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# Design and Testing of a Heat Transfer System in the Constant Nusselt Number Range



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ABSTRACT

This paper presents the Thermal Control subsystem design for the Primate mission of the NASA Biosatellite program. A subsystem is described that provides temperature control for the fuel cell power source, cryogenic gases, miscellaneous liquids and the Gas Management Assembly. The latter provides control of the gaseous environment in the primate compartment. The trade-offs to determine the subsystem and component requirements are presented as are component, subsystem breadboard and system thermal vacuum test results. The component test program verified that all components met their requirements with the exception of one heat exchanger. Fortunately, the system requirements could be and were relaxed. The subsequent breadboard and system Thermal Vacuum tests verified that the Thermal Control subsystem met all of the system requirements.

ACKNOWLEDGMENT

A subsystem, such as the one in this report, could not have been accomplished by a single individual. Therefore, the author, who was the responsible subsystem engineer, wishes to acknowledge the support of the following individuals: P. Shelley who was responsible for the major components and was of invaluable assistance to the author, L. Pochettino and R. Ebersole for the analytical support and L. Henry for performing the test program.

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## Introduction

The Biosatellite Project was undertaken by NASA for the purpose of implementing a program of biological experiments in the environment of space.

The primate mission of the Biosatellite Project was to determine the effects of prolonged weightlessness of the behavior, the cardiovascular system, the metabolic functions, and the central nervous system of a primate for an orbital duration period of up to thirty days. The spacecraft attitude was random during orbit with a limitation on the tumbling rate to minimize the "g" forces on the primate.

The spacecraft consists of a re-entry heat shield, capsule (primate area), thrust cone and adapter as shown in Figures 1 and 2. The major subsystems required for the spacecraft are Thermal Control, Gas Management Assembly, Electrical Power and distribution, Gas Storage, Attitude Control, Telemetry-Tracking and Command, Separation and Life Support. The capsule is pressurized with a standard atmosphere throughout the mission and is the only part of the spacecraft recovered. The other parts of the spacecraft are subjected to a vacuum environment. All of the Thermal Control S/S equipment is mounted in the adapter with one exception which is located in the capsule.

The Thermal Control Subsystem was required because several of the major subsystems required temperature controlled coolant to provide heating or cooling. The spacecraft power is supplied by a fuel cell which is operated by cryogenically stored oxygen and hydrogen and as a by-product generates heat and water. The fuel cell requires temperature controlled coolant to function properly and to remove the generated heat. The by-product water is processed and some used for the primate and the remainder is available to the Thermal Control S/S. The capsule air temperature and humidity are controlled by the Gas Management Assembly utilizing coolant from the Thermal Control S/S as a heat sink. As temperature controlled coolant was available, it was used to heat the cryogenically stored gases and other liquids rather than using heaters. In addition it was used to control the temperature of the Pace/Rho experiment (named after the principal investigators, Drs. Pace and Rho) which analyzed the primate's urine for creatine, creatinine and calcium.

A) System Requirements

The Thermal Control subsystem (TC S/S) was required to provide temperature controlled coolant and remove heat from the Fuel Cell, the air to liquid heat exchanger of the Gas Management Assembly (GMA) and the Pace/Rho urine analysis experiment. In addition the TC S/S was required to heat the cryogenically stored gases and prevent the liquids contained in the adapter section from freezing. The liquids contained in the adapter are water (fuel cell by-product), urine and metabolic water. The detail requirements are given in the table below:

| | Flow Rate (lb/hr) | Coolant Inlet Temp. °F | Heat Load Btu/hr |
|---|---|---|---|
| Fuel Cell | 50 minimum | 40 to 75 | 200 to 375 |
| GMA | 50 minimum | 45 ± 3 | 175 to 480 |
| Pace/Rho | | 40 to 85 (1) | 0 to 21 |
| Cryogenics | | 45 to 105 | 0 to 55 (2) |
| Liquids | | greater than 35°F | |

(1) Pace/Rho mounting plate temperature

(2) Heat loss from the coolant (negative heat load)

Maximum allowable power usage - 28 watt at 26 ± 5 VDC

Maximum allowable weight - 55 lbs

It was requested that a concerted effort be made to reduce power usage as the spacecraft was power limited.

The environment was defined as a sink temperature ranging from -154°F to +40°F for the best obtainable optical coating.

The sink temperature is defined as that temperature to which the spacecraft radiates and is calculated using the heat fluxes from the sun and the earth and the spacecraft attitude. The variation in the sink temperature is caused by variations in the spacecraft-sun relationship, spacecraft - earth relationship and degradation of the optical coating due to ultra-violet from the sun. The sink temperature variation during one orbit for the hot, nominal and cold cases are shown in figure 3. A further discussion of heat fluxes can be found in Appendix A.

The GMA coolant inlet temperature requirement was based on the capsule relative humidity requirement and resulting dew point temperature rather than the capsule temperature requirements.

The fuel cell coolant temperature requirement was bracketed between the need for low temperatures to extend its life but high enough to prevent the water in the Fuel Cell from freezing. A more detailed discussion of the above can also be found in Appendix A.

B) Subsystem Design

The two major considerations in the design of the subsystem in addition to the system requirements, were providing flexibility and the unknown in the design of the components. The two major system interfaces, GMA and Fuel Cell, were to be designed and built by vendors and their initial requirements were not hard and changed constantly. Forcing the vendor to stay with his initial requirements would have had a greater program schedule and cost impact then providing flexibility in the Thermal Control S/S which was an in-house design.

The low flow rate and resulting low Reynolds number meant that the component vendor had to extrapolate their data as they had not previously designed equipment for this Reynolds number range. As a result, it was determined that a complete component test program was required to determine the components operating characteristics at the design point and off design points. The component test program in addition to providing data also gave us a feel for operating a subsystem at these low flows. A control analysis was attempted and found not to be possible due to lack of existing data. Therefore, it was decided that breadboard testing would be used to determine the control parameters and that whatever control system was used must be flexible and be adjustable during breadboard testing.

The maximum sink temperature (figure 3) indicated that for a portion of an orbit the radiator outlet temperature would be above that required to maintain the GMA inlet temperature. Therefore a topping device would be required to dissipate all or part of the heat load from the GMA.

The design of the subsystem was a series of tradeoffs based on the system requirements and the above design considerations. Below is a summary of the major tradeoffs and following this the full description of each tradeoff is given.

a) Dual vs Single Loop

The first tradeoff was to determine the basic subsystem configuration. In the single loop configuration shown in figure 5, the GMA and Fuel Cell are tied together while in the dual loop, figure 6, the two are independent of each other. The dual loop was chosen so that the GMA and Fuel Cell were independent giving the flexibility desired. In addition the water usage was less with the dual loop.

b) Regeneration vs Radiator By-Pass

This tradeoff considered two methods of controlling the coolant temperature downstream of the radiator. For regeneration a heat exchanger would be used to take heat from the coolant entering the radiator and transfer it to the coolant leaving the radiator. In the radiator by-pass method the coolant temperature would be controlled by by-passing hot coolant around the radiator and mixing it with the radiator outlet coolant. The regenerative method was chosen as in the by-pass method the coolant temperature could reach its pour point.

c) Control System

It was determined in this tradeoff that a mechanical valve actuated by temperature changes in the coolant, could be used to control the Fuel Cell inlet temperature as its required accuracy was only $\pm 15^{o}F$. The GMA inlet temperature requirement of $\pm 3^{o}F$ dictated an electrical control system. A digital system was chosen over an analog system as it would required less power, would meet the accuracy requirements, transient requirements and its control parameters could be varied during breadboard testing.

d) Topping Device

As previously mentioned, a topping device is required to dissipate the GMA heat load during peak solar periods. Two devices were evaluated: a boiler using excess fuel cell by-product water and a fusion thermal storage device. The boiler was chosen because there was sufficient water, it was lighter in weight and had been proven feasible in the Mercury and Gemini programs. In addition, if several extreme hot orbits were encountered, the fusion device would not meet its requirements while the boiler would.

Following is a detailed description of the major tradeoffs which were summarized above and some additional tradeoffs which had been performed. The results of these tradeoffs determined the design of the subsystem.

a) Dual vs Single Loop

The first tradeoff was to determine the basic configuration of the subsystem. The two possible configurations were a single loop, figure 5 or a dual loop shown in figure 6. The single loop has one pump and the GMA and Fuel Cell would be interelated. The dual loop has two pumps and the GMA and Fuel Cell are not related. The heat is transferred from one loop to the other and then radiated to space in the dual loop. Following is a more detailed description of each configuration.

1) Single Loop

The coolant from the radiator is heated in the regenerative heat exchanger and the outlet temperature is controlled by the digital control system to meet the GMA requirement. The coolant temperature would be controlled by the modulating valve bypassing the flow through the regenerative heat exchanger. The coolant from the modulating valve goes through the topping device and then to the GMA. To maximize the Fuel Cell inlet temperature, the pump was placed between the GMA and the Fuel Cell. The Fuel Cell inlet temperature therefore depends on the GMA heat load and the heat absorbed from the pump. The cryogenic heat exchangers are downstream of the Fuel Cell where the coolant is warmest and gas heating most efficient. After the cryogenic heat exchangers, the coolant enters the regenerative heat exchanger where it gives up some of its heat before entering the radiator.

2) Dual Loop

The coolant after leaving the radiator enters the interloop heat exchanger where it absorbs the heat from the GMA transported by the other loop. The coolant is then further heated in the regenerative heat exchanger to meet the requirements of the Fuel Cell. The coolant temperature is controlled by the mechanical valve bypassing the flow through the regenerative heat exchanger. The coolant from the valve goes to the pump and then to the Fuel Cell. After the Fuel Cell, it goes to the cryogenic heat exchanger and then to the regenerative heat exchanger where it gives up some of its heat before entering the radiator. In the other loop the GMA inlet temperature is maintained by the modulating valve controlling the flow through the interloop heat exchanger. After the GMA, the coolant flows through the topping device and then into the interloop heat exchanger where it transfers the GMA heat load to the other loop. When the topping device is required, the Fuel Cell loop can be decoupled from the GMA loop and the topping device only dissipates the GMA load.

In the single loop the Fuel Cell inlet temperature is dependent on the GMA, Pace/Rho and Pump heat loads, the range of Fuel Cell inlet temperatures can be calculated from an energy balance and is shown in Appendix A.

The computed range of Fuel Cell inlet temperature (50°F to 71°F) was not compatible with the original requirements of 75°F to 105°F. A further discussion of the changes in the Fuel Cell requirement is in Appendix A. This calculation also graphically shows the the dependence between the GMA heat loads and the Fuel Cell inlet temperature.

In the single loop the topping device would have to dissipate the Fuel Cell heat load in addition to the GMA heat load. This would result in a 75% increase in the heat load for the topping device. As a boiler was chosen for the topping device, it would result in a 75% increase in water usage which is not available. A Heat of Fusion topping device would require a 75% increase in weight and the regeneration cycle would be marginal. The dual loop requires six additional components with a combined weight of 5.3 lbs and a power penalty of 1.0 watts. Following is a table of the additional components.

| Qty | Name | Weight (lbs) | Power (watts) |
|---|---|---|---|
| 1 | Interloop Heat Exchanger | 3.5 | None |
| 2 | Check Valve | 0.5 | None |
| 1 | Filter | 0.6 | None |
| 2 | Additional Pump Section | 0.2 | 1.0 |

The only new component would be the interloop heat exchanger. The dual loop was chosen because it provided flexibility between the GMA and Fuel Cell requirements and early in the program it was very doubtful whether the Fuel Cell requirements could be lowered to that of the single loop. In addition with the single loop additional water would have to be carried for the topping device which would offset the weight of the additional components in the dual loop.

b) Regenerative vs Radiator By-Pass

This tradeoff considered two methods of maintaining the coolant temperature for the Fuel Cell during the colder cases. In the radiator bypass method, coolant would be by-passed around the radiator to maintain the required temperature. In the regenerative method a heat exchanger would take heat from the coolant entering the radiator and transfer it to the coolant leaving the radiator.

The reduction of flow through the radiator in the by-pass method does not reduce the heat transfer to space as we are in the constant Nusselt number range, as will be shown later. In the regeneration method the radiator inlet temperature is reduced resulting in a lower transfer to space which is a function of temperature to the forth power as this is radiation heat transfer. The lower heat transfer plus full flow result in a higher radiator outlet temperature. In the hot case there would be no regeneration and therefore the heat transfer would be identical for both methods. In the radiator by-pass method the coolant outlet temperature would be close to its pour point and therefore the regenerative method was chosen. The calculations for the radiator out temperature during the cold case for the radiator by-pass method is shown in Appendix A.

The computed radiator temperature of -137$^{\circ}$F is below the pour point of the coolant which is -120$^{\circ}$F. The above calculations were an approximation and the outlet temperature would be higher than that shown. Preliminary testing in this range showed that flow was unstable and that restarting the radiator required a higher temperature environment than anticipated. This was probably due to the thermal time constant as flow wants to re-establish itself in the center of the tube which is furthest from the increasing heat flux. As a result of this tradeoff, the regeneration system was chosen.

c) Radiator Configuration

This tradeoff determined the design of the radiator. The problem was to design a radiator which was large enough to handle the maximum case within the available water supply and maintain the outlet temperature in the cold case within the capability of a regeneration heat exchanger. A completely circumferential radiator was chosen because the spacecraft is randomly oriented in space and this would assure that the radiator sees fluxes from both the sun and earth as well as from space. The radiator area was sized for the maximum thermal load at a sink temperature of 0$^{\circ}$F, as the total mission time above this sink temperature would not exceed the water available for the boiler. The radiator area was also checked to assure that at the maximum sink temperature of 40$^{\circ}$F and maximum fuel cell load, the maximum fuel cell inlet temperature was not exceeded.

A computer program was used to find the radiator area which divided the radiator into 36 parts. The results of the program was a plot of radiator outlet temperature vs heat load for specific sink temperatures and radiator areas. As an example figure 7 shows radiator outlet temperature vs heat load for a 24 sq ft radiator and 0$^{\circ}$F sink temperature. The temperature drop through the tube was neglected as it is very small when compared to the film drop. The equations used for the computer program are in the Appendix.

With a constant radiator efficiency the requirements of both the hot and cold case could not be met. A radiator sized for the hot case would result in a cold case radiator outlet temperature of below -45°F. It therefore was necessary to reduce the radiator efficiency during the cold case, which was achieved by increasing the fin length during the cold case. The resulting design consisted of four parallel tubes equally spaced brazed to a aluminum sheet. During the hot case the four tubes are used and during the cold case only one end tube is used. This results in a large increase in the fin length during the cold case while the increase in flow has virtually no effect on the convention heat transfer as we are in the constant Nusselt number range. The computed radiator efficiency is 94% for the hot case and 57% for the cold case. The calculations are shown in the Appendix.

To assure that the maximum efficiency could be obtained the radiator had to be designed to achieve equal flow in the four tubes. Even though the radiator would function in a zero g environment head effects were taken into account to permit thermal balance testing with the spacecraft in the vertical attitude. Ground cooling tubes were also brazed to the radiator to permit ambient testing and pre-cooling of the radiator prior to launch. The ground cooling tubes were brazed to the outboard side and the flight tubes on the inboard side to protect them from handling damage which could cause leaks or flow unbalance.

The tube diameters were chosen based on pressure drop during the cold case and the space restriction between the spacecraft and the launch shroud. The pressure drops for the hot and cold cases are plotted in figure 8.

d) Thermal Control S/S Insulation from the Spacecraft

The anticipated maximum variation of the spacecraft skin temperature was 0°F to 100°F. As the hot and cold spacecraft skins would coincide with hot and cold radiator temperatures, the heat exchanger between the Thermal Control S/S and the spacecraft skin had to be minimized. A heat exchanger of less than 50 Btu/hr was established as a goal. As a result of this, all components in the Thermal Control S/S were thermally insulated from the spacecraft skin by mounting them on plastic sub structure and covering them with super insulation blankets. The super insulation blankets consist of multiple layers of aluminized mylar and have an effective emissivity of 0.01 when fastening techniques and end effects are included. All coolant tubing was mounted with plastic tube clamps and covered with super insulation blankets. The liquid lines which had to be maintained above freezing were traced with warm coolant lines and then both lines covered with a superinsulation blanket forming a oven. The subsequent thermal balance tests have verified that the above thermal insulation resulted in a heat exchange of less than 50 BTU/hr.

e) Control Systems

This trade off determined the type of control system that was required to maintain the GMA and Fuel Cell inlet temperatures within their requirements. As previously stated, a control analysis could not be done and therefore a requirement for an active control system was that it be flexible and be adjustable during breadboard testing. An additional criteria was to minimize the power required for the control system.

The fuel cell inlet temperature requirement of a control band of 35°F is well within the capability of a mechanical valve actuated by temperature changes in the coolant. In fact this type of valve is normally used for control bands as small as 10°F. The advantage of a mechanical valve is that is requires no electrical power and therefore it was chosen to control the fuel cell inlet temperature.

The GMA inlet temperature requirement of $\pm 3$°F accuracy dictated that an electrical control system be used. Both digital and analog control systems were considered. With an analog system the valve is driven by a servo motor requiring continuous power but would result in a continuous adjustment of the flow through the interloop heat exchanger. Early estimates by valve vendors indicated that seven watts would be required to drive the valve. In the digital system the valve would be driven by a stepping motor which would only require power when a change in the valve position is required. The power required to pulse a stepping motor of the size required is 10 watts for a duration of 10 milliseconds.

The digital system was chosen for the following reasons:

1) Tests on the fuel cell for the Gemini program showed that for our step changes in power the coolant outlet temperature would change less than 1/2°F per minute. Also it was estimated that the maximum rate of change from the GMA was less than 1°F per minute and would primarily be due to primate metabolic load changes. Therefore the predominate transient is due to the environment which is 7°F per minute in the radiator outlet temperature and occurs when the spacecraft crosses the sun line. As a result, during the eclipse portion of the orbit (approximately 40 minutes) little or no temperature change would occur and with a digital system no power would be required for the valve.

The maximum transient of 7°F per minute results in a required flow rate change of 3 lb/hr in a minute. This is a small rate of change and can be handled readily by a digital system.

f) Topping Devices

As previously mentioned, a topping device is required to dissipate the GMA heat load during peak solar periods. Two devices were evaluated: a boiler using excess fuel cell by product water and a fusion thermal storage device. The fusion thermal storage device will be discussed first.

The device would consist of a heat exchanger containing the fusion material through which the coolant would flow. During the hot phase, the fusion material would absorb heat from the coolant by going from a solid phase to a liquid phase. During the cold phase when an additional heat load can be placed on the radiator, the material would be regenerated. The logical choice for a fusion material would be water with a heat of fusion of 144 Btu/lb. The two disadvantages are its 10% volume change during phase changes and its melting point of 32°F which would not permit sufficient regeneration time during the eclipse portion of a hot orbit. Tetradecane (a paraffin wax) with a melting point of 40°F and heat of fusion of 98 Btu/lb could be used as its higher melting point would permit sufficient regeneration time. The concept is simple, but the design must allow for volumetric changes while maintaining good thermal contact between the fusion material and heat exchanger surface. The design must also include the differences in thermal conductivities between the liquid and solid state of the material. To incorporate the fusion thermal storage device in the subsystem, valving would be required to have it in the GMA loop, when required, for operation and in the radiator loop when being regenerated. Based on a 450 Btu/hr GMA heat load, 45 minutes of operation and a 50% safety margin, 5.2 lbs of fusion material is required. The estimated weigh is shown below:

| | Total Weight |
|---|---|
| Fusion Material | 5.2 lbs |
| Heat Exchanger | 3.5 lbs |
| Valves | 4.5 lbs |
| Additional excess water storage tank weight | 2 lbs |
| | 15.2 lbs |

The boiler dissipates the GMA heat load by boiling fuel cell product water and venting the steam to space. A water balance had to be established between the water produced and the primate needs. The minimum fuel cell water generation is 2.4 lbs/day of which the primate requirements are 1.1 lb/day, therefore leaving 1.3 hrs/day for the boiler. The allowable boiler operating time can now be calculated:

$$Q = W h_{fg} \qquad (1)$$

Q = heat dissipated

= boiler efficiency (90%)

W = Pounds of water available for boiler (0.054 lb/hr)

hfg = heat of vaporization (1070 Btu/lb)

Q = 52.1 Btu/hr

As the water balance is for the 30 day mission, a average GMA thermal load shall be used rather than a peak load.

GMA heat load = 350 Btu/hr

$$\text{Allowable boiler operation} = \frac{52.1}{350} \times 100 = 14.9\%$$

As can be seen in figure 4, the average time that the sink temperature is above $0^oF$, the radiator design point, is less than 15%. The feasibility of using a boiler in space was proven in the Gemini and Mecury program. The primary problems are associated with the zero "g" environment. In a one "g" field, the vapor rises due to its lighter weight, therefore the water stays in contact with the heat transfer surface. As differences in relative weights do not exist in a zero "g" field, other forces such as capillary forces must be used to assure water at the heat transfer surface. The estimated weight of the boiler was 7 pounds based on the previous designs.

The boiler was chosen as there was sufficient water, it was lighter in weight and had been proven feasible in previous space programs. In addition if several extreme hot orbits were encountered, the fusion device would not be able to regenerate while the boiler this would be no problem.

g) Cryogenic Boil-off Vs Minimum Power Load

This trade-off showed that a heater could be used to increase the minimum fuel cell power with no effect on fuel comsumption. Below a minimum power load the cryogents would boil-off to space due to the heat leak into the tanks. A heater bonded to the coolant lines results in a doulbe heat input as the Fuel Cell is 50% efficient, therefore the heat input would be the sum of the heater power plus increased fuel cell dissipation. This additional heat input would result in an increase in the radiator outlet temperature during the cold cases if the heater were thermostatically controlled. Hydrogen is the critical fuel and therefore the trade-off will be shown with hydrogen. The demand rate below which $H_2$ venting occurs is 0.011 lb/hr. The fuel cell power output at this consumption rate is 110 watts as can be seen in figure 9. The minimum spacecraft power requirement was 100 watts and therefore a 10 watt heater can be used to maintain a fuel cell power of 110 watts consistent with the $H_2$ flow rate.

h) Single vs Redundant Pump

The use of a backup pump in the subsystem was required by reliability considerations as the apportioned reliability of the subsystem was 0.9761 while the estimated reliability of the pump was 0.9560. All other components in the subsystem had an estimated reliability of 0.9870 and higher. Therefore, to approach the reliability apportionment, and as the pump reliability was significantly lower than any other component, a redundant pump was added. The redundant pump required the addition of 4 check valve and two differential pressure switches. The check valves are required to prevent back flow through the non operating pump. The differential pressure switch is required to determine pump failure and initiate automatic switch over to the backup pump. The reliability estimate for the use of a backup pump and the additional hardware was 0.9985 compared to a reliability estimate of 0.9560 for a single pump. The additional weight due to the added components is as follows:

| | |
|---|---|
| 1 pump | 4.0 lbs |
| 4 check valves | 0.44 lbs |
| 2 differential pressure switches | 0.60 lbs |
| Additional tubing and miscellaneous | 0.50 lbs |
| | 5.54 lbs |

As the coolant system is mission critical, the substantial reliability increase warranted the additional weight.

C) Component Requirements

As the basic subsystem has now been defined, the component requirements could be established and basic component designs traded off to determine the best basic design for this particular application. As the component designs firmed up, there was a constant interplay amoung the subsystem and the other components therein. Changes in other subsystems as they were firming up also had their effect on the component requirements.

The component requirements given in the following section are the final requirements except where noted.

a) Pump

This was considered the most critical component in the subsystem because of its power usage was a significant portion or the total spacecraft power generated and as previously stated the spacecraft was power limited. Preliminary power requirements for the pump were 25 watts and we were requested to work with the pump vendors to reduce this power requirement. The problem was that none had made a coolant pump for the low flow rates that we wanted and therefore the power estimates were based on extrapolations. Fortunately, by the time we had completed the subsystem tradeoffs, a pump vendor had completed development tests on the Lunar Excursion Module pump and even though it was twice our flow rate, he could from these results, guarantee a lower power requirement.

1) Requirements

The minimum flow and power requirements were placed upon the subsystem and the remainder were requirements from within the subsystem. The pressure for the pump was determined by estimating the pressure drop for each component in the loop and then adding a safety factor. The estimated pressure drops were then used as the requirement for each component. Following is a summary of the requirements.

| | GMA Loop | Fuel Cell Loop |
|---|---|---|
| Maximum Flow | 50 lb/hr | 50 lb/hr |
| Maximum Pressure Head | 13 psid | 33 psid |
| Minimum Coolant Inlet Temp. | 45$^{o}$F | 45$^{o}$F |

| | GMA Loop | Fuel Cell Loop |
|---|---|---|
| Maximum Coolant Inlet Temp. | 75°F | 75°F |
| Suction Pressure | 20.5 $\pm$ 1.5 psia | 20.5 $\pm$ 1.5 psia |
| Nominal Pressure | 8 psid | 15 psid |
| Nominal Coolant Inlet Temp. | 60°F | 60°F |
| Voltage | 26 + 5/-3.5 VDC | |
| Power at Nominal Conditions | 16 watts | |
| Power at Maximum Conditions | 22 watts | |
| High Pressure Relief Valve | 50 + 5/-0 psid | |

2) Tradeoff's

A gear pump, centrifical pump and vane pump were evaluated. The centrifical pump flow vs pressure drop characteristics could not meet the requirements as a pump designed to meet the 50 lb/hr flow rate at maximum pressure and minimum coolant inlet temperature, would require greater than 16 watts power at the nominal conditions. A gear pump for this pressure range would have a very small flow variation, so that the pump could be designed for almost constant flow. The gear pump could not meet the nominal power requirement as the internal frictional losses are too high. While the vane pump flow-pressure characteristics are not as flat at the gear pump, the internal friction losses are lower and it could meet our requirements.

A. C. and D. C. motors were evaluated for the pump. The fuel cell generates D. C. power, therefore no inversion and resulting loss would be required for a D. C. motor. The problem with D. C. motors is that they require commutation such as brushes or other sophisticated commutation methods, such as light rays. Commutation is a life and reliability problem and in addition does not permit a flooded motor design which will be discussed later. For the A. C. motor the available D. C. must be inverted to A. C. which results in a power penalty of up to 20%. In addition, start-up of a small single phase A. C. motor under load requires a special design. The trade off resulted in a pump - two phase AC motor - inverter combination designed by the pump vendor who could optimize the three sections for the maximum efficiency. The solid state inverter is more reliable than any commutation system and the two phase motor permits easy start-up.

A flooded motor vs dry motor with dynamic seal was also evaluated. In the flooded motor design the Coolanol would be allowed into the motor section. The advantage would be that all seals would be static and the heat from the motor would be absorbed by the coolanol which is a dielectric and therefore would not present electrical problems. The disadvantage is higher windage losses due to the fact that Coolanol would be between the rotor and starter. In the dry motor design, a dynamic seal on the shaft would be required between the pump and motor. The disadvantages are the inherent lack of reliability of a dynamic shaft seal, heat dissipation from the motor and increased shaft drag due to the seal. The flooded motor design was chosen as a result of this tradeoff.

3. Component Description

The pump section contains two vane elements, one for each loop, which are centrifugally extended. The pump bearings utilize the coolanol for lubrication and the motor section contains a 400 cycle - 2 phase A. C. motor which directly drives the vane elements. The rotor and starter were canned to provide a smooth surface to reduce windage losses. The inverter section contains solid state circuitry which provides 28 V - 400 Cycle - 2 phase power from the 26 volt D. C. input. The motor, inverter and pump sections are housed in a hermetic container.

b) Control System

1) Requirements

A subsystem tradeoff had determined that a digital system was best for our application, also that the system must have flexibility and be adjustable during breadboard testing. The system requirement on the controls was that it shall maintain 45 $\pm$ 3°F at the GMA inlet. Following is a list of the detail requirements:

Null point shall be 45 $\pm$ 0.5°F

Maintain GMA inlet temperature at 45 $\pm$ 3°F

Dead band shall be between 0.1°F and 0.3°F

Maximum coolant transient 7°F/minute

Provide boiler on and off logic

2) Tradeoff's

The major tradeoff was determining the null point tolerance and dead band. This was determined experimentally during breadboard testing using the following criteria:

1) GMA inlet temperature shall be 45 $\pm$ 1.5°F with a transient of 7°F/minute at the radiator outlet.
2) The control system shall not overshoot more than two pulses.
3) The control system shall not pulse more than once in 15 minutes under steady state conditions.

3) Component Description

Modulating Valve - The valve stem is driven through a gear train by a stepping motor and requires 1400 pulses from open to close. The valve is of the spindle type and as one port closes the other port opens.

Controller - The controller contains all the electronics and is of a cord wood module design. A bridge circuit with the thermistor remotely located in the coolant line measures the GMA inlet temperature error. The pulse rate is determined by the magnitude of the error with a maximum pulse rate of 200 pulses/sec. The error signal is decreased in time by a R-C circuit reducing the pulse rate and preventing overshooting. Therefore, for a step change the pulse rate starts at a rate equivalent to the error and then is decreased by the R-C circuit.

C) Heat Exchangers

1) Requirements

The requirements for the interloop and regenerative heat exchangers were generated within the subsystem. The pressure drop through the heat exchangers affected the pump pressure requirements and therefore the estimated values in determining the pump requirements were used as the requirement for the heat exchanger. The effectiveness of the regenerative heat exchanger is coupled to the radiator area and resulting water usage. The regenerative heat exchanger must be effective enough to maintain the fuel cell inlet temperature during the cold case with a radiator area selected for the hot case.

Following is a detail list of the requirements and the calculation to determine the resulting required effectiveness.

| | Interloop | Regenerative |
|---|---|---|
| Heat Transfer (Btu/hr) | 480 | 1500 |
| Loop A | | |
| Flowrate (lb/hr) | 55 | 50 |
| Inlet Temp. ($^oF$) | 38 | -29 |
| Max. Pressure Drop (psi) | 1.5 | 5.0 |
| Loop B | | |
| Flowrate | 55 | 50 |
| Inlet Temp. ($^oF$) | 66 | 63 |
| Max. Pressure Drop (psi) | 2.0 | 4.0 |

The effectiveness of both heat exchangers can now be calculated as follows:

$$E = \frac{Q}{(W\ cp)\ min\ \ (T_{hot\ in} - T_{cold\ in})} \qquad (2)$$

Eq. 12.18 b Heat, mass and momentum transfer by Rohsenow & Choi

| | | | Interloop H.E. | Regenerative H.E. |
|---|---|---|---|---|
| Q | = | Heat Transferred (Btu/hr) | 480 | 1500 |
| W | = | Flow Rate (lb/hr) | 55 | 50 |
| Cp | = | Specific heat (Btu/lb $^oF$) (Monsanto Chemical Co.) | 0.438 | 0.416 |
| T hot in ($^oF$) | | | 66 | 63 |
| T cold in ($^oF$) | | | 38 | -29 |
| E | = | Effectiveness | 0.711 | 0.784 |

2) Tradeoff's

To meet the pressure drop and heat transfer requirements of the regenerative heat exchanger a straight fin design was used. The disadvantage of this fin configuration is that with no crossflow if a passage is blocked by air the heat transfer area of that passage is lost. A stripped fin configuration was used for the interloop heat exchanger as this fin met the heat transfer and pressure drop requirements. This fin permits cross flow and therefore complete passages will not be blocked by air.

The heat exchangers were designed as flat plates rather than compact heat exchangers as the flat plates could be manufactured without introducing fluxes or brazing salts into the coolant passages. Fluxes or salts trapped in the heat exchangers can never be totally removed and would eventually cause fouling in the coolant system. The disadvantage is that a greater volume is needed, but by standing the heat exchangers on end a minimum of mounting surface is required.

3) Component Description

Both components are flat plate counterflow liquid to liquid heat exchangers. The regenerative heat exchanger has eight passes and the interloop has five. The manifolding between passes is done internally and is critical in the regenerative heat exchanger to assure good flow distribution.

d) Temperature Controller Valve

As previously shown in the control system tradeoff, the Fuel Cell inlet temperature could be maintained with a mechanical valve.

1) Requirement

The valve shall maintain an outlet temperature of $60 \pm 5^{o}F$ with one inlet varying from $85^{o}F$ to $55^{o}F$ and the other inlet from $65^{o}F$ to $-29^{o}F$. The Fuel Cell inlet requirement was $40^{o}F$ to $75^{o}F$ but a much tighter requirement was placed on the valve as this was the ideal temperature for the Fuel Cell to maximize its life. The maximum allowable pressure drop through the valve is 1.0 psi at a flow rate of 60 lb/hr and the maximum temperature rate of change at either inlet is $2^{o}F$/minute.

2) Tradeoff's

In addition to the mechanical valve a two-way solenoid valve was considered. A temperature sensor located at the outlet of the solenoid valve would determine which part of the valve to open. The electronics for this would be simple and by using a latching valve the average power could be less than one watt. The disadvantages are that there would be step changes in temperature which would be detrimental to the Fuel Cell and some power would be required.

The mechanical valve is controlled by the expansion and contraction of a temperature sensitive liquid contained in a bellows. This provides a smooth temperature control and requires no power. This valve is not as accurate nor as rapid to respond to transient changes but would meet our requirements and therefore was chosen.

3) Component Description

The valve is basically a cylinder and the internal mechanisms consist of two concentric bellows containing the temperature sensitive liquid. The outer bellows is rigid and the inner bellows is attached to the valve stem and is displaced by the expansion and contraction of the fluid. The coolant flows around the couter bellows from top to bottom and the convolution act as heat transfer fins.

e) Boiler

In the subsystem tradeoff's it had been determined that a boiler shall be used as the topping device. This tradeoff is to determine the basic type of boiler to be used.

1) Requirement

The boiler shall maintain an outlet temperature of 45 $\pm$ 3$^{o}$F under the following conditions:

| | | |
|---|---|---|
| Maximum heat load | - | 480 Btu/hr |
| Minimum heat load | - | 175 Btu/hr |
| Coolant flow rate | - | 55 $\pm$ 5 lb/hr |
| Transient | - | 40 Btu/hr/min |

The efficiency of the boiler shall be 90%, that is, water usage vs the heat dissipated. The boiler is allowed to exceed 48°F for the first two minutes of operation.

2) Tradeoff's

Three basic types of boilers were evaluated: Sublimation, Dynamic and Wick. Two development contracts were let, one each for the Dynamic and Wick type boilers and the Wick boiler was eventually chosen as a result of evaluating the designs which evolved from the development contracts. Following is a detailed description of the three basic types of boilers:

Sublimation Boiler

This boiler consists of a porous plate filled with water with the outer surface exposed to space. The water is ice at the interface as the space pressure is below the water triple point of .09 psia. The heat required to sublimate the ice is transferred from the coolant via fins inserted into the porous plate and the exposed surface area can be sized for the range of heat loads to be handled, within reason. Water is constantly fed into the inboard side of the porous plate and therefore no water control system is required and the coolant outlet temperature would be controlled by a bypass valve. This boiler design is the simplest of the three and is not affected by a zero "g" environment. The disadvantage is that the water remaining in the porous plate when the boiler turns off, would be sublimated by external heat fluxes from the earth and sun. The water lost as a result of this was estimated to be 0.05 lbs per operation. The available water as previously shown was 1.3 lbs/day. The boiler expected usage was once per orbit or 16 times per day. The water lost therefore is:

$$16 \times .05 = 0.8 \text{ lb/day}$$

Therefore, 60% of the available water is wasted and as a result this boiler concept could not be used.

Dynamic Boiler

This device consists of two heat transfer surfaces with water in the inside and coolant on the outside. The pressure and therefore the boiling temperature is controlled by a valve sensing the coolant outlet temperature. The maximum valve orifice is sized for the maximum heat load and therefore the maximum steam flow. The valve maintains a 45 $\pm$ 3°F coolant outlet temperature by sensing the outlet temperature and varying the boiler temperature. The valve which meters the water into the boiler measures the coolant inlet temperature which is a measurement of the heat load. The maximum valve orifice is sized again for maximum heat load. In addition a water solenoid valve is required to shut off the water during non-operational periods. With the two valves sensing as explained above it is possible to inject more water than required or vice versa during transient periods. The boiler core volume was therefore increased beyond that required for steady state.

The volume increase is small considering that the maximum water flow rate is 0.5 lb/hr. The steam path was made torturous to assure that water droplets carried by the steam would inpinge upon heat transfer surfaces and boiloff rather than be carried out to space along with the steam.

A boiler of this design was built and tested, and showed an inherent start-up time problem. The initial test results indicated a start-up time of 12 minutes versus specified maximum of 2 minutes. The problem was in establishing a well distributed water flow in the core. The mal-distribution resulted in reduced boiling area and therefore the outlet temperature could not reach 48°F. The mal-distribution was verified by building a Plexiglas model and using red colored water. The water inlet manifold was redesigned several times and wicking was used to improve the distribution. The best that could be achieved was a start up time of 8 minutes.

The other problem was control of the water inlet flow. The flow rates of 0.2 lb/hr to 0.5 lb/hr required an extremely small orifice in the valve. The valve in the boiler tested had upstream orifices and in addition, control orifices which were 7 mils in diameter. The control orifices were subjected to clogging which did occur several times during test. When monitoring the water flow, it was noted to be erratic. This occurred as a one mil change in orifice dia would result in a 30% change in flow rate. As a result of these problems, this boiler design was abandoned.

c) Wick Boiler

This boiler consists of a wick inner core from which the water is boiled off. As the water boils off, drying the wick, capillary action pumps more water up. The wick is spot welded to a inner core, the other side of which has fins to transfer the heat from the coolant. The vapor valve orifice is sized for the maximum heat load and the valve is cycled to maintain the coolant outlet temperature at $45 \pm 3^{o}F$. A thermistor measures the coolant outlet temperature and another thermistor the inlet temperature, this information is used to control the vapor valve. The water flow is controlled on a batch basis and enters a small reservoir from which the wicks pump it up to the boiling area. The water level is measured by a capacitance gauge which opens the water valve when the water falls below a predetermined level. The reservoir is actually a narrow annulus which will hold the water during the zero "g" environment. A boiler of this design was built and tested and has proven to be successful. The test results will be discussed in a later section.

d) Component Description

The wick type boiler is basically a 6 1/2" dia cyclinder 6" long. A inner cyclindrical core contains the wicks, steam compartment, water reservoir and the water level sensor. The coolant flows between the inner core and the outer cylinder in a counter flow direction. A vapor valve is located at the top of the cylinder and is driven by a rotary solenoid. The electronic control package which controls the vapor valve and water valve is bolted to the bottom of the cylinder. Two thermistors are spot welded to the inner core one to measure coolant outlet and the other to measure the coolant inlet. The location of the thermistors were determined by instrumenting the inner core and determining the most representative locations. The water valve is clamped to the outer shell of the boiler.

D) COMPONENT TEST RESULTS

As had been previously stated, a component test program was required as the low Reynolds number resulted in the component vendor's extrapolating their data to design our components. The test programs were designed to determine the operating characteristics of the components at there design point and at other possible operating conditions. In general all components met or exceeded their requirements and in cases where problems were uncovered, corrective action could be taken early in the program schedule. The test program also was a good learning period for us before starting the more sophisticated subsystem and system tests.

a) Pump

1) Summary of Results

The two development pumps tested, met or exceeded the design requirements. One pump required 13.2 watts and the other 13.5 watts at the design point versus 16 watts allowable. The minimum flow rate for both pumps was 51 lb/hr versus 50 lb/hr specified at the worst conditions of minimum voltage, maximum pressure differential and minimum coolant temperature inlet. The maximum power requirement did not exceed 19 watts vs the 22 specified under the worst conditions of maximum voltage, maximum pressure differential and minimum coolant inlet temperature.

2) Test Description

The test set-up consisted of two loops each containing a heat exchanger to control temperature, hand valve to impose the required pressure drop, accumulator to maintain pump suction pressure and a variable D. C. voltage power supply The instrumentation in each loop consisted of calibrated flow meters, inlet and outlet pressure gauges and thermocouples mounted on the tubing and pump housing. Voltmeters, ammeters and watt meters measured the electrical characteristics. A test schematic is shown in Figure 11.

3) Test Results

Design Condition

Coolant Temperature - 60°F

Fuel Cell Pressure Differential - 15 psi

GMA Pressure Differential - 8 psi

D. C. Voltage - 28 VDC

Suction Pressure - 20.5 psia

| Parameter | Pump #1 | Pump #2 |
|---|---|---|
| GMA | 59 p.p.h. | 57 p.p.h. |
| Fuel Cell Flowrate | 52 p.p.h. | 54 p.p.h. |
| Power | 13.2 watts | 13.5 watts |
| Maximum Power | 20.1 watts | 21.7 watts |

Coolanol Temperature - 45°F

Fuel Cell Pressure Differential - 33 psi

GMA Pressure Differential - 13 psi

D. C. Voltage - 31 VDC

Suction Pressure - 20.5 psia

| Parameter | Pump #1 | Pump #2 |
|---|---|---|
| GMA Flowrate | 51.5 p.p.h. | 51.0 p.p.h. |
| Fuel Cell Flowrate | 50.6 p.p.h. | 51.0 p.p.h. |
| Power | 14.6 watts | 14.8 watts |

b) Heat Exchangers

1) Summary of Results

Development tests of the interloop heat exchanger showed that it would transfer 524 Btu/hr versus the 480 Btu/hr specified. The pressure drop in loop A was higher than specified 2.95 psi vs 1.5. Decreasing the pressure drop would have required a redesign which could have been accomplished as the heat transfer exceeded the minimum required. The additional 1.5 psi was acceptable as the regenerative heat exchanger pressure drop was below the specified minimum.

The development test of the regenerative heat exchanger showed that the maximum pressure drop in either loop was 2.7 psi while the maximum specified was 5 psi. The heat transfer was marginal with test data ranging from 1640 Btu/hr to 1730 Btu/hr vs 1500 Btu/hr specified. The results showed that the test had to be run carefully with the heat

## 1) Interloop Heat Exchanger

| Test | Loop A (Fuel Cell) Flow Rate lb/hr | Loop A Temp Inlet °F | Loop A Temp Outlet °F | Loop A Pres. Drop Psi | Loop A Heat Transfer Btu/hr | Loop B (G.M.A.) Flow Rate lb/hr | Loop B Temp Inlet °F | Loop B Temp Outlet °F | Loop B Pres. Drop Psi | Loop B Heat Transfer Btu/hr |
|---|---|---|---|---|---|---|---|---|---|---|
| Design Point | 55.2 | 38 | 60.1 | 2.95 | 520 | 54.9 | 66 | 43.9 | 1.92 | 528 |
| Max. Pressure Drop | 60.0 | -36.3 | -7.8 | 11.8 | 684 | 24 | 55 | -18.5 | 2.31 | 736 |
| Off Design Point | 57.0 | -41 | -18 | 12.1 | 563.7 | 12 | 75 | -31 | 1.8 | 540 |

## 2) Regenerative Heat Exchanger

| Test | Loop A (Fuel Cell) Flow Rate lb/hr | Loop A Temp Inlet °F | Loop A Temp Outlet °F | Loop A Pres. Drop Psi | Loop A Heat Transfer Btu/hr | Loop B (G.M.A.) Flow Rate lb/hr | Loop B Temp Inlet °F | Loop B Temp Outlet °F | Loop B Pres. Drop Psi | Loop B Heat Transfer Btu/hr |
|---|---|---|---|---|---|---|---|---|---|---|
| Design Point | 50 | -24 | 57 | 1.31 | 1700 | 50 | 68 | -13 | 1.32 | 1730 |
| Max. Pressure Drop | 50 | -29 | 48 | 2.58 | 1588 | 50 | 63 | -16 | 1.07 | 1690 |
| Off Design Point | 60 | -25 | 39 | — | 1577 | 60 | 51 | -14 | — | 1645 |
| Off Design Point | 75 | -26 | 34 | — | 1837 | 75 | 44 | -13 | — | 1783 |

exchanger and the tubing well insulated and equilibrium established for at least one hour. The criteria for a good test run was that the heat transfer calculated on the hot side and the cold side balanced within 5%. The measurement of the temperature was extremely critical and three (3) thermocouples were placed into the coolant at each location and the average of the three used.

2) Test Description

The interloop heat exchanger was tested using two separate circuits each with its own pump and this permitted independent flow control for each loop. The thermocouples were located in the coolant approximately 3" from the ports and the flow was measured using specially calibrated flow meters. The test schematics is shown in Figure 12. The coolant lines and the heat exchanger were covered with closed cell foam insulation. The test setup for the regenerative heat exchanger was identical except that a single circuit was used so that the flow through both sides was identical. In calculating the heat exchanger effectiveness, the error of flow measurements was therefore eliminated. The pressure drop was measured using a "U" tube mercury manometer attached to tee's at the inlet and outlet ports. The system was deaerated prior to start of test by use of a vent valve and the absence of air was verified by using high flow rates and observing the flow through the flow meters. Air in the system affects the performance of the regenerative heat exchanger because it can block the heat transfer passages. Ten micron absolute filters were also installed in the system to assure that the heat exchangers would not be blocked by particule contamination.

c) Temperature Control Valve

1) Summary of Results

The valve met all of the requirements as specified in section C-3 and maintained an outlet temperature of 56°F to 63.5°F over the full range of inlet temperatures. The maximum pressure drop was 0.5 psi measured at an inlet temperature of -29°F at port A and 55°F at port B.

2) Test Description

A schematic of the test setup is shown in figure 13. The inlet and outlet temperatures were measured with thermocouples inserted into the tubing. The outlet temperature was measured 36" from the valve to assure that the fluid was not stratified and that a true average fluid temperature was measured. The valve and lines were insulated with 1" thick foam insulation and with the outlet temperature within 15°F of ambient the heat leak was negligible and therefore no error was introduced by the remote temperature location. The pressure differential was measured with a mercury "U" tube connected between the inlet port and the outlet ports.

3) Test Results

Inlet Temp. (°F)

| Port A | Port B | Outlet Temperature (°F) | Flow Rate (lb/hr) | Pressure Drop (psi) |
|---|---|---|---|---|
| -29 | 55 | 55 | 56 | 0.48 |
| -29 | 60 | 56 | 54 | 0.42 |
| 0 | 65 | 60 | 55 | 0.39 |
| 20 | 65 | 60.5 | 54 | 0.37 |
| 50 | 85 | 63.5 | 56 | 0.25 |
| 64 | 85 | 64 | 57 | 0.22 |

d) Two Position Valve

1) Summary of Results

The valve met its requirements with the exception that the flow distribution ranged from 60% to 69% rather than 70% to 80%. As a result of this, the internal valve orifice was resized and the remainder of the valves met this requirement.

Due to schedule limitations, this valve was not reworked but an external orifice was added to raise the flow distribution to 70% to 80%. The flow distribution range was less than 10% and therefore the orifice could be used.

2) Test Description

A schematic of the test setup is shown in figure 14. The U tube manometer in addition to measuring the pressure differential, was also used to assure that the inlet pressures to ports A and B were within 0.02 psi. If the inlet pressures were not equal, the flow distribution would not be a function solely of the valve. The coolant temperatures were measured with in line thermocouples but as no temperature mixing occurred, the outlet thermocouple was located near the outlet. The cross port leakage was measured by maintaining the inlet coolant below -12°F, opening the closed port and collecting the coolant leakage in a calibrated breaker.

3) Test Data

| Outlet Temp. (°F) | Total Flow lb/hr | Port A Flow lb/hr | Flow Dist. (%) | $\Delta$P Inlet to Outlet (psi) |
|---|---|---|---|---|
| 85 | 55 | 38 | 69 | 0.20 |
| 60 | 55 | 36 | 64.5 | 0.23 |
| 45 | 55 | 34 | 62 | 0.25 |
| 18 | 55 | 33 | 60 | 0.31 |
| 2 | 55 | 29 | 53 | 0.34 |
| -12 | 55 | 0 | -- | 0.85 |
| -12 | 60 | 0 | -- | 0.95 |
| -36 | 55 | 0 | -- | 1.1 |

e) Boiler

1) Summary of Results

The tests on the final configuration boiler showed that it met all of its requirements as specified. The boiler maintained the outlet temperature between 44°F and 48°F over the full heat load range. The outlet temperature reached 48°F in 1.8 minutes from start-up with the maximum heat load and a heat load variation of 40 Btu/hr/min resulted in an outlet temperature change of less than 1°F. The thermodynamic efficiency of the boiler was greater than 90% for all cases tested.

Initial testing on the boiler revealed several problems. One was leakage of the water valve which resulted in the boiler flooding in about 24 hours if it was not used. Since the boiler could be off for days during cold environments, the water valves had to be redesigned. A problem was also found in the thermistor to boiler core attachment. A poor joint results in the boiler core freezing as the sensor would read warm and keep the vapor valve open. This problem was corrected by changing the method of attachment to spot welding.

2) Test Description

A schematic of the test equipment is shown in figure 15. The tests were run with the spacecraft vent tube attached to the boiler and using processed fuel cell product water. The boiler assembly was placed in a vacuum chamber for the tests to assure a pressure of less than 0.2 inches Hg at the boiler exit. The vent line was cooled with liquid nitrogen to simulate space environment and no evidence of freezing was found in the vent tube.

3) Test Results

| Temp IN ($^{o}$F) | Temp OUT ($^{o}$F) | Flow Rate (lb/hr) | Heat Load Btu/hr | Start Up Time (min) | Water Usage (cc/hr) |
|---|---|---|---|---|---|
| 67 | 47 | 60 | 525 | 1.8 | 236 |
| 62 | 44 | 60 | 450 | 1.7 | 168 |
| 52 | 44 | 60 | 175 | 1.5 | 108 |
| 61 | 44 | 60 | 445 | 1.7 | 160 |

f) Temperature Measurement

Early in the component test program, test results were inconsistent when measuring coolant outlet temperatures when a valve was mixing coolants at widely different temperatures. Small adjustments of a valve resulted in up to 10$^{o}$F temperature changes at the outlet. A thermodynamic balance of the flow change showed that the outlet temperature change should not exceed 1$^{o}$F. The thermodynamic location in the tube was checked and found not to be touching the tube. Further investigation showed that one of the characteristics of a low Reynolds number flow is the persistence of a pronounced radial temperature gradient in the liquid, following the mixing of different temperature coolants. To verify this, three thermocouples were placed in the coolant lines at one point. One was placed at the center and the other two, 1/3 of the distance from the wall on either side of the center. Temperature differences of up to 40°F were measured 1" from a valve which mixed coolants of 75°F and -30°F. At a distance of 12" from the valve, the difference was reduced to 5°F and at 36" from the valve the difference was less than 1°F. As a result, temperatures at mixing valve outlets were always measured 36" from the valve. In addition at least 2 thermocouples were placed at each point to assure that no difference existed.

## E) Breadboard Testing

As previously stated, the next step in the test program was the breadboard tests. This test for the first time, put all the components together and proved that they would work together as a subsystem. The breadboard was also used in lieu of a control analysis to determine the control parameters for the Thermal Controller which controls the GMA inlet temperature. The breadboard testing included steady state and transient conditions.

a) Test Objectives

Following are the detail test objectives:

1) Determine the Thermal Controller null point tolerance and dead band such that the GMA inlet temperature will be $45 \pm 1.5^{o}F$ under all conditions. In addition the control system shall not over shoot more than 2 pulses during transient and it shall not pulse more than once in 15 minutes under steady state conditions.
2) Verify that the fuel cell inlet temperature will be controlled within the specified limits of $40^{o}F$ to $75^{o}F$ during all conditions.
3) Compare component breadboard data with its bench test data.
4) Determine system pressure drops.
5) Determine pump power and flow.
6) Determine overall loop response during radiator transients, GMA heat load transients, and Fuel Cell heat load transient
7) Determine the radiator outlet temperature at the boiler turn on point for various GMA heat loads.

b) Summary of Results

The breadboard tests verified that when all the components were put together as a subsystem, they would meet these requirements. The concept of using the breadboard to determine the control characteristics worked quite well and we were able to home in after several attempts.

The subsystem showed itself to be very stable under transient radiator conditions and heat load changes. Following is a more detailed summary:

1) With a dead band of $\pm$ 0.3°F about the null no pulsing resulted in steady state with up to a 3°F change in radiator outlet temperature and during transient conditions.
2) The Fuel Cell inlet temperature was maintained between 51°F and 66°F for all steady state and transient conditions.
3) The maximum pressure drop in the Fuel Cell loop was 31 psi and 6 psi in the GMA loop.
4) The maximum pump power was 16.5 watts at 26 VDC which occurred during the cold case steady state run.
5) The minimum flow in the Fuel Cell loop was 53 lb/hr and the minimum flow in the GMA loop was 50.5 lb/hr which again occurred in the cold case.
6) The Fuel Cell inlet temperature and GMA inlet temperature were controlled within their specified limits for all transient combinations including radiator outlet temperature, GMA heat load change and Fuel Cell heat load change. The transient results are shown in figures 18 and 19.
7) The boiler turn on point at which the radiator temperature is above 34°F and the modulating valve is in the full heat exchanger position was found to be a radiator outlet temperature of 40°F for a GMA heat load of 485 Btu/hr.

3) Test Specimen

A schematic of the test loop is shown in figure 16. All components in the loop were engineering development hardware which had previously undergone component level testing but the loop did not include a boiler, as the unit had not been completed. The tubing diameters and lengths were as close as possible to that which would be used in the flight spacecraft. All tubing was covered with a 1/2" thick foam insulation, the components were covered with 1" thick closed cell foam insulation and mounted on textolite insulation. The spacecraft side of the radiator was identical to the actual configuration and the other side of the radiator had 100 ft. of cooling tubes brazed to it and a 10 KW heater bonded on. Cooling was achieved by circulating the coolant through a bath which was maintained at -100°F utilizing liquid nitrogen. A combination of radiator bypass control and heater power was used to control the radiator outlet temperature. The GMA and Fuel Cell heat loads were simulated using heaters bonded to an aluminum plate to which coolant tubing was brazed.

4) Instrumentation

Thermocouple tee's were located in the loop as shown in figure 16. Each thermocouple tee contained three (3) 36 gauge copper-constanten thermocouples, one located at the center of the tee and the other two 1/8 inch away from center on each side. Thermocouples were located on the components as shown in figure 17. The flow was measured using flow meters calibrated with Coolanol 25 at 25°F increments, as the changes in Coolanol viscosity, affects the flow meter performance. The pressure across the pump was measured with pressure differential transducers having a range of 0 to 50 psi and a 0 to 15 psig pressure gauge was used to measure the coolant accumulator pressure. The pumps and thermal controller were supplied by D.C. power supplies having a voltmeter with a 0-40 VDC range with a accuracy of 0.05% of full scale, the ampere meter range was 0 to 1 amp with an accuracy of 0.25% of full scale. A oscillscope was used to measure the amplified temperature error signal of the GMA sensor. This permitted monitoring the dead band and the pulses to the modulating valve.

5) Test Procedure

To assure that each test run was valid the following had to be verified prior to start of the run:

a) Pump voltage within 0.2 VDC of that specified.
b) Gas pressure on the coolant accumulator at 5.5 $\pm$ 0.5 psig.
c) The three thermocouples at each location shall be within 2°F.
d) The GMA and Fuel Cell power simulation was measured electrical and then the heat input to the coolant calculated by measuring the temperature raise and the coolant flow. These two had to agree within 5%.
e) A complete heat balance of the system was made and the sum of the heat inputs had to be within 10% of the heat transferred in the radiator.
f) A stable radiator outlet temperature as defined by a change of less than 1°F in 30 minutes.

6) Discussion of Test Results

The GMA and Fuel Cell inlet temperature response as a function of both radiator temperature transients and step heat load changes are shown in figures 18 and 19 As shown in figure 19, the Fuel Cell inlet temperature varied between 46°F and 62°F (40°F to 75°F spec valve). The GMA inlet temperature as shown in figure 18 varied between 44°F and 46°F (42°F to 48°F spec valve). As no boiler was included in the breadboard, temperature control could not be maintained when the radiator outlet temperature was above 39°F at which point the system would normally go into boiler mode. Prior to start of this test, the loop was at equilibrum for three hours with a radiator outlet temperature of -42°F, Fuel Cell thermal load of 200 Btu/hr and a GMA thermal load of 165 Btu/hr. When the radiator temperature was increased, a step thermal load increase of 335 Btu/hr in the GMA was made. Conversly, when the radiator temperature was decreased, the heat loads were reduced to the original level by a step change.

In figure 19, inlet B is the bypass around the Regenerative Heat Exchanger and inlet A is the flow from the heat exchanger. As can be seen from fig 19, during the cold cases there was no bypass flow and stagnant coolant temperature is being measured. This can be seen by the drastic temperature drop at 40 minutes and also that inlet A and fuel cell inlet no longer coincide after this time.

F) Thermal Balance Test

This was the final engineering test of the subsystem, which was mounted in a development spacecraft in its flight configuration. The spacecraft was placed in a thermal vacuum chamber and subjected to the expected environmental extremes with the subsystem and its immediate interfaces being fully operational. The subsystem met all system requirements and during the transient phases it was demonostrated to have a large thermal time constant resulting in a reduced temperature range for the Fuel Cell.

a) Test Objectives

The overall objective of the test was to evaluate the design and prove that it would meet the system requirements when functioning in a spacecraft and subjected to environmental extremes. The detail objectives are listed below:

1) Determine radiator effectiveness.
2) Demonstrate the adequacy of the subsystem to maintain the GMA inlet temperature of $45 \pm 3^{o}F$.
3) Evaluate the performance of the subsystem during transient conditions.
4) Demonstrate the adequacy of the subsystem to maintain the Fuel Cell coolant inlet temperature within the required limits.
5) Demonstrate the ability of the subsystem to maintain the GMA coolant inlet at $45 \pm 3^{o}F$ during boiler operation.
6) Determine the heat leak from the subsystem to its surroundings.
7) Demonstrate the adequacy of the flight sensor locations to measure the coolant temperature.
8) Determine the Fuel Cell heater design.
9) Demonstrate the Fuel Cell heater design.
10) Demonstrate the adequacy of the subsystem to maintain the liquid filled components above freezing.
11) Demonstrate that the Pace/Rho cooling plate can be maintained between the required temperature limits.

b) Summary of Thermal Balance Test Results

The subsystem met all system requirements during all test phases. The GMA inlet temperature was maintained between 43°F and 47°F during both normal and boiler mode of operation and the Fuel Cell inlet temperature was maintained between 43°F and 63°F during all test phases.

The transient phases of the test demonstrated that the subsystem has a large thermal time constant therefore the temperature rate of change was significantly smaller than had been anticipated. This additional thermal damping had a positive effect on the system because it reduced the operating temperature range of the coolant system during transient environments. A mal-distribution of flow in the radiator resulted in a 88% effectiveness vs the design goal of 96% but the subsystem still met its requirement due to the large thermal time constant. Following is a list of other test results.

1) The Pace/Rho cooling plate was maintained between 43°F and 63°F.
2) All liquid lines and storage tanks were maintained above freezing.
3) The heat leak from the coolant system to the spacecraft was less than 40 Btu/hr.
4) The use of flight temperature sensors on the external surface of the tubing was verified as an acceptable method to measure coolant temperatures. The temperature difference between sensors on the outside of the tubing and those in the fluid did not exceed 2°F during the steady state phase and was 3°F during the transient phase.
5) The average pump power was 14.5 watts, maximum was 15.1 watts and minimum was 12.2 watts. The power for the thermal control assembly was less than 5 watts.

c) Test Specimen

All components of the Thermal Control S/S were mounted in a development spacecraft which consisted of a forebody (heat shield) capsule, thrust cone and adapter. All components not part of the Thermal Control S/S were thermally simulated in their shape, surface finish, mounting and thermal dissipation.

Mounting brackets, coolant tubing, interface surfaces, super insulation and optical coatings were as close to flight configuration as possible. The Fuel Cell was simulated with a component that was identical except heaters were used as a heat load. A water tank was located in the adapter to feed the water to the primate simulator and the tank was pressurized with $N_2$ to control feed rate and therefore the latent heat load.

d) Instrumentation

The location of the thermocouples in the coolant loop is shown in figure 20. The thermocouples were read out with a automatic scanning system coupled to a GE 225 computer and a model 35 Teletype Printer. With this system 400 thermocouples could be printed out in engineering units within four minutes of starting the scan. The computer was programmed to compute the average temperature for selected groups of thermocouples and their deviation. The fast data response time was required during the transient test to assure real time control of the test. Also, a special test panel was built to control all the subsystem operational components.

e) Test Procedure

Prior to placing the test spacecraft into the Thermal Vacuum Chamber, all parts of it were tested to assure that it would function properly in the chamber. After the chamber had been pumped down and the cryopanel activated, all component simulators were set for Phase I conditions as per Table 1. When the radiator outlet temperature reached $32^{o}F$, the radiator flux heaters were set for 60 watts/quadrant and the sink canister temperatures were set for Phase 1 conditions as shown in Table 2. After the canisters had stabilized, the pre-programmed radiator heat flux/time control system for the four radiator quadrants were activated and a five minute monitoring cycle of all test sensors was started. The test was continued until the temperature stabilized, that is, one orbit agreed with the preceeding orbit. After stabilization, the boiler was inhibited as a failure mode test. The starting sequence for Phases II and III were identical to Phase I. Phase I was the hot phase, Phase II was the cold Phase. During Phase II, transients heat loads were used in the capsule to simulate primate psychosomatic experiments. During Phase III, cold phase, whenever the Fuel Cell heat turned on the Fuel Cell simulated heat load was increased to account for the additional power usage.

f) Test Results

1) Fuel Cell Temperatures

The Fuel Cell inlet temperature was between 59°F and 62°F during the maximum case, 58°F and 59°F during the nominal case and 42°F to 47°F for the minimum case. The tighter inlet temperature control during the maximum and nominal cases results from the temperature control valve maintaining the inlet temperature at 60 ± 5°F while during the minimum case the valve inlet temperatures were outside of the valve control band and therefore the Fuel Cell inlet temperature was coupled closer to the environment. During the maximum case and inhibited boiler mode, the Fuel Cell inlet temperature varied between 64°F and 68°F which was within the 75°F maximum requirement. The maximum temperature raise of the coolant through the Fuel Cell was 11°F.

2) GMA Inlet Temperature

The GMA inlet temperature was controlled between 43°F and 46°F during normal operating and 46°F to 47°F during boiler mode. The effect of mode switching can be seen in the 50°F and the 42°F point. The 50°F point occurred during boiler pull down and was within the two minute allotted period and the 42°F point occurred at boiler off when the switching of the coolant valve allowed a cold slug to come out of the interloop heat exchanger. Neither of these short excursions had any effect on the capsule air temperature.

3) Water Usage

As can be seen in figure 21, there is adequate water available for the boiler operating time measured during this test. This is despite the fact that the radiator flow was not properly distributed. During a special steady state test phase, the boiler turn on sink temperature was found to be -24$^{o}$F for maximum heat loads and -12$^{o}$F for nominal heat loads. A review of the transient test data shows that the boiler turn on occurred 18 minutes later for the maximum case and 32 minutes later for the nominal case based upon the above mentioned test data. The turn off time log was 5 minutes for the maximum case and 2 minutes for the nominal case. This difference in time logs between increasing and decreasing environments resulted in the boiler on time being shorter than would be anticipated from steady state results. The shorter time lag during decreasing environments is caused by the radiator rejecting heat to a rapidly decreasing sink temperature. The time lag during the increasing environment is caused by the system storing part of the heat energy, therefore rejecting a smaller quantity and allowing the sink temperature to be higher. The water usage was within the requirements because of the difference between the time lags.

4) Two Position Valve

The valve did not close the three tubes of the radiator during the cold portion of the first three cycles. As a result, the radiator effectiveness did not decrease and the outlet temperature was colder than expected. The valve functioned properly after the first three cycles and a post test inspection showed that a gasket interfered with the valve operation and then was sheared off.

5) Regenerative Heat Exchanger

The regenerative heat exchanger appeared to perform well throughout the test. Analysis of the data showed its effectiveness to be 84% verses the component requirement of 80%.

6) Interloop Heat Exchanger

Analysis of the data showed a 86% effectiveness versus the component requirement of 75%. The increased effectiveness helped to minimize the boiler on-time as a higher radiator outlet temperature was possible before full heat exchanger flow was required on the GMA side.

7) Temperature Control Valve

The valve maintained an outlet temperature of 57 $\pm$ 1 1/2°F during the hot and nominal cases and was highly effective in dampening out the temperature transients. In the cold phase the valve inlet temperatures were below those specified and therefore the valve could not maintain the outlet temperature. The input transient to one port of the valve was 23°F and the valve damped it to 3°F transient.

G) Conclusions and Recommendations

a) Meeting of Requirements

The subsystem has met all of its requirements throughout an extensive vehicle test program for both the qualification spacecraft and the flight spacecraft. The spacecraft test program included a Thermal Vacuum Test with a live primate and operating fuel cell. During this test, the subsystem easily met its requirements and showed itself to have a significant margin.

The GMA inlet temperature requirement of $\pm$ 3°F has never caused a problem and, except for short (less than 2 minutes) mode switch transients, the inlet temperature has been $\pm$ 1.5°F. The final fuel cell inlet temperature requirement of 40°F to 75°F has been met but the initial requirement of 75° to 105°F would have required a 89% effective regenerative heat exchanger. This was the initial goal but was not achieved even after two redesigns. The low flow rate caused maldistribution in the heat exchanger and in addition the manufacturers do not have data on their fin performance for this Reynolds number and therefore they had to extrapolate existing data. To achieve a 89% effectiveness would have required a separate development program with a significant cost and schedule impact on the overall program.

The minimum flow requirements for both loops have been achieved in all cases and did not impose a severe restriction on the pump vendor. The high pump efficiency resulted in reducing the maximum subsystem power from 28 watts to 20 watts. This 8 watts decrease resulted in a 6% overall vehicle power reduction which now is the margin on the cryogenic fuel supply.

b) Test Program

The test program starting at component level, then breadboard and finally a subsystem test integrated with the spacecraft has proven to be invaluable. The low Reynolds' number flow resulted in many of the components being designed using extrapolated data and therefore a good test at the component level was required. The component level tests permitted evaluating them with a minimum of external interference, use of specialized instrumentation and testing of design conditions peculiar to the component. This testing also permitted us to perfect our test techniques as mistakes in the test set-up could be easily corrected due to its accessibility.

The breadboard testing combined the subsystem for the first time and permitted evaluation of the subsystem with a minimum of external effects. The test was designed so there was easy accessibility to the variable control functions in the Thermal Controller and they were optimized during this test. This could not have been done at the component level nor during spacecraft Thermal Balance Tests. It also was possible to use flow meters during the breadboard tests to evaluate the pump performance in the subsystem. The other flowmeters permitted evluation of the Regenerative Heat Exchanger - Temperature Control Valve system. Flowmeters could not be used in the Thermal Balance Tests as any remote indicating flow meter would introduce a significant pressure drop. The Thermal Balance test was of course the final engineering proof of the design. During this test all external effects were introduced and the performance of the subsystems evaluated. The test was done early in the overall cycle and as a result system changes have occurred and will continue to occur, which have an effect on the subsystem. With the data from the test, the changes can be evaluated and a positive statement made as to the effect of the change on the subsystem. In summary, the importance of the complete test program cannot be overemphasized in achieving a subsystem design which met its requirements.

c) Higher Flow Rate

The flow rate of 55 lb/hr resulting in a 100 or lower Reynolds Number created both equipment design and test problems which have been previously discussed. The low flow rate was chosen to conserve power as it was the limiting item in the system design. Pump power is a function of efficiency, flow rate and pressure drop. A increase in flow rate would cause a power increase and a pressure drop increase results in a further power increase. The pressure increase could be minimized as a portion of the pressure drop in the heat exchangers was to maintain flow distribution which could be kept constant for the higher flow rates. Pump-Motor efficiences increase with increased hydraulic output and this is especially true for our low flows. It would be the recommendation of the author that for a new system a detailed tradeoff be made of hydraulic output versus pump-motor effiencies to achieve the lowest power usage and yet avoid the problems associated with the extremely low Reynold's numbers.

## Afterward

Prior to publication of this report, the Biosatellite spacecraft was launched from the Eastern Test Range, Cape Kennedy. The Thermal Control Subsystem met all of its systems requirements as specified on pages 1 and 2. The Fuel Cell inlet temperature was maintained between 50°F and 55°F except when the Pace/Rho experiment was turned off at which time it reached 46°F. This is well within the required temperatures of 40°F to 75°F. As the Pace/Rho Experiment and the Fuel Cell Controller are controlled by the same coolant, their baseplate temperature would have been within the same range. No heater power was required throughout the mission indicating that the subsystem has a 20 watt heater power reserve for minimum load/minimum environment conditions.

The GMA Inlet Temperature was 44.0° $\pm$ 0.5°F verses the specified range of 42°F to 48°F. The liquids in the adaptor were kept from freezing as there was no evidence of blockage in the Water, Urine and Metabolic Water Systems. Because of the lower heat load, the Boiler was not required to dissipate the heat from the GMA. The passive control system maintained the electronic components within their specified temperature limits.

## Appendix A

A) Heat Fluxes

The spacecraft is randomly oriented during the mission and is rate limited minimizing the gravity forces on the experiment. A orbital analysis showed that the spacecraft would be coning about the velocity vector (nose forward) at the lower attitudes (125 to 150 miles) and coning about a gravity gradient (nose down) above 150 miles. The analysis also showed that the spacecraft would not be stable due to the constant shifting of the center of gravity as fuel and attitude control gases are consumed. The range of heat fluxes were then found for the attitudes considered and the sink temperatures calculated using the following equation:

$$T = \left[ \frac{\frac{\alpha}{\epsilon} (S + A) + E}{\sigma} \right]^{1/4} \qquad (1)$$

T = Sink temperature $^{o}R$

$\alpha$ = Solar absorptivity of surface -0.20 $\pm$ 0.03

$\epsilon$ = Emissivity of surface - 0.86 $\pm$ 0.03

S = Solar heat flux for vehicle attitude - Btu/hr

A = Solar heat flux reflected from earth for spacecraft attitude - Btu/hr

E = Heat flux from earth for spacecraft attitude - Btu/hr

$\sigma$ = Stefan-Boltzmann constant $0.178 \times 10^{-8}$

B) Fuel Cell

The Fuel Cell was similar to that used on the Gemini Program where it had an operational life requirement of 14 days. The ground rules were to do nothing that would result in a design change of the Fuel Cell which dictated the use of Coolanol and a minimum flow rate of 50 lb/hr. Extending the life of the Fuel Cell from 14 to 30 days required decreasing the Gemini inlet temperature of 100 $\pm$ 20°F. The minimum Fuel Cell coolant temperature was unknown in the early stages of the program as water separation problems were encountered at coolant temperatures below 70°F and therefore the initial requirement was 75°F to 105°F. The large unknown in the Fuel Cell temperature requirement had a significant effect on the system design. The range of Fuel Cell heat loads include both electrical load variations and degradation of the Fuel Cell over its mission life.

C) Gas Management Assembly (GMA)

One of the functions of the GMA is to maintain the capsule atmosphere at $75 \pm 5^{o}F$ and a relative humidity of $55 \pm 15\%$. The coolant inlet temperature requirement of $45 \pm 3^{o}F$ was dictated by the humidity requirement rather than the dry bulb temperature. The minimum dew point temperature which occurs at $70^{o}F$ and 70% R.H. is $59^{o}F$. The $48^{o}F$ maximum coolant inlet temperature is required to achieve a $59^{o}F$ dew point.

D) Fuel Cell Inlet Temperature - Single Loop Configuration

In the single loop configuration the Fuel Cell inlet temperature is dependent on the GMA, Pace/Rho Urine Analysis Experiment and Pump heat loads. The range of Fuel Cell inlet temperatures can be calculated from an energy balance and is shown below:

$$T1 = T2 + \frac{1}{WCp} (Q2 + Q3 + Q4) \qquad (2)$$

$T_1$ = Fuel Cell Inlet Temperature ($^{o}F$)

$T_2$ = GMA Inlet Temperature ($^{o}F$)

W = Coolant Flow Rate (lb/hr)

Cp = Coolant Specific Heat $\left(\frac{BTU}{(lb - {}^{o}F)}\right)$

Q2 = GMA Heat Load (BTU/Hr)

$Q_3$ = Pace/Rho Heat Load (BTU/Hr)

$Q_4$ = Pump Heat Load (BTU/Hr)

To determine the maximum Fuel Cell inlet temperature the following values were used:

T2 = Maximum GMA inlet temperature $48^{o}F$

W = Minimum Flow Rate 50 lb/hr

Cp = $0.44 \frac{BTU}{lb-{}^{o}F}$ Monsanto Chemical Company

$Q_2$ = Maximum GMA Heat Load $480 \frac{BTU}{Hr.}$

$Q_3$ = Maximum Pace/Rho Heat Load 21 BTU/Hr

$Q_4$ = Pump Heat Load Assuming 14 watt

Pump power and 10% efficiency 43 Btu/hr

Maximum Fuel Cell Inlet Temperature = $70.8^{o}F$

To determine the minimum Fuel Cell inlet temperature the following values were used:

$T_2$ = Minimum GMA Inlet Temperature $42^{o}F$

W = Maximum Estimated Flow Rate 60 lb/hr

Cp = $0.44 \frac{BTU}{lb\text{-}^{o}F}$ Monsanto Chemical Company

$Q_2$ = Minimum GMA Heat Load $175 \frac{BTU}{Hr.}$

$Q_3$ = Minimum Pace/Rho heat load 0 Btu/Hr

$Q_4$ = Pump Heat Load 43 BTU/hr

Minimum Fuel Cell Inlet Temperature = $49.5^{o}F$

E) Radiator Inlet Temperature

Following is the calculation for the radiator outlet temperature if the radiator by-pass method were used to control the radiator outlet temperature.

1) Radiator Inlet Temperature

The radiator inlet temperature would equal the Fuel Cell outlet temperature if no cryogenic gas heating was required:

$$T_2 = T_1 + \frac{Q_1}{WCp} \quad (3)$$

$T_2$ = Radiator outlet temperature ($^{o}F$)

$T_1$ = Fuel Cell inlet temperature = $75^{o}F$

Q = Fuel Cell Heat Load = $200 \frac{Btu}{He.}$

W = Coolant Flow Rate = 50 lb/hr

Cp = Specific Heat of Coolanol = $0.44 \frac{Btu}{lb\text{-}^{o}F}$

$T_2$ = $83^{o}F$

2) Reynolds number of coolant in radiator

$$\text{Reynolds Number} = \frac{DV}{\epsilon\ r} \quad (4)$$

D = Radiator tube I.D. = 0.0283 ft.

V = Coolant Velocity = 0.143 ft/sec

$\rho$ = Coolant Density = 58 lb/ft$^3$ Monsanto Chemical Co.

$\mu$ = Coolant Visconsity = 0.0404 lb/ft-sec

Viscosity at -30$^{\circ}$F, assumed average viscosity for first approximation.

Reynolds Number = 17

Note the extreme low Reynolds number

3) Prandtl Number of Coolant in Radiator (5)

Prandtl Number = $\frac{Cp}{k}$

Cp = 0.45 $\frac{Btu}{lb\text{-}^{\circ}F}$

= 0.0404 lb/ft sec

k = Thermal Conductivity = .08 $\frac{BTU - ft}{hr - ft^2 - {}^{\circ}F}$

Monsanto Chemical Co.

Prandtl Number = 818

4) Nusselt Number

The Nusselt Number for this low flow rate can be found in Figure 7.20 of "Heat, Mass and Momentum Transfer" by Rohsenow and Choi which plots Nusselt number as a function of:

$$\frac{X}{D} \quad \frac{1}{Re} \quad Pr \qquad (6)$$

X = Distance from inlet of radiator = 94 in. at center of radiator

D = I.D. of tube = 0.34 in.

Re = Reynolds number = 17

Pr = Prandtl number = 818

$$\frac{X}{D} \quad \frac{1}{Re \; Pr} = 0.0199$$

From figure 7.20 Nusselt No. = 4.4

From figure 7.20 the Nusselt No. at a infinite length from the tube entrance for uniform wall temperature is 3.66.

A Nusselt No. of 3.66 will be used as it is conservative.

Nusselt No. $= \frac{hD}{k} = 3.66$ (7)

h = Coefficient of heat transfer $\left(\frac{BTU}{Hr\text{-}ft^2\text{-}^oF}\right)$

D = I.D. of tube = 0.34 in.

k = Thermal Conductivity = .08 $\frac{BTU\text{-}ft}{hr\text{-}ft^2\text{-}^oF}$

h = 10.4 $\frac{BTU}{hr\text{-}ft^2\text{-}^oF}$

5) Temperature Differential from Coolant to Radiator Sheet

The temperature drop across the tube wall and the radiator sheet are neglected for this tradeoff. The temperature differential therefore is dependent only on the coolant film coefficient of heat transfer.

$Q = h A \Delta t$ (8)

Q = Minimum system heat load = 399 $\frac{Btu}{Hr}$

h = Coefficient of heat transfer = 10.4 $\frac{Btu}{hr\text{-}ft^2\text{-}^oF}$

A = Tube heat transfer area - 0.34 I.D.

Tube attached to 60" dia. radiator - 1.39 $ft^2$

$\Delta t$ = temperature drop ($^oF$)

$\Delta t$ = 27.6$^oF$

6) Average Radiator Sheet Temperature

With the heat load and the sink temperature known, the average radiator temperature can be found.

$Q = \quad EA\ (T_1{}^4 - T_2{}^4)$ (9)

Q = Minimum system heat load = 399 $\frac{Btu}{Hr}$

= Stephen-Balzman constant = $0.178 \times 10^{-8}$

= Radiator efficiency = 0.60 for the cold case as shown in the next section

E = Emissivity = 0.85

A = Radiator area = 24 $ft^2$ - see Section (F)

$T_2$ = Minimum sink temperature = -154$^oF$ - See page 1

$T_1$ = Average radiator temperature ($^oF$)

$T_1$ = -55$^oF$

7) Average Coolant Temperature

Knowing the average radiator sheet temperature and the temperature gradient from the coolant to the radiator sheet, the average coolant temperature can be found.

$T_2 = T_1 + \Delta T$ (10)

$T_2$ = Average coolant temp -°F

$T_1$ = Average radiator sheet temp. (-55°F)

$T$ = Temperature gradient (28°)

$T_2$ = -27°F

8) Coolant Radiator Outlet Temperature

The radiator outlet temperature now can be found as the inlet and average coolant temperature are now known:

$$T_2 = \frac{T_1 + T_3}{2} \qquad (11)$$

$T_2$ = Average Coolant Temperature (-27°F)

$T_1$ = Coolant inlet temperature (83°F)

$T_3$ = Coolant outlet temperature -137°F

F) Radiator Area

The following two equations were used in the computer program to determine the radiator outlet temperature vs heat load for specific sink temperatures and radiator areas:

$$Nu = Nu + \frac{K_1 \left(\frac{D}{X}\right) Re \; Pr}{1 + K_2 \left(\frac{D}{K}\right) Re \; Pr^{N}} \qquad (12)$$

Equation 7.71 - "Heat Mass and Momentum Transfer" by Rohsenow and Choi

With the Prandtl number of 818 as previously shown, the case of parabolic velocity and uniform temperature was used to find the constants on page 166 of the above reference.

Nu = 3.66

$K_1$ = 0.0668

$K_2$ = 0.04

N = 0.0667

Re = Reynolds number

Pr = Prandtl number

D = I.D. of tube = 0.34 in

X = Distance from entrance (in)

Press Information

GENERAL ELECTRIC
Re-entry & Environmental
Systems Division
3198 Chestnut St., Philadelphia, Pa. 19101

Alan D. Johnson
Phone: 215 823-3203

FOR IMMEDIATE RELEASE

Artist's concept shows 15-pound monkey orbiting the Earth in a Biosatellite spacecraft. Some of the major parts of the spacecraft are called out. General Electric's Space Re-entry Systems Programs developed the Biosatellite spacecraft and integrated its experiments under the direction of the National Aeronautics and Space Administration's Ames Research Center. The Space Re-entry Systems Programs organization is part of GE's Re-entry and Environmental Systems Division. Specific objective of the primate mission is to gain insight into basic physiological phenomena including identification of possible hazards to man of prolonged space flight. The Macaca Nemestrina monkey, now in orbit, will remain in space for up to a month before recovery.

- 30 -

PLEASE CREDIT: General Electric Company
Space Re-entry Systems Programs
Re-entry and Environmental Systems Division
Philadelphia, Pa.

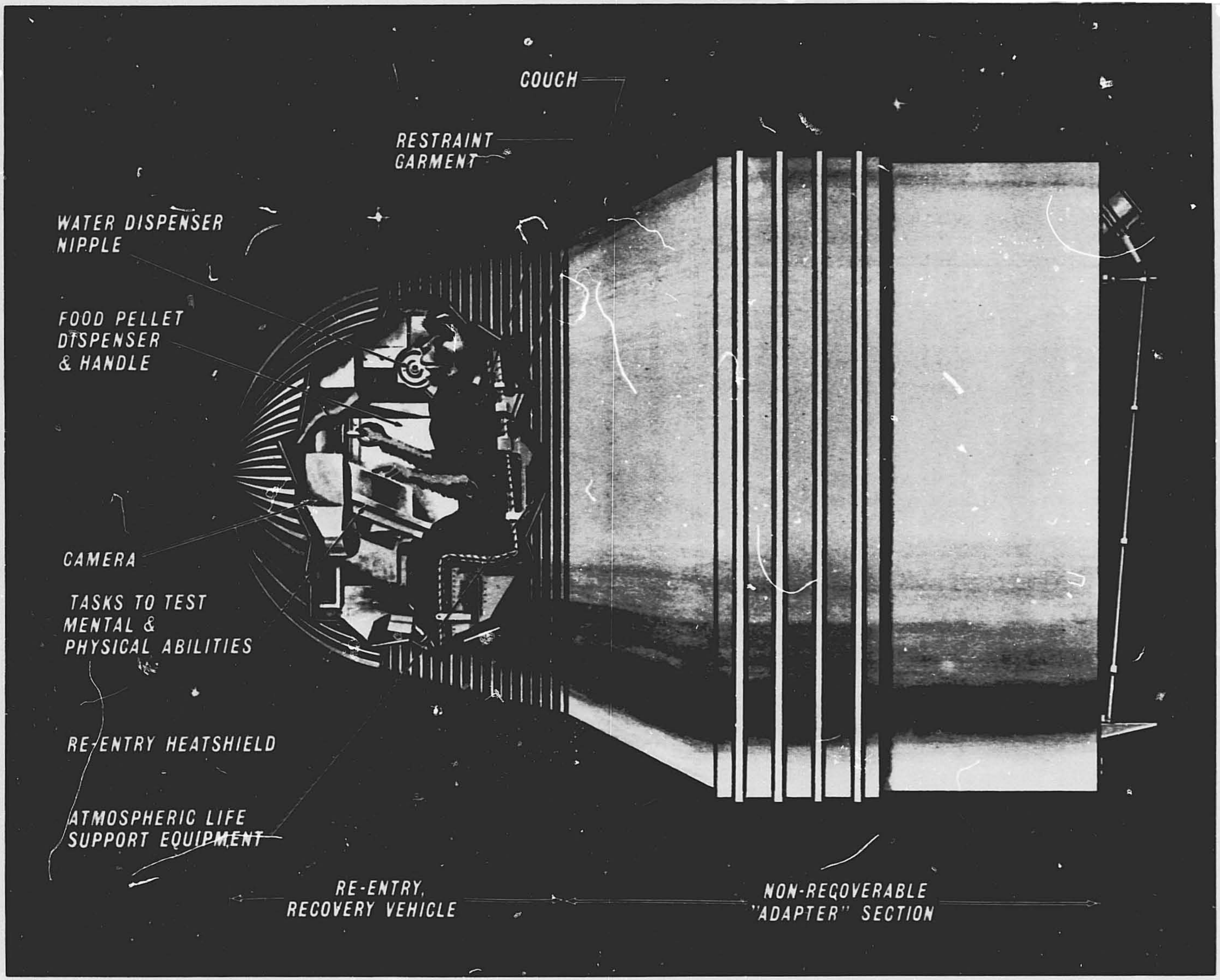
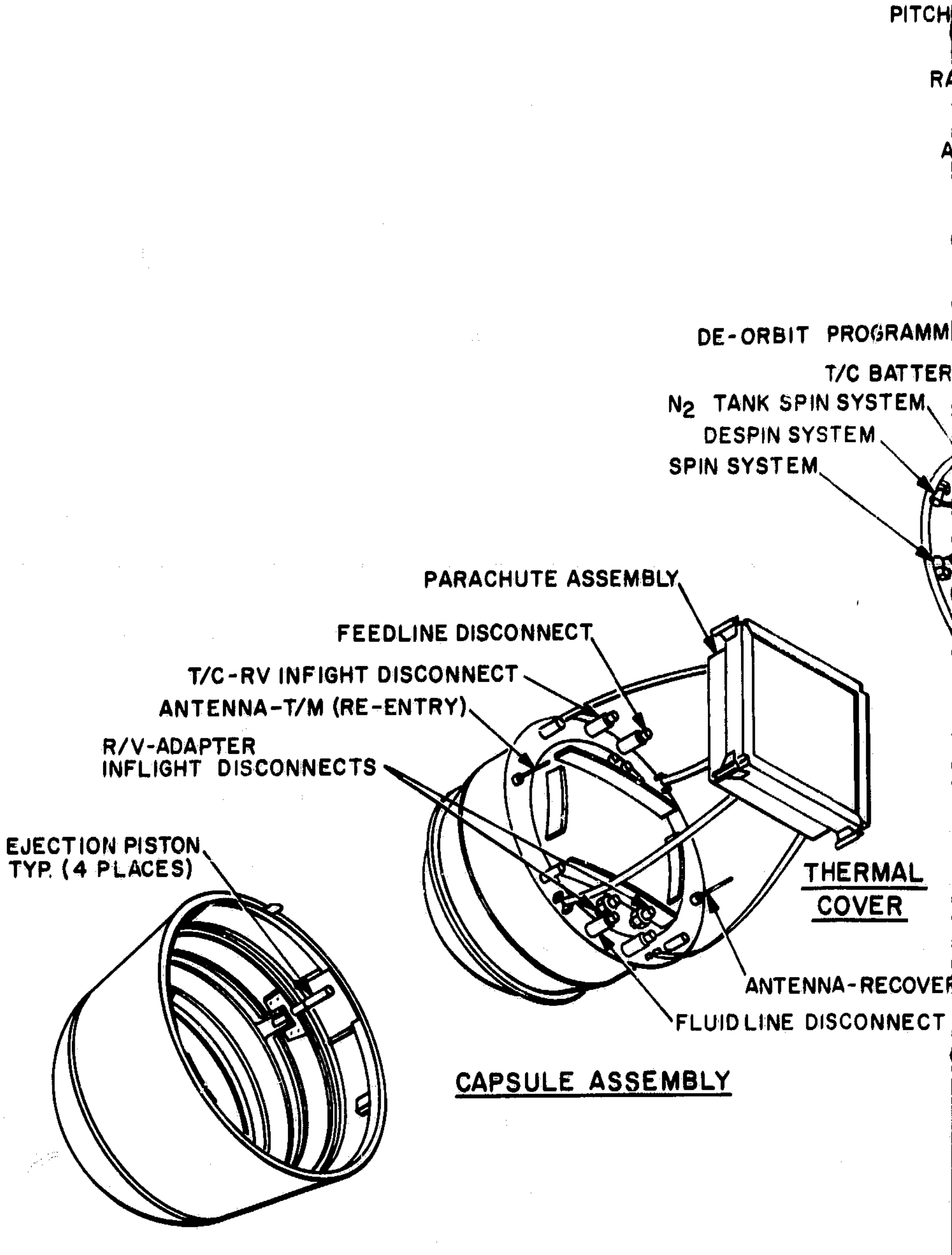

COUCH
RESTRAINT GARMENT
WATER DISPENSER NIPPLE
FOOD PELLET DISPENSER & HANDLE
CAMERA
TASKS TO TEST MENTAL & PHYSICAL ABILITIES
RE-ENTRY HEATSHIELD
ATMOSPHERIC LIFE SUPPORT EQUIPMENT
RE-ENTRY, RECOVERY VEHICLE
NON-RECOVERABLE "ADAPTER" SECTION

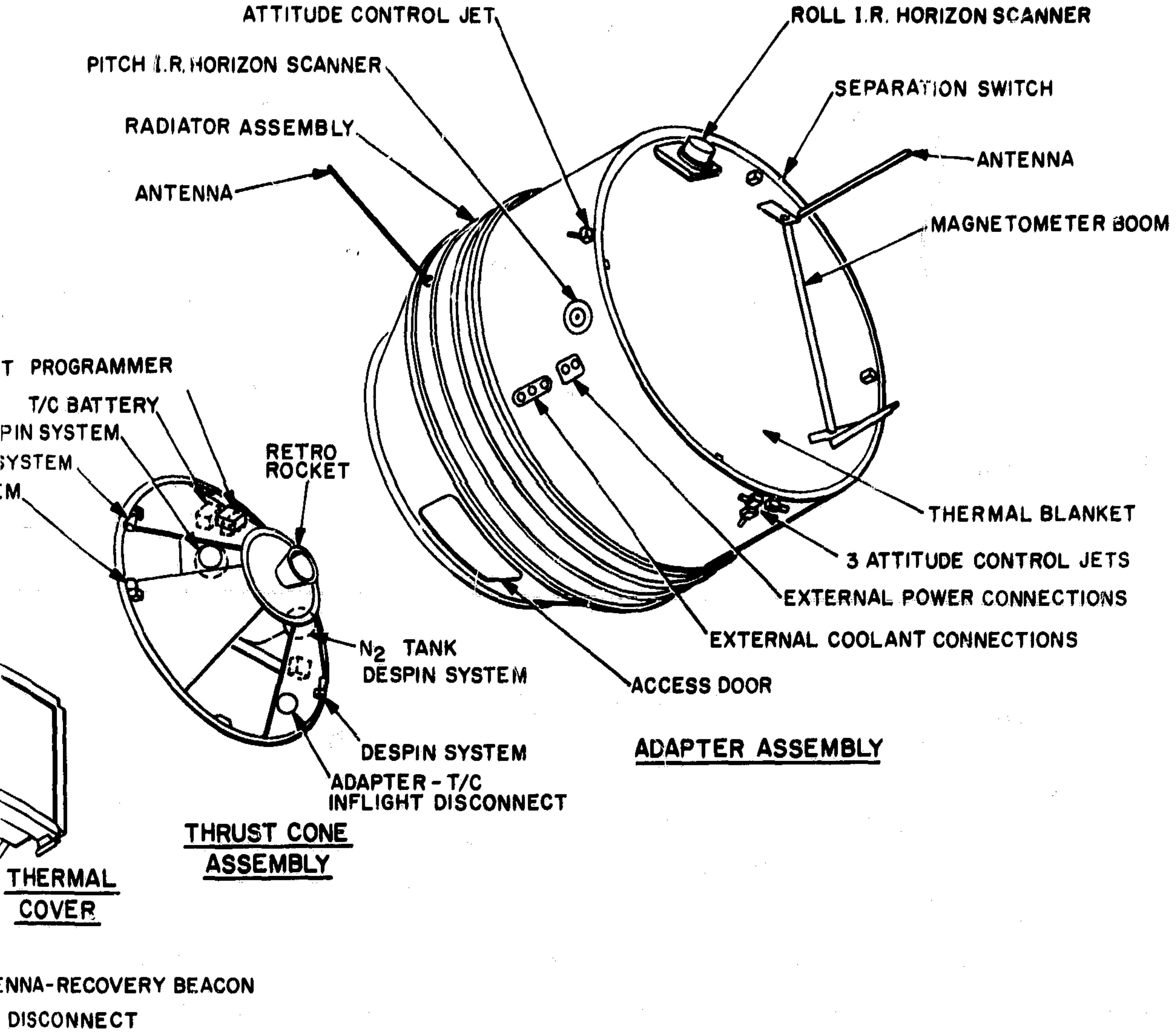
FIG. 1

**BIOSATELLITE SPACECRAFT-E**
**PRIMATE MISSION C**

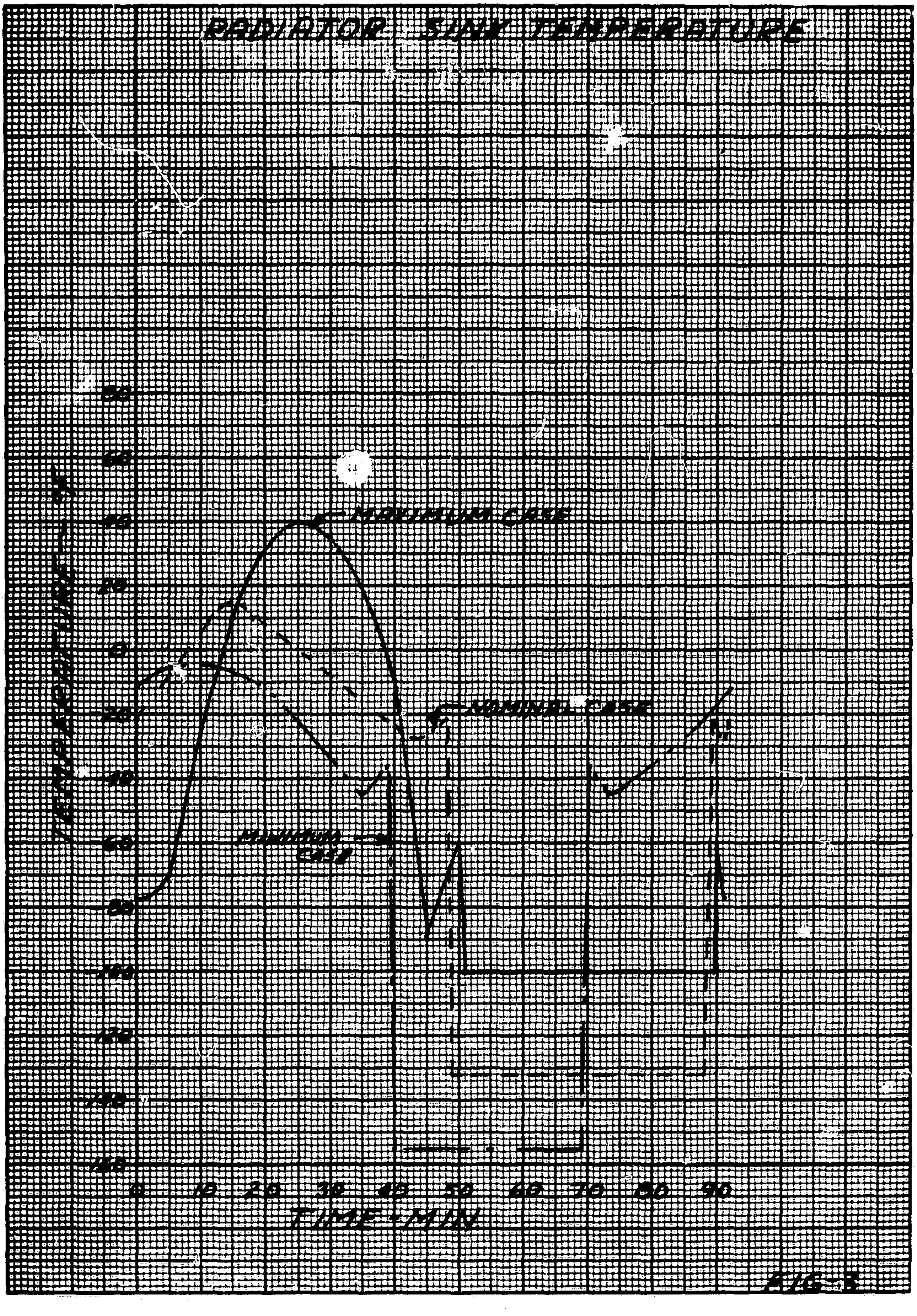



**FOREBODY ASSEMBLY**

ECRAFT - EXPLODED - ISOMETRIC VIEW
E MISSION CONFIGURATION



359-11G


RADIATOR SINK TEMPERATURE
MAXIMUM CASE
NOMINAL CASE
MINIMUM CASE
0
10
20
30
40
50
60
70
80
90
TIME - MIN

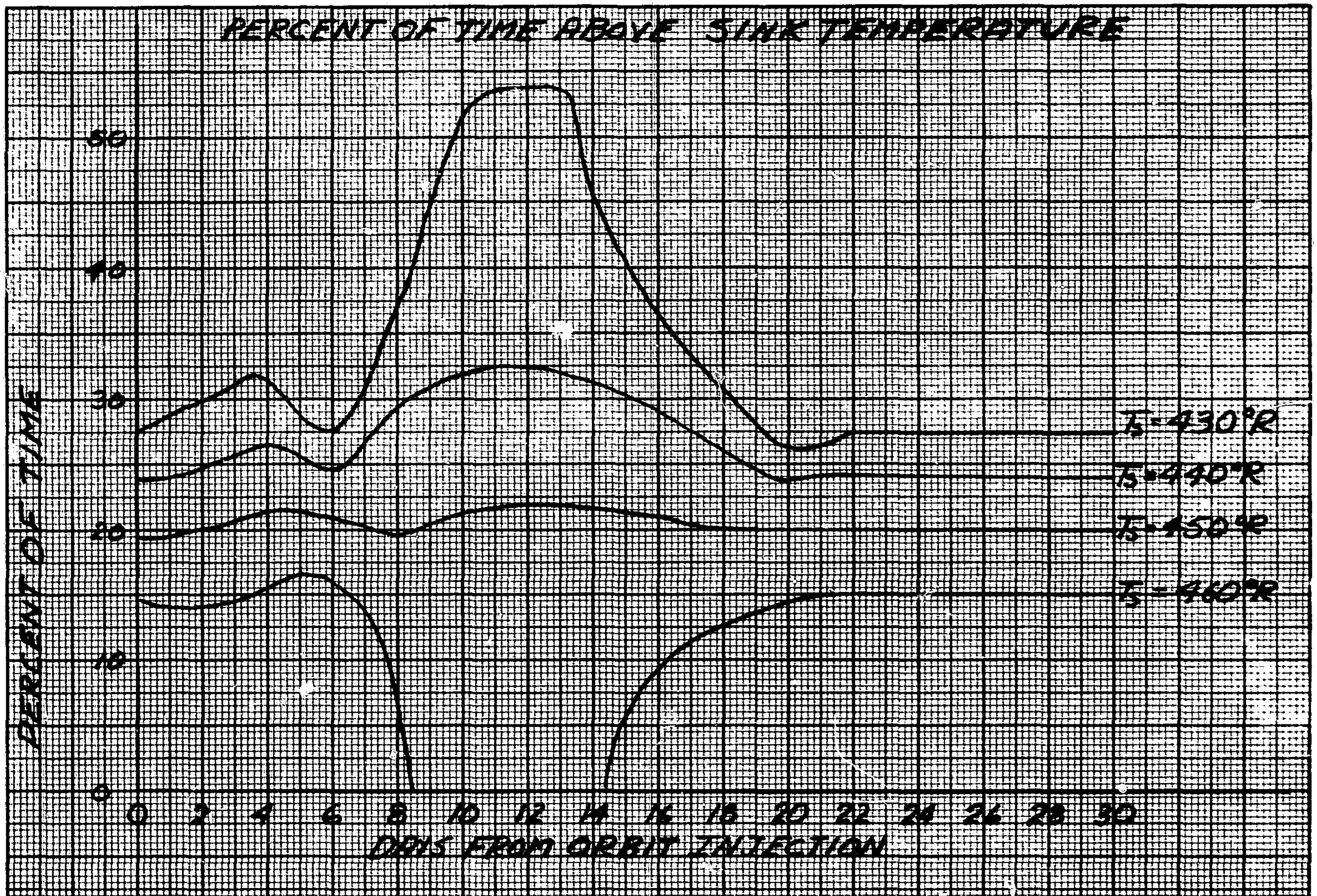
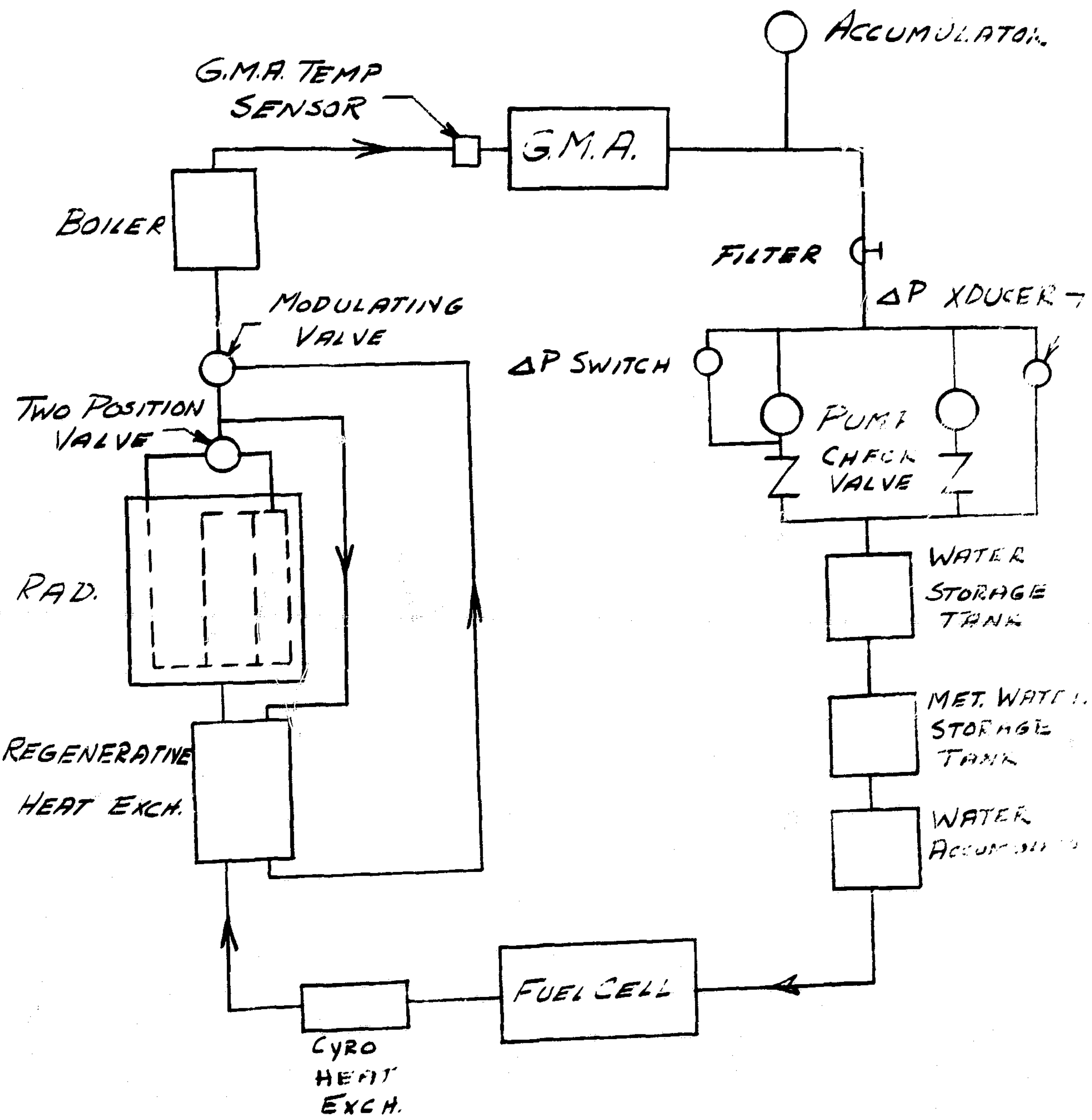

PERCENT OF TIME ABOVE SINK TEMPERATURE
PERCENT OF TIME
DAYS FROM ORBIT INJECTION
0
10
20
30
40
50
0 2 4 6 8 10 12 14 16 18 20 22 24 26 28 30
$T_S$ = 430°R
$T_S$ = 440°R
$T_S$ = 450°R
$T_S$ = 460°R

FIG. 4




SINGLE LOOP

FIG

# DUAL LOOP

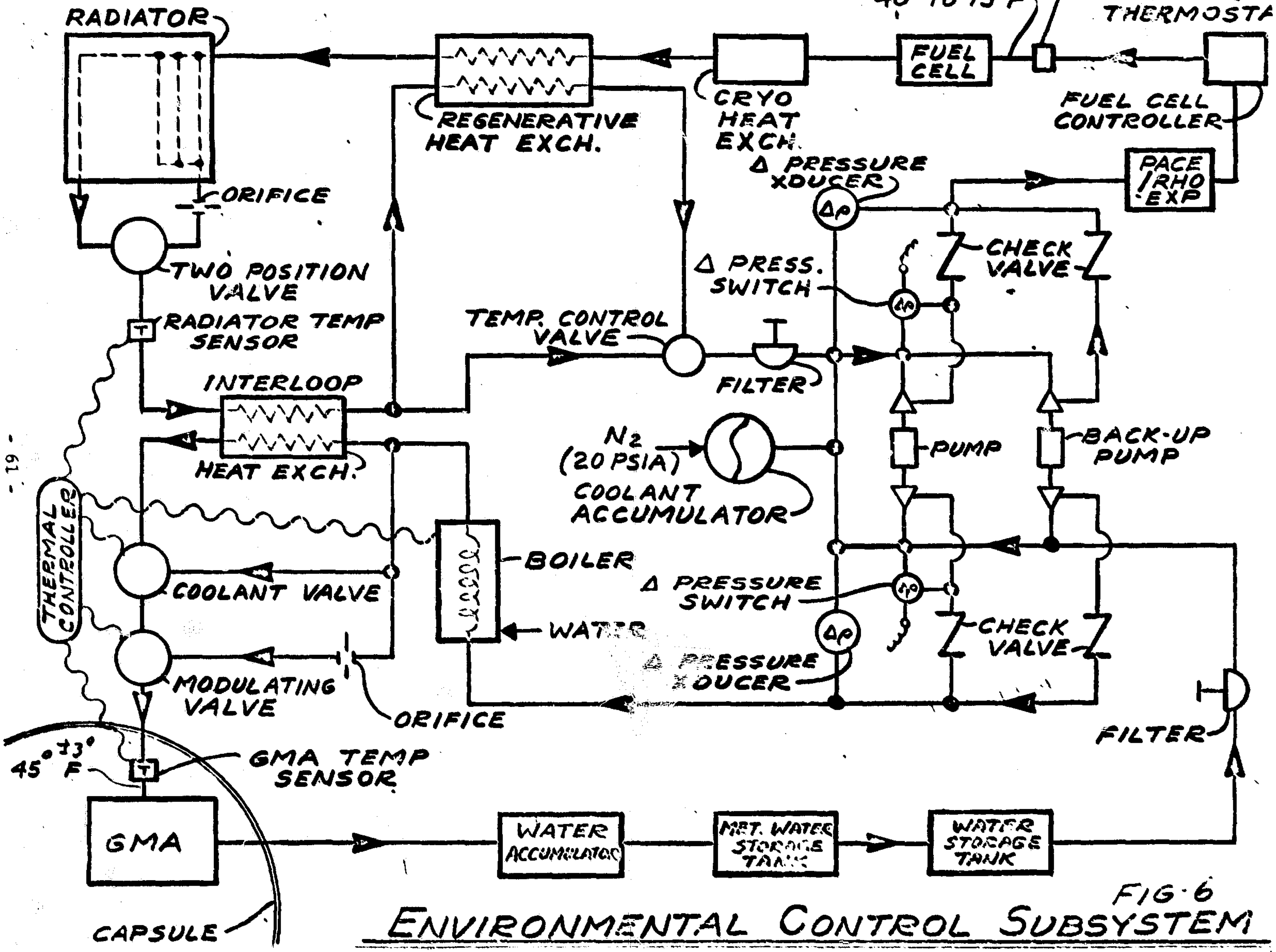



FIG. 6

ENVIRONMENTAL CONTROL SUBSYSTEM

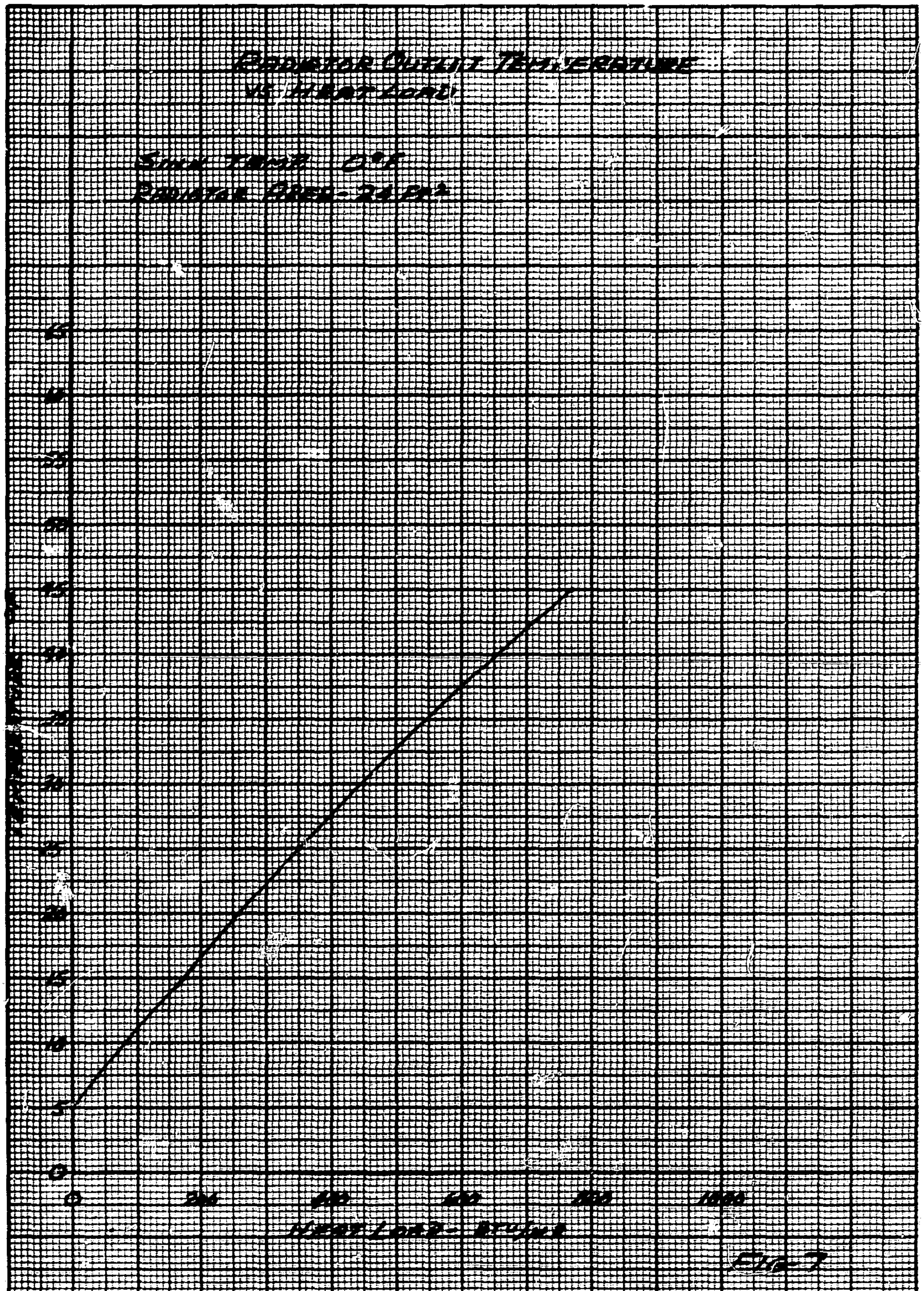

RADIATOR OUTLET TEMPERATURE
VS HEAT LOAD
SINK TEMP 0°R
RADIATOR AREA - 24 FT²
0
5
10
15
20
0
200
400
600
800
1000
HEAT LOAD - BTU/HR

FIG 7

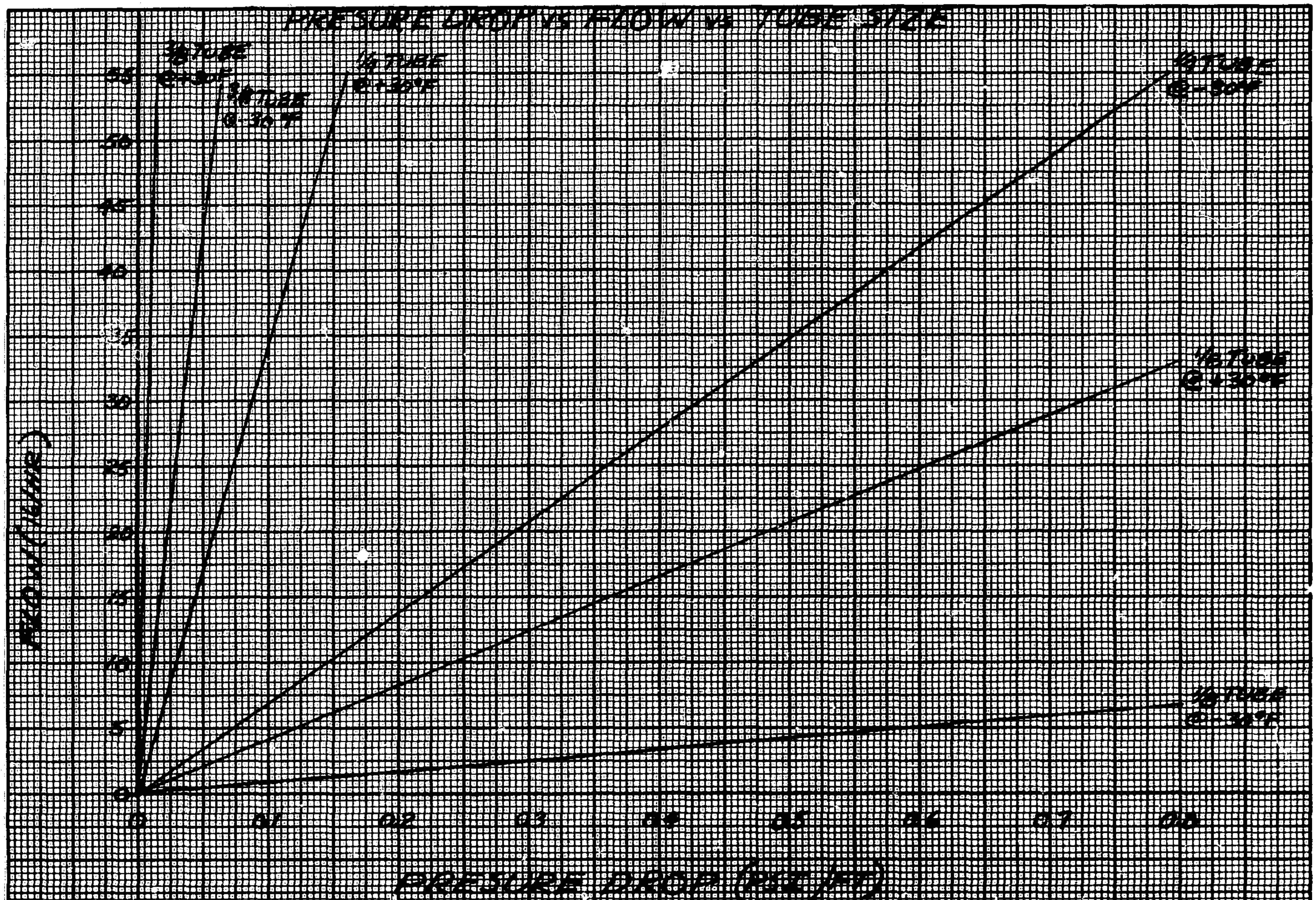
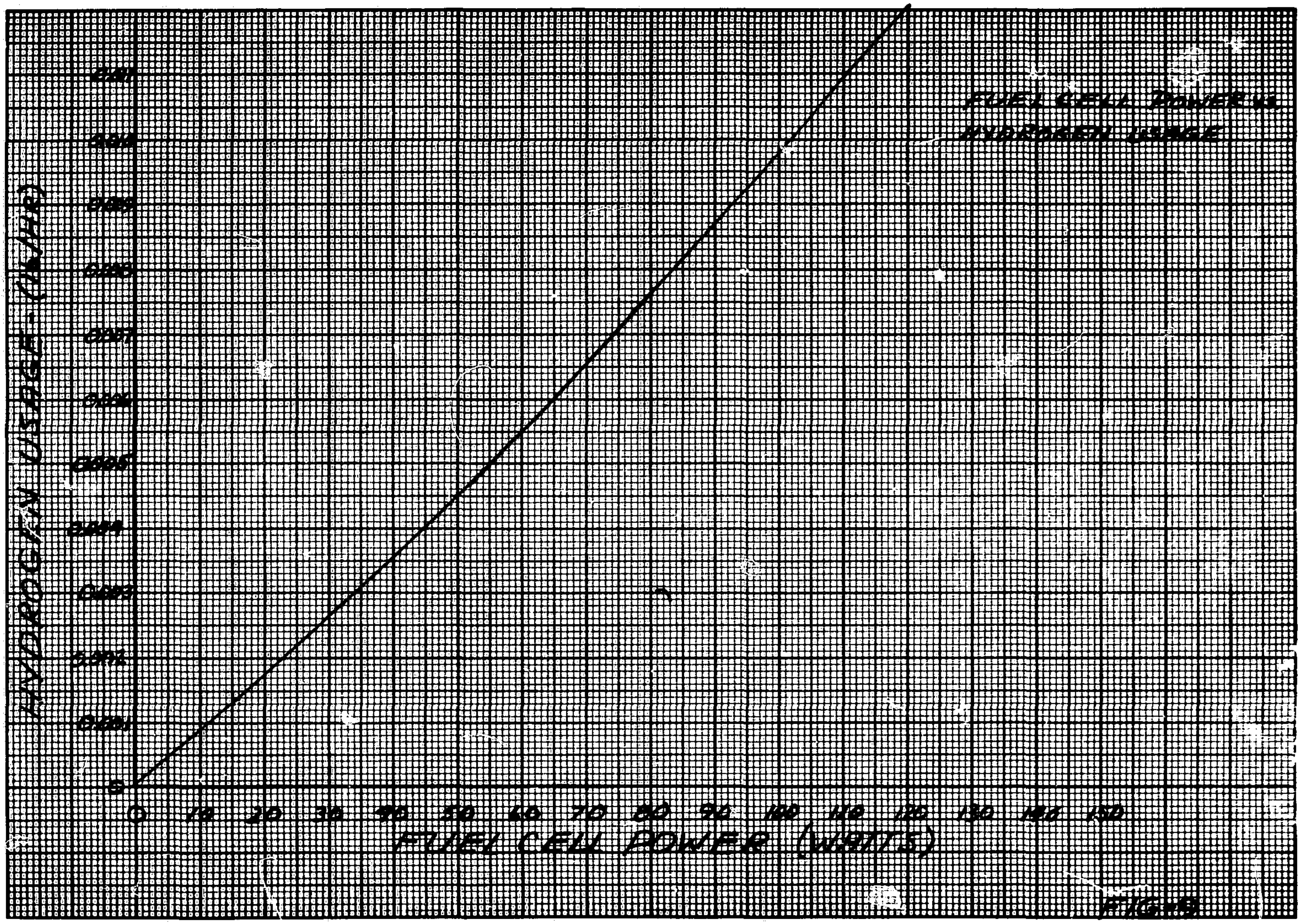

PRESURE DROP vs FLOW vs TUBE SIZE
0
5
10
15
20
25
30
35
40
45
50
55
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
PRESURE DROP (PSI/FT)

FIG-8


FUEL CELL POWER vs
HYDROGEN USAGE
HYDROGEN USAGE - (lb/HR)
0
.001
.002
.003
.004
.005
.006
.007
.008
.009
.010
.011
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150
FUEL CELL POWER (WATTS)

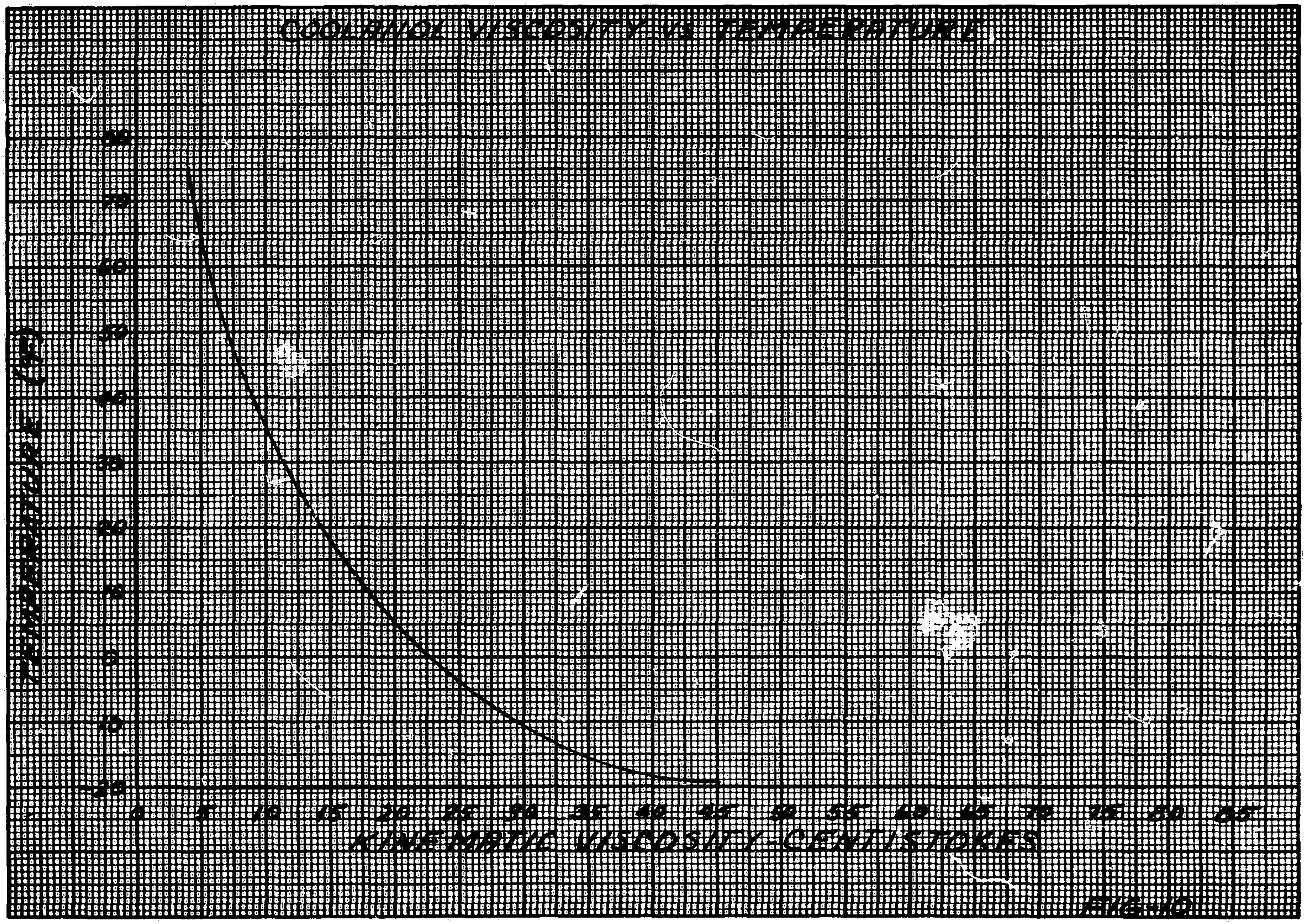
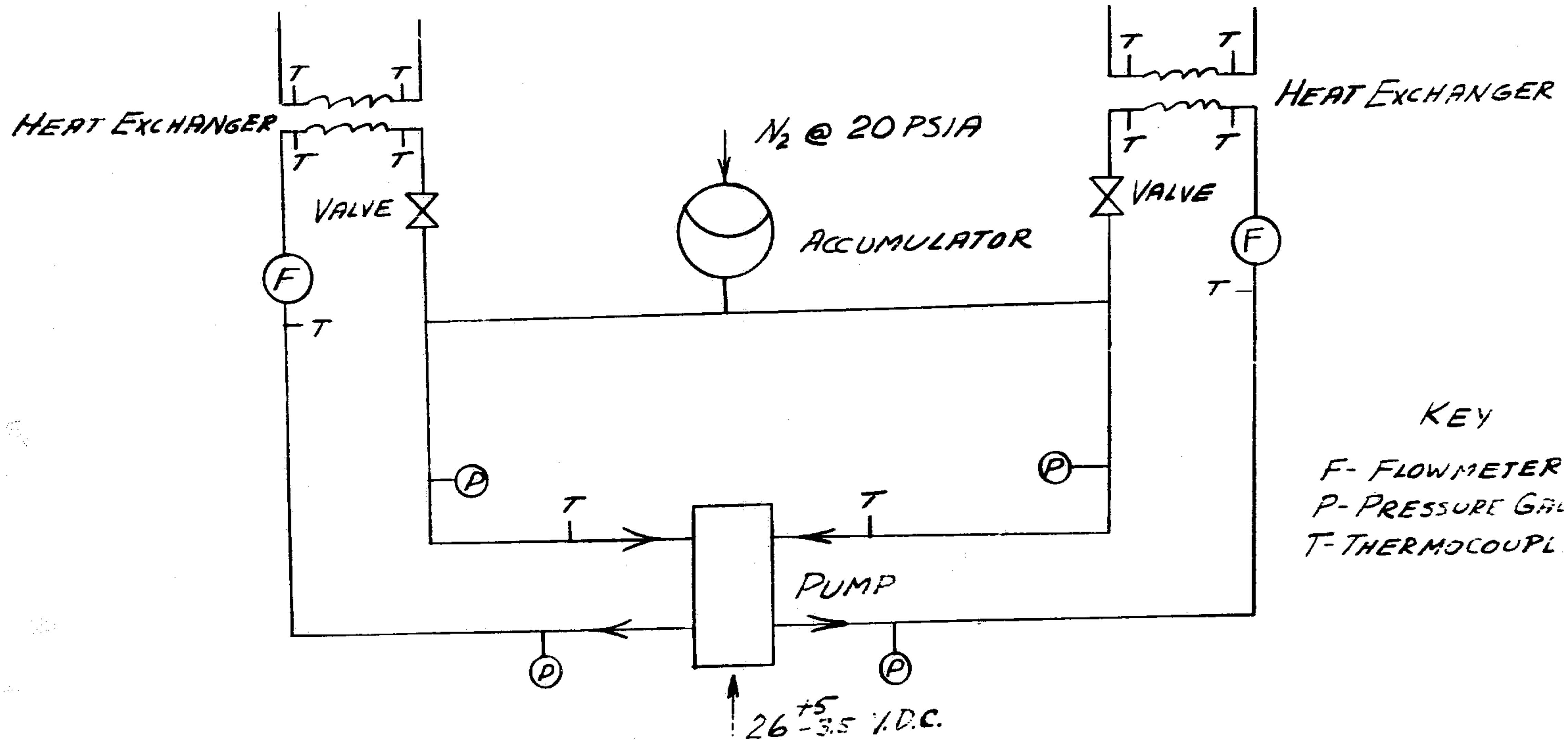

COOLANOL VISCOSITY VS TEMPERATURE
TEMPERATURE
80
70
60
50
40
30
20
10
0
10
-20
0 5 10 15 20 25 30 35 40 45 50 55 60 65 70 75 80 85
KINEMATIC VISCOSITY-CENTISTOKES

FIG-10




PUMP TEST SCHEMATIC

# HEAT EXCHANGER TEST SCHEMATIC

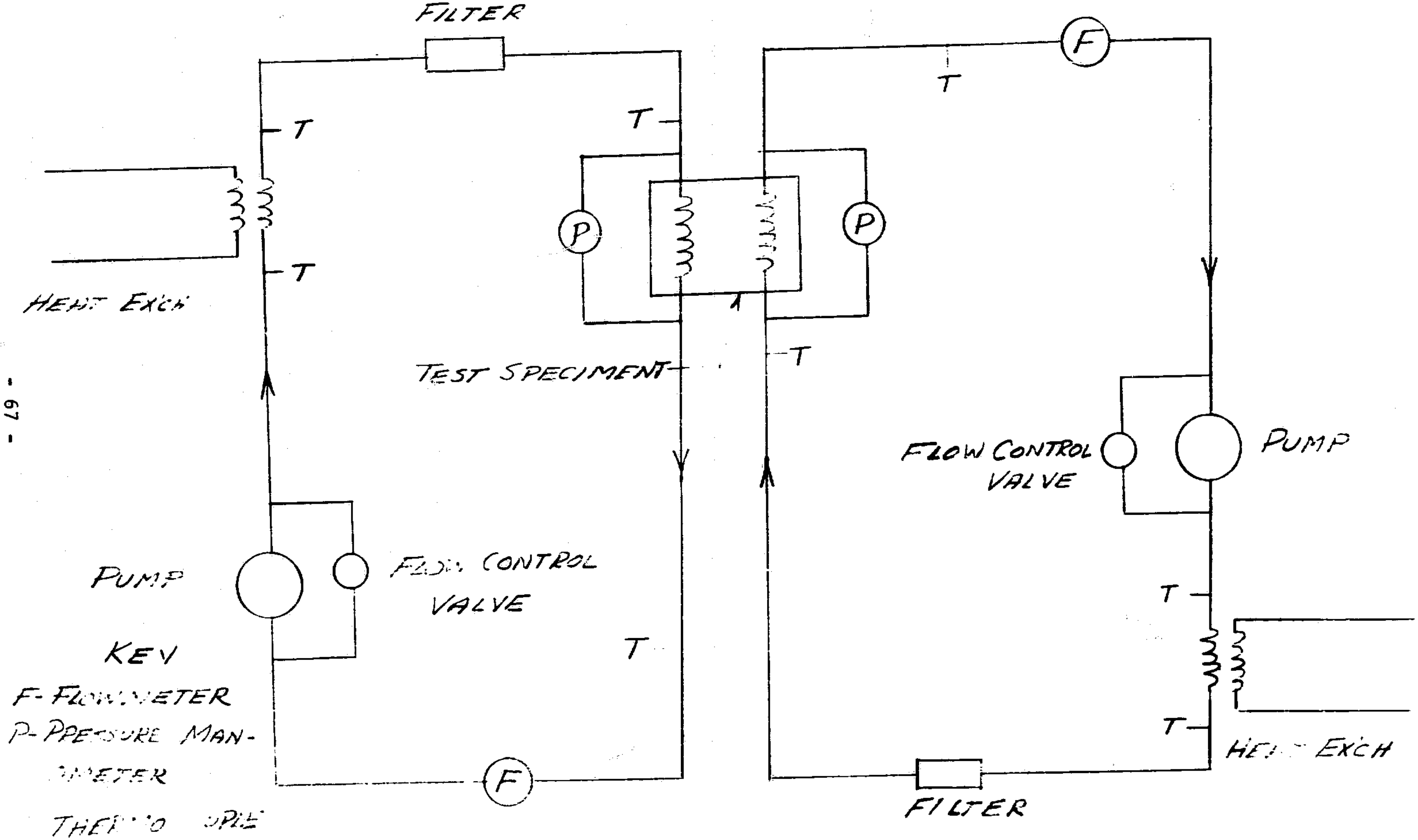
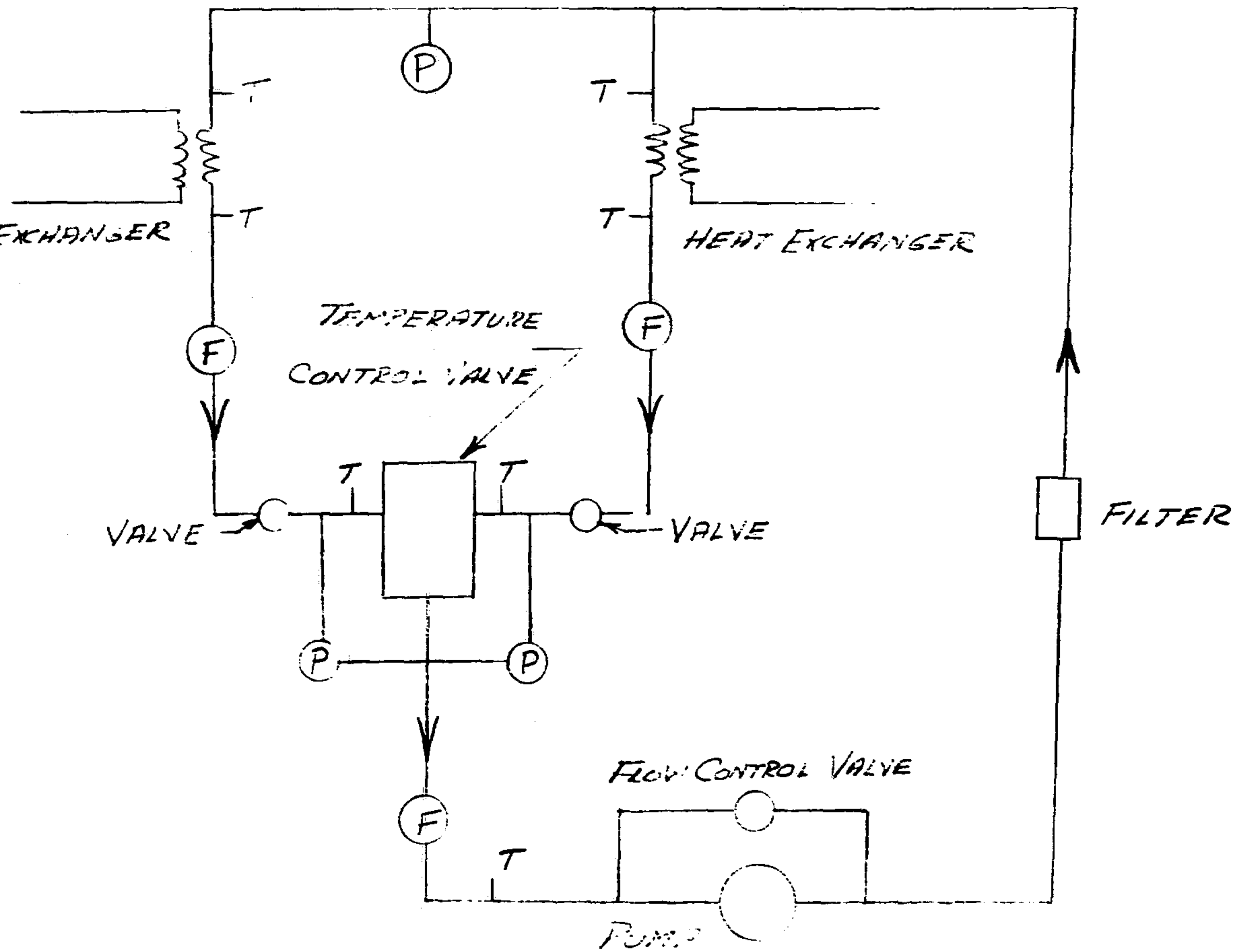



FIG-12

# TEMPERATURE CONTROL VALVE TEST SCHEMATIC

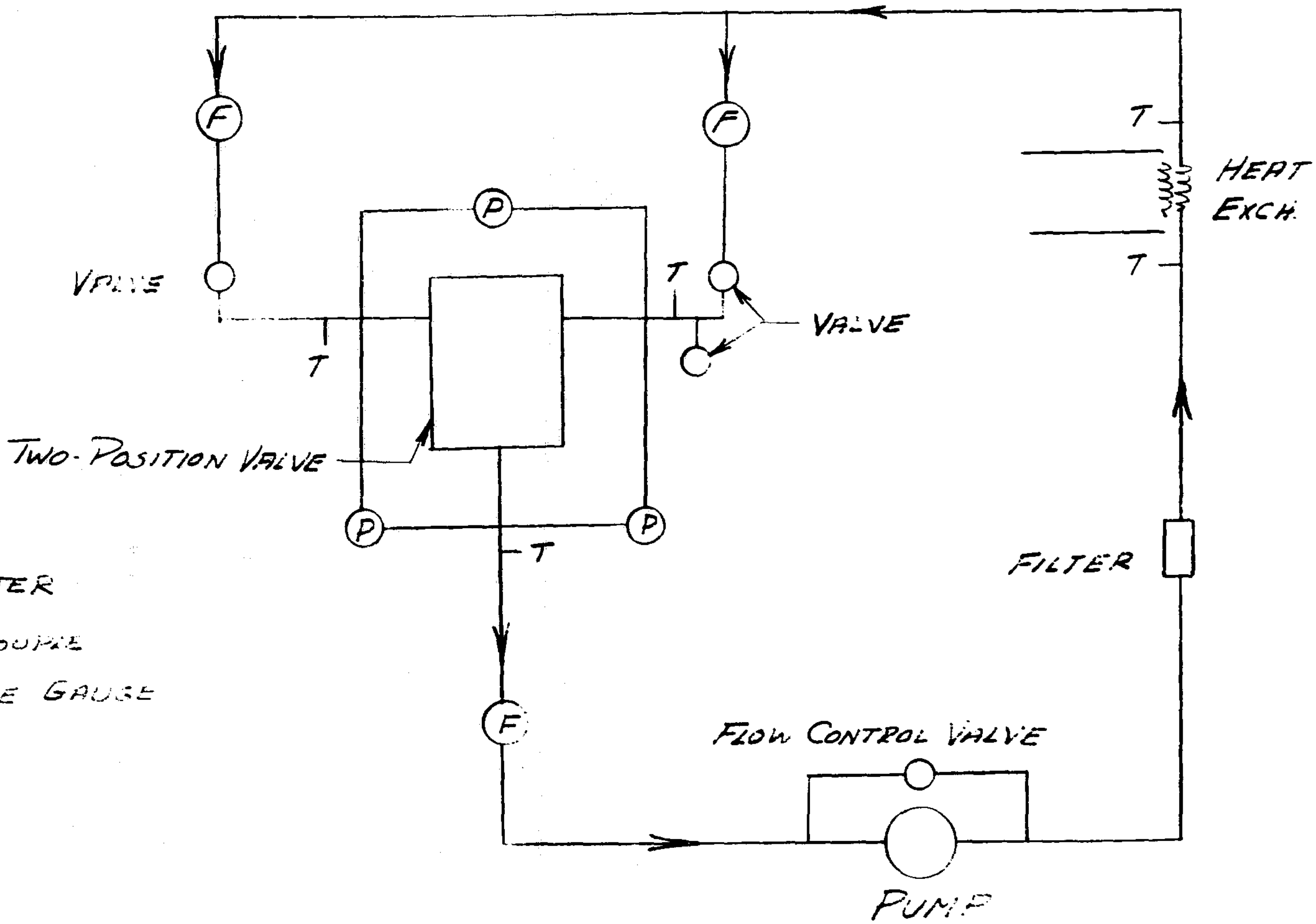



KEY
F- FLOWMETER
P- PRESURE GAUGE
T- THERMOCOUPLE



Fig-17

TWO POSITION VALVE TEST SCHEMATIC




KEY

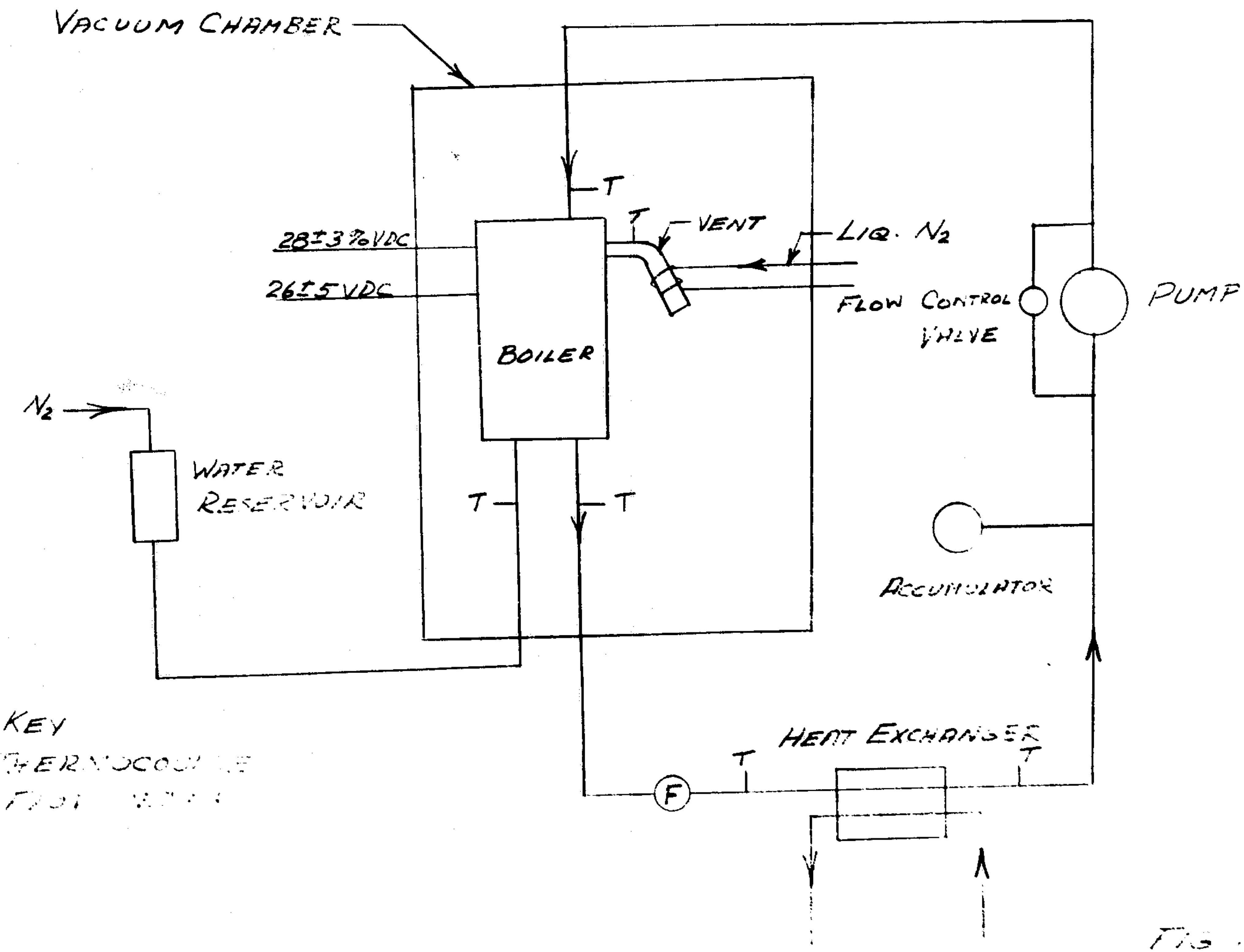
F- FLOWMETER

T- THERMOCOUPLE

P- PRESSURE GAUGE

FIG-14

# BOILER TEST SCHEMATIC




KEY

FIG.

BREADBOARD SCHEMATIC

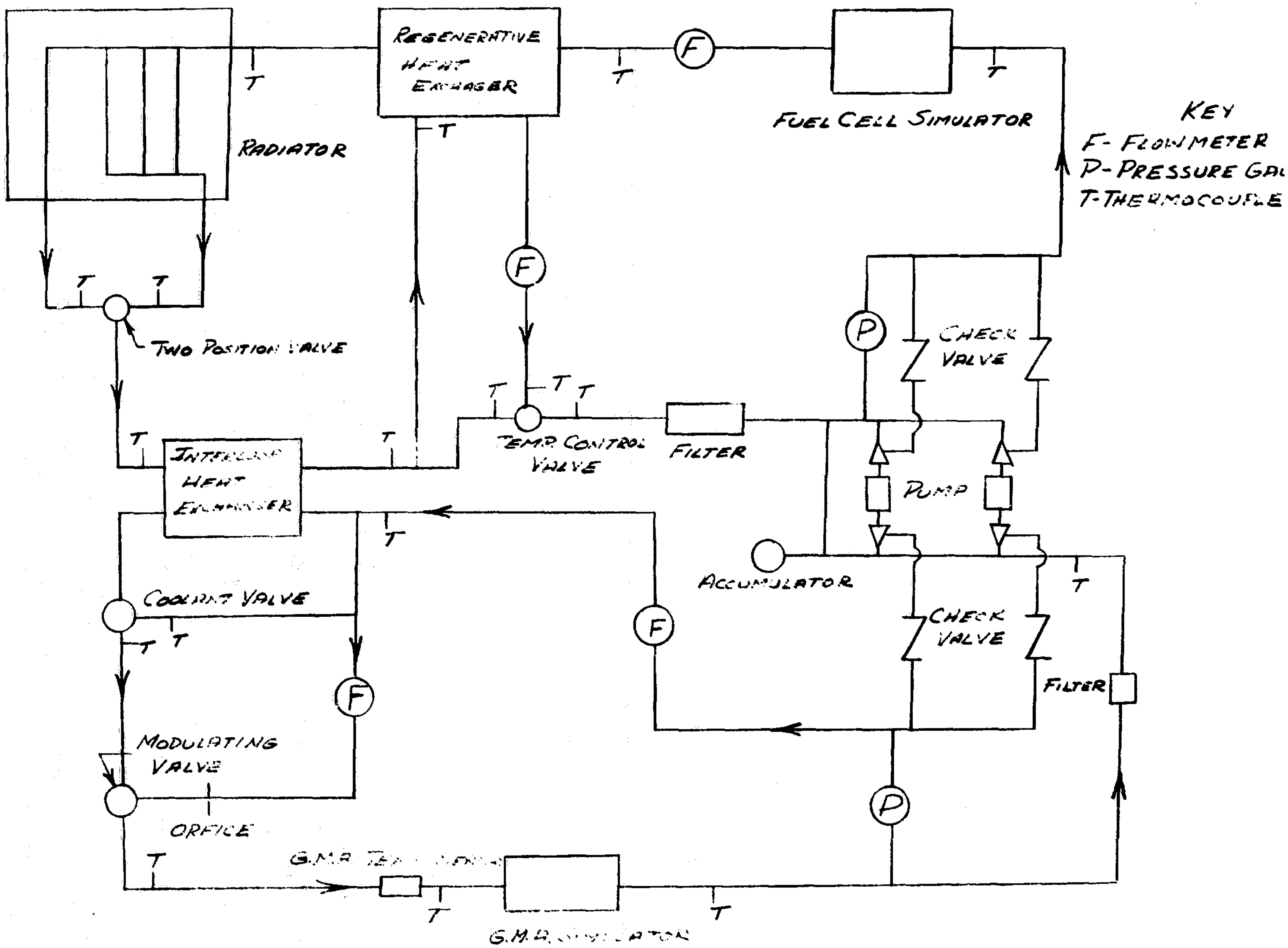
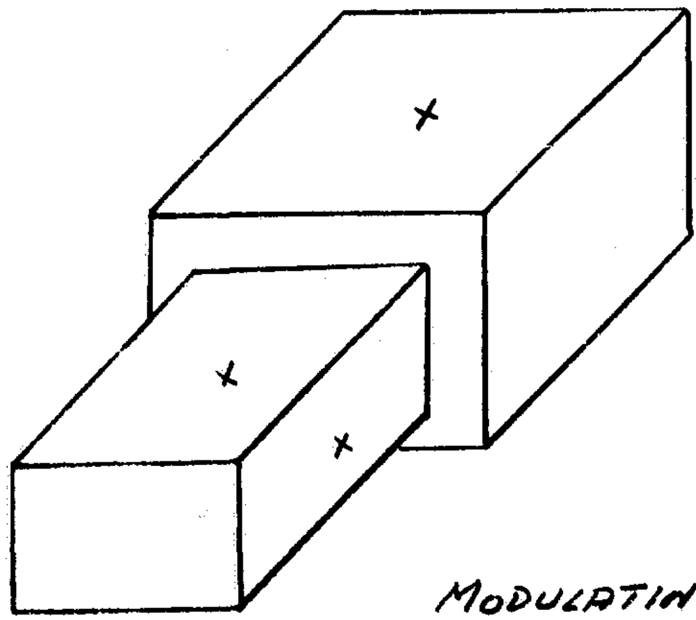


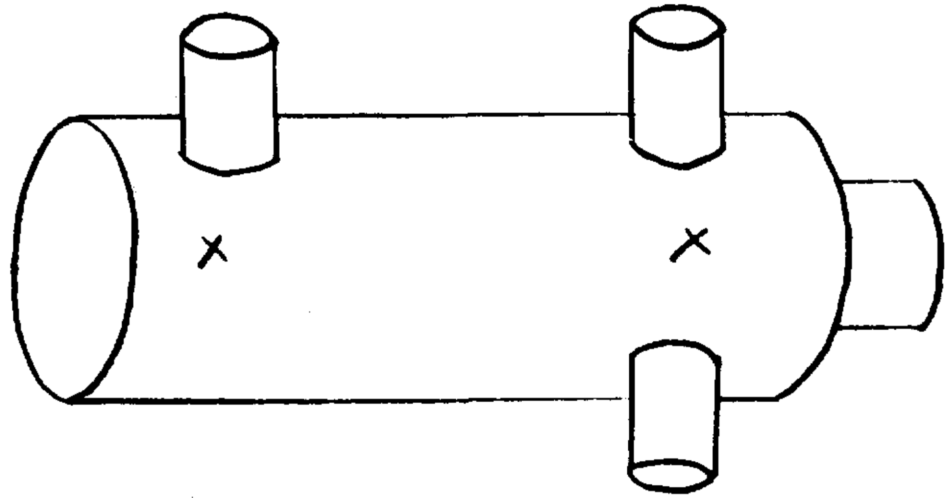

FIG. 16

KEY

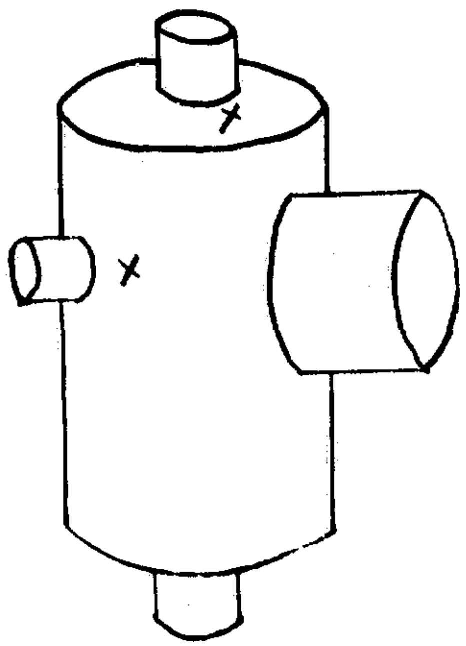
X- THERMOCOUPLE



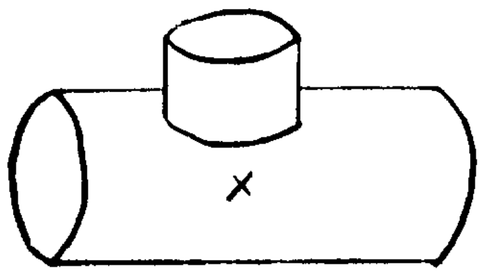
MODULATING VALVE



TWO-POSITION VALVE & TEMPERATURE CONTROL VAL



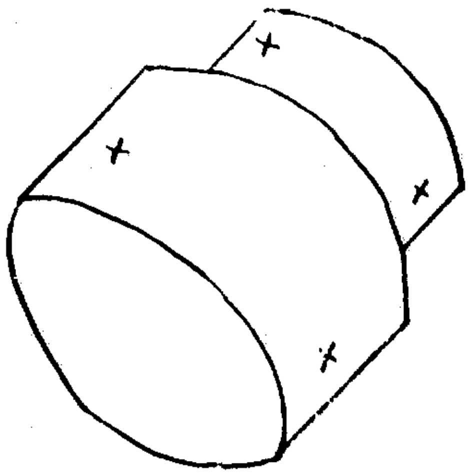
COOLANT VALVE



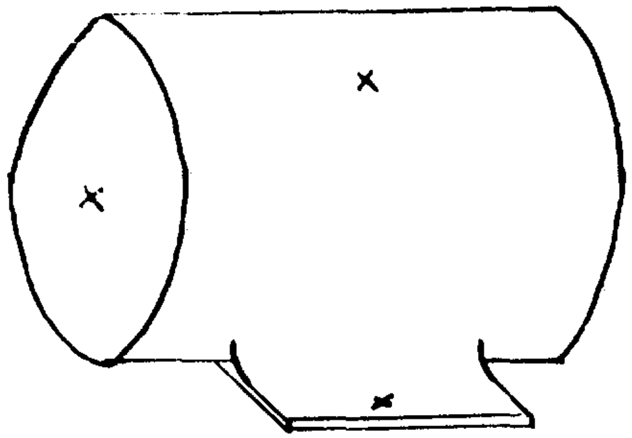
G.M.H TEMPERATURE SENSOR

KEY

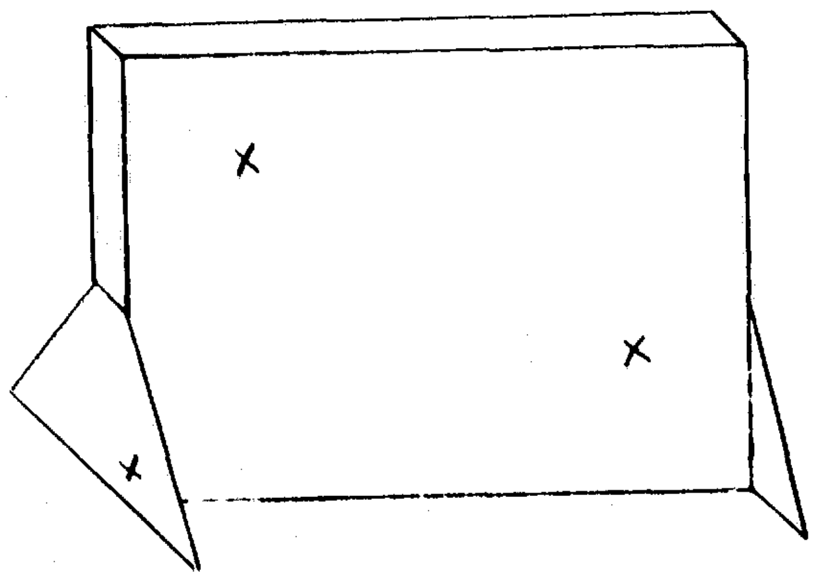
X-THERMOCOUPLE



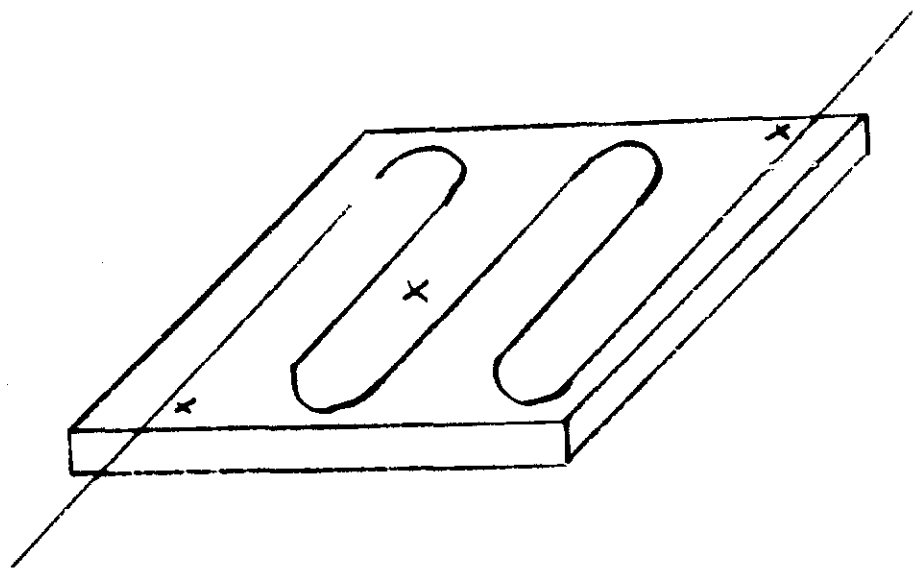
PUMP



FUEL CELL SIMULATOR

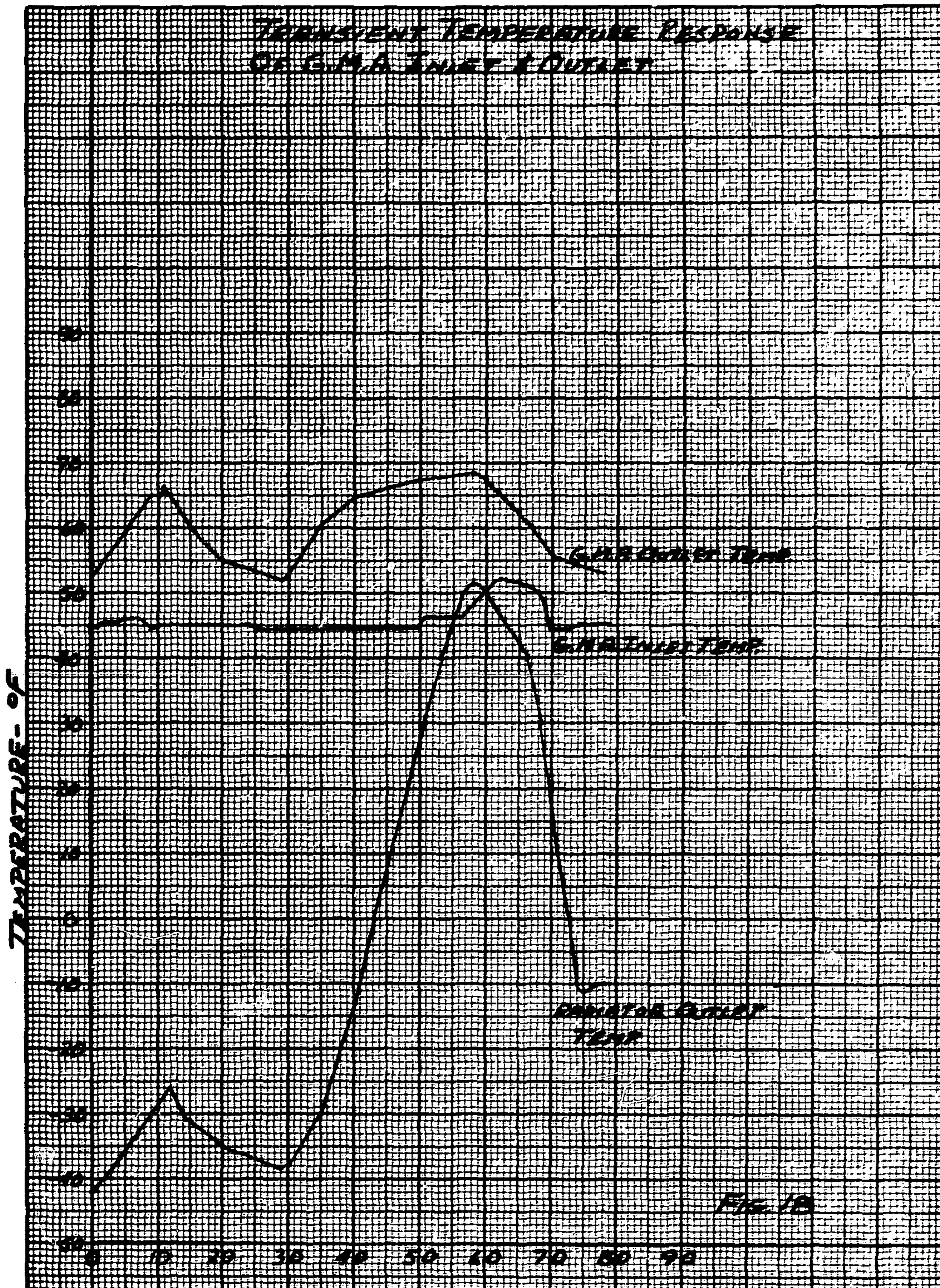


HEAT EXCHANGERS

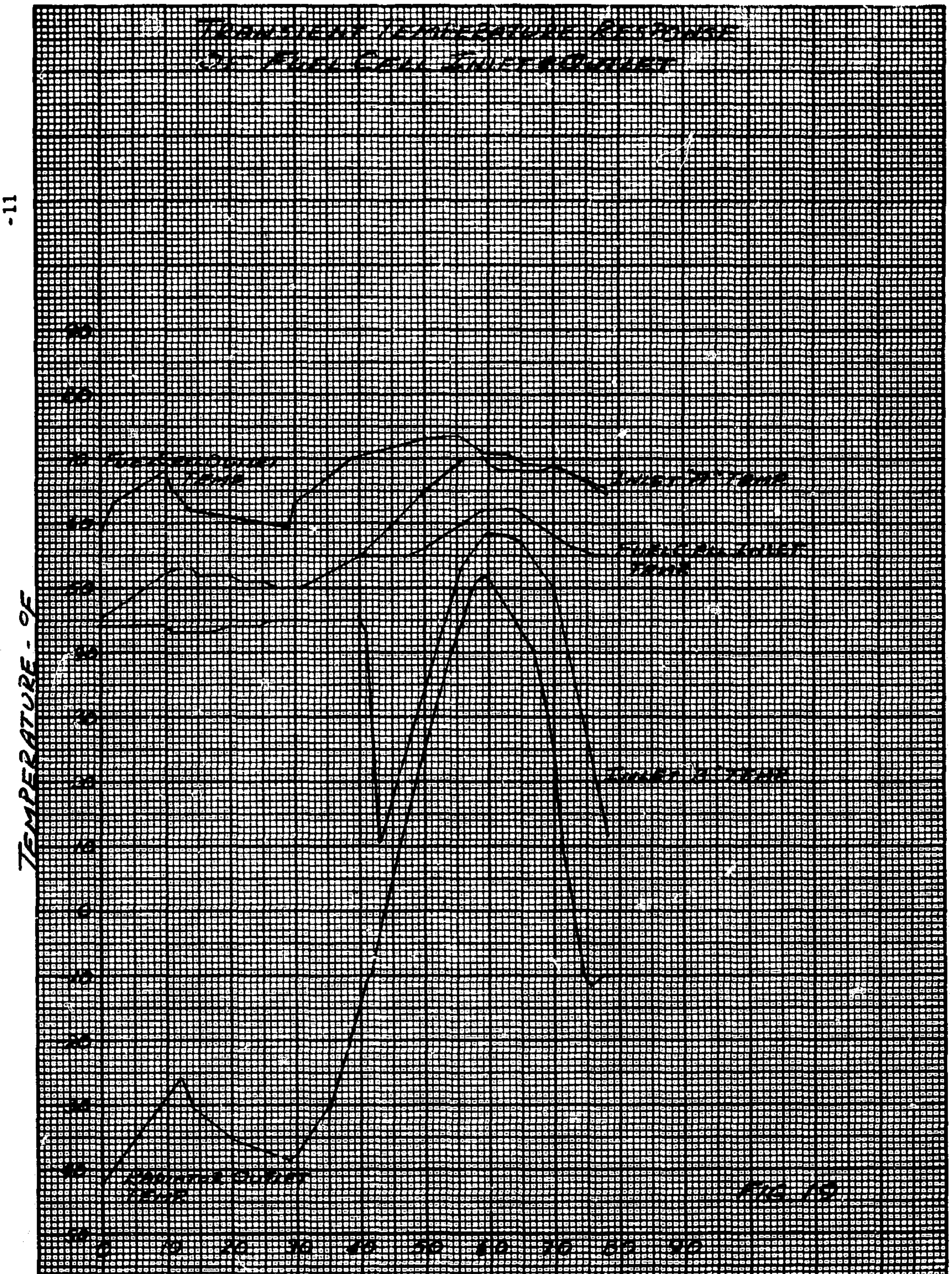


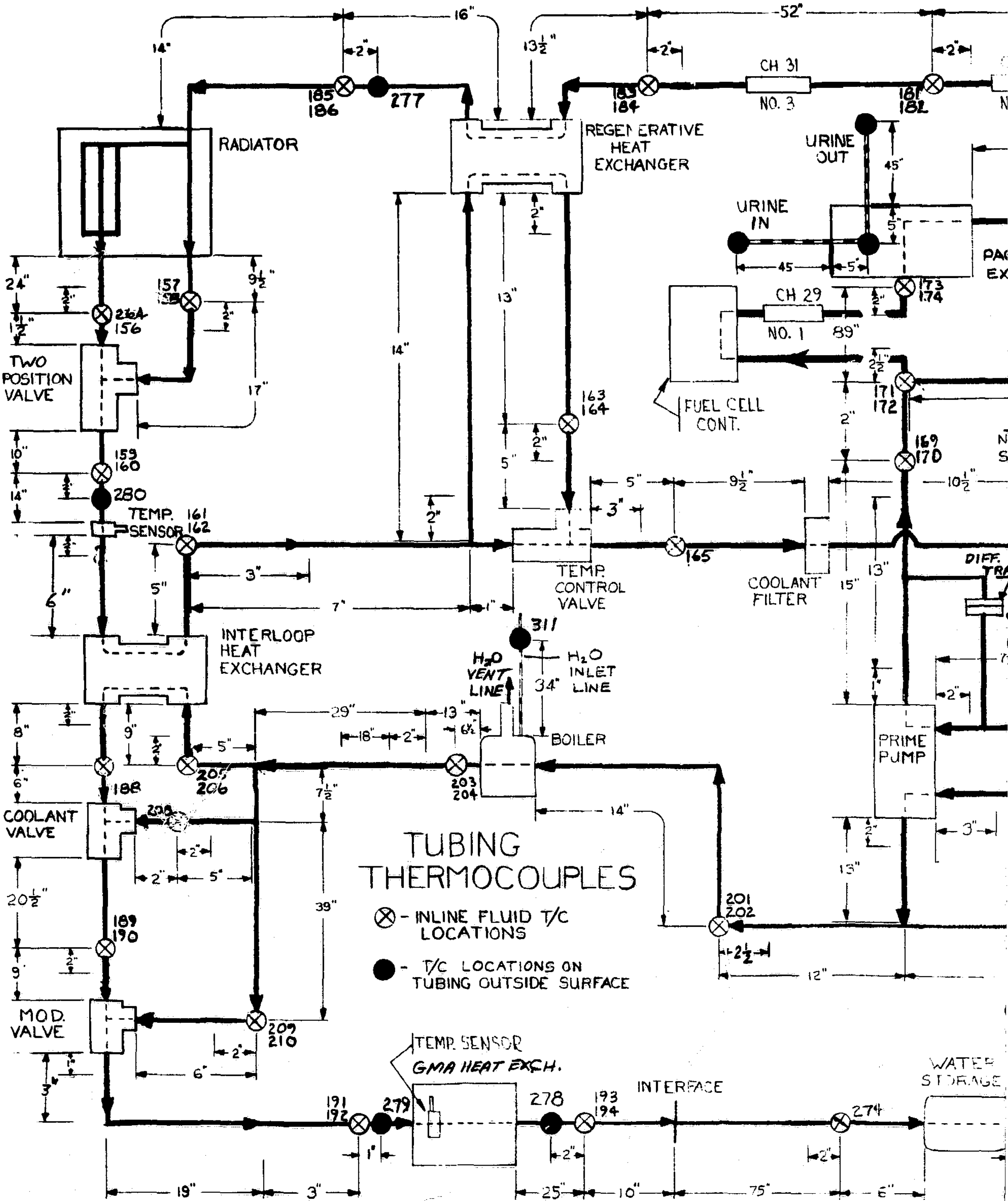
G.M.A. SIMULATOR

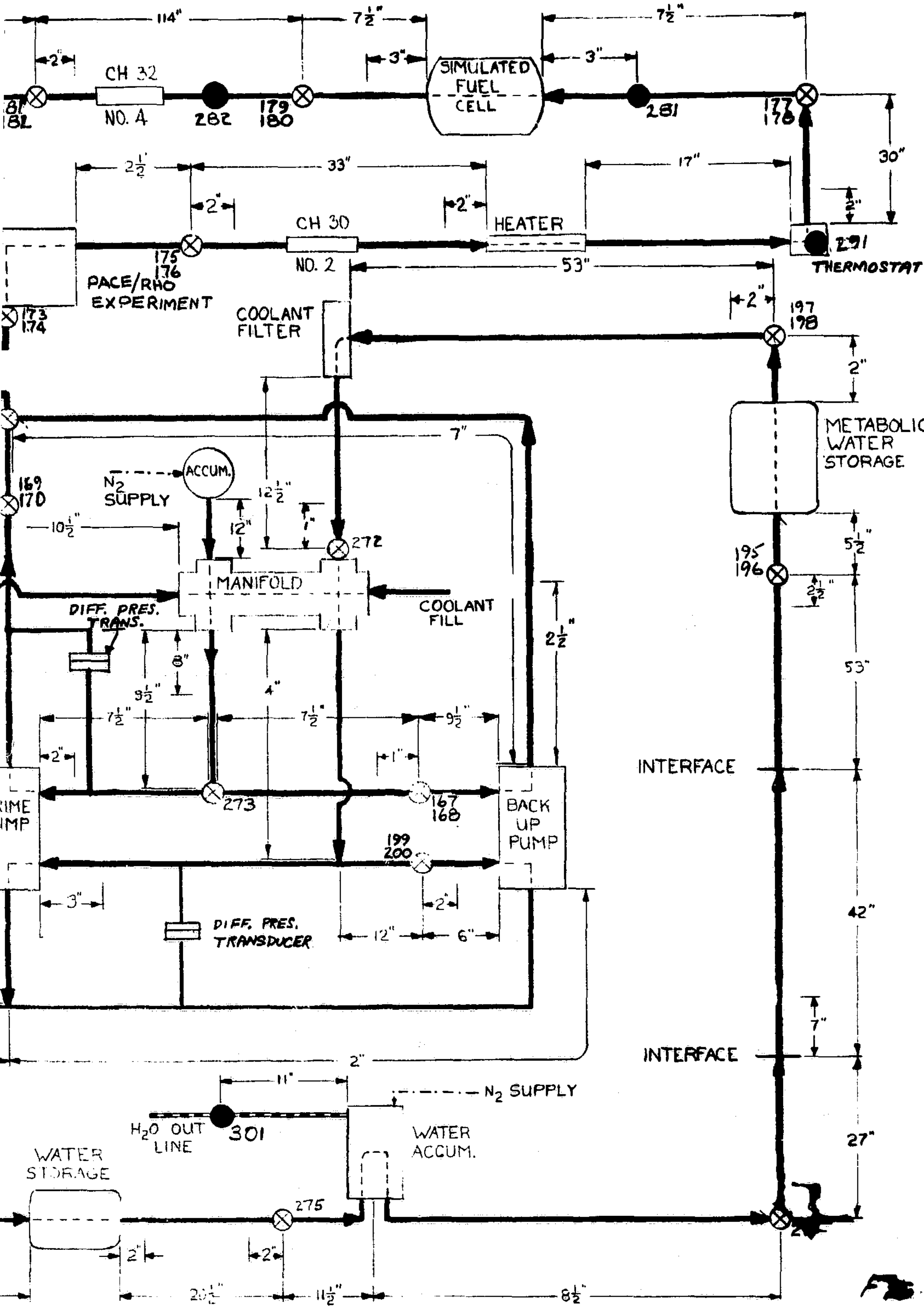
Fig. [illegible]


TRANSIENT TEMPERATURE RESPONSE
OF G.M.A INLET & OUTLET
G.M.A OUTLET TEMP
G.M.A INLET TEMP
RADIATOR OUTLET
TEMP
FIG. 18
TEMPERATURE - °F
TIME - MINUTES
90
80
70
60
50
40
30
20
10
0
-10
-20
-30
-40
-50
0
10
20
30
40
50
60
70
80
90

-11




TUBING THERMOCOUPLES
⊗ - INLINE FLUID T/C LOCATIONS
● - T/C LOCATIONS ON TUBING OUTSIDE SURFACE
RADIATOR
REGENERATIVE HEAT EXCHANGER
CH 31
NO. 3
URINE OUT
URINE IN
CH 29
NO. 1
FUEL CELL CONT.
TWO POSITION VALVE
TEMP. SENSOR
TEMP. CONTROL VALVE
COOLANT FILTER
INTERLOOP HEAT EXCHANGER
H2O VENT LINE
H2O INLET LINE
BOILER
PRIME PUMP
COOLANT VALVE
MOD. VALVE
TEMP. SENSOR
GMA HEAT EXCH.
INTERFACE
DIFF.
185
186
277
183
184
181
182
157
158
264
156
173
174
171
172
169
170
163
164
159
160
280
161
162
165
311
205
206
203
204
188
208
201
202
189
190
209
210
191
192
279
278
193
194
274




TUBING T/C LOCATIONS

FIG. 20

FOLDOUT FRAME 2

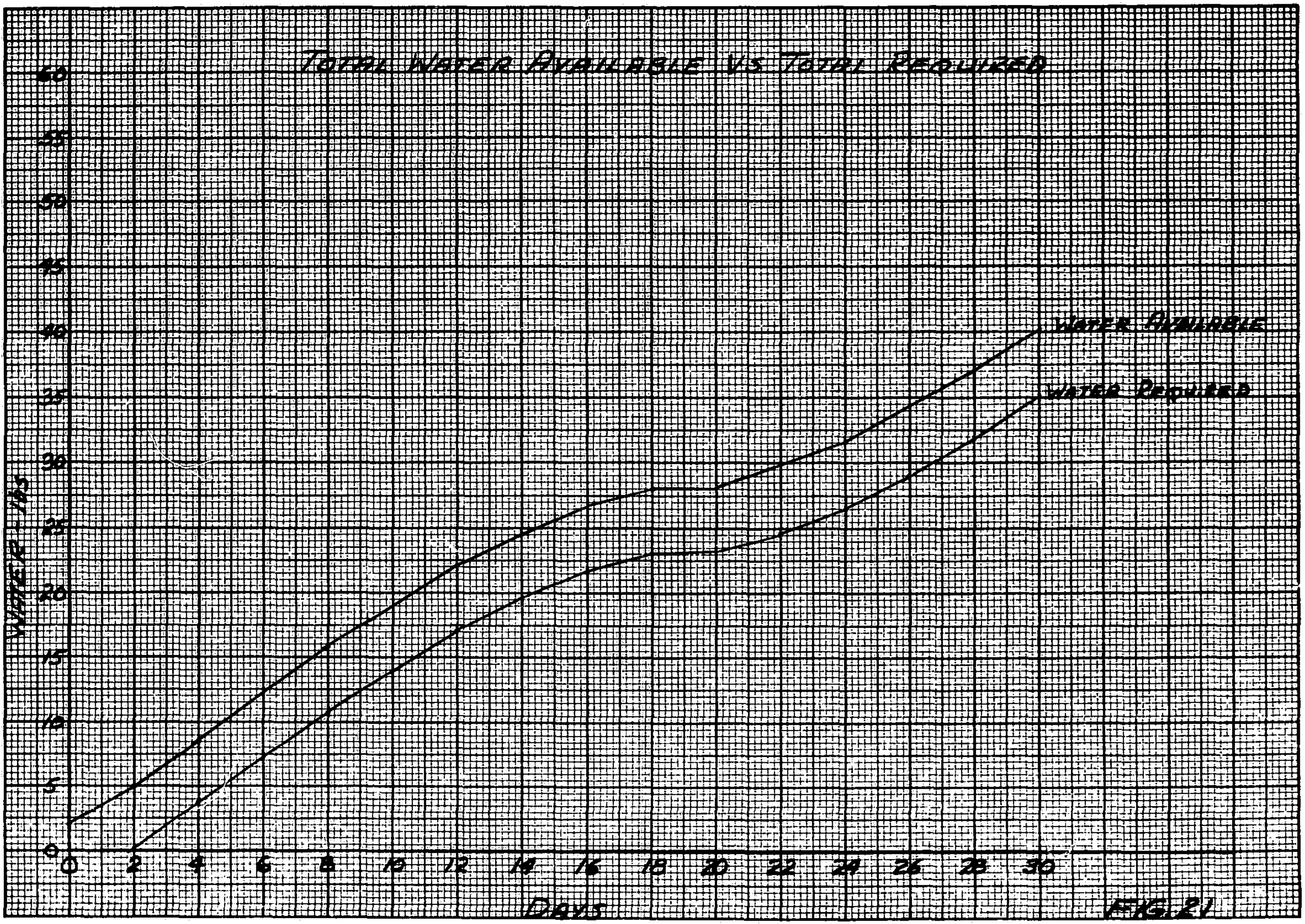

TOTAL WATER AVAILABLE VS TOTAL REQUIRED
WATER - lbs
0
5
10
15
20
25
30
35
40
45
50
55
60
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
DAYS
WATER AVAILABLE
WATER REQUIRED

FIG. 21

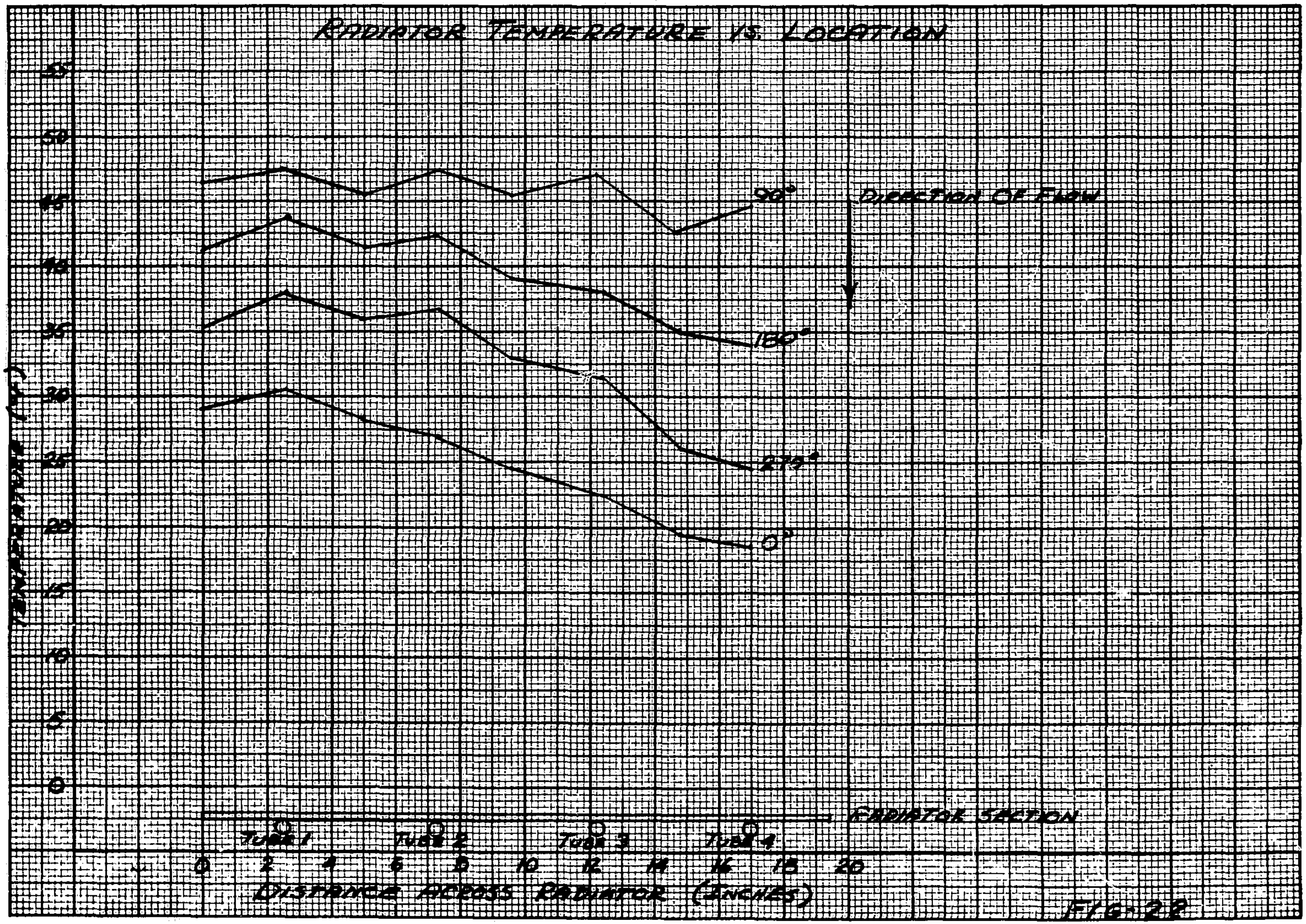

RADIATOR TEMPERATURE VS. LOCATION
55
50
45
40
35
30
25
20
15
10
5
0
90°
180°
270°
0°
DIRECTION OF FLOW
RADIATOR SECTION
TUBE 1
TUBE 2
TUBE 3
TUBE 4
0
2
4
6
8
10
12
14
16
18
20
DISTANCE ACROSS RADIATOR (INCHES)

FIG. 22

TABLE I

COMPONENT HEAT DISSIPATION - WATTS

| Component Name | Phase I | Phase II | Phase III |
|---|---|---|---|
| R/V Power Supply | 9.5 | 9.5 | 9.1 |
| Power Controller - R/V | 2.9 | 2.9 | 2.9 |
| Camera and Lights | 4.5 | 4.5 | 2.9 |
| Recorder | 3.3 | 3.3 | 0 |
| Signal Conditioner - R/V | 10.8 | 10.8 | 4.9 |
| GMA Master Controller | 5.0 | 5.0 | 0 |
| LIOH Canister | 9.4 | 9.4 | 4.0 |
| Primate | 36.3 | 36.3 (1) | 13.5 |
| Pschomotor Display | 0 | (2) | 0 |
| Fuel Cell/Cryogenics | 109 | 94 | 59 |
| Fuel Cell Controller | 6.0 | 6.0 | 0 |
| Pace/Rho | 10.0 | 10.0 | 6.5 |
| Inverter Power Supply | 34.0 | 34.0 | 16.0 |
| Battery | 8.0 | 8.0 | 0 |
| Rate Gyro | 16.0 | 16.0 | 11.0 |
| A/C Programmer | 8.1 | 8.1 | 0 |
| Jet Controller | 4.4 | 4.4 | 0 |
| Power Controller | 4.0 | 4.0 | 4.0 |
| Magnetometer Programmer | 3.4 | 3.4 | 0 |
| Storage Programmer | 4.4 | 4.4 | 0 |
| Transmitter | 1.2 | 1.2 | 0 |

| | | | |
|---|---|---|---|
| Tracking Beacon | 1.8 | 1.8 | 0 |
| Multi-coder | 0.6 | 0.6 | 0.23 |
| I.R. Pitch Scanner | 5.0 | 5.0 | 0 |
| I.R. Roll Scanner | 5.0 | 5.0 | 0 |
| Primate/LIOH Water Vapor (cc/hr) | 21 | 21 | 3 |

TABLE II

CANISTER ZONE TEMPERATURE - $^oF$

| Canister Zone | Test Phase | | |
|---|---|---|---|
| | 1 | 2 | 3 |
| Capsule | 106 | 75 | 45 |
| Adapter - Conical Section | 55 | 43 | 31 |
| Adapter - Cylinderic Section | 66 | 37 | 8 |
| AFT Cover | 66 | 37 | 8 |
| Rate Gyro | -16 | -80 | -144 |
| Jet Controller | 29 | -25 | -79 |
| Power Controller | 32 | -15 | -62 |
| Inverter Power Supply | 34 | -38 | -111 |
| Fixture | 66 | 37 | 8 |