Model tested by GSFC is similar, with some modifications, to the CPL-1 Model. These systems uses ammonia as the working fluid. The evaporator zone consists of two individual evaporators and six evaporators that are integrated into a common cold plate. Each evaporator consists of an axially grooved aluminum tubing outer shell and an internal wick structure made of high density polyethylene porous material. Both the vapor header and the liquid line are made of smooth walled aluminum tubing bent into several passes to provide an adiabatic transport length of approximately 10 meters. At the condenser end, six parallel condenser tubes were clamped to a cold plate connected to the coolant loop. A subcooler, which provides additional heat sinking at the condenser exit, is also installed. Later, some modifications were made on the CPL-1 test loop in order to achieve the objectives of the planned tests. Details of these modifications and other related information are given in Ref.[4].

A general assessment of the test data gathered on CPL-1 and CPL-2 Engineering Models was completed during Task I activities. Following is a brief summary of this assessment, and a review of expected as well as some unexplained transient data trends exhibited during test runs.

1. Successful start-up of the CPL-1 was obtained when the "standard" start-up procedure was followed, i.e., the condenser sink was kept at 10°C and the reservoir at 25°C while heat input of 100 watts was applied to each of the evaporators.



NOTE:

- Each evaporator assembly consist of 12" active length CPL Pump(s) and aluminum cold plates equipped with heaters and cooling coils
- The CPL Breadboard is also set up to evaluate non-condensible gas getters which are not shown in this schematic.

Figure 1. Schematic of the CPL Engineering Model (OAO Corporation)

CPL-2 start-up at 0°C to 13°C condenser sink, 25°C reservoir temperatures at 100 watts power input to each evaporator, however, exhibited slight inlet oscillations in evaporators E1, E3, and E8. Under apparently similar conditions all isolator and evaporator inlet temperatures displayed oscillatory behavior in some test runs. At 45°C reservoir, 18°C condenser sink temperatures and 100 watts power input to each evaporator wide fluctuations were observed in all isolator and inlet temperatures of the CPL-2 test systems. Similar oscillations occurred during start-up in asymmetric heating and condenser cycling tests of the CPL-2 system.

Although Ref.[4] noted some difficulties in start-up of CPL-1 test system when evaporator E-1 had additional mass, no oscillatory start-up behavior was reported. For this condition a new start-up procedure was developed for CPL-1. The new scheme was reported to be successful in all tests.

2. <u>Transport Limit Test Data</u>

2.1 <u>CPL-1 Test System</u>

Transport limits at 15°C, 25°C, and 35°C reservoir temperatures were obtained when temperatures of evaporators E5 and/or E6 continued to rise without any sign of reaching an equilibrium. These evaporators could be recovered by reducing the respective power inputs. No inlet deprime of any evaporator was seen in these tests. It was also observed that for operating temperatures of 15°C and 25°C there existed some threshold power inputs at which E5 and E6 temperatures were higher than the others. The threshold power was 600 watts/evaporator at 25°C and 400 watts/evaporator at 15°C. As the power input increased beyond these limits, temperatures of E5 and E6 kept rising without bound. Following further testing it was concluded that capillary pumping capability was the limiting factor in the CPL-1 transport performance and E5 and E6 were the weakest evaporators. On the other hand, at 45°C reservoir temperature transport limit was always accompanied by an inlet deprime of E3 and no recovery of this evaporator was possible by simply reducing the power input to E3. No temperature drift in other evaporators occurred in the 45°C test.

In reservoir cycling test, at 45°C, evaporator temperature profiles were fairly uniform. As the reservoir temperature was reduced to 25°C and below, temperatures of E5 and E6 were higher than the others by 3°C to 5°C and the temperature profile was no longer uniform. As the reservoir increased from 15°C toward 45°C, a uniform temperature profile of the evaporators was restored for temperatures above 30°C. No oscillatory temperatures trends were reported in these test runs.

2.2 CPL-2 Test System

Although general characteristics of the data trends of CPL-2 transport limit tests agreed with that of the CPL-1 test system, varying level of temperature oscillations were observed in CPL-2 test runs conducted under various operating condition.

In Figures 2-3 isolator, inlet, and pump temperature variations of evaporators 1 and 2 are displayed for transport limit test runs conducted at 25°C reservoir and 0°C chiller temperatures. Half way in these tests the chiller temperatures were changed as recorded in the figures. These data displays a definite oscillatory behavior in isolator and inlet temperatures in portions of the recordings. Temperature variations of E1 recorded in earlier test runs are shown in Fig. 4. Although amplitudes of the oscillations in this figure are smaller, the general transient data trend is similar. During these runs E8 and to some extent also E7 displayed similar oscillatory temperature behavior. In contrast, only slight fluctuations in temperature of E3 and other evaporators were observed as shown in Fig. 5. At the intermediate power inputs, from 100 W to 500 W, temperature variations of evaporators E1, E2 and E8 were rather erratic. When, however, the power in each evaporator is increased to 600 watts, temperature oscillations were diminished almost completely in all evaporators. At 850 W to 900 W levels, all pump temperatures started to warm up, E2 deprimed 8 minutes later, and the other pumps exhibited temperature increase characteristics toward pump dry-up.

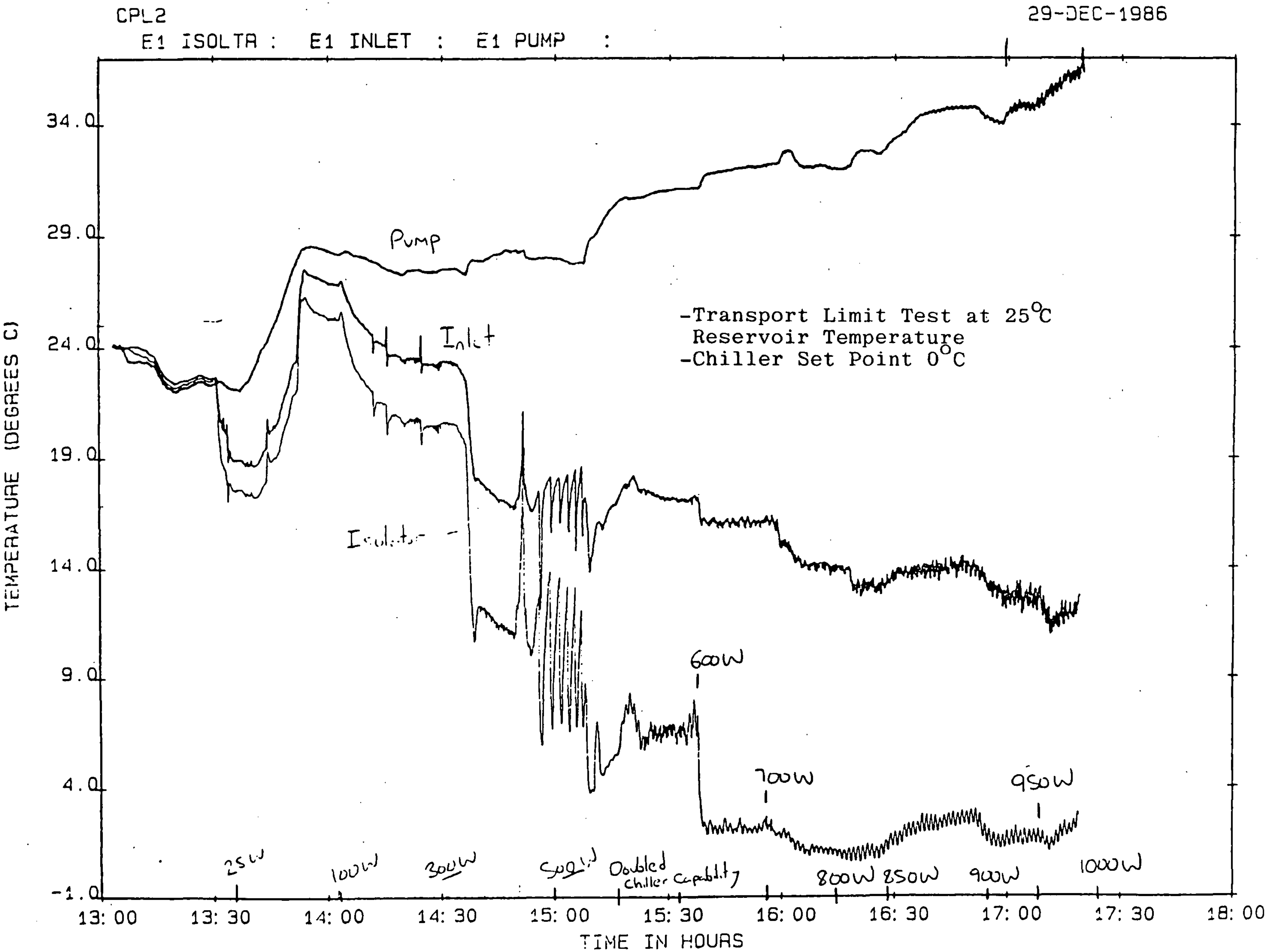


Figure 2. Evaporator 1 Temperature Variations

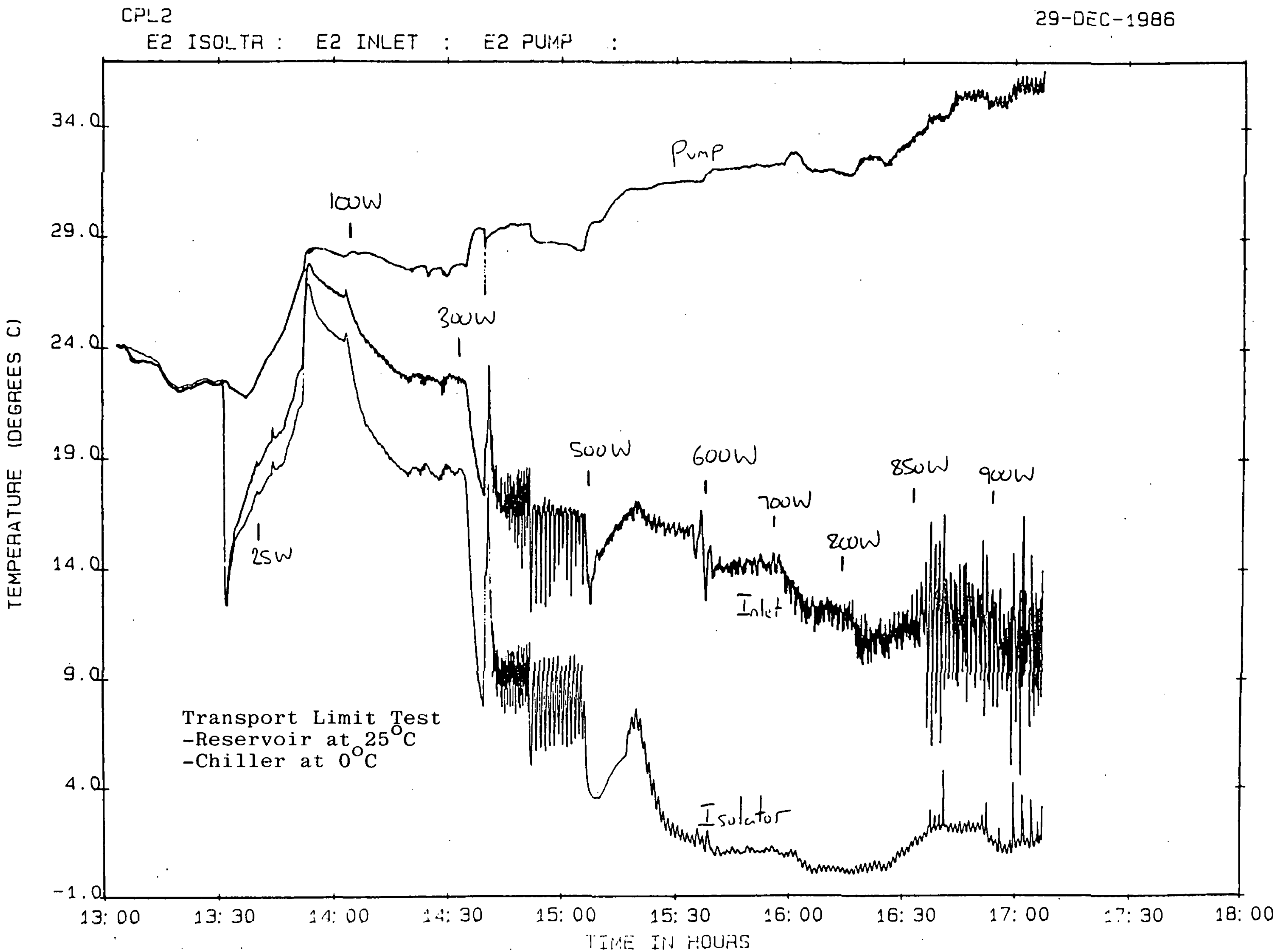


Figure 3. Evaporator 2 Temperature Variations

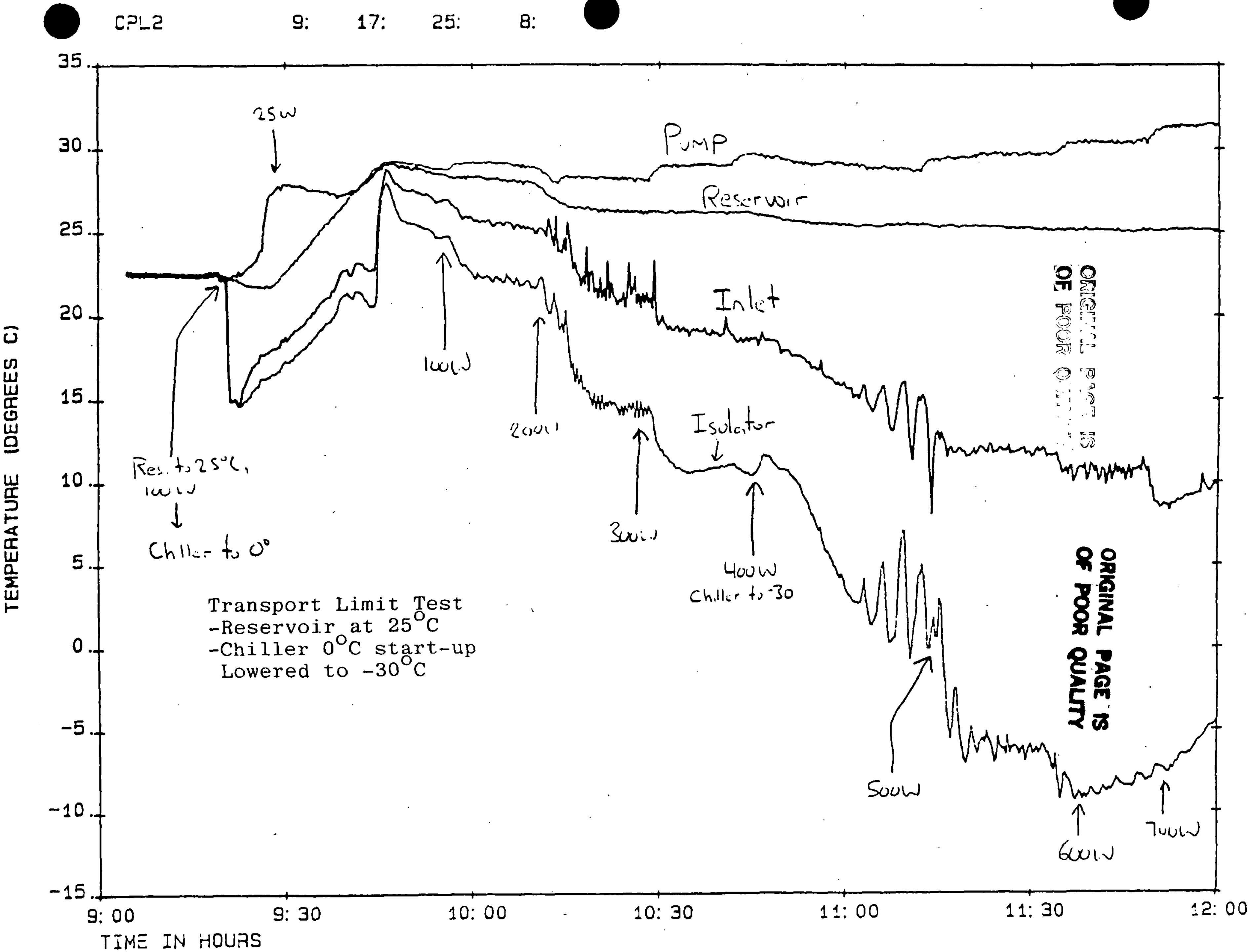


Figure 4. Evaporator 1 Temperature Variations

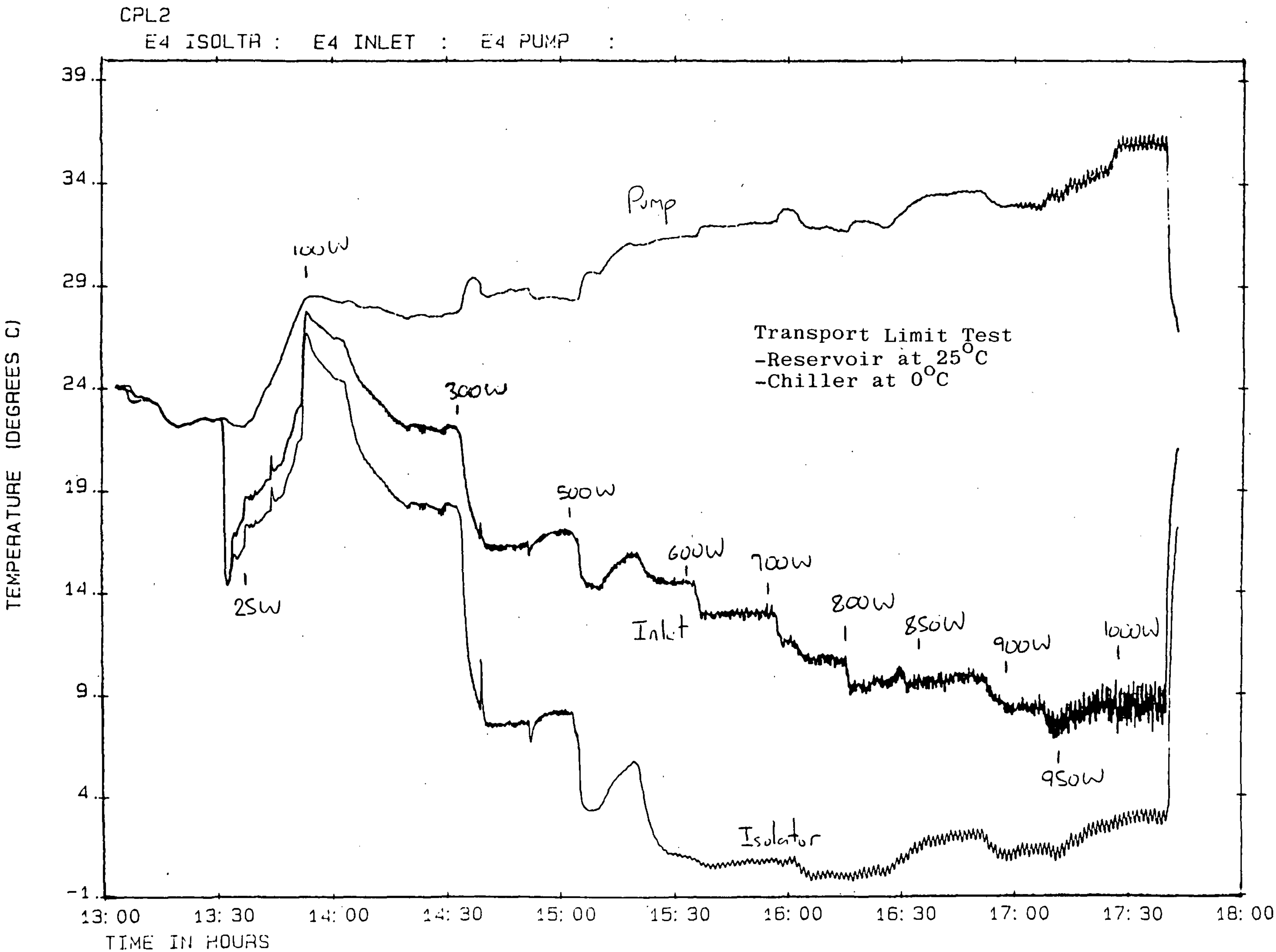


Figure 5. Evaporator 4 Temperature Variations

When the transport limit test runs were conducted at $45^{o}C$ reservoir and $18^{o}C$ condenser temperatures, wide temperatures oscillations were recorded, at 100 W, in all isolators and inlets. Within the power input range from 300 W to 700 W, oscillations in E2, E7, and E8 tended to decrease and diminish. This trend was delayed in E1 until power was increased to about 700 W. When the total system load was 5600 W, E2 deprimed with $16^{o}C$ of subcooling. In an earlier test run, conducted under almost identical reservoir and condenser temperatures, E2 deprimed when the total CPL evaporator power was raised from 6.4 kw to 6.8 kw.

3. <u>Subcooling Requirement Test Data</u>

CPL test data indicated that when there was insufficient subcooling, the evaporators experienced inlet deprimes. In a CPL-2 test run, at $25^{o}C$ reservoir and $14^{o}C$ subcooler outlet temperatures 100 W was applied to all eight evaporators. After 70 minutes under these conditions, the condenser chiller set point temperature was increased $1^{o}C$ every 12 minutes. After 40 minutes, E1 deprimed with a subcooler outlet temperature of $18^{o}C$, i.e. $7^{o}C$ of subcooling. With decreased subcooling, varying degrees of temperature oscillations appeared in all evaporator isolators and inlets, except E6 which remained very stable throughout. During the last 30 minutes of testing, however, both isolator and inlet temperature fluctuations diminished to near negligible values before E1 deprimed. Temperatures of E6 tended to run from $2^{o}C$ to $3^{o}C$ cooler than that of other evaporators. During these test runs although the chiller outlet temperature cycled about 3°C every minute to maintain a constant set point temperature, a negligible cycling in temperatures was observed as the evaporator section was approached.

Another subcooling performance test with a 45°C reservoir temperature and a 4 kw total load was conducted on the CPL-2 test system. When the subcooler outlet temperature was increased suddenly from 37°C to 41°C (decrease in subcooling from 8°C to 4°C) E1 deprimed. During these tests, oscillations did not appear in the E1 inlet until the subcooling was reduced to about 17°C. In another test run, oscillations of up to 8°C were observed in E1's isolator and inlet temperatures even with as much as 35°C of subcooling. E2, which deprimed first in this test, displayed 1-2°C oscillations in its inlet with from 35-25°C of available subcooling. When the subcooling was decreased below 25°C, E2 inlet became more erratic, oscillating 4-5°C until it deprimed. Some fluctuations in the E8 inlet temperature were seen in these tests. The other six evaporators, E3-7, displayed negligible oscillations in both isolator and inlet temperatures.

4. Other CPL Test Data

Additional CPL performance data were collected under other test conditions, such as: Asymmetric heat input applied to evaporators, heat load sharing, low power limit, system turn-down ratio, diode function of the condensers, sudden change of the reservoir set point, pressure priming under load. Since the initial stage of the present investigation involves identification of data trends in the transient CPL operation, the foregoing test data review will be an appropriate starting point for this purpose. As the work progress, rest of the available test data will be used, as needed.

The CPL test data exhibit the following rather distinctive transient performance characteristics:

(a) The earlier transient test data gathered on CPL-1 engineering model did not display the kind of oscillatory temperature behavior observed in more recent CPL-2 test runs. When the reservoir set-point temperature of CPL-1 was changed suddenly, evaporator inlet temperature variation displayed a transient characteristics which can be considered oscillatory. Similar behavior was also observed during the reservoir cycling tests when the evaporator E3 suffered an inlet deprime. Otherwise, the reported CPL-1 test data were free of any noticeable fluctuations in isolator and evaporator inlet temperatures.

(b) Review of the CPL-2 test data indicated that, under various operating conditions, isolator and inlet temperatures of some of the evaporators displayed, so far unexplained, oscillatory transient behavior.

(c) Although increased subcooling seemed to reduce and even to diminish these oscillations under most of the test conditions, in CPL-2 condenser cycling tests at constant power input, temperature fluctuation persisted in all evaporators when the condenser chiller temperature was reduced from 10°C to -40°C. In contrast, oscillations were diminished in most evaporators as the chiller temperature was raised back to 10°C. These seems to indicate that the subcooling influences the transient behavior in a manner combined with effects of some other factors.

(d) The CPL-2 test data also seemed to indicate that level of the power inputs to evaporators, and the magnitude of the step changes in these inputs influence degree of the temperature fluctuations. Within certain power input ranges, for instance, amplitude of the fluctuations displayed a diminishing trend.

(e) It is known that presence of non-condensible gases in the heat pipes can lead to oscillatory transient behavior. Although in CPL-2 tests runs no immediate evidence of existence of such a situation was determined, it is important that this possibility should be taken into consideration.

(f) The test data available in literature indicate that rapid heat pipe start-up, rapid condenser cooldown, and rapid reservoir set-point temperature change all influence the transient heat pipe performance significantly. Therefore, assessment of the CPL test data should consider implications of these operational factors on the system transient performance characteristics.

(g) Under at least externally similar testing conditions, local instantaneous transient or even time-averaged transient performances of apparently similar evaporators displayed different test data characteristics. Reasons for such deviations have not been readily explainable. It is expected that internal thermal-fluid mechanic characteristics of parallel evaporators have significant influence on the test data collected. Such flow characteristics, on the other hand, are affected to varying degrees by the CPL system design, presence of non-condensible gases and their distribution within the CPL system, as well as by differences in the physical design parameters of the evaporators. Although evaporators undergo a variety of static performance tests, similarity on the basis of these tests may not always imply similar dynamic performance characteristics.

(h) The CPL test data related to the evaporator inlet deprime in the low power limit tests, as well as the high power input operation leading to dry-out are rather consistent. Therefore, this data should be very useful in studying the physical mechanisms leading to such critical conditions.

(i) For a complete test data assessment other factors such as, details of the test data acquisition procedures, unaccounted parasitic heat exchanges between the test systems and the environment, etc. should also be considered. It was reported that some of the CPL test data was suffered from the parasitic heat exchanges.

II. TRANSIENT CPL PERFORMANCE - (Preliminary Study)

In recent years, the subject of transient heat pipe behavior has received considerable attention [1,5-7]. Authors of Ref. [1] concluded that when the capillary structure is fully wetted, heat pipe dependent operating parameters, such as pressure and temperatures, working fluid flow rate, change smoothly in response to changes in independent parameters, such as power input, and thus the heat pipe exhibits a quasi-steady state behavior. However, under adverse conditions, such as boiling in the evaporator section, account must be taken of internal fluid dynamics. As a result, the quasi-steady state model can no longer be appropriate, predictions become much more difficult. Evidently there exists a need, as noted in [1], for detailed internal measurements which can be used as a basis for developing mathematical models which can be used for predicting the transient behavior. Some of the basic transient heat pipe performance investigations were reviewed in Ref. [1].

When the transient CPL test data is evaluated against the information available in transient two-phase flow literature, one notices the striking similarities between the transient oscillatory behavior displayed in both CPL and the two-phase flow systems. It is known that in addition to the capillary, the sonic, and the entrainment limits, the heat pipe performance is also limited by the boiling heat flux limit. Although boiling in the heat pipe wick and the associated heat flux has been the subject of many investigations, the main emphasis in these studies was prediction of the critical maximum heat flux. Implications of the wick boiling conditions on the transient heat pipe performance have not been investigated properly.

1. <u>Wick Boiling Effects on Transient Heat Pipe Performance</u>

In the current literature, prediction of the heat pipe critical boiling heat flux limit is based on criteria similar to those which apply to nucleate pool boiling from planar surfaces. On the other hand, it is almost universally agreed that boiling in a porous structure starts at significantly lower heat flux densities and temperature differences than does in a large liquid volume. For this reason, it is evident that the physical models using the mentioned criteria cannot be appropriate for predicting the heat pipe critical boiling heat flux limit.

Even more important is the fact that, except the critical conditions of a heat pipe inlet deprime or dry-out, a possible occurrence of boiling inside the heat pipe wick under other conditions has been discounted in theories using the conventional boiling models. Because, these models lead to the conclusion that the critical liquid superheat required for boiling initiation is rather high, which can only be reached beyond the heat pipe capillary limitation. As soon as the capillary limit is exceeded, the liquid flow into the wick will be insufficient to match the level of flow rate required by the power input; temperatures will increase and the transition from evaporation to boiling will be initiated, leading to a heat pipe dry-out. Although such a conclusion is correct when the heat pipe performance is controlled by the capillary limitation, possibility of boiling initiation at much lower fluid superheats can change the entire heat pipe transient performance characteristics.

Oscillatory flow instabilities may occur whenever heat is added to a flowing liquid which undergoes a phase change. The parametric effects of such instabilities, observed experimentally in conventional two-phase flow systems, have been studied by many investigators. Although there exists no such study in the heat pipe literature, one should not dismiss the possible occurrence of some form of flow instabilities in heat pipe applications. In fact, in view of the oscillatory transient behavior observed in some of the CPL tests, one is tempted to assume that the transient CPL operation could have involved the conditions which could lead to some form of flow instabilities. The following presentation deals with this consideration.

Two-phase flow instabilities can be static or dynamic in nature. A flow is subject to a "static instability" if, when the flow conditions change by a small step from the original steady state conditions, another steady state is not possible in the vicinity of the original state. The threshold of instability can be predicted by using steady state laws. A static instability can lead either to a different steady state condition or to an oscillatory behavior [8]. A flow is subject to a "dynamic instability" when inertia and other feedback effects have an essential part in the process.

Static two-phase flow instabilities include, among others, the Ledinegg instability and chugging. Ledinegg instability involves a sudden change in the flow rate to a lower value. It occurs when the curve of steady state system pressure drop (Δp) versus mass flow rate (G) curve (internal) has a negative slope, as shown in Fig. 6. The criterion for this instability is given by

$$\left(\frac{\partial \Delta P}{\partial G}\right)_{inter.} \leq \left(\frac{\partial \Delta P}{\partial G}\right)_{ext.} \tag{1}$$

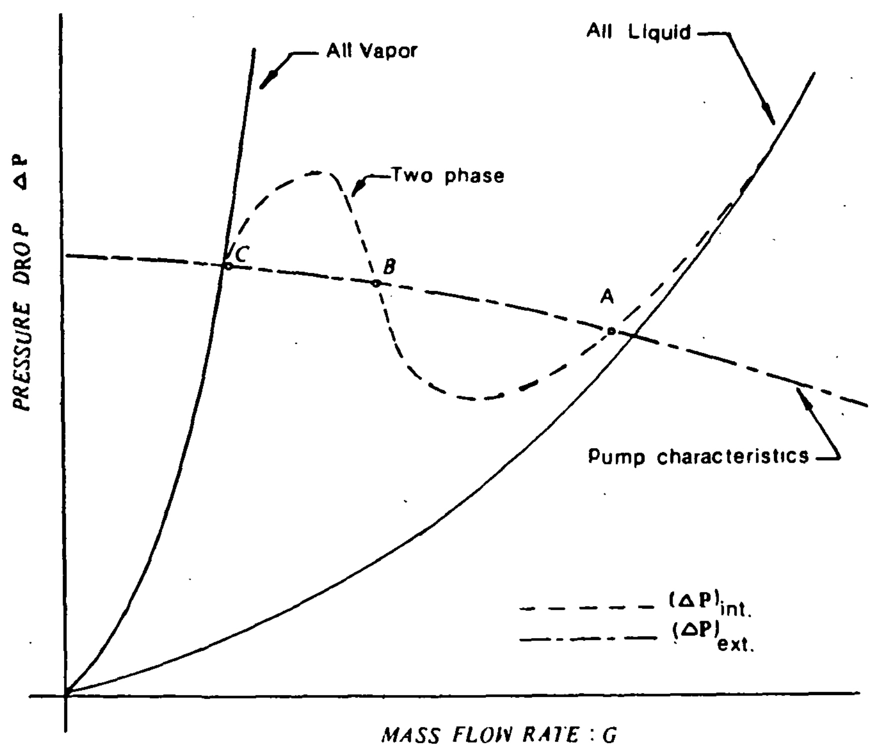


Figure 6 - Two-Phase Flow Pressure Drop Characteristics

$\left(\frac{\partial \Delta P}{\partial G}\right)_{ext.}$ is slope of the steady state pump characteristic curve.

The steady state pressure drop in a heated flow passage with boiling can be expressed as

$$\Delta p_{in} = \Delta P_i + \Delta P_c + \Delta P_f + \Delta P_g + \Delta P_e \tag{2}$$

where the right-hand terms are the inlet, convective acceleration, frictional, gravitational, and exit pressure drops, respectively. Because of the convective acceleration due to boiling, the decrease in the flow rate does not necessarily correspond to a decrease in the passage pressure drop. If the condition given by Eq. (1) is satisfied, the flow becomes unstable. Then, referring to Fig. 6, operation at point B is unstable. A slight change in the flow rate at B accelerates the flow to a new stable operating conditions either at point A or point C. This instability tendency is enhanced at low subcoolings.

The term "chugging" involves static phenomenon displaying a cyclic behavior. The cycle consists of heat storage in the liquid during an incubation period, nucleation, expulsion of vapor from the passages, and re-entry of the liquid. With the liquid return, the subcooled nonboiling condition is restored, and the cycle starts over again.

Dynamic two-phase flow instabilities, such as acoustic oscillations, and density wave oscillations have much higher frequencies than frequencies of oscillations observed in the CPL tests. Therefore, during the initial stage of the present work such instabilities will not be considered. Pressure drop oscillations is a secondary dynamic instability phenomenon triggered by a static instability. This type oscillations are associated with operation on the negative sloping portion of the Δp versus mass flow curve. Since a typical period of pressure drop oscillations is 40 seconds, such fluctuations are also considered in the preliminary study.

Tentative explanation of thermal-fluid mechanic mechanisms of the transient CPL behavior is presented in the following sections. Collection of the test data for supporting validity of the proposed physical models will start very soon.

2. <u>Temperature Oscillations in CPL Tests</u>

Earlier it is noted that boiling in a porous structure starts at significantly lower heat flux densities and liquid superheats than does in a large liquid volume. Therefore, it is reasonable to assume that presence of vapor nucleation inside the porous heat pipe wick may lead to the type of static flow instabilities discussed above. Such instabilities can display an oscillatory behavior consisting of repetitive liquid superheat, nucleation, vapor expulsion from the porous wick, and re-entry of the liquid. This cyclic flow phenomenon will cause fluctuations in the pump temperatures. Since heat conduction along the evaporator inlet pipe will change in response to the cyclic flow conditions, isolator temperatures will also experience fluctuations.

As noted in [3], the onset of boiling in a wick-type structure is one of the least understood mechanisms of physics of boiling. Although it is generally agreed that boiling in a porous structure begins at much lower heat flux and liquid superheat than does boiling in a large volume, different hypotheses have been formed to explain the experimental results. According to Ref. [9], there are sections within the porous layer with a ready phase interface to serve as boiling nuclei. The dimensions of these nuclei are close to those of the maximum pores of the given porous structure. The existence of such nuclei within the wick leads to a decrease of the liquid superheat and associated heat flux necessary to start boiling. Presence of the noncondensible gases helps formation of the nucleation centers.

Reference [3] considers separately two cases: flooded wick type walls and surfaces with capillary charging of the wick, and hypothesizes that activation of surface nucleation sites depends not only on the superheat of the heating surface, but also on properties of the thermal boundary layer. With thicker layer nucleation sites are activated at lower ΔT. The wick contacting the heating wall interferes with convection of liquid and increases the thickness of the thermal boundary layer. As a result, a much lower superheat is needed for activating nucleation sites in the case of a porous than in that of a nonporous wall.

As the second case, if the wick is charged by capillary action, thickness of the thermal boundary layer, and consequently ΔT, is significantly affected by the thickness of the wick layer. For thicker wicks (more than a few pore diameter thick) data on ΔT are similar to those for flooded wick-type walls; thermal boundary layer increases with increasing wick thickness thus leading to lower ΔT necessary for boiling start. Although Ref. [3] assumes that probability of early activation of nucleation sites in pores is very low compared to the activation of the sites of the heating wall itself, it noted that pore sites of the wick may also be activated due to large vapor volumes from the moving, surface-formed, vapor bubbles.

Figure 7 shows the general form of the temperature distribution in porous wick of a CPL evaporator. Temperatures in the region adjacent to the heating surface are higher. Following the analysis given in [9], it is assumed that a vapor bubble grows at the heating surface into pores of the wick, as shown in the figure. The vapor pressure in this bubble is larger than the pressure of the surrounding liquid by the amount of the capillary pressure arising in these fine pores,

$$P_{vn} = P_{\ell} + \frac{2\sigma}{R_{vn}} \tag{3}$$

where, R_{vn} is average radius of the vapor bubble. If the mean capillary pressure in the evaporation zone open to the vapor passages is given by $(2\sigma/R_{sb})$, then the liquid and vapor pressures in the heat pipe evaporation zone is given by,

$$P_{v} = P_{\ell} + \frac{2\sigma}{R_{sb}} \tag{4}$$

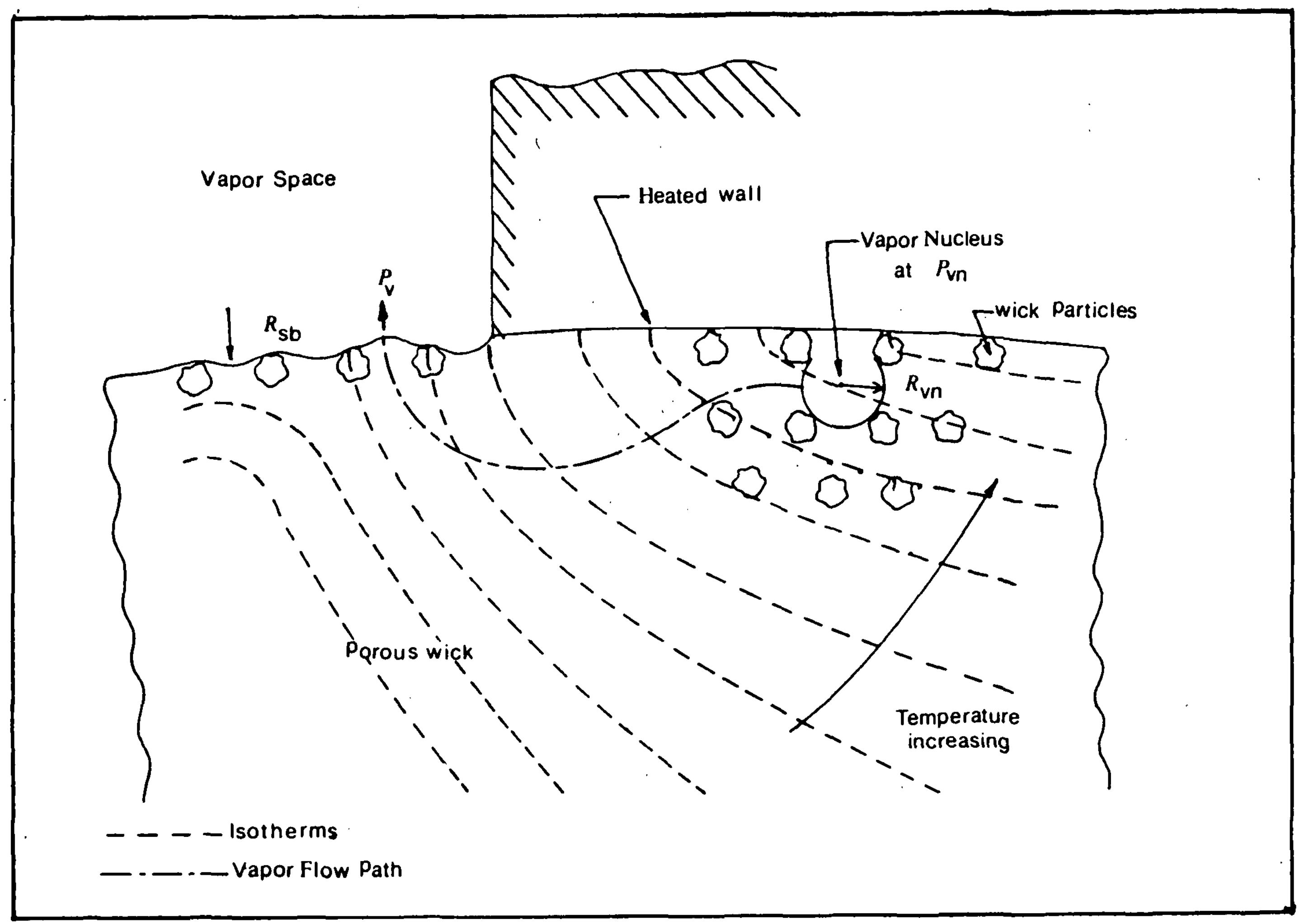


Figure 7 - Vapor Nucleation and Wick Temperature Distribution at the Moment of Boiling

Combining these equations, the following relation is obtained for the equilibrium condition.

$$P_{vn} - P_v = \frac{2\sigma}{R_{vn}} - \frac{2\sigma}{R_{sb}} \tag{5}$$

If the equilibrium is disturbed, this relation becomes an inequality. When a nucleation site is activated, the left side of this relation becomes larger than the right side, and when the bubble collapses the direction of the inequality changes to the opposite.

For fixed values of P_v and R_{vn}, the condition of equilibrium of Eq. (5) may be disrupted both due to an increase of P_{vn} (caused by an increase in the liquid superheat) and due to an increase of $2\sigma/R_{sb}$ (caused by an increase of the power input). These two effects make the left side of Eq. (5) larger than the right side, leading to bubble growth and boiling in the region adjacent to the heating wall.

So long as no boiling exists at the heating surface-porous wick interface, heat is conducted through the liquid soaked wick to the liquid menisci open to the vapor passages, and heat is removed by evaporation. With the incipience of boiling, however, vapor accumulates inside the wick underneath the heating surface. When P_{vn} reaches a level to overcome the hydraulic resistance of the flow path between the vapor region and the open vapor flow passages, the vapor will be expelled from the wick into the vapor passages. Next the subcooled liquid refills wick portion underneath the heating surface and the heat storage, superheat, nucleation cycle starts over again. Evidently, such a cyclic process displays characteristics of "chugging" discussed in the earlier section. The described process would lead to temperature and pressure drop oscillations of low frequency in the evaporator section.

Referring to Eq. (5), P_{vn} is affected by the degree of subcooling, whereas $2\sigma/R_{sb}$ changes with the power input. For this reason, the mentioned cyclic phenomenon is controlled by the combined actions of liquid subcooling and the power input. Boiling at high power inputs, for instance, can be avoided by increased subcooling. If the pore sites inside the wick are activated due to movement of large volumes of vapor (at high power inputs), a stable vapor flow path can be formed which disrupts the liquid flow to the evaporation region, leading to a dry-out. Under this condition, although the porous wick is still saturated with liquid, flow of the liquid to the capillary interface is prevented. Therefore, the normal operation can be recovered by reducing the power input. At low power inputs the amount of liquid flow into the porous wick is very small. Therefore, incipience of boiling can quickly dry the wick and lead to an inlet deprime. It is believed that the physical models based on the boiling phenomenon described above can be used to represent and explain both the inlet deprime and the dry-out processes.

Starting from Eqs. (4,5), Ref. [9] presented an analysis which led to an equation giving the heat flux, q_{sb}, for the start of local boiling in heat pipe wicks:

$$q_{sb} = \frac{2/R_{vn}}{\delta/(Mk\alpha) + (F_h \ell)/(NF_w K_w)} \tag{6}$$

where, δ is the average effective length between the heated wall and the capillary layer open to the vapor passage; k, effective thermal conductivity; F_h, heating surface area; F_w, wick liquid flow area; K_w, wick permeability, and ; ℓ , length of the liquid flow path. Parameter α accounts for the influence of the thermal contact resistance of the wick with the heating wall and,

$$M = \sigma T_v/\lambda\rho_v \quad , \quad N = \sigma\lambda\rho_\ell/\mu_\ell$$

σ, surface tension; T_v, ρ_v vapor temperature and density; λ, latent heat of evaporation; ρ_ℓ ,μ_ℓ liquid density and dynamic viscosity.

Although it is expected that the magnitude of R_{vn} is close to those of the maximum pores of the porous structure, investigators of Ref. [9] were note successful to determine R_{vn} by using their boiling model and the available test data. It was mentioned earlier that, according to Ref. [3], the thickness of the wick has significant influence on q_{sb}. The disregard of this factor in Ref. [9], therefore, was claimed to be the reason why the published experimental data could not be analyzed properly.

In the present investigation an effort will be made to introduce an improved model of the boiling mechanism for predicting the heat flux at the start of wick boiling. Validity of this model will be checked by using the published test data and the data to be obtained on the CPL test system. It should be noted that direct effect of subcooling is not evident in Eq. (6). This equation refers to the conditions when a nucleation site R_{vn} is activated. It is known, however, that nucleation at a given site will be delayed with increased subcooling. The modified boiling model should also take into account the subcooling effects. For this purpose, it is believed that the analysis given in Ref. [10] will be very useful.

In the study of the transient CPL performance it seems that use of an appropriate mathematical model is needed for predicting the heat flux and the temperature difference at the start of wick boiling. As mentioned earlier, such a model can also be used for predicting the inlet deprime and the heat pipe dry-out conditions.

The immediate objectives of the present investigation is to evaluate the transient CPL performance by considering only the static type flow instability conditions. It is possible that progress of the present work may indicate a need for consideration of dynamic type instabilities in the transient CPL operation.

3. CPL Test Loop

Details of the test loop is given in Ref. [11]. Schematic of CPL Test Loop is shown in Fig. 8. The system uses Freon 11 as the working fluid. The evaporator zone consists of two identical

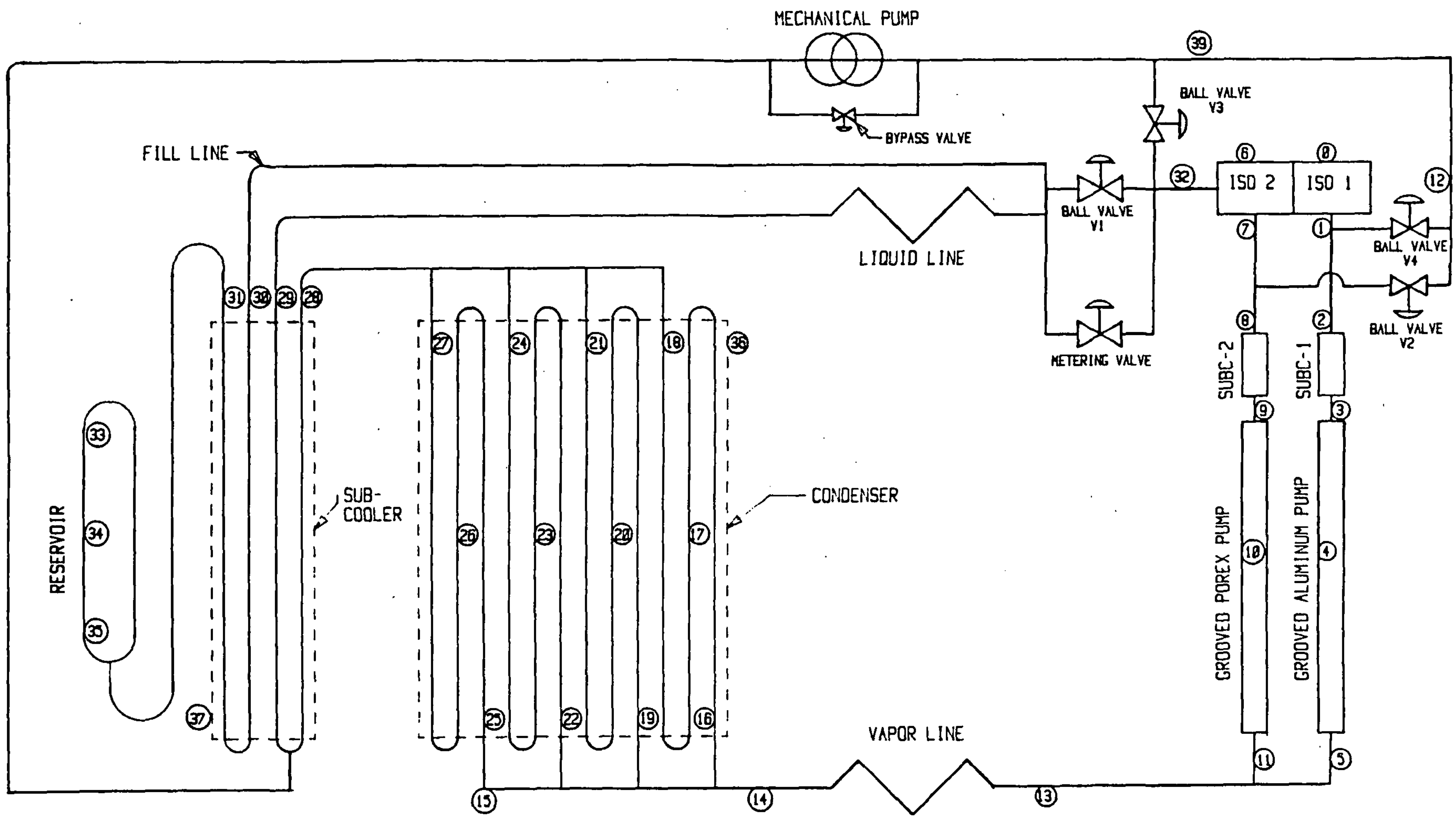


Figure 8. CAPILLARY PUMPED LOOP
GSFC/DYNATHERM

individual evaporators. Each evaporator consists of an axially grooved aluminum tubing outer shell and a POREX insert, fit tightly inside the grooved shell. The loop contains 10 meters each of liquid and vapor flow passages. Both lines are bent into several passes. The liquid line is made of 3/8" O.D. x 0.035" wall aluminum tubing. The vapor line is made from 1" O.D. x 0.049" wall aluminum tubing.

The condenser is made from 5/8" thick aluminum plate which contains semicircular grooves on one side for six parallel coolant legs, and on the other side for four parallel condenser legs which run in a transverse direction. Each leg makes three serpentine passes across the plate. The coolant legs are connected on either end to a coolant manifold which in turn is connected to the mechanical cooling system. There are two subcooling passes for the condensed liquid and the liquid supply from the reservoir.

The reservoir which is used for controlling the loop temperature is connected to the loop by a tube exiting at its bottom. The connecting line makes two passes over the cold plate and connects to the loop at the point where the liquid line enters the liquid manifold of the evaporators. The isolators, shown in Fig. 8, provide a capillary barrier to vapor flow through the liquid manifold, preventing the failure of one evaporator from affecting the other evaporator. A six inches long liquid subcooler is clamped to the liquid inlet to each evaporator between the liquid manifold and the evaporator. Subcoolers are connected to the refrigerated cooling bath.

The test loop is instrumented as shown in Fig. 9. A total of 37 copper-constantan thermocouples were installed for temperature measurements. Thermocouples were attached to the surfaces with aluminum tapes. A special small diameter (1/16") fast response temperature probe was inserted into the liquid space at the inlet side of one of the evaporators for measuring the liquid temperatures. Pressure transducers were installed both at the inlet and the outlet of the same evaporator for pressure measurements. Other details of the test loop is given in Ref. [11].

4. Data Acquisition System

As mentioned above, the test loop is instrumented with 37 copper-constantan thermocouples and 32 of these are selected to be connected to two 16 channel MetraByte EXP 16 multiplexer boards with a gain of 1,000. Outputs of the pressure transducers are fed to a third multiplexer board with a gain of unity. The output voltages of both pressure transducers are also fed to a differential amplifier (gain of 47) with its output fed to the third multiplexer board for differential pressure measurement.

Three watt transducers measure the electrical power fed by three variable transformers to the evaporator and reservoir electric heaters with their outputs also connected to the third multiplexer board. The power to each heater can be regulated by variable transformers and interrupted manually or automatically

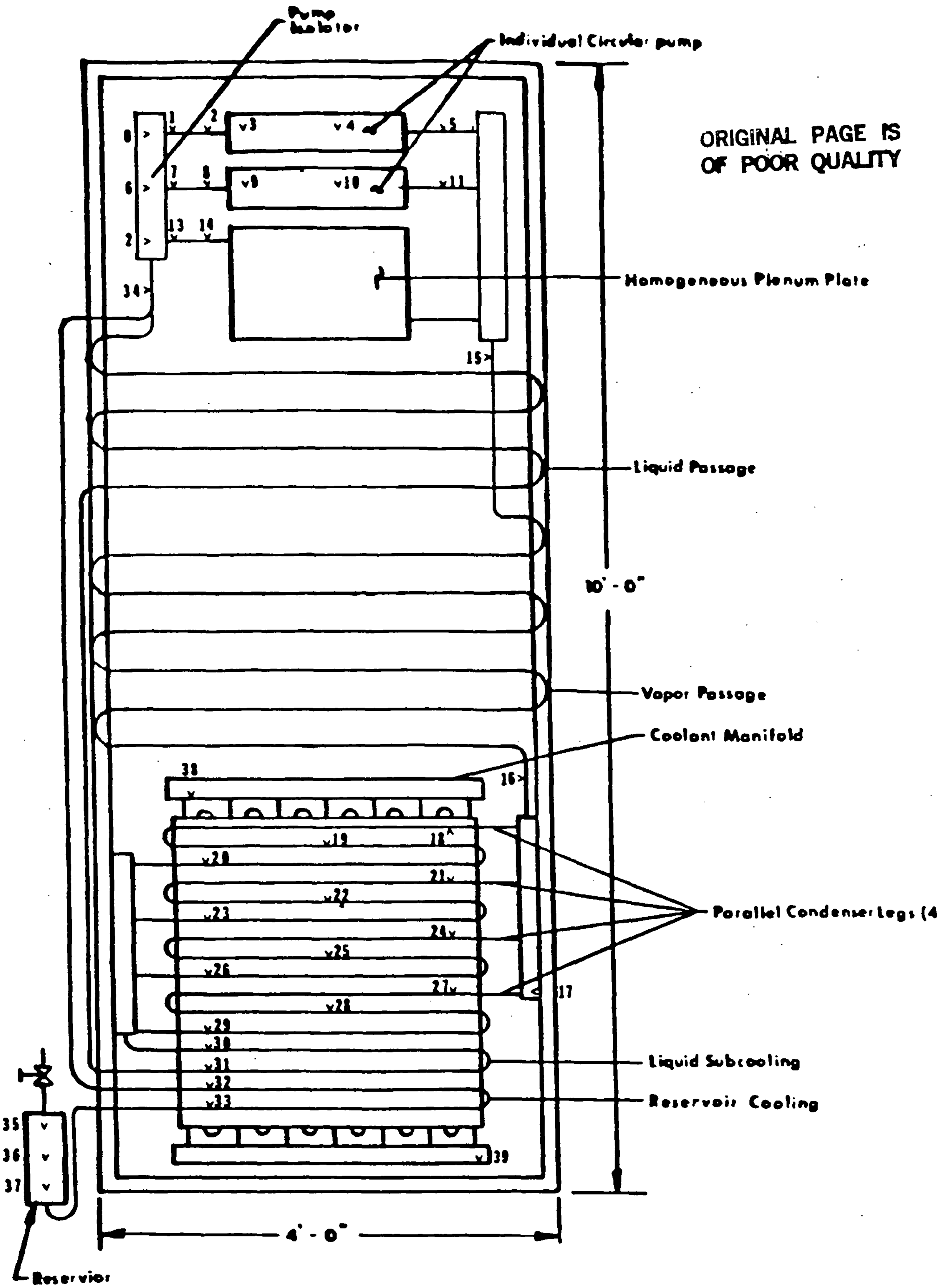


Figure 9. Schematic of CPL Test Loop
(Dynatherm Corp.)

by the computer data acquisition programs via a computer controlled relay.

The three multiplexer boards are connected to a MetraByte DASH 8 high speed A/D converter and timer/counter board for the IBM PC microcomputer. DASH 8 board installed inside an IBM PC/XT does the A to D conversion as required by the software.

Presently, there are four computer programs which have been developed to run the data acquisition hardware as well as to compute and display some pertinent variables and store data to the hard disk. Each of these programs, however, was written for a specific task. CPL1 was developed for monitoring the global test loop performance. It reads all data channels simultaneously and displays on the screen a table of 13 key variables and a color plot of 5 temperatures, writes to the hard disk all data from the multiplexer boards, performs thermostatic control of reservoir temperature based on input reference temperature, and shuts off power to each evaporator when temperature exceeds set points. The CPL1 program presently runs at 3 seconds per scan but should be able to achieve 1 Hz scan rates when compiled.

TRACE program was written to monitor a single channel at the highest possible speed. It reads any channel of the three multiplexing boards, simultaneously displays on screen 3 key temperatures and a full screen plot of the selected variable, and performs thermostatic control of reservoir temperature based on input reference temperature. TRACE runs at up to 27 Hz scan rate.

XPLOT program is similar to TRACE but can track 2 variables.

It runs a little slower but allows real comparison between 2 variables, such as two temperatures, a temperature and a pressure, etc.

The fourth program CPL was written to diagnose the data acquisition system. It reads and converts variables from all data channels and displays them on the screen in a table with the appropriate units. CPL runs at about 1 Hz scan rate.

Further developments include real time printing of screen display in CPL1 and TRACE, Fast Fourier Transform spectral decomposition on TRACE, and a post processing data analysis program.

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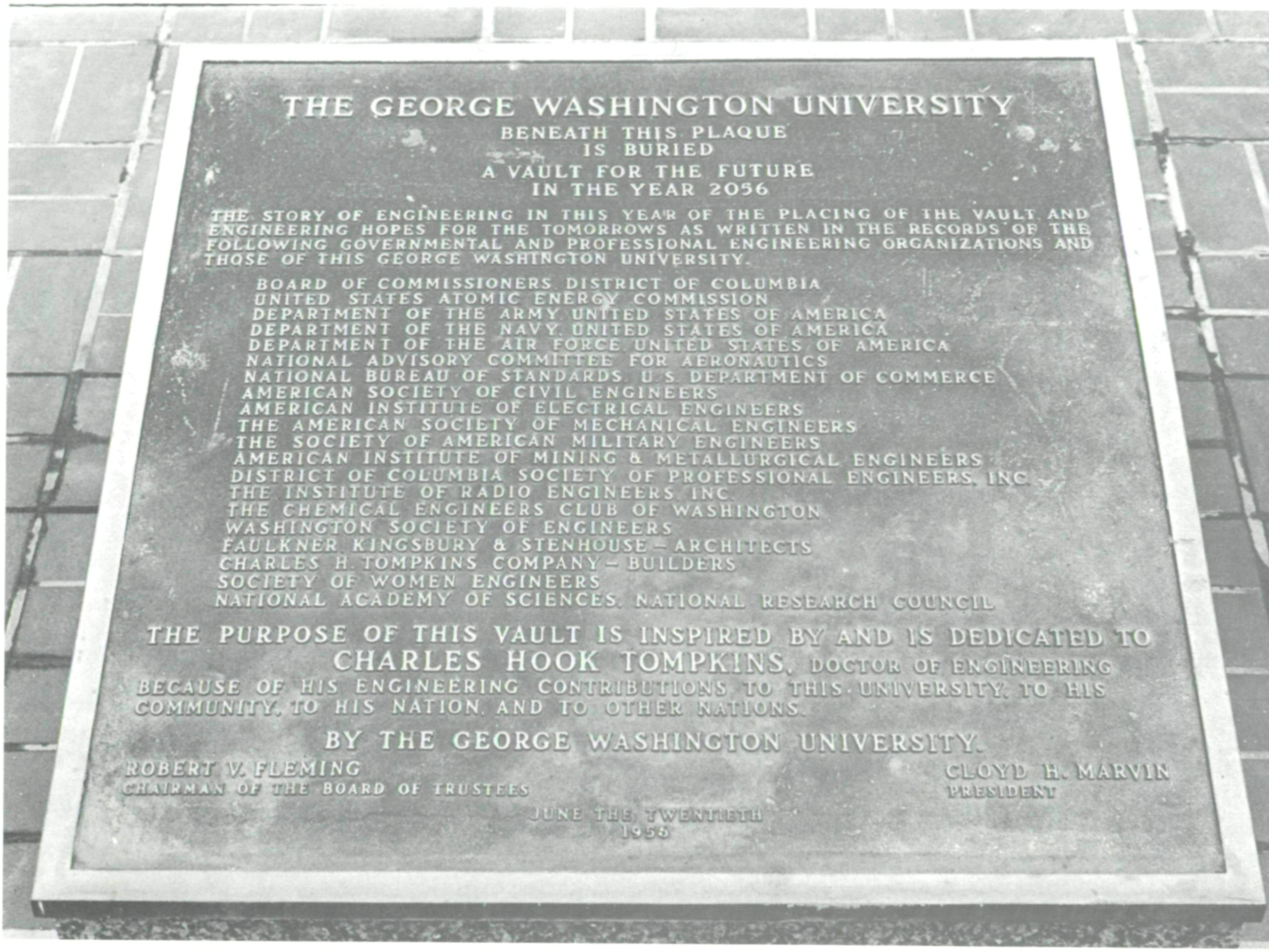

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