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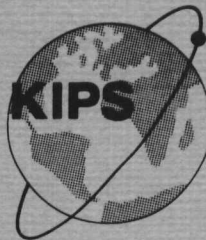
KIPS

KILOWATT ISOTOPE POWER SYSTEM

"KIPS CONTROL SYSTEM SELECTION"

TOPICAL REPORT

KIPS

Sundstrand Advanced Technology Operations

ROCKFORD, ILLINOIS 61101
unit of Sundstrand Corporation

SUNDSTRAND

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KIPS

KILOWATT ISOTOPE POWER SYSTEM

"KIPS CONTROL SYSTEM SELECTION"

TOPICAL REPORT

Prepared for the
U.S. ENERGY RESEARCH AND
DEVELOPMENT ADMINISTRATION
UNDER CONTRACT EX-76-C-16-3071

Prepared by: Sundstrand Advanced
                Technology Group

NOTICE

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SUMMARY

This Control System topical report covers basic control require-
ments, selection of control system and a recent review of an
electromechanical approach to the flow control valve.

Section I covers the basic control requirements for Organic
Rankine cycle systems, particular requirements for an isotope
fueled space power system, and special requirements imposed by
launch, Shuttle deployment and spacecraft requirements. Various
control devices which can be used to meet system requirements are
discussed.

In Section II, various combinations of control functions and devices
are presented with comments as to the suitability of each for the
intended application. This is essentially a review of the selection
process used to pick the present KIPS control system. The formal
trade-off matrix, component description, and system selection, as
prepared for Design Reviews 2 and 3, is included as Appendix A to
the report.

Section 3 covers the recently completed design of an electronic-
electromechanical flow control valve and compares this approach
to the thermal bulb-hydro-mechanical flow control valve baseline.
The results of this comparative study indicate that the present
configuration is preferable to an electrical valve.

1.0 KIPS Control System Requirements

The KIPS control system must be designed to accommodate a wide variety of requirements. Some of these are fundamental control functions required on any Organic Rankine cycle system, some are necessitated by the particular requirements of the changing heat rate of the isotope heat source used in the KIPS System, and additional functions are necessitated by spacecraft requirements. These control system requirements are discussed in the following sub-sections. Those noted by (A) are primary control functions required of any ORC*, those designated (B) are imposed by the isotope heat source and those designated (C) are additional but no less important requirements imposed by the ground and space environments, launch load considerations, and spacecraft torque requirements.

1.1 Turbine Inlet State Point Control

1.1.1 Basic Requirements (A)

A Rankine cycle system needs to have some control system to maintain turbine inlet conditions at the selected state point. Keeping turbine inlet conditions relatively close to the design point insures that the system will be able to produce the required power and that the turbine will always be supplied with superheated vapor to avoid erosion damage from partially vaporized working fluid. The flow rate through the turbine nozzles is proportional to turbine inlet pressure and inversely proportional to the square root of temperature. Thus, for constant turbine nozzle area, only two of the three parameters, flow, temperature and pressure need to be controlled.

* Organic Rankine Cycle

The necessity for flow control can best be illustrated by assuming that all other parameters of the system are held constant and noting the effects of a temperature perturbation on the system stability if no flow control is used.

A slight decrease in turbine inlet temperature increases the vapor density at the turbine inlet and permits greater flow through the turbine nozzles. Increased flow through the constant heat rate heater results in a further decrease of vapor temperature and even higher nozzle flow. This process cascades until the turbine is flooded with liquid and can produce no power.

A positive temperature perturbation starts a cascading process in the opposite direction, resulting in a system overtemperature shutdown.

For the KIPS ORC system where there is only a small variation in system heat input due to isotope decay, sufficient control on turbine inlet conditions can be achieved by throttling flow and permitting the small corresponding change in pressure to occur. A pressure control or a flow limiter can provide constant flow rate, but will not provide adequate turbine inlet temperature control because the 7% decay in heat source input is of the same order of magnitude as the working fluid superheat. Small variations in heat input lead to large changes in temperature. Therefore, if a flow limiter or pressure regulator is used as the primary means of turbine inlet state property control, supplemental temperature control is also required. This supplemental control can be achieved by using heat pipes or shutters to control heat input rate or by diverting working fluid within the loop to shunt excess heat to the radiator.

1.1.2   Temperature Control Requirements for Isotope Decay (B)

As the isotope heat source decays, less heat is delivered to
the KIPS boiler. There are basically two approaches to the problem
of maintaining constant turbine inlet temperature as the isotope
decays. In the first approach, flow is kept constant and heat is
diverted. In the second, flow is permitted to vary to absorb all
available heat.

It is possible to keep the flow rate constant at a value
corresponding to EOL* condition and control turbine inlet temperature,
via shutters, thermal shunts, or heat pipes on the isotope heat
source. This approach limits the PSC* output power to EOL condi-
tions even when additional heat is available. It also means that the
isotope heat source will have additional thermal losses throughout
the heat source life due to the presence of the thermal shunt device,
further reducing PCS output power. An alternate method for dissipating
excess heat at BOL* condition is to shunt fluid around the
regenerator. The regenerator connects the flow loop to the heat re-
jection loop so excess heat can be dissipated through the regenerator-
condenser interface.

One technique for accomplishing the regenerator short circuit
is to divert vapor flow around the regenerator. The vapor flow can
be taken from a point either before or after the turbine. It is also
possible to accomplish the same result by diverting the regenerator
liquid flow around the regenerator, thus causing the regenerator to
operate at a higher temperature and shunting excess heat via the
higher regenerator vapor outlet temperature.

* EOL   End of Life
  PCS   Power Conversion System
  BOL   Beginning of Life

The second approach consists of using all of the available heat at all times and adjusting boiler flow rate as required to maintain constant turbine inlet temperature. This approach permits maximum possible power output at all times and is, therefore, preferred if it can be easily accomplished.

1.2 CRU* Speed Control (C)

CRU speed control is not a primary requirement for a system with D.C. output power as alternator frequency is not critical. The speed control requirements are imposed by two conflicting requirements associated with the KIPS missions.

It is desirable to operate at a high rotative speed during launch shock and vibration loads.

Analysis of the CRU bearings indicated that extra bearing capacity would be required to handle launch and orbital transfer shock and vibration loads. The jet condenser jets also required extra pressure to achieve higher velocities and deflect less under these conditions. Building in extra capacity in these components was deemed undesirable since this extra capacity was normally not required and would result in higher parasitic losses and lower system efficiency.

By letting the CRU speed vary as load changes, the CRU seeks a speed which limits turbine input power to that required by the system and removes the need for active control of turbine power. The frequency wild system also aids in the system acceptance of full load on transients since the speed is high at part load and the CRU inertia can be used to insure that overload transients can be applied without substantial voltage drop due to speed variations.

* Combined Rotating Unit (Turbo-Alternator Assy.)

Spacecraft torque minimization requirements prohibit the use of frequency wild operation once orbit has been attained. Reasonably close control of CRU speed is required to insure that the spacecraft remains stable. The KIPS control system must then provide a dual mode feature, i.e. frequency wild or high speed operation for launch and speed control at best efficiency point for sustained spacecraft use. Speed can be controlled either by adjusting the CRU input power or its output power. Since turbine inlet temperature is usually kept constant to maintain superheated vapor conditions, turbine input power is controlled by adjusting flow or pressure ratio. Flow can be adjusted directly by varying turbine nozzle area or diverting some flow past the turbine. Flow can also be adjusted indirectly by a throttling valve which reduces turbine inlet pressure and causes a reduction of flow rate through fixed area turbine nozzles. A throttling valve at the turbine outlet will cause a power reduction by backpressuring the turbine and reducing the available turbine pressure ratio.

If a flow throttling valve is used to control temperature, the speed control function can be accommodated indirectly by bypassing the flow on the liquid side of the regenerator when excess power is available and speed begins to increase. As liquid is bypassed, boiler inlet temperature drops, turbine inlet temperature drops and flow is reduced by the temperature control valve to bring turbine inlet temperature back up to the desired value. This flow reduction reduces turbine power and prevents further speed increase.

Turbine speed can be also controlled by adjusting the CRU output power. An electrical load bank can be used to dissipate excess power. This technique removes excess heat from the power producing loop by converting the heat into work and converting the work back into heat

outside of the loop on a high temperature radiator. The same
effect is accomplished by loading the CRU hydraulically by means of
the system pump or a separate hydraulic drag disc. However, the
heat produced by this means must be rejected by the system radiator.

1.3 Electrical Output Controls

1.3.1 Voltage Regulation (C)

A voltage regulator is required to maintain the output voltage
within required limits. Voltage regulation is typically accomplished
by controlling alternator field current. Several types of voltage
regulator circuits can be used.

PROPORTIONAL VOLTAGE REGULATORS: Shunt and series regulators are
simple with low parts count, but have the major drawback that they
are dissipative. Dissipation causes higher losses in semiconductors
and resistors lowering reliability. Therefore, dissipative propor-
tional voltage regulators were avoided because of their low reliability.

MAGNETIC AMPLIFIER REGULATOR: The magnetic amplifier acts as a
phase controlled switch on the rectified AC field supply voltage
to supply a variable DC voltage to the exciter field. The magnetic
amplifier can be considered a switch in series with the exciter
field that can be turned on at various phase angles of the rectified
input voltage half sine waves, thus producing a variable DC.
This regulator is simple with few parts; its drawback is the weight
of the magnetic amplifier which must handle the total field current.

PULSE WIDTH MODULATED REGULATORS: The pulse width modulating
regulators can be put into two classes, forced and Bang-Bang
Modulation.

1.3.1 Continued

a) Pulse Width Modulation: In pulse width modulation a switch in series with the field is operated at either the ripple frequency of the field power source or at a selected frequency derived from an oscillator. The duty cycle of the modulation frequency is varied by voltage error to maintain the output voltage. This regulator is more complicated than the others considered and is more applicable to systems requiring exceptionally good voltage regulation.

b) Bang-Bang Pulse Width Modulation: In the Bang-Bang system the switch in series with the field is allowed to modulate at the system natural frequency and have a duty cycle dependent on the loop dynamics. This is a simple regulator with voltage regulation characteristics less refined than the forced pulse width modulation scheme.

1.3.2 Overload Protection (C)

Provision must be made in the control system to prevent an electrical overload from occurring. With any type of P.C.S. an electrical overload can cause control problems by the attempt to operate outside the expected operating range. When the system heat input is fixed an attempt to extract more power than is thermodynamically possible can cause severe system disruptions.

In the case of the ORC system, an electrical overload causes the CRU to slow down in an attempt to match turbine torque with the increased torque requirements of the alternator. This speed decrease will eventually reduce feed pump pressure to a point where the turbine flow is reduced. The system will then overheat, causing the PCS system to shutdown.

1.3.2 Continued

Overload prevention can be achieved by design of the alternator such that it saturates and cannot deliver excess power. An attempt to overload the system results in rapid voltage drop which limits the power so that an overload cannot occur. This approach usually compromises the generator efficiency at design conditions, and cannot easily be used when several design power levels are required from the same generator.

With a constant output voltage P.C.S. as required for the KIPS system, current limiting can be used to provide overload protection. Current sensors can be incorporated into the generator controls to limit the current to the design value or to provide short circuit protection by disconnecting the load when an overload occurs. Current limiting can also be used to partially unload the generator in the event of a CRU underspeed condition caused by a partial failure of some other system component. If the maximum alternator field current is just sufficient to carry full load at rated speed and output current is limited to that required to provide rated power at rated voltage, a decrease in speed below minimum design speed will result in a linear decrease in voltage and output power. The turbine power does not decrease as rapidly with speed as the generator output, so a lower speed balance point can be attained and the system will continue to supply some electrical power instead of shutting down.

The major drawback to the use of current limiting for KIPS is the varying output power capability with time as the isotope heat source decays. A fixed current linear setting would always limit the electrical output to E.O.L. conditions. An adjustable current limit would require some means of determining what the maximum load

1.3.2   Continued

capablitiy of the PCS is at any set of operating conditions and
setting the current limit appropriately. Current limiting is accomplish-
ed by cutting back on alternator field current as required to main-
tain constant output current during electrical overloads. Thus
output voltage will drop and CRU speed will increase. Thus current
limiting violates the requirement for constant speed operation.

CRU speed is an excellent indicator of system overload. As
excessive load is applied and the speed sags, the underspeed
signal can be used to reduce or cut off alternator field current
to limit the output capability of the alternator and thus prevent
continued speed reduction. In this way overloads are prevented
from harming the PCS and constant speed is maintained.

1.4   Heat Rejection Control (C)

The primary means of P.C.S. heat rejection is via the system
radiator. The radiator must be large enough to reject BOL waste
heat at full load and the highest heat sink temperature. At other
operating conditions the radiator is oversized and it is desirable
to limit the heat rejected to maximize P.C.S. efficiency; this
capability is particularly desirable at E.O.L. when the P.C.S.
output power is lowest.

It is also very desirable to have some means of controlling radiator
outlet temperature when the radiator auxilliary cooling loop is
used for system heat rejection when the P.C.S. is installed in the
Shuttle Bay. P.C.S. control of the radiator temperature eliminates
the need for active Shuttle control of the cooling temperature.

A third function of the radiator heat rejection control is to
permit subcooling of the radiator metal to act as a heat sink when
the KIPS system is launched using a Titan vehicle. This heat

1.4 Continued

sink permits the KIPS system to continue running during the critical
launch period when the radiator is shrouded.

2.0 Control System Selection

It can be seen from the description of control requirements
in section 1 that there are many possible techniques to provide each
of the control requirements. The objective in selecting a control
system is then to perform all of the control system functions with
a minimum number of devices. The following discussion is an attempt
to step through the requirements to show what types of controls are
most likely to lead to a simple system.

A good starting point is the requirement that the PCS must
operate both with the speed controlled for spacecraft stability
and frequency wild for launch. The speed wild capability should be
arranged so that it is also a viable backup operating mode in the
event of a speed control system failure. Since the change from
speed wild to constant speed mode must be made after orbital trans-
fer, a remote electrical signal must be used to provide mode
switching. Figure 1 shows the various techniques for providing speed
control with comments on the effects of each technique on system
performance. The electrical load bank becomes the obvious choice
when tight speed control is required and has the other advantages
of easiest remote switching, minimal effect on E.O.L. performance
and ease of adding required overload protection. The electric load
bank was therefore selected as the means of system speed control.

Tree structure:

- **SPEED WILD AND CONST. SPEED REQUIREMENT**
  - **OUTPUT POWER CONTROL**
    - ELECT LOAD BANK
    - HYD LOAD BANK
  - **INPUT POWER CONTROL**
    - HEAT DUMP AS HEAT SOURCE
      - SHUTTERS
      - HEAT PIPES
    - HEAT SHUNT WITHIN LOOP

	ELECT LOAD BANK	HYD LOAD BANK	SHUTTERS	HEAT PIPES	HEAT SHUNT WITHIN LOOP
DUAL MODE	EASY REMOTE SWITCHING	ELECTRO-MECHANICAL INTERFACE REQUIRED	ELECTRO-MECH REQ	SWITCHING REQUIRED	ELECTRO - MECH INTERFACE REQ
SPEED CONTROL	EXCELLENT	GOOD	POOR	POOR	VERY POOR
EFFECT ON PCS EFFICIENCY	NONE	SLIGHT DECREASE	CONSIDERABLE HEAT LOSS WHEN NOT IN USE		NONE
OTHER EFFECTS	HIGHER GAIN PERMISSIBLE DUE TO COMPENSATION CAPABILITY. BETTER SPEED CONTROL	TIME LAG IN FILLING AND EMPTYING COULD LEAD TO INSTABILITY	EXPOSED MOVING PARTS		LARGE THERMAL EXCURSIONS ON REGEN AND TUBING NOT ABLE TO ACCEPT FULL LOAD IMMEDIATELY AFTER NO LOAD OPERATION

The electric load bank speed control sub-system consists of a speed detection circuit, a means of switching power to the load bank, and the load bank resistors.

Two frequency discriminator circuits were studied, a passive tuned circuit and an integrated circuit discriminator. The passive tuned circuit was selected based on lower parts count, lower weight, and higher reliability.

Three types of PLR* loading circuits were considered, an AC power switching circuit using SCR's, a Bang-Bang DC power switching circuit using transistor switches, and a pulse width modulated DC power switching circuit using transistor switches. The first option was unsuitable because of the low reliability of the SCR's. The second option was eliminated due to excessive speed variation which creates undesirable torque effects on the spacecraft. The selected PWM circuit has a high modulating frequency so speed variations are of a low amplitude and high frequency.

System turbine inlet temperature control was then investigated using the chart of Figure 2. As discussed in Section 1.0, direct control of temperature by throttling flow will produce inherently adequate control of turbine inlet pressure and can be designed to control temperature to any desired operating range. The other approach, using a flow limiter or pressure regulator for gross control during frequency wild operation and some means of shunting excess heat by use of a diverter valve, or shutter, provides distinctly inferior temperature control because of component thermal lags, and prevents the system from accepting full load immediately when operating at no load in the speed wild mode.

No simplification of the mechanism is evident from the use of

PLR: Parasitic load resistor

```
                    ┌──────────────────┐
                    │ TURBINE INLET    │
                    │ TEMP + PRESS     │
                    │ CONTROL          │
                    └──────────────────┘
```

| FLOW THROTTLING TO MAINTAIN CONST. INLET TEMP | PRESSURE OR FLOW REGULATOR WITH HEAT INPUT CONTROL |

| FLOW LIMITER FOR GROSS CONTROL | FAST ACTING SINGLE STAGE FLOW CONTROL | THERMAL SHUNT ON HEATER | REGENERATION BYPASS (VAPOR SIDE) | REGEN BYPASS (LIQ.SIDE) |

FLOW TRIM FOR FINE CONTROL

STEADY STATE OPERATION	GOOD TEMPERATURE CONTROL	POOR TEMPERATURE CONTROL DUE TO THERMAL LAG
FREQ. WILD OPERATION	CAN BE DESIGNED TO TRACK SPEED ADEQUATELY	SEVERE TEMPERATURE SWINGS
	FULL POWER AVAILABLE AT ALL TIMES	DELAY IN ACCEPTING FULL LOAD

2.0 Continued

the latter technique so a flow throttling valve was selected to provide temperature control.

Various flow control valves were investigated during the early months of the KIPS program to insure that the valve would be as simple and reliable as possible. Electrically operated valves were eliminated early in the studies because of the expected low reliability of the long chain of components required. The electrical valve requires a thermal sensor, electronic control circuits, an amplifier to produce valve drive current, and an electro-mechanical valve. The temperature control function is then also dependent on a continuous and uninterruptable supply of electrical power. Section 3 of this report covers a more recent re-examination of the use of an electrical temperature control valve.

A fluidic temperature control valve appeared to be quite attractive and within the state of the art. The fluidic valve should require no moving parts if temperature sensing is accomplished via a fluid viscosity sensitive capillary tube. This approach was pursued in conjuction with Harry Diamond Laboratories. As the KIPS detailed model of the steady state operating points for the system was completed an operating map of detailed requirements for the fluidic valve revealed that higher valve gain and faster response than originally anticipated were required. As additional amplifier stages were added the valve passages became more complicated, heat losses through the temperature sensor increased and valve pressure drop increased. In addition it was found that to properly start and shut down the system a check valve must be added to the fluidic valve vent line.

Alternate valves were then investigated in which direct thermal expansion of bimetallic discs or hydraulic displacement of a liquid

2.0   Continued

filled thermal bulb could be used to provide flow throttling.

The trade-off and selection process used to pick the best valve configuration prepared and presented in the Data Pac for KIPS Design Review No. 2 (Ref. 1). Further evaluation of the selected valve design was included in Design Review No. 3. (Ref. 2).

The revised selection matrix and discussion is included in this report as Appendix A.

As the result of these trade-off studies a 2 stage flapper valve was selected for temperature control. The first stage uses a redundant bellows assembly to actuate a flow limiting flapper nozzle and the second stage uses redundant liquid metal (NaK) filled temperature sensors to actuate a reference setting flapper nozzle. The valve is able to respond rapidly via the flow limiter to keep turbine flow constant under all operating conditions and can make gradual corrections to the flow setting to keep turbine inlet temperature constant.

During the second Design Review, reservations were expressed concerning several features of the valve design. The primary concern was the NaK filled thermal sensors. It was suggested that Sundstrand take a fresh look at an electrically driven valve to evaluate the potential benefits of static electronic components and a simple valve as opposed to the dual flapper, bellows, and NaK filled sensor of the current design. This recent trade-off is covered in Section 3.

Having selected a parasitic load bank for speed control with provision for underspeed alternator unloading to prevent output electrical faults from affecting the P.C.S., and a flow throttling valve to control turbine inlet temperature, the basic system controls are complete. The system can operate either frequency wild or at constant speed as desired by switching off the speed control circuit and turbine inlet temperature is controlled under all

2.0   Continued

operating conditions.   To complete the control system a voltage
regulator and heat rejection control device are required.

Voltage control is provided by the voltage regulator which
controls alternator field current.   The voltage regulator responds
rapidly to load transients to maintain close control of output
voltage even with rapid load fluctuations.   Frequency wild opera-
tion mandates the use of a voltage regulator to control output
voltage.   Alternator power output capability increases with speed
so as speed increases, alternator field current must be decreased to
maintain voltage control.   A straight forward switching regulator
circuit was selected over a proportional regulator due to the fact
that switching regulators dissipate much less power than proportional
regulators and lower power dissipation increases reliability.

Solid state switching components were compared to a system using
a saturable reactor to perform the switching function.   Solid state
switching provided advantages in both reliability and weight.
Details of the electronic controller trade-off studies are contained
in Appendix A.

A radiator bypass valve was selected as the best method of
controlling system heat rejection.   The valve provides radiator
outlet temperature control during a Titan launch, Space Shuttle
Bay operation, and deployed spacecraft operation.   The valve has
the very desirable feature of failing "safe" i.e. in the closed
position so that a failure does not greatly reduce PCS Output power
capability; a 40 watt electrical output reduction occurs if the
bypass valve fails closed.   The selection process for the bypass
valve is covered in Appendix A.

A PCS system block diagram showing the type and location of
the various controls is shown in Figure 3.

BOILER

TEMP

ALTERNATOR

TURB

REGEN

JET COND

ACCUMULATOR FILTER

RAD BYPASS

VOLT REG.

UNDERSPEED

SPEED CONTROL

RECTIFIERS

FLOW CONTROL VALVE

RADIATOR

POWER SWITCHING

CUSTOMER LOAD

SYSTEM PUMP

LOAD BANK

CRU BEARINGS
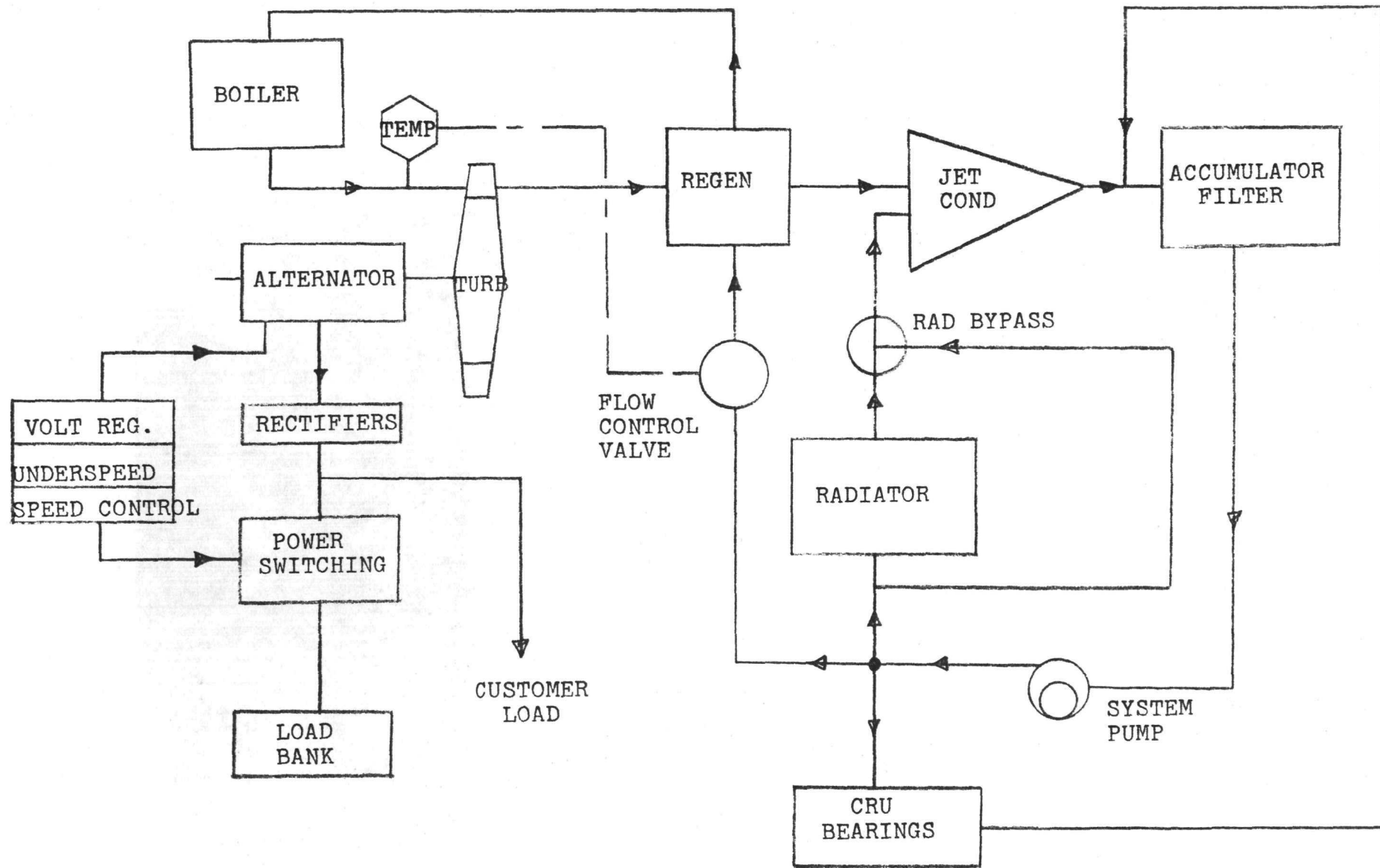
FIGURE 3 KIPS PCS BLOCK DIAGRAM

3.0 Electronic vs. Mechanical Flow Control Valve

The purpose of this section is to compare an electromechanical flow control valve with the present hydromechanical flow control valve design to see if a reliability advantage can be achieved by changing to an electrical control system.

The major advantage of an electrical control system is in the simplification of the valve mechanism, particularly deletion of the NaK filled sensor and the bellows assemblies. Accordingly a simple single stage flapper valve was designed using a flexure pivot and a dry coil to achieve the simplest possible throttle valve. This valve is shown in Figure 4. Flow characteristics are displayed on Figure 5. It is immediately obvious that the mechanical part of the electrically driven valve is considerably less complicated and therefore more reliable than the present mechanical valve of Figure 6. The critical questions are whether the added mechanical reliability is sufficient to cover the rather complicated electrical drive circuit and temperature sensing electronics package associated with the electrical valve, and the provision of an uninterruptable power supply for the electrical valve.

The second question is of particular importance for the KIPS power system. Any KIPS flow control valve must be continuously operating to maintain correct turbine inlet temperature. During system start up it is assumed that external electrical power is available to run the electrical temperature control valve until the C.R.U. is up to speed and providing power. In the event of a user short circuit after the system has been placed in operation, the KIPS alternator output voltage drops and control power is lost. With an electrical temperature control valve the P.C.S. will become unstable during a short circuit and will shut down either due to excessive
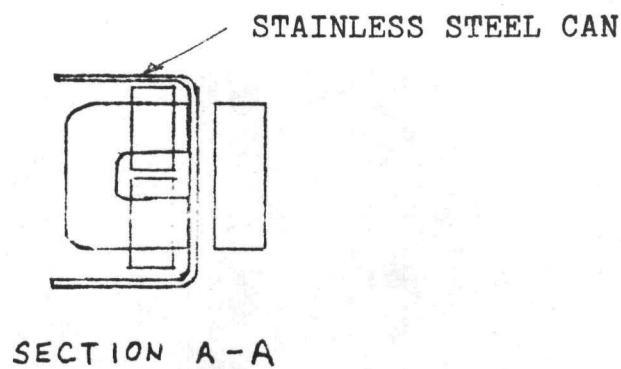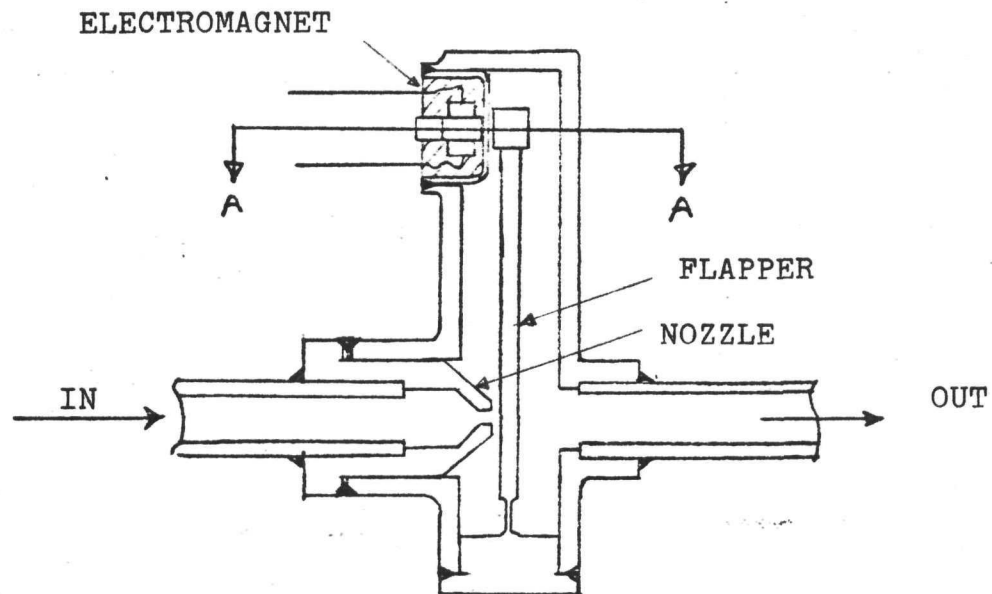
ELECTROMAGNET

FLAPPER

NOZZLE

IN →

OUT

A   A

STAINLESS STEEL CAN

SECTION A-A

ELECTRO-MECHANICAL FLOW CONTROL VALVE

FIGURE 4

FLOW CHARACTERISTICS
OF
SINGLE STAGE FLAPPER
VALVE

REDUNDANT
BELLOWS

DUAL BELLOWS
TO PREVENT NAK LEAK

REDUNDANT
TEMPERATURE
SENSORS

CONTROLLED
LEAK

INLET

OUTLET

KIPS FLIGHT SYSTEM FLOW CONTROL VALVE

FIGURE 6
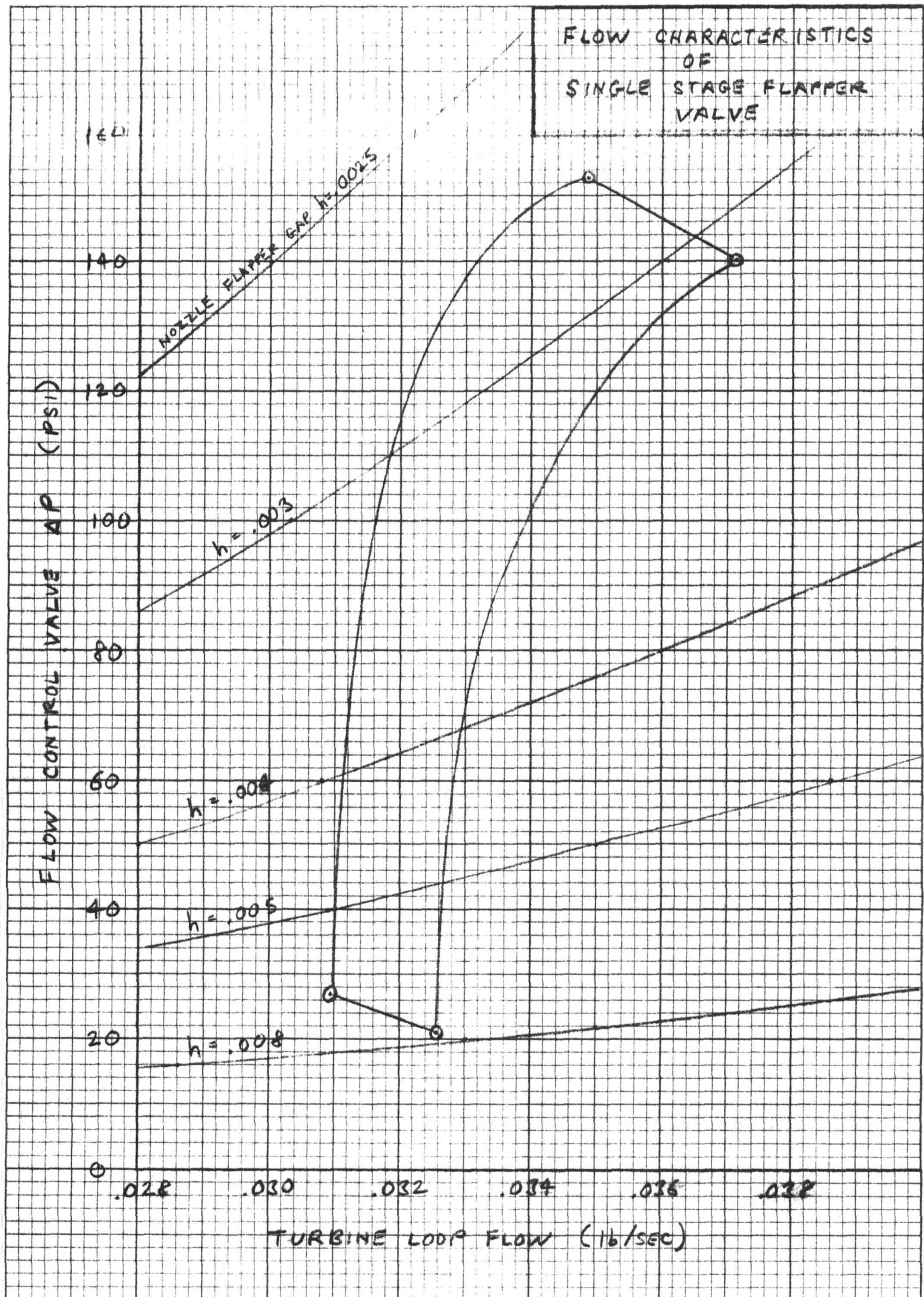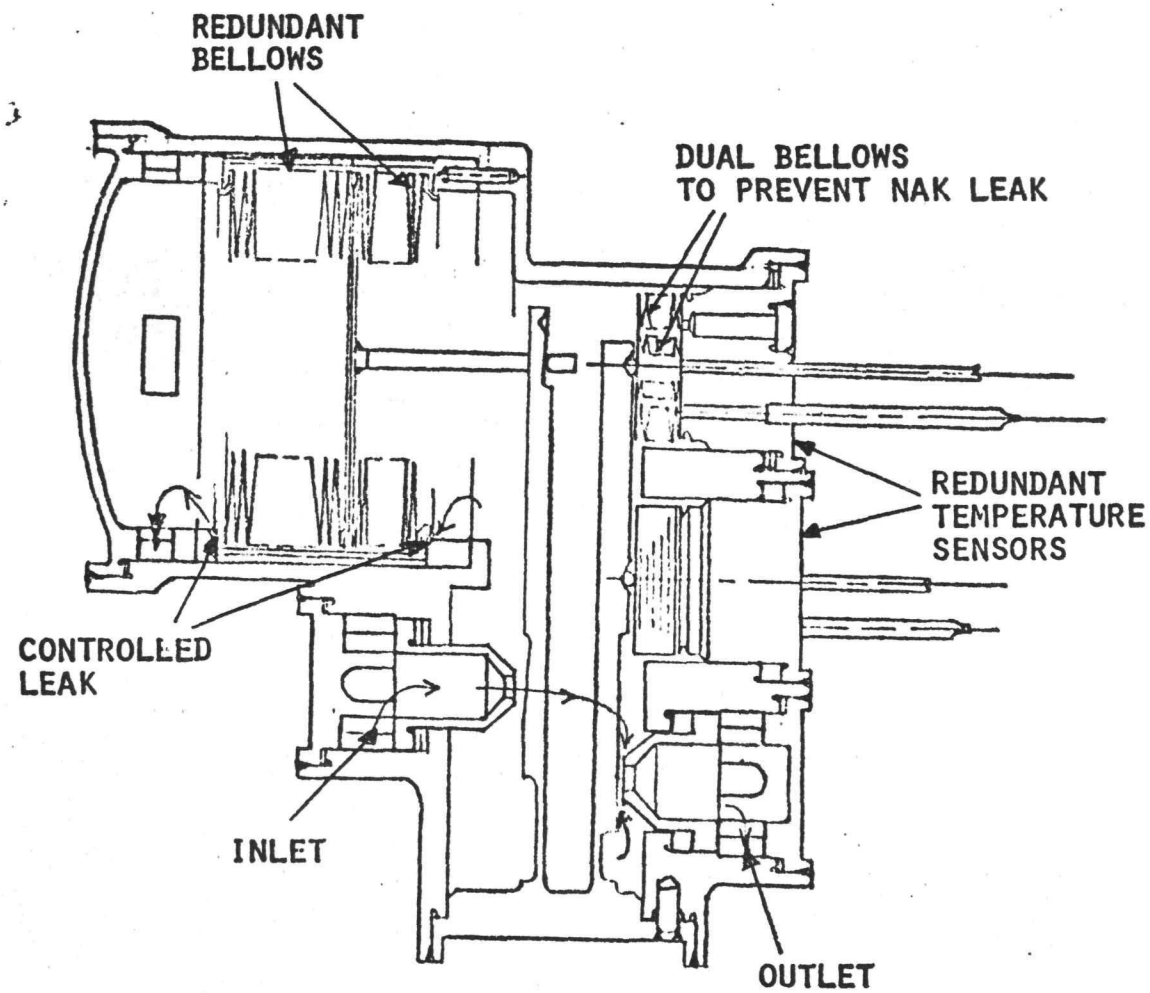
3.0 Continued

Flow which causes the turbine to run wet and eventually floods out
the jet condenser or due to insufficient flow which causes an overtempera-
ture shutdown.

Similar problems occurs in the event of a user electrical over-
load. As the overload causes the CRU to slow down, the existing
controller sheds load by removing field excitation current from the
alternator. Alternator output power then goes to zero until the CRU
speeds up and residual magnetism is sufficient to re-energize the
alternator field. If the overload is still present, the CRU slows
down and the field is again de-energized. No electrical output
power to operate the control valve is available when the alternator
field current is shut off. The electrical valve is therefore
incompatible with the present KIPS control system because the
present system does not provide uninterruptable power. Three possible
ways of providing independent temperature control power were investiga-
ted.

3.1 Uninterruptable Power Supply For Electromechanical Valve

One approach is to add a battery to the system to provide
temperature control power during electrical bus faults. A brief
investigation (Ref. 3, 4, 5) into battery life and reliability revealed
that a battery was totally unsuitable as a means of providing
backup power for the controller.

A silver-zinc battery could be stored in the unactivated state
for long periods but would require an electrical signal and time to
actuate and would not be easily reusable in the event of a second
fault several months or years later.

Ni-Cad batteries are often used on spacecraft and considerable
developement has gone into the design of high reliability batteries.

3.0 Continued

Reference 1 states that "the average service life of a Ni-Cad
battery can approach five years", not an encouraging statement for
a system with a 7 year mission life. In addition Ni-Cd batteries
cannot be operated at temperatures as high as the lowest KIPS fluid
temperature so additional battery thermal conditioning would be
required. The battery could not be floated on the 28 VDC bus but
would require a constant current power supply for trickle charging
and a state of charge sensing circuit for recharging the battery
after use.

A second possiblity for providing uninterrupted power to the
temperature controller is to redesign the CRU to add a permanent
magnet generator which would operate independently of the main
alternator to supply control power. This option, although far
better than a battery, adds additional complexity to the system.

The third possibility requires a controller redesign to
disconnect the user from the Power bus in the event of a short
circuit or overload. Current transformers could be used to sense
user current and switches used to disconnect the power in the event
of a short circuit. Mechanical switches are not desirable because
the number of moving parts will most likely exceed those of the
mechanical valve we are trying to replace.

A large transistor could be used as a switch but no JANTXV
transistors of the required 50-70 amp capacity are available.
Transistors this large have been built but have unknown reliability.
An SCR could also be used as the switch but they have a 1.2 volt
voltage drop when "on" resulting in a 55 watt power loss for a 1300
watt system. This not only amounts to a constant 4% reduction in
available output power but also necessitates active cooling of the

3.0 Continued

SCR. SCR's in general have inadequate reliability for a large life
mission, particularly at these high current levels.

Of these possible sources for uninterruptable power for the
electric valve, the P.M.G. is the most feasible and was used in
making up the electrical temperature control valve block diagram.

3.2 Temperature Sensors For Electromechanical Valve

The electrical temperature sensor used to sense turbine inlet
temperature for the electromechanical valve was selected from
three possible types.   Thermistor, thermocouple, and resistance
temperature devices were considered.

The first device to be considered, the thermistor, has a
maximum temperature limitation of 200°C or 392°F which is far
short of the 650°F turbine inlet temperature. The thermistor was
judged not suitable and not given further consideration.

The thermocouple does not suffer from this temperature limita-
tion. It is however, a thermoelectric device generating a millivolt
level electrical signal proportional to the temperature. The
thermocouple can be modeled as a voltage source in series with a
large value resistor. The input amplifier must therefore have a
very high input impedance to prevent loading down the thermocouple
output. This combination of low level signals with high input
impedance makes the circuit very susceptible to noise. This noise
susceptibility and the other problems associated with the high
input impedance make the thermocouple sensor undesirable for this
application.

The third device to be considered was the platinum resistance
temperature device, commonly referred to as a platinum RTD. It is
a coil of platinum wire whose resistance changes as a function of

3.0 Continued

temperature. It is a resistance device like the thermistor rather than a thermoelectric device like the thermocouple, and is useable at temperatures exceeding 1200°F. From considerations of operating environment and circuit noise susceptibility, the platinum RTD is the most suitable electrical temperature sensor.

Investigation revealed that platinum resistance temperature devices (RTD's) are suitable in space for periods of 10 years or more (Ref. 3). The RTD used with the electro-mechanical valve is assigned a failure rate of 2.0 failures per million hours based on data supplied by manufacturers. This data lists a minimum failure rate of 5 failures per million hours for a commercial aircraft application. Conservatively, a factor of 2.5 is used for a failure rate of 2.0 failures per million hours.

3.3 Electronic Controller Design Approach For Both Systems

3.3.1 Electronic Controller for Mechanical Valve Schematic

Reliability of the KIPS electronic controller was of prime importance in its design. Two design concepts were explored in an effort to maximize reliability while keeping equipment cost, weight and size reasonable. The first used a triple redundant design for all circuits with two of the three required to be operating for success. This resulted in a controller with a reliability greater than .99 for five years. This reliability was achieved at the expense of increased cost, weight and size.

The second method used a minimum parts design approach. The circuits were simplified and all parts were carefully selected and derated. Redundancy was retained in the parasitic load resistors (PLR's) and their transistor switches. The use of three PLR's

3.1.1 Continued

sized for 0.5 PU load reduces the switching transistors' current by
a factor of two and provides redundancy in the PLR's which have the
highest assigned failure rate in the control system. This system
provides a reliability of .982 while keeping the cost, weight and
size of the controller reasonable. For a completely triple redundant
system the cost, weight and size would be roughly three times as
large.

3.3.2 Electronic Controller For Electromechanical Valve

    The KIPS controller for the electro-mechanical valve option
employs redundancy in two areas. The parasitic load resistor and
switching transistor circuitry is the same configuration used in
the mechanical valve version. Three identical circuits are provided
with two of the three required to be operating for system success.
The second area to employ redundancy is the temperature control
circuitry. Because of the high failure rate of the platinum resis-
tance temperature device (2 failures/10^6 hours), redundancy is
required if a controller reliability greater than 0.9 for five years
is to be achieved. The configuration used has three identical
temperature control circuits with two of the three required for
system success. Using redundancy in these two areas a controller
reliability of .9599626 for five years is achieved.

3.3.3 Comparison of the Block Diagrams For The 2 Valve Approaches

    The resulting block diagrams for the electromechanical and
hydromechanical systems are shown in Figures 7 and 8 respectively.

    A list of the significant differences between the two systems is
presented in Figure 9. The hydromechanical valve has a pressure
regulator function to maintain flow control during speed transcents

KIPS CONTROL SYSTEM

ELECTRO-MECHANICAL

VALVE OPTION

FIGURE 7

SPEED CONTROL

3 ∅ HOMOPOLAR GENERATOR

3∅ ARMATURE STAR CONNECTED 1130 Hz

SOLID ROTOR

FIELD WINDING

TURBINE

TRANSFORMER PRIMARY

PASSIVE TUNED CIRCUIT FREQUENCY DISCRIMINATOR

PASSIVE TUNED CIRCUIT FREQUENCY DISCRIMINATOR

TRIANGLE WAVE GENERATOR

Σ

TRANSISTOR COMPARATOR

SOLID STATE POWER SWITCH

SOLID STATE POWER SWITCH

SOLID STATE POWER SWITCH

THREE ELEMENT PLR

OVERLOAD PROTECTION

COMPARATOR N < 95%

TUNING CAPACITOR BANK

3∅ FULL WAVE RECTIFIER

± 28 VDC

L-C FILTER

EXTERNAL LOAD

VOLTAGE REGULATOR

CURRENT SWITCHING AMPLIFIER

ZENER REFERENCE

Σ

RESISTOR

RESISTOR

FROM PUMP

Flow Limiter

TEMP Control

TO REGEN

Thermal Sensor

Thermal Sensor

KIPS CONTROL SYSTEM
MECHANICAL VALVE

FIGURE 8

Function	Hydromechanical	Electromechanical
Temp. Sensing	Double Redundant NaK Filled Bulbs	Triple Redundant Platinum Resisting Bulbs
Motive Power	NaK Expansion For Temp. Portion Fluid Feedback on Double Redundant Bellows For Gross Flow Limiting	Single PMG Triple Redundant Drive Circuits Single Electromagnet on Flapper Nozzle
Cooling Requirement	None	Cold Plate (Existing)
Power Consumption	None	10 Watts
Response Time Constant For Speed Perturabation	Milliseconds	No Valve Response
Response Time Constant For Temp. Perturbation	10 Seconds	2 - 3 Seconds Minimum
Type of Flow Control	Flow Limit With Temp. Bias	Direct Temp Control of Flow
System Stability	Good	Presumed Adequate

FIGURE 9 VALVE FUNCTIONAL COMPARISON

RELIABILITY BLOCK DIAGRAM

ELECTRO-MECHANICAL FLOW CONTROL VALVE

*

```
PMG                 Rectifiers          Temperature Control      Electromagnet      Control Valve
λ=.026×10⁻⁶         λ=.00526×10⁻⁶       λ=2.1165×10⁻⁶           Coil               Flapper
R₁=.998862          R₂=.99977           R₃=.911463              λ=.0096×10⁻⁶       λ=1.0×10⁻⁹
                                                                 R₄=.99958          R₅=.999956
                                        Temperature Control
                                        λ=2.1165×10⁻⁶
                                        R₃=.911463

                                        Temperature Control
                                        λ=2.1165×10⁻⁶
                                        R₃=.911463
```

$$PMG: \lambda = .026 \times 10^{-6}, \quad R_1 = .998862$$
$$Rectifiers: \lambda = .00526 \times 10^{-6}, \quad R_2 = .99977$$
$$Temperature\ Control: \lambda = 2.1165 \times 10^{-6}, \quad R_3 = .911463$$
$$Temperature\ Control: \lambda = 2.1165 \times 10^{-6}, \quad R_3 = .911463$$
$$Temperature\ Control: \lambda = 2.1165 \times 10^{-6}, \quad R_3 = .911463$$
$$Electromagnet\ Coil: \lambda = .0096 \times 10^{-6}, \quad R_4 = .99958$$
$$Control\ Valve\ Flapper: \lambda = 1.0 \times 10^{-9}, \quad R_5 = .999956$$

2 of 3 required

$$R = 3R_3^2 - 2R_3^3$$
$$R = .977872$$

* $\lambda = 2.0 \times 10^{-6}$ for platinum resistance bulbs.

$$R = .998862 \times .99977 \times .977872 \times .99958 \times .999956$$

$$R = .9761 \quad (5\ years)$$

FIG 10

RELIABILITY BLOCK DIAGRAM
MECHANICAL FLOW CONTROL VALVE

Pressure Motor Bellows
$\lambda_1 = 100.4 \times 10^{-9}$
$R = .99998$

Pressure Motor Bellows
$\lambda_1 = 100.4 \times 10^{-9}$
$R = .99998$

Inlet Nozzle Flapper
$\lambda_2 = 1.0 \times 10^{-9}$
$R = .999956$

Temp. Sensor
$\lambda_3 = 100.4 \times 10^{-9}$
$R = .99998$

Temp. Sensor
$\lambda_3 = 100.4 \times 10^{-9}$
$R = .99998$

Outlet Nozzle Flapper
$\lambda_4 = 1 \times 10^{-9}$
$R = .999956$
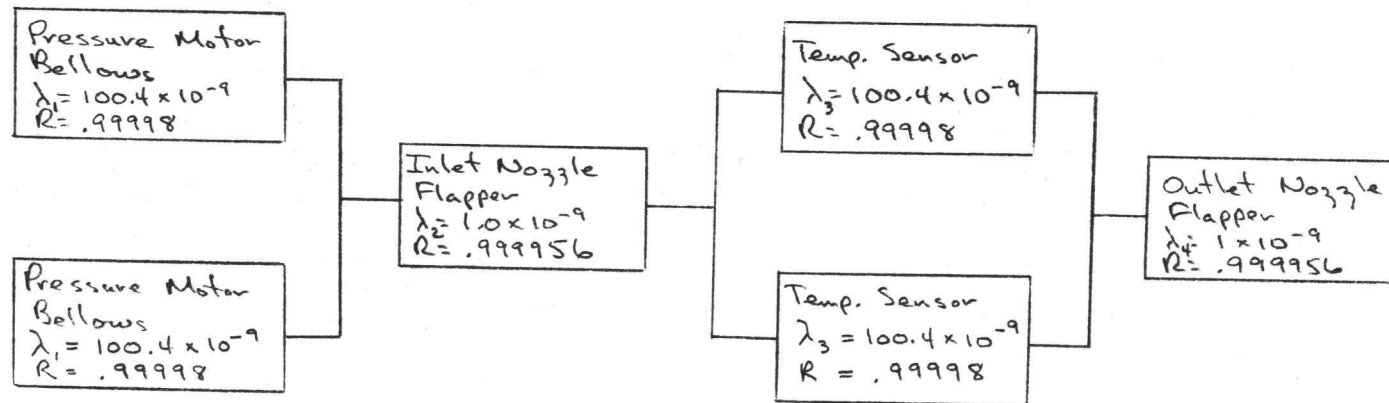
$$R = \left[2e^{-\lambda_1 t} - e^{-2\lambda_1 t} \right] \left[e^{-\lambda_2 t} \right] \left[2e^{-\lambda_3 t} - e^{-2\lambda_3 t} \right] \left[e^{-\lambda_4 t} \right]$$

$R = .999872 \quad (5 \text{ years})$

FIG. 11

3.3.3 Continued

and a relatively slow responding temperature control function. The
electromechanical valve does not have the pressure regulator
function. It is assumed that the more rapid temperature response
of the electrical temperature sensor can make flow corrections
sufficiently fast so that instability does not occur during speed wild
operation.

The increased circuitry required for the electro-mechanical valve
option slightly increases the controller size, weight, cost, and power
consumption. Increase in size, weight and cost is estimated to be
10 to 20%. Power consumption increase for the temperature control
circuit and electro-mechanical valve is estimated to be on the order
of 10 watts, evenly divided between the two.

3.4 Reliability Comparison of Both Systems

Figures 10 and 11 show the reliability block diagrams for the
electro-mechanical valve and controls and the mechanical valve. The
failure rates for the mechanical valve are as presented in the Design
Review #3 data package (See Section 3.4.1 of this report). The failure
rates for the electro-mechanical valve controls and electro-magnetic
coil are MIL-HDBK-217B rates, except for the platinum bulb rate
which is not listed in MIL-HDBK-217B. Rationale for selection of an
appropriate failure rate for the platinum RTD is discussed in
section 3.2.

The lesser reliability of the electro-mechanical valve and controls
is due primarily to the relatively high failure rate estimated for
the platinum resistance bulbs. However, even if the platinum bulb
reliability was unity (λ = 0) the reliability of the remainder of
the electro-mechanical valve and controls would be .99809 which is less

3.4   Continued

than the .99987 reliability calculated for the mechanical valve.

The hydromechanical valve is therefore preferred for inclusion
in the KIPS Flight System.

Additional supporting data on the reliability of the bellows,
NaK filled sensor, and the operating temperature of the electronic
controller has been collected and is discussed in the following sub-
sections.

3.4.1   KIPS Component Operating Temperature and Derating

Because of coolant temperature limitations, the KIPS controller
will be operating at an ambient temperature of approximately 75°C.
This is well below the maximum operating temperature of all controller
components. Some questions have been raised about the effect of this
temperature on semiconductor and IC reliability.

A review of some of the available literature was made to determine
what temperature derating guidelines are commonly used for high
reliability spacecraft applications. The NASA funded "Long-Life
Assurance Study for Manned Spacecraft Long-Life Hardware", NASA-CR-128908
lists the temperature derating criteria shown in Table 1 for IC's,
transistors, and diodes. The 75°C operating ambient temperature is
within the 85°C limit of Marshall Space Flight Center (MSFC) for
IC's. Likewise the 102°C maximum transistor junction temperature
is below the 110°C limit of MSFC, GSFC, and Motorola and well
below the 130°C limit of Martin. The maximum KIPS diode junction
temperature of 104°C does exceed the 100°C limit recommended by
MSFC, GSFC, and Grumman but is below the maximum of 110°C recommended
by motorola and 130°C recommended by Martin. It should be remembered
that these are recommendations only and not absolute limits which if
exceeded, will result in sudden loss of reliability.

3.4.1 Continued

	KIPS	MSFC	Martin	Motorola	Grumman	GSFC
I.C. – T ambient Max.	75°C	85°C	–	–	–	–
Transistor–T_j Max.	102°C	110°C	130°C	110°C	–	110°C
Diode – T_j Max.	104°C	100°C	130°C	110°C	100°C	110°C

Table 1 – KIPS temperature vs. limits of various space agencies and contractors.

3.4.1 Continued

To further reinforce this statement that the reliability of the KIPS controller is adequate with a cold plate temperature of 75°C, reliability predictions were made at 75°C and 40°C for comparison. MIL-HDBK-217B, "Reliability Prediction of Electronic Equipment" was used for both predictions. The component failure rates in MIL-HDBK-217B are based on component internal or junction temperature which are derived from case temperature and electrical stress. For most controller components case temperature will be equal to cold plate temperature. Because of higher power levels and thermal resistance from case to cold plate, the actual case temperatures of the PLR switching transistors, and rectifier diodes will be higher than the cold plate. Actual case temperatures of these components were used in both cases. These yielded reliabilities of .98647836 and .98194596 for 40°C and 75°C respectively. The increase in reliability at the lower temperature is small, less than 0.5%.

3.4.2 Nak Sensor Data

A data base search has been conducted to support the use of NaK filled temperature sensors in the KIPS mechanical valve. Use of NaK filled temperature sensors has been found in commercial and industrial applications. NaK filled pressure sensors, somewhat similar to the mechanical valve temperature sensor, were found to be used in reactor component experimental loops. Also found were data for bellows liquid metal seals used in reactors and reactor experiments. The temperature and pressure sensor and bellows data is presented in the following paragraphs. Also presented are the results of a Sundstrand study concerning corrosion of thermal sensor materials by NaK.

Sundstrand has contacted several manufacturers who use NaK

3.4.2 Continued

filled capillary temperature sensors in commercial and industrial

applications. Figure 12 describes the NaK temperature sensors,

their applications and experience.

References 6, 7, and 8 discuss the use of NaK filled capillary

pressure transducers in high temperature alkali metal test loops

at General Electric, The pressure transducer described in these

references contain a slack diaphram which separates the process

fluid from the Na filled capillary tube. The pressure of the

process fluid is transmitted across the slack diaphram through the

NaK filled capillary tube to a bourdon tube. This capillary is

316 stainless, 0.125 inch OD x 0.050 ID x 25 feet long. In

comparison the capillary for the KIPS temperature sensor is 304

stainless, 0.062 inch OD x 0.026 inch ID x 7 inches long.

The transducers accumulated 72,650 hours during the referenced

loop tests. Operating times and approximate operating tempera-

tures for the transducers are as follows:

	Number of Transducers	Loop Time (Hours)	Transducer Temp. ($^\circ$F)
Cb-1Zr Pumped Sodium Loop (Ref. 6)	1	2,650	650
Cb-1Zr Rankine System Corrosion Test Loop (Ref. 7)	4	5,000	2000 1840 1350 800
T-111 Rankine System Corrosion Test Loop (Ref. 8)	5	10,000	2000 1925 1350 800 800

NAK FILLED TEMPERATURE SENSOR EXPERIENCE

APPLICATION	SENSOR DESCRIPTION	EXPERIENCE	COMMENTS
Honeywell Controls, Models T683 & T654 Temperature Controllers	S.S. capillary, .025 ID, .093 O.D. Copper bellows. Silver brazed and heliarc welded throughout. NaK sensors used up to $1200^{O}F$.	15 years 300 units/year	18 month warranty QA manager says "very good success".
General Electric Appliance Division, Self-Cleaning Ovens	304 SS capillary, .020 ID, .062 OD. Oven cleaning temperature is $880^{O}F$	Field Experience: 6 years 3/4 million units/year Test Experience: 1000 hours @ $900^{O}F$ 3500 hours cycling $350 - 450^{O}F$	When tested above $1000^{O}F$, capillary plugging was experienced.
Temperature sensor for night storage heaters. Sensor produced by Essex International, Mansfield, Ohio	Diaphram with 321 SS capillary .020 ID, 18 - 45 in. length. Sensor is set for $1300 - 1400^{O}F$ Sensors experience temp. change from $500^{O}F$ to $1300^{O}F$ daily.	50,000 sensors shipped to England since 1972.	Used in brick heating ovens to heat bricks overnight at low electric rates for day-time heat supplys. No service problems reported by sales/service concern.

FIG 12

3.4.2 Continued

The reference 6 loop transducer was filled with commercially pure
NaK (analysis of NaK sample indicated 5 ppm oxygen level). The reference
7 loop transducers were filled with high-purity NaK (analysis of two NaK
samples indicated an average 3.8 ppm oxygen level). A sample of Nak
used to fill the reference 8 transducers was analyzed to contain less
than 3 ppm oxygen.

Operation of the pressure transducers in the three loops is described
as trouble-free.

Additional information about NaK filled capillary pressure sensors
was obtained from Westinghouse (Ref. 9). Westinghouse uses Na filled
sensors for pressure measurement in high temperature sodium streams.
Approximately 10 pressure sensors, manufactured by Taylor Instrument
Company, have been used for about 10 years. An estimated 20,000
total hours have been logged for the transducers, with the high time
transducer having about 3,000 hours. The capillaries are stainless steel,
0.058 inch ID x 25 ft. long. The transducers are used at up to
1200°F. The only failures have been capillary breaking due to
handling damage.

Reference 10 presents the results of a literature search for
information about the design and operating experience of bellows
in liquid metal systems. Examples are cited of metallic bellows
in a molten metal environment in a number of reactors and reactor
experiments. With the exception of one example of a hydraulically
formed structural bellows, all other examples are for edge welded
type bellows. The report (Ref. 10) states that bellows requiring
long stroke and/or low spring rate are virtually always of the welded
type. In general, the bellows described in the report are unlike
the NaK sensor bellows in size and environment. Most of the example

3.4.2 Continued

bellows seals described were exposed to sodium vapor at up to 1150°F

a liquid sodium at 1200°F. However, one bellows of interest is

incorporated in the expansion compensator units of the SNAP 10A

Nuclear Power Unit. These bellows are welded metallic bellows.

They use 0.010 in, thick metallic discs, are of AM-350 steel and

see liquid NaK at 750°F in service. No experience data is provided

in the report. The KIPS temperature sensor bellows is of AM-350, but

is immersed in a fluid at approximately 220°F. A primary value of

the reference 10 report is that it provides an extensive bibliography

of references relating to design and operating experience of bellows

in liquid metals.

This bibliography may be useful for locating additional operating

experience data, particularly for the SNAP 10A expansion compensator

bellows.

3.4.3 Thermal Sensor Corrosion

A study was conducted concerning the corrosion of thermal

sensor materials by NaK. The conclusion of the study was that corro-

sion of thermal sensor materials should not present problems in

achieving the seven year life. Details of this study are presented

in Appendix B.

The following is also provided as supplementary information,

based on telephone conversation and correspondence.

J. H. Devan, Oak Ridge National Lab, advised that there would

be no compatability problem between AM-350 stainless steel and

NaK between 220°F and 650°F. Mr. Devan also advised that dissolu-

tion of precipitate in NaK occurs at about 800°C (1472°F).

Correspondence with W.E. Reuther, Argonne National Lab, on the

3.4.3 Continued

topic of the effects of NaK on stainless steel bellows, advised
that NaK would not be expected to interact in any significant way
with stainless steel bellows at $110^{\circ}C$ ($230^{\circ}F$).

B. C. Huguelet, Argonne National Lab, advised that there are
no appreciable corrosion or embrittlement problems with NaK and
stainless steel at $700^{\circ}F$ as long as the welds are good and free
from impurities; corrosion starts at about $600^{\circ}C$ ($1112^{\circ}F$).

3.4.1   MECHANICAL CONTROL VALVE RELIABILITY

Figure 13, which is extracted form the Design Review #3 data
package, shows the reliability estimation for the KIPS mechanical
control valve.

Figure 14 shows the block diagram for the pressure motor bellows
seals. Either bellows seal may fail with only minor consequences.

Figure 15 shows the thermal sensor. Either thermal sensor bellows
may fail without leaking NaK into the working fluid loop. Two
temperature sensors are contained in each control valve so that, in
the event of a primary (inner) bellows failure in one sensor, the
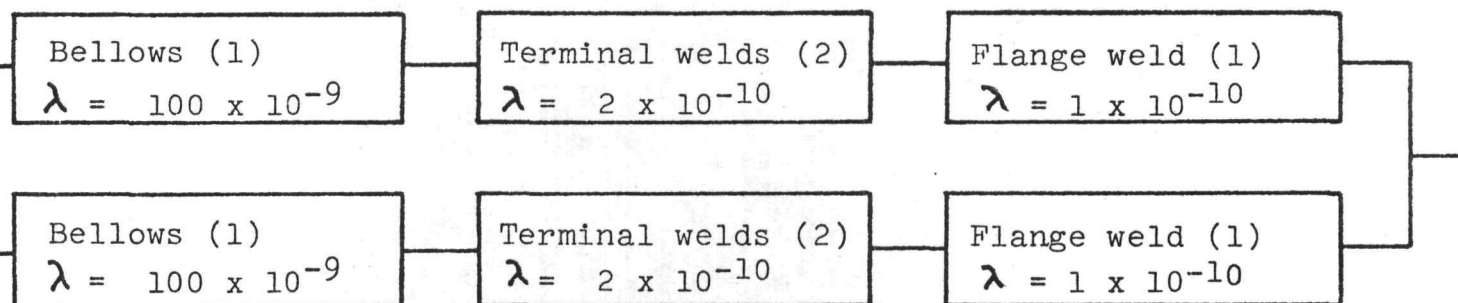other temperature sensor will carry the control valve function.

COMPONENT	FAILURES PER 10^9 HOURS		REMARKS
	ALLOCATED	ESTIMATED	
Flow Limiter	22.8	2.9	Moving parts in the flow limiter are: - 2 redundant temperature sensors; - 2 redundant bellows seals; - 2 nozzle flappers.

Each temperature sensor provides dual bellows containment so that, in the event of the primary bellows failure, Nak is not released in the system.

KIPS power capability is not degraded due to the failure (leakage) of one temperature sensor. The estimated failure rate of the thermal sensor was calculated by modeling (1 of 2 may fail) the thermal sensors in parallel. Using a bellows failure rate of 0.1×10^{-6} (AVCO Reliability Engineering Data Series - Failure Rates) the failure rate for the pair of temperature sensors was found to be 0.43×10^{-9}. The AVCO Failure rate is conservative compared to MCBF of 6.9×10^{-8} cycles (90% confidence) based on Metal Bellows Company Data for bellows of the same material and designed using the same design approach. It is estimated that each KIPS temperature sensor will experience 2555 thermal cycles.

The estimated failure rate for dual redundant bellows seals was also calculated by modeling the bellows in parallel. The estimated failure rate for each bellows is 0.1×10^{-6} (AVCO Reliability Engineering Data Series Failure Rates). The estimated failure rate for the redundant bellows configuration was calculated to be 0.43×10^{-9}.

The nozzle flappers are designed for a failure probability of $< 1 \times 10^{-9}$.

$$\lambda_{Total} = (0.43 + 0.43 + 1 + 1) \times 10^{-9}$$
$$\lambda_{Total} = 2.86 \times 10^{-9}$$

-43-

FIG 13

FIG 14

PRESSURE MOTOR BELLOWS REDUNDANCY
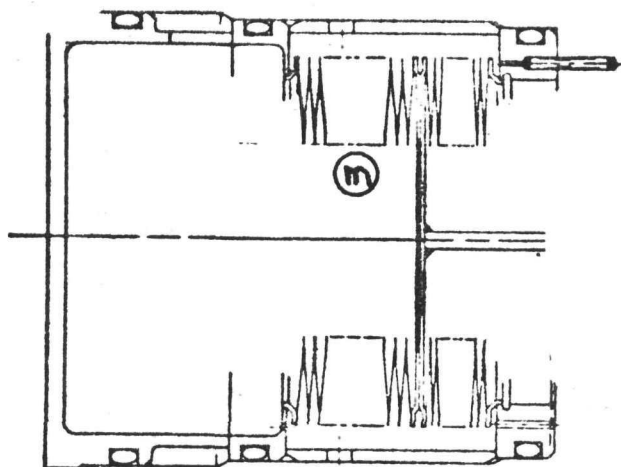
- Either bellows may fail with only minor consequences.

- System tolerant of minor bellows leakage.

- Access to both sides of bellows for cleaning.

- Helium leak check of bellows assembly.

Bellows (1) $\lambda = 100 \times 10^{-9}$	Terminal welds (2) $\lambda = 2 \times 10^{-10}$	Flange weld (1) $\lambda = 1 \times 10^{-10}$
Bellows (1) $\lambda = 100 \times 10^{-9}$	Terminal welds (2) $\lambda = 2 \times 10^{-10}$	Flange weld (1) $\lambda = 1 \times 10^{-10}$

$$R_T = 2e^{-\lambda t} - e^{-2\lambda t} = e^{-\lambda_T t}$$

$$\lambda_T = 0.43 \times 10^{-9} \text{ failures per hour}$$

THERMAL SENSOR BELLOWS REDUNDANCY
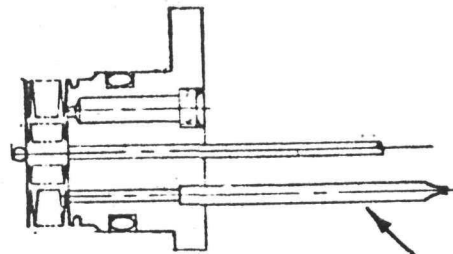
Helium leak check tube,
crimp and weld after
vacuum bake.

● Analysis shows no NaK corrosion problem with sensor parts.

Thermal sensor Reliability $= e^{-\lambda t} = .9956$ (5 years)

$= 100.4 \times 10^{-9}$ failures per hour

(Redundant thermal sensor will allow continued system operation.)

System Reliability Impact (no NaK in fluid loop) $= 2e^{-\lambda t} - e^{-2\lambda t}$

$= .99998$ (5 years)

equivalent $\lambda = .43 \times 10^{-9}$ failures per hour

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APPENDIX A

Control System Selection as Presented For Design Review #3

3.0   ELECTRICAL CONTROLS (PDR)

   3.1   Tradeoffs - Block Diagrams, Reliability, Performance
          Weight, Single Point Failure Considerations
   3.2   Baseline Components
   3.3   FSCD and GDS Selection (Similarity and Differences)
   3.4   Breadboard Test Results
   3.5   Further Study and Test Plans
   3.6   Margins of Safety and Reliability
   3.7   Generator Terminal Block
   3.8   Critical Characteristics

4.0   FLUID CONTROLS (PDR)

   4.1   Turbine Flow/Temperature Control Valve

          4.1.1   Function
          4.1.2   Tradeoffs
          4.1.3   Selected Design, Analysis
          4.1.4   Similarity to Flight System
          4.1.5   Test Plan
          4.1.6   Reliability
          4.1.7   Quality Control, Cleaning, Critical Characteristics
          4.1.8   Appendix

   4.2   Radiator Bypass Valve

          4.2.1   Function and Need
          4.2.2   Tradeoffs
          4.2.3   Selected Design and Analysis
          4.2.4   Similarity to Flight System
          4.2.5   Test, Plans
          4.2.6   Filter
          4.2.7   Reliability
          4.2.8   Quality Control, Cleaning, Critical Characteristics
          4.2.9   Appendix

3.0   ELECTRICAL CONTROLS (PDR)

3.1   TRADEOFFS

Tradeoffs were made between various controller configura-
tions. Our intent was to determine the least complex
controller that has a probability of greater than .99 of
achieving the mission life of 5 years. This reliability
criteria was pursued rather than a no single point failure
philosophy because it produced a less complex and lower
weight system with an acceptable mission risk. Block
diagrams of various configurations that were reviewed are
discussed below. The reliability numbers that are given
are the probability of achieving the 5 year mission life.

3.1.1 SPEED CONTROL

1. The baseline voltage regulator of Fig. 1 has one frequency
discriminator, one comparator, three power switches and three
PLR's. If the PLR's are sized such that all three are re-

quired for full load the reliability is .8739218. If the sizing is changed such that any two PLR's are capable of full load then the reliability is increased to .99265277.

2. Fig. 2 shows a speed control where each PLR is driven by a separate frequency discriminator, comparator, and power switch combination. Two PLR's are capable of handling the full load. The reliability is .994073.

3. The speed control of fig. 3 has three parallel connected frequency discriminators driving three sets of parallel connected comparators which control three sets of series connected power switches. Two PLR's are capable of full load. The reliability is .994644.

4. Fig. 4 shows three parallel connected frequency discriminator and comparator combinations, each capable of driving two PLR's through series connected power switches. Two PLR's are capable of full load. The reliability is .994192.

3.1.2 VOLTAGE REGULATOR

1. The baseline voltage regulator with overload protection is shown in figure 5. The reliability is .99737289.

2. The voltage regulator of figure 6 consists of two of the baseline voltage regulators connected in parallel with only one regular required for proper operation. The reliability is .9956431.

3. Fig 7 shows a voltage regulator which consists of three baseline voltage regulators, with each one driving a separate field in the generator. Only two of the fields, and associated circuitry, are required for proper operation. The reliability is .999104.

3.1.3 TUNING CAPACITOR

Figure 8 shows the baseline tuning capacitors and a tuning capacitor system that could be used for a no single point failure philosophy.

Reliabilities are:

Baseline	.999999
Fused Redundant	.9999412

3.1.4 RECTIFIER

The baseline rectifier and other rectifier configurations that could increase rectifier reliability are shown in fig 9. The reliabilities are as follows:

Baseline	.999107
Redundant	.9999994
Fused Redundant	.99988461
Combination	.9999922

3.1.5 CONCLUSIONS

The desired controller must have an overall reliability of .99. Since the PLR's individually have been assumed to have the lowest reliability of all the controller compo-nents, the first step in increasing the baseline controller reliability is to choose the configuration where the PLR's are sized such that two PLR's are capable of full load. With this change, the baseline controller will have a reliability of .990045. Any additional changes that would significantly increase controller reliability would also significantly increase system weight. A description of the block diagrams for the baseline system appears in the following section.

3.2.1   BASELINE SPEED CONTROL

A block diagram of the speed control is shown in fig. 1.
This concept relies on the turbine being capable of
accelerating a load slightly less than 1.0 per unit (P.U.)
through the rated frequency of 1122Hz, and also that a
load slightly greater than the 1.0 P.U. will be too great
for the turbine to hold at that frequency. Consequently,
whenever the generator output frequency exceeds 1122 hz,
a load of at least 1.0 P.U. is added to the load. The
turbine speed will start to decelerate, regardless of the
actual useful load. The parasitic load is removed once
the generator output frequency is less than the rated
frequency of 1122 hz. The parasitic load is comprised of
three parasitic load resistors (PLR's) each drawing .5PU
load such that two PLR's are capable of providing full
load. Having three PLR's with any two capable of handling
full load greatly increases the reliability of the system
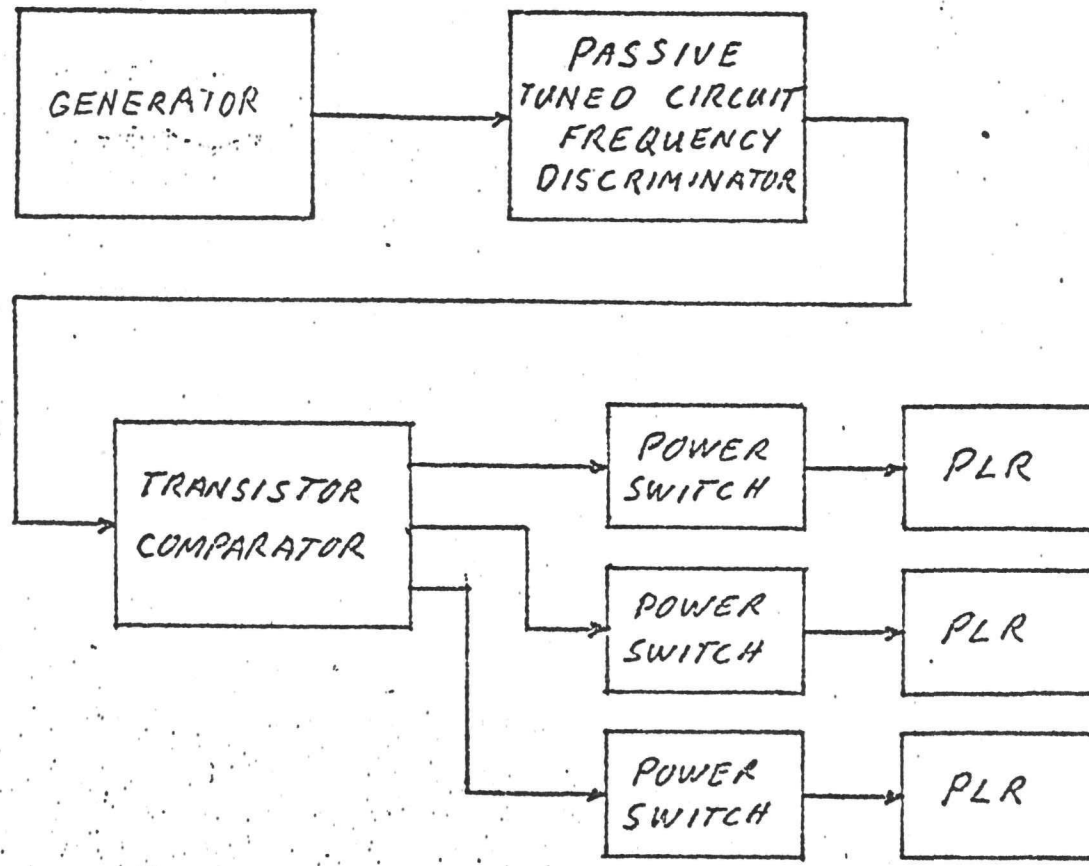since a short circuited PLR is unlikely.

The frequency discriminator consists of a passive tuned
circuit which produces a DC output voltage proportional to
system frequency near the resonant frequency of the tuned
circuit. Figure 10 shows the general form of the frequency
discriminator transfer characteristic. The system operates
slightly to the right of the f_o point. The comparator is
set to switch at the voltage proportional to 1122 Hz. This
comparator turns the power switches on or off, applying or
removing power to the PLR's.

3.2.2 BASELINE VOLTAGE REGULATOR

A block diagram for the voltage regulator is shown in figure 5. This voltage regulator is a switching type to minimize the regulating losses. Overload protection is included to prevent an excessive load condition from causing the turbine to decrease in speed to a point where the system would cease operation. The switching point of the regulator is set by the zener reference and bias circuit. When the output voltage is low, the switching amplifier is on, causing maximum field current to flow in the field winding, increasing the generator voltage. When the output goes higher than the set point, the switching amplifier is turned off, allowing the field current and generators output voltage to start to decay. The overload protection uses a frequency discriminator similar to the speed control frequency discriminator, and a comparator to detect when turbine speed falls below 95%. When this is detected, the switching amplifier is turned off, decreasing the generator output voltage and lowering the generator load. The generator output voltage will not decrease below a small residual voltage which is sufficient to operate the frequency discriminator, comparator, and switching amplifier. The reduction in generator load prevents the system from ceasing operation.
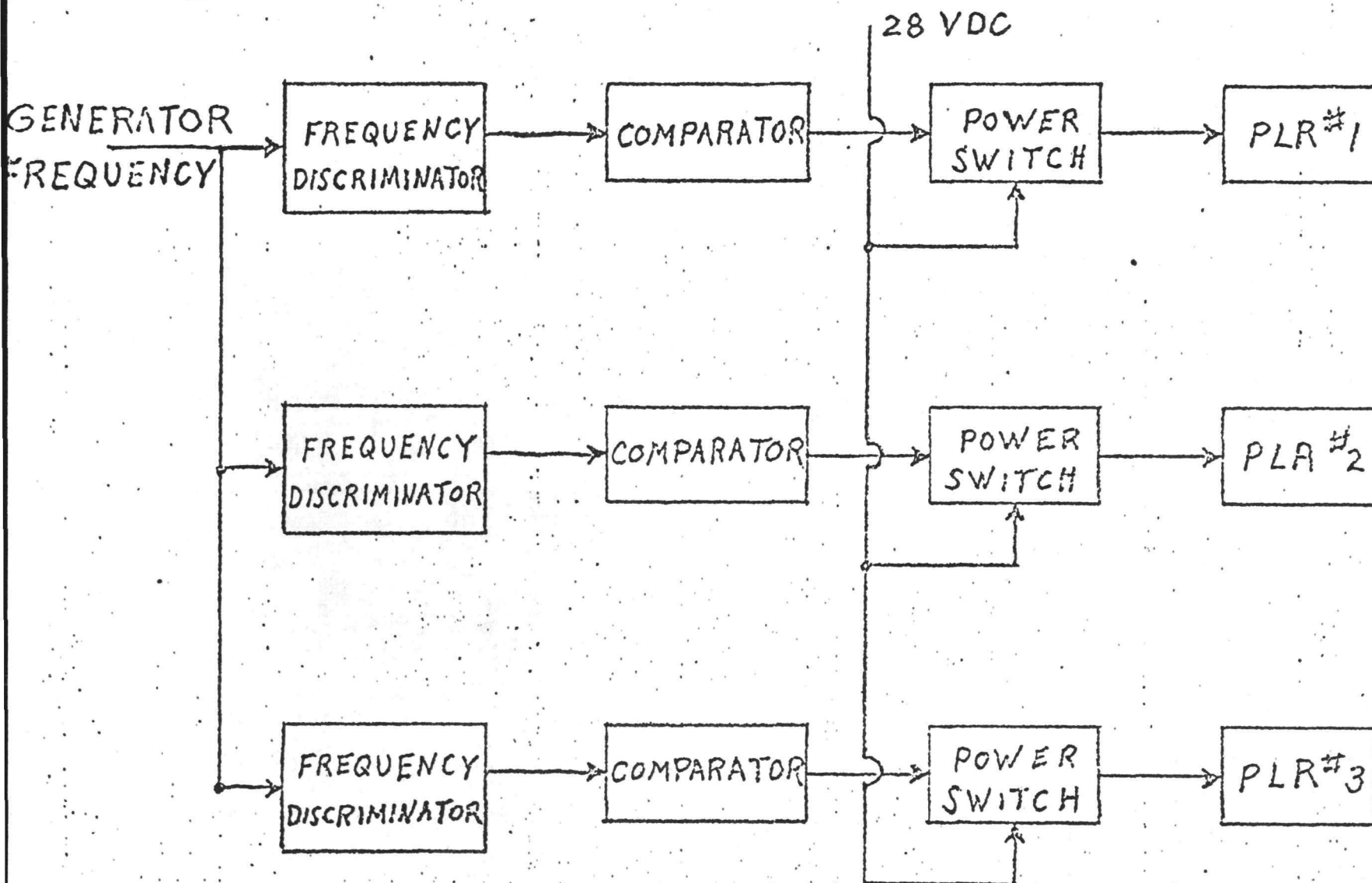
FIG. 1

SPEED CONTROL

```
┌──────────────┐          ┌──────────────────┐
│              │          │     PASSIVE      │
│  GENERATOR   │─────────▶│  TUNED CIRCUIT   │
│              │          │    FREQUENCY     │
│              │          │  DISCRIMINATOR   │
└──────────────┘          └──────────────────┘


     ┌───────────────┐        ┌──────────┐      ┌──────────┐
     │               │───────▶│  POWER   │─────▶│   PLR    │
     │  TRANSISTOR   │        │  SWITCH  │      │          │
     │  COMPARATOR   │        └──────────┘      └──────────┘
     │               │        ┌──────────┐      ┌──────────┐
     │               │───────▶│  POWER   │─────▶│   PLR    │
     └───────────────┘        │  SWITCH  │      │          │
                              └──────────┘      └──────────┘
                              ┌──────────┐      ┌──────────┐
                              │  POWER   │─────▶│   PLR    │
                              │  SWITCH  │      │          │
                              └──────────┘      └──────────┘
```
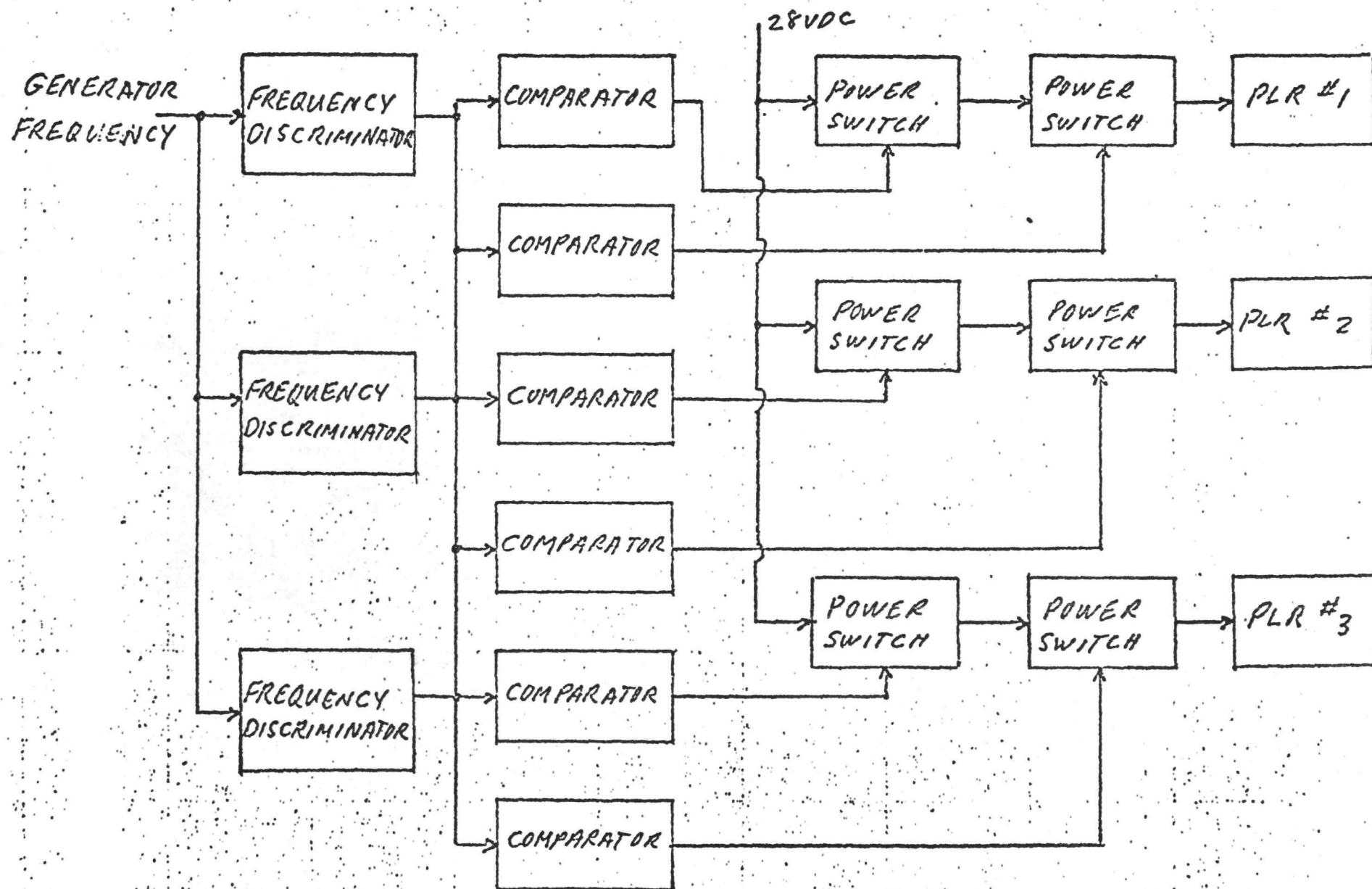
2 PLR's CAPABLE OF FULL LOAD

FIG 1

9/2/76

SPEED CONTROL

2 PLR'S CAPABLE OF FULL LOAD
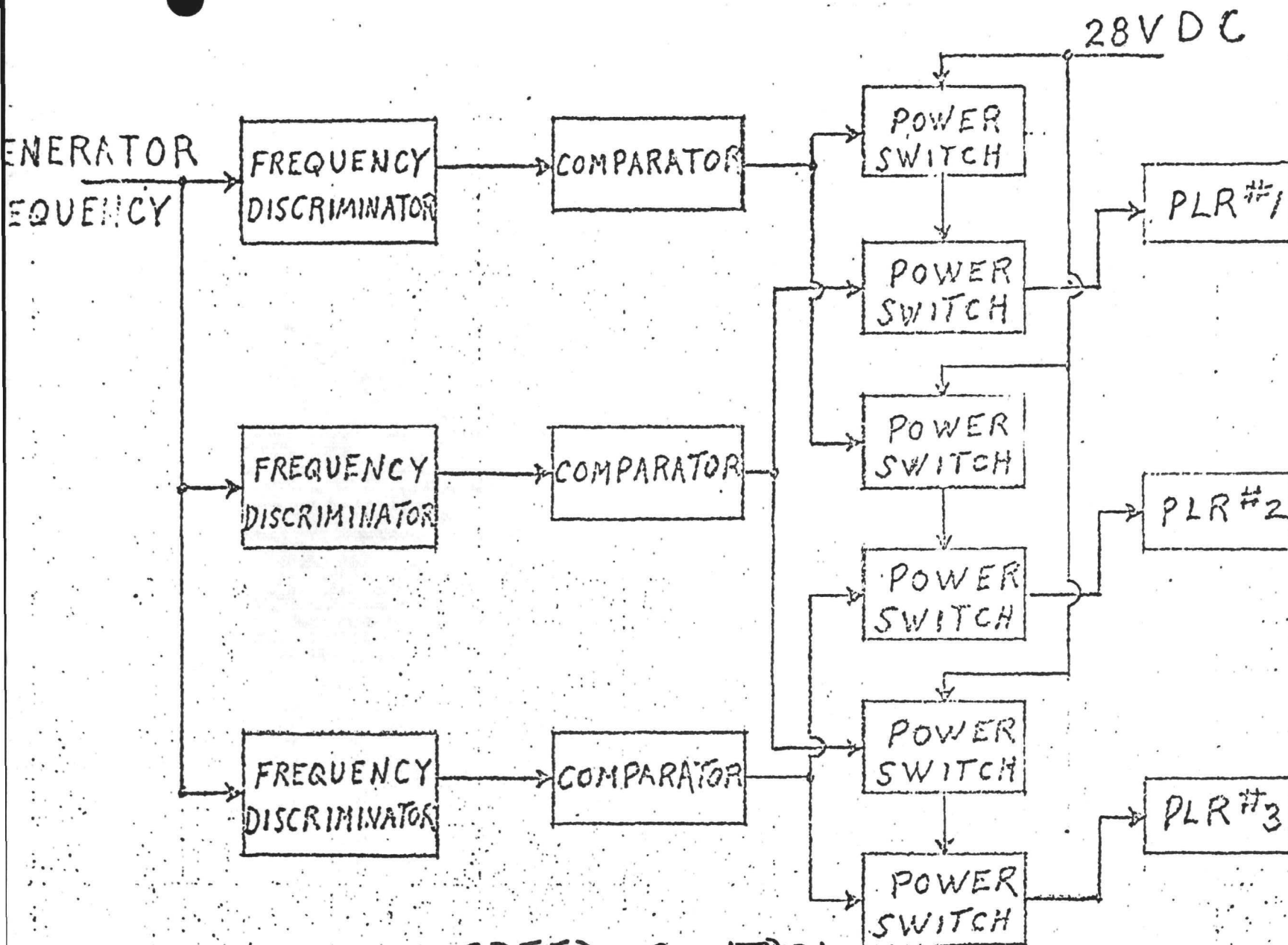
FIG. 2

SPEED CONTROL
2 PLRs CAPABLE OF FULL LOAD
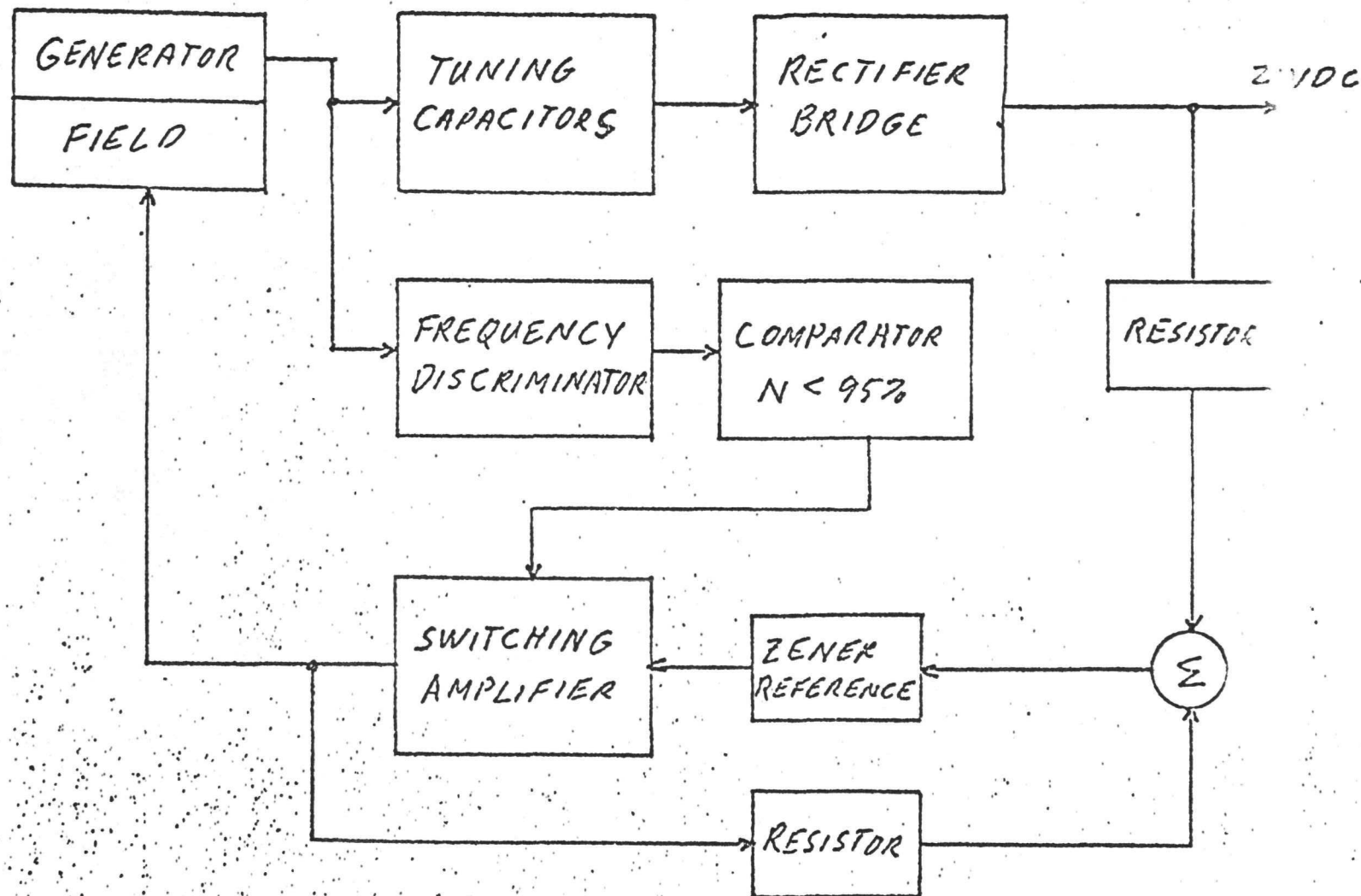FIG 3

SPEED CONTROL

2 PLR'S CAPABLE OF FULL LOAD

FIG. 4

GENERATOR

FIELD

TUNING CAPACITORS

RECTIFIER BRIDGE

$2\ VDC$

FREQUENCY DISCRIMINATOR

COMPARATOR $N < 95\%$

RESISTOR

SWITCHING AMPLIFIER

ZENER REFERENCE

Σ

RESISTOR

FIG 5   VOLTAGE REGULATOR   9/2/76
WITH
OVERLOAD PROTECTION
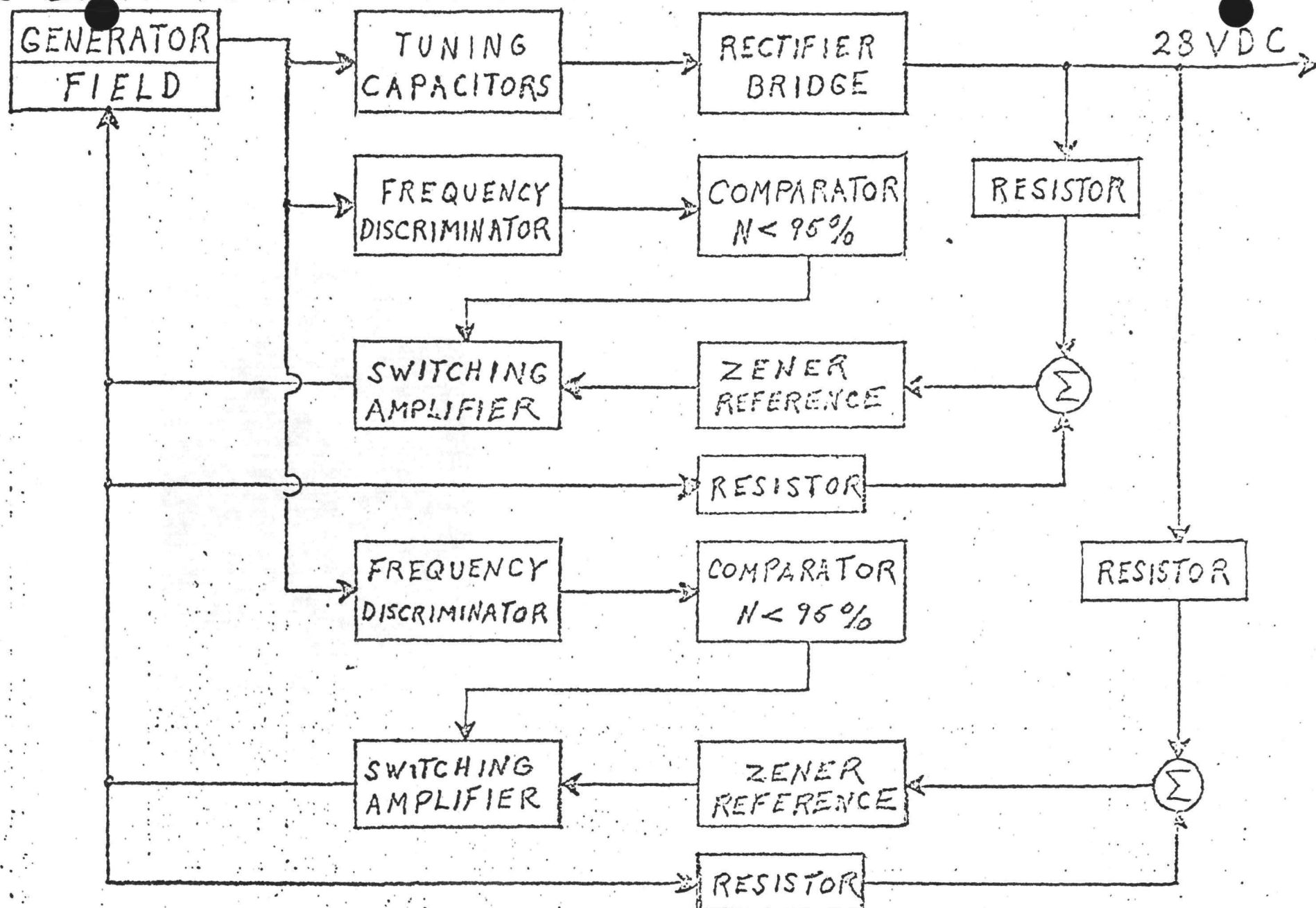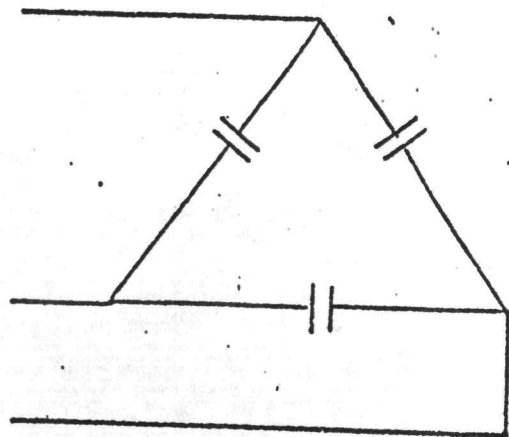
FIG 6   VOLTAGE REGULATOR
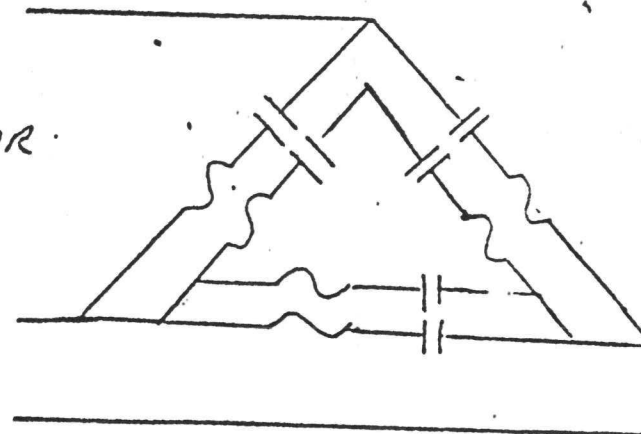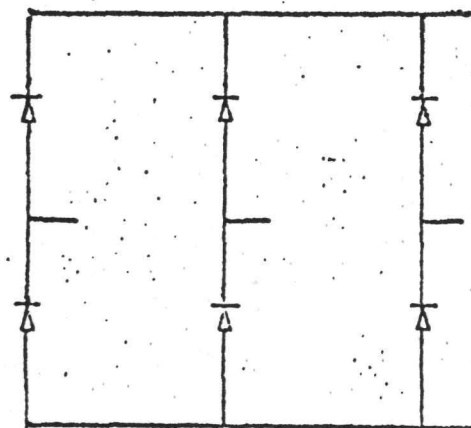
FIG. 7   VOLTAGE REGULATOR
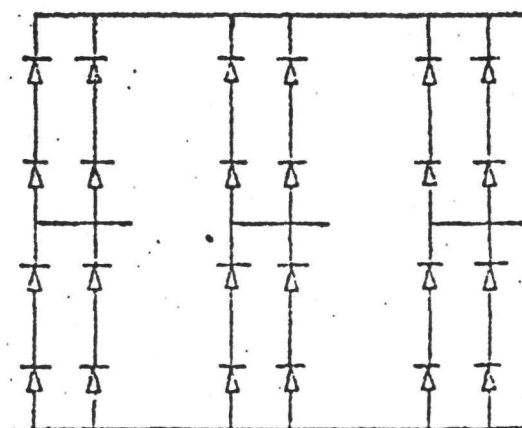
GENERATOR
LEADS

BASELINE

GENERATOR
LEADS

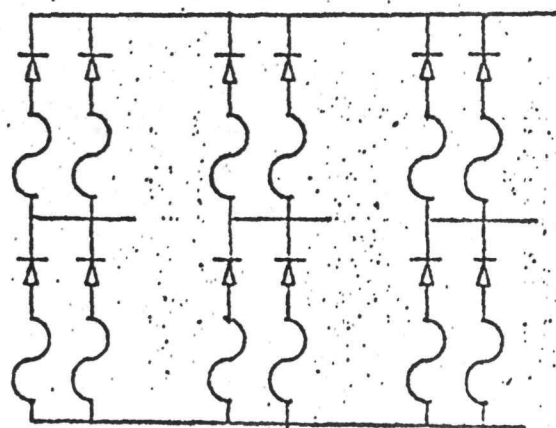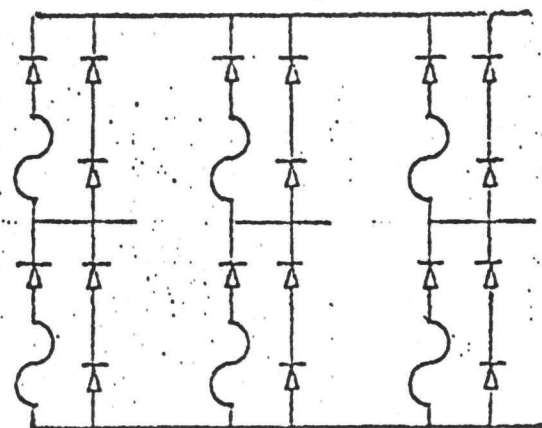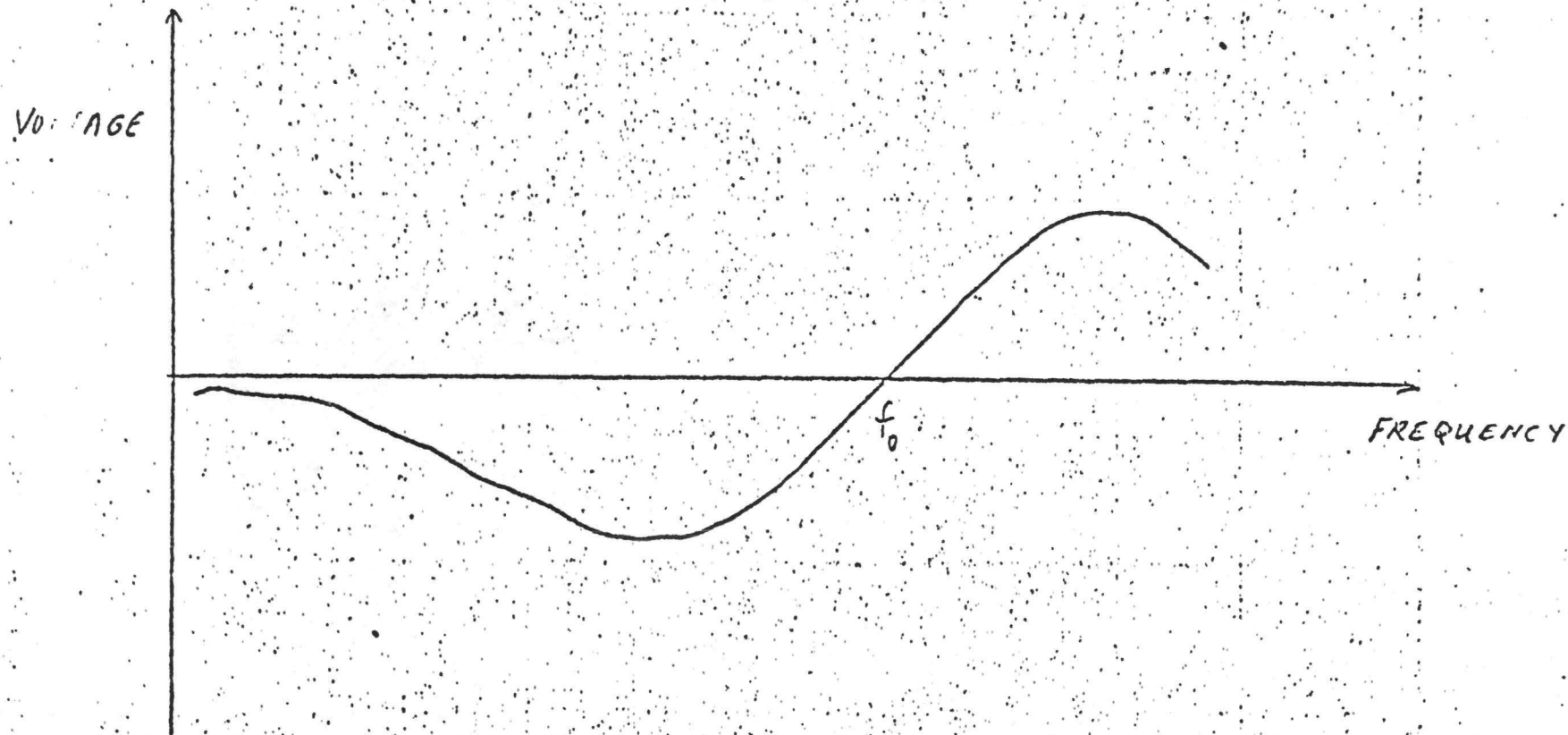FUSED REDUNANT

FIG. 8 · TUNING CAPACITORS

BASELINE

REDUNDANT

RECTIFIER

FUSED REDUNDANT FIG. 9 COMBINATION

VOLTAGE

FREQUENCY

f_0

FREQUENCY DISCRIMINATOR
TRANSFER CHARACTERISTICS
FIG 10

3.2.3   CIRCUIT SCHEMATIC

The present schematic of the controller is shown in Figure 11. Recent
testing of the controller with the KIPS alternator showed that several
adjustments to the circuit design, as presented in the PDR data package,
were necessary.

1.   A pulse-width-modulation (PWM) technique is now
     incorporated in the speed control. This helps to
     stabilize the output voltage.

2.   An LC filter is used for additional smoothing of the
     output voltage waveform.

3.   Inductors are used to speed the switching times of the
     PLR drive transistors.

4.   Both speed detectors are driven from a single transformer
     to save space.

FIGURE 11

KIPS ALTERNATOR CONTROLLER SCHEMATIC

REV. D   11-16-7_
D. FLYKMAN
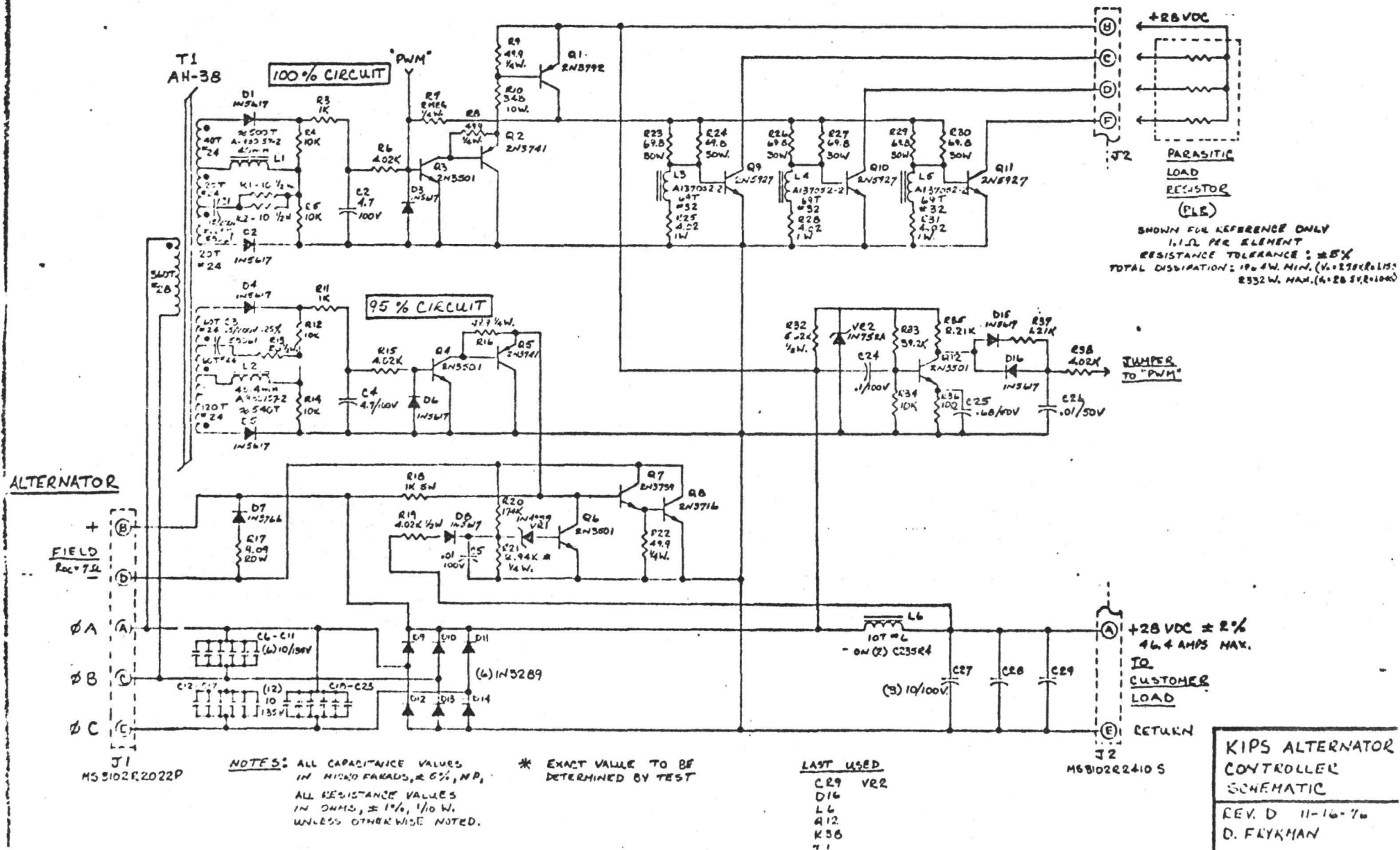
3.3   FLIGHT SYSTEM CONCEPTUAL DESIGN AND GROUND

         DEMONSTRATION SYSTEM SELECTION

A major guideline for the GDS controller design is its similarity
to the FSCD controller. The GDS electrical circuit will be identical
to the flight circuit in function, performance, and redundancy. One
desirable difference is the addition of two GDS test connectors that
provide access to circuit test points. These are necessary to measure
circuit operation, efficiency, and to allow optimization of the speed
control and voltage regulator under closed-loop conditions. These
test connectors and additional wiring increase the package size and
weight slightly.

The PLR for the GDS is likely to be a convective breadboard unit of
wirewound resistors rather than the radiative titanium unit which will
be used for the flight units.

The reliability levels of the GDS electrical components will be as
close as possible to the flight design levels. In each case, the
GDS component will have the same basic part number (such as 1N3766)
as the flight component, but its reliability level may be lower as
forced by long procurement lead times.

3.4   BREADBOARD TEST RESULTS

The KIPS breadboard controller, corresponding to the PDR circuit
design, has been tested with the KIPS alternator. The voltage
regulation, with no speed control, is very stable and produces a
nearly ideal output waveform. The field current is switched on the

3.4   BREADBOARD TEST RESULTS (Continued)

ripple that exists on the output voltage.

The speed control, as originally proposed, causes severe voltage
transients as the PLR switches on and off. This action also disturbs
the voltage regulation. To solve this problem a pulse-width-modulation
(PWM) approach was tried. It was found that PWM at a high frequency
(6700 Hz) can reduce the ripple voltage and the value observed was
4.5 volts (RMS). A filter will be added to attenuate the ripple to
about 2 volts (RMS). Final testing of the complete controller, as
shown by the Figure 11 schematic, has not been done.

The PLR used for this testing is a resistive load bank that dissipates
heat by convection.

The tuning capacitors have produced several desirable features. The
residual voltage, when the field current is zero, is increased due to
the capacitance. This is shown in Figure 12. The high voltage at the
95% speed point aids the speed detector operation when the speed is below
that point since the field is kept off. Also, the voltage (V_{L-L}) wave-
form is smoothed significantly. A power factor of .87 has been measured
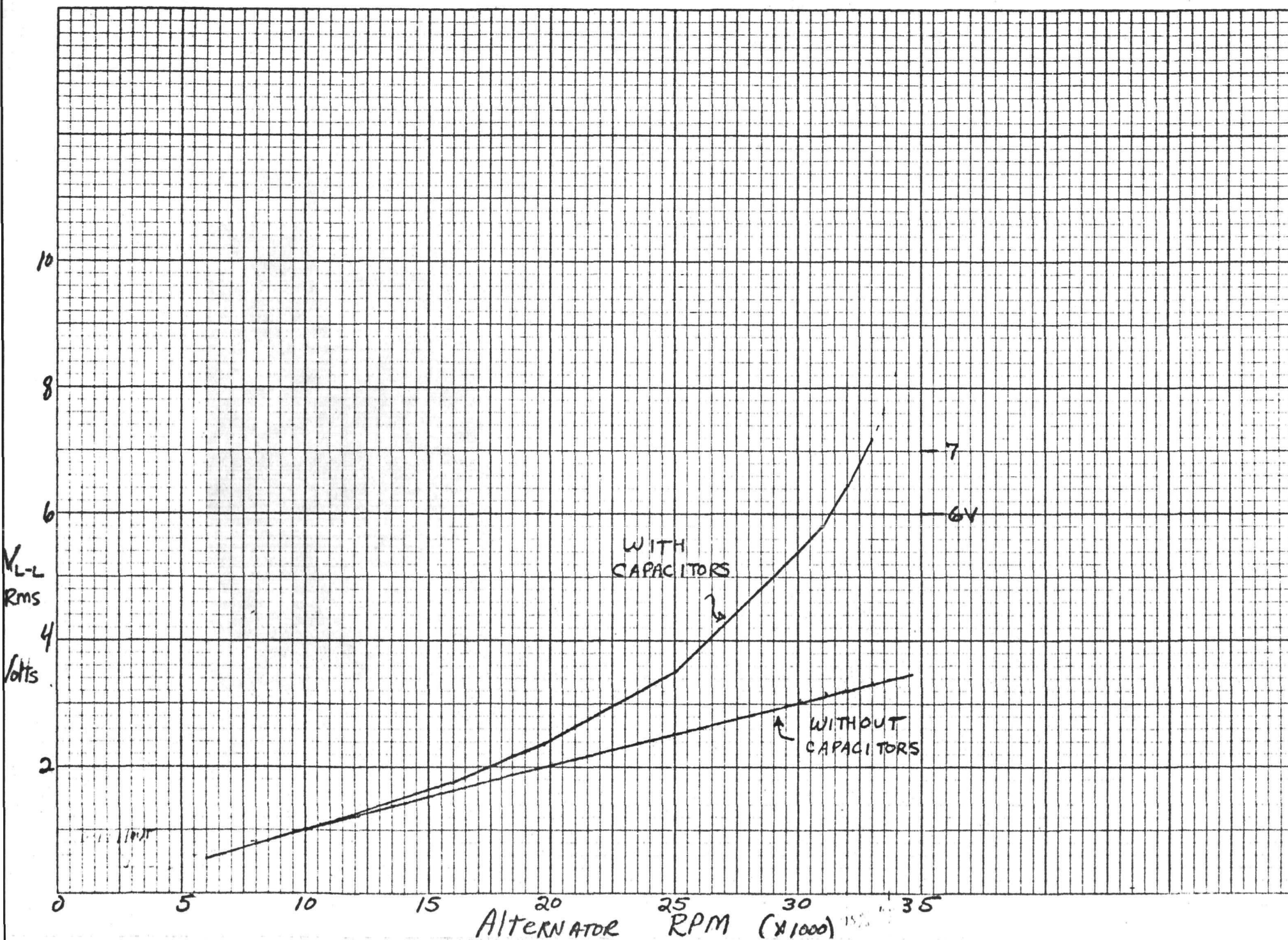with an alternator load of 1200 watts.

3.5   FURTHER STUDY & TEST PLANS

A worst case analysis will be made for the electrical circuits.
Proper operation will be assured for the expected range of component
parameters after a five year mission.

Figure 12. KIPS Alternator Residual Voltage

3.6   MARGINS OF SAFETY AND RELIABILITY

All components that are used in the controller have stress levels
that are well below the rated stress. For example, the resistors
will never have over 50% of their rated power dissipation and
typically will be less than 25%. The capacitors are operated at
a third or less of their rated working voltage. The wire sizes are
selected for about 2000 Amperes/inch^2 (Max) which is a conservative
figure. The semiconductor devices are also operated well below their
rated stress levels.

Using the expected stress levels during system operation and MIL-HDBK-
217B as a guide for the failure rates of the components, the controller
reliability has been calculated to be .987 for the five year mission
life.

3.7   GENERATOR TERMINAL BLOCK

The generator terminal block provides for termination of both control
and output power leads. Glass and metal construction forms a hermetic
seal. Threaded fastners provide clamping force to make the electrical
connections.

External connections are isolated by separators which provide electrical
insulation.

3.7.1 MATERIALS AND PROCESSES, CONTROLLER

The controller for GDS will be fabricated with brazed aluminum.
Brazing will also attach the cooling tubing to the package.

The FSCD units' materials and processes will be selected after
GDS development testing.

3.8 CRITICAL CHARACTERISTICS

Nearly all of the electronic components have corresponding Military
Specifications that cover the necessary characteristics. Each device
is inspected by either the vendor or Sundstrand to meet or exceed these
detailed characteristics. In this way, the circuit elements are described
to a high degree. The parts list is given in Figure 13.

QTY	SUNDSTRAND PART NO.	VENDOR PART NO.		DESCRIPTION		MILITARY SPECIFICATIONS
4		JANTXV2N3501	1*	TRANSISTOR		MIL-S-19500/366A
1		JANTX 2N3739				/402
2		JANTXV2N3741				/441
1		JANTXV2N3716				/408B
1		JANTX 2N3792		↓		/379B
3	58060	JANTX 2N5927				↓ /440
9		JANTXV1N5617	2*	DIODE		MIL-S-19500/411
1		JANTX 1N3766		RECTIFIER		/297A
1		JANTXV1N4959		ZENER DIODE 11V 5%		/356B
6		JAN 1N3289		RECTIFIER		↓ /246D
1		JANTXV1N752A	*	ZENER DIODE 5.6V 5%		/127F
2	58061-22	THIIT-0031		CAPACITOR	.5MF 200V 1/4%	
2		M83421/01-3317S			4.7MF 100V 5%	MIL-C-83421
1		-2089S	*		.01 50V 5%	
1		-3089S			.01 100V 5%	
1		-2245S	*		.68 50V 5%	
1		-3173S	*		.1 100V 5%	
21		↓ -3353S		↓	10MF 100V 5%	↓
1	58058			TRANSFORMER		MIL-T-27
1	58059-1			INDUCTOR, PRECISION	39.95MH 1/2%	A930157-2 CORE
1	58059-2			↓	45.43MH 1/2%	↓
3		†		INDUCTOR	2.5MH	A137052-2 CORE
1		†		INDUCTOR, FILTER		

\# PWM Circuit: Preliminary Quantity
† To Be Determined

FIGURE 13

QTY	SUNDSTRAND PART NO.	VENDOR PART NO.	DESCRIPTION	MILITARY SPECIFICATIONS
1		RER65F3480R	RESISTOR 348 Ω 10W	MIL-R-39009
6		RER75F69R8R	69.8 Ω 30W	
1		RER70F9R09R	9.09 Ω 20W	
1		RER60F1001R	1K Ω 5W	
2		RNC70H10R0FS	RESISTOR 10 Ω 1/2W	MIL-R-55182
4		RNC55C1002FS	10K 1/10W	
2		RNC55C1001FS	1K 1/10W	
2		RNC55C4021FS	4.02K 1/10W	
4		RNC65C49R9FS	49.9 1/4W	
3		RWR81S4R02FS	4.02 1W	
1		RNC70C20R0FS	20 1/2W	
1		RNC70C4021FS	4.02K 1/2W	
1		RNC55C1743FS	174K 1/10W	
1		RNC65CXXXXFS	SELECT 2.94K 1/9W	
1		RNC65C2004FS	2.0 MEG 1/4W	
1		RNC70C5621FS *	5.62K 1/2W	
1		RNC55C3922FS *	39.2K 1/10W	
1		RNC55C1002FS *	10K 1/10W	
1		RNC55C1211FS *	1.21K 1/10W	
1		RNC55C1000FS *	100 1/10W	
1		RNC55C2211FS *	2.21K 1/10W	
1		RNC55C4021FS *	4.02K 1/10W	

ALL RESISTORS ARE ± 1%

QTY	SUNDSTRAND PART NO.	VENDOR PART NO.	DESCRIPTION	MILITARY SPECIFICATIONS
1		MS3102R2022P	CONNECTOR	
1		MS3102R2410S	CONNECTOR	

FIGURE 13 (CONT)

* PWM Circuit: Preliminary Quantity

4.0   FLUID CONTROLS

4.1   TURBINE FLOW/TEMP CONTROL VALVE

4.1.1   Function and Need for Flow Control Valve

The location of the valve in the system is illustrated in Fig.
4.1. Figure 4.2 presents data for the speed governed mode of
operation that illustrates the dependence of turbine inlet
temperature on turbine loop flow. A large change in turbine
inlet temperature results from a small change in turbine loop
flow. The three curves presented represent different levels
of heat flow from the isotope heat source at different times
in the life of the unit. It is evident from the figure that
some means of indexing flow during the life of the unit is re-
quired. If no means of flow biasing were available, the unit
would drop off in turbine inlet temperature as the isotopes
decayed. The drop off in turbine inlet temperature would re-
sult in the unit running wet and a premature shut down of the
unit.

The requirement that the unit run non-speed governed compounds
the turbine loop flow control problem. The data presented on
Fig. 4.3 is for the non-speed governed case and illustrates
the range of flows and throttling pressure drops the valve
must control over. Once again the information is presented at
three levels of isotope heat input. The curves presented are
for a turbine inlet temperature of 650°F. Around each of
these lines is a small band (not shown) representing the ac-
ceptable range of operating temperatures. The size of this
band can be estimated from Figure 4.2.Speed varies along each
of the curves as a function of the load applied. In order to
stay within the acceptable range of operating temperatures
under all these conditions some sort of closed loop temperature
controller is required.

Alternate mechanisms were considered, however, it must be re-
membered that only approximately five percent of the heat input
is into superheat. As a result turbine inlet temperature de-
creases rapidly as flow increases. Conversely a relatively
small decrease in flow causes a relatively large increase in
turbine inlet temperature. Thus, for a given level of heat
input, flow must be kept within a narrow band. Alternative
mechanisms considered to control flow could not meet the system
requirements and were therefore discarded.

Figure 4.3 defines the flow and pressure drop envelope that the
valve is required to stay within. The valve must be able to
control flow over the approximate range of .027 lbm/sec. to
.037 lbm/sec. and vary pressure drop over the range of 20 psi
to 130 psi. The range which the valve must hold temperature
within was independently defined as 637°F - 670°F. The tem-
perature range was chosen in order to preclude running wet at
the low temperature extreme and to minimize degradation of the
working fluid at the high temperature extreme.
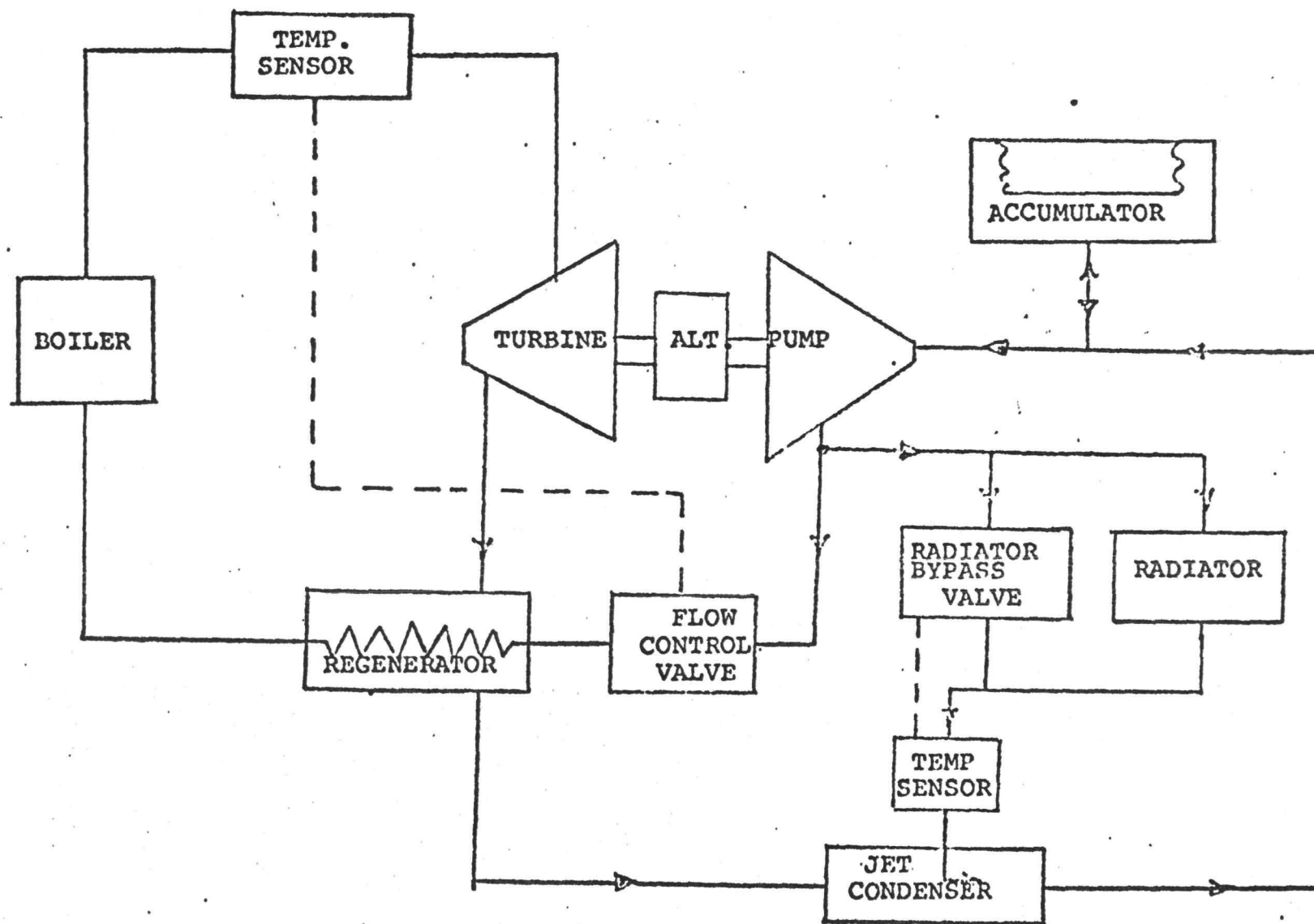
FIGURE 4.1

K.I.P.S. FLUID LOOP CONTROLS

FIG. 4.1

FIGURE 4.2

KIPS FLOW CONTROL REQUIREMENTS

Turbine Inlet Temp. vs Turbine Loop Flow
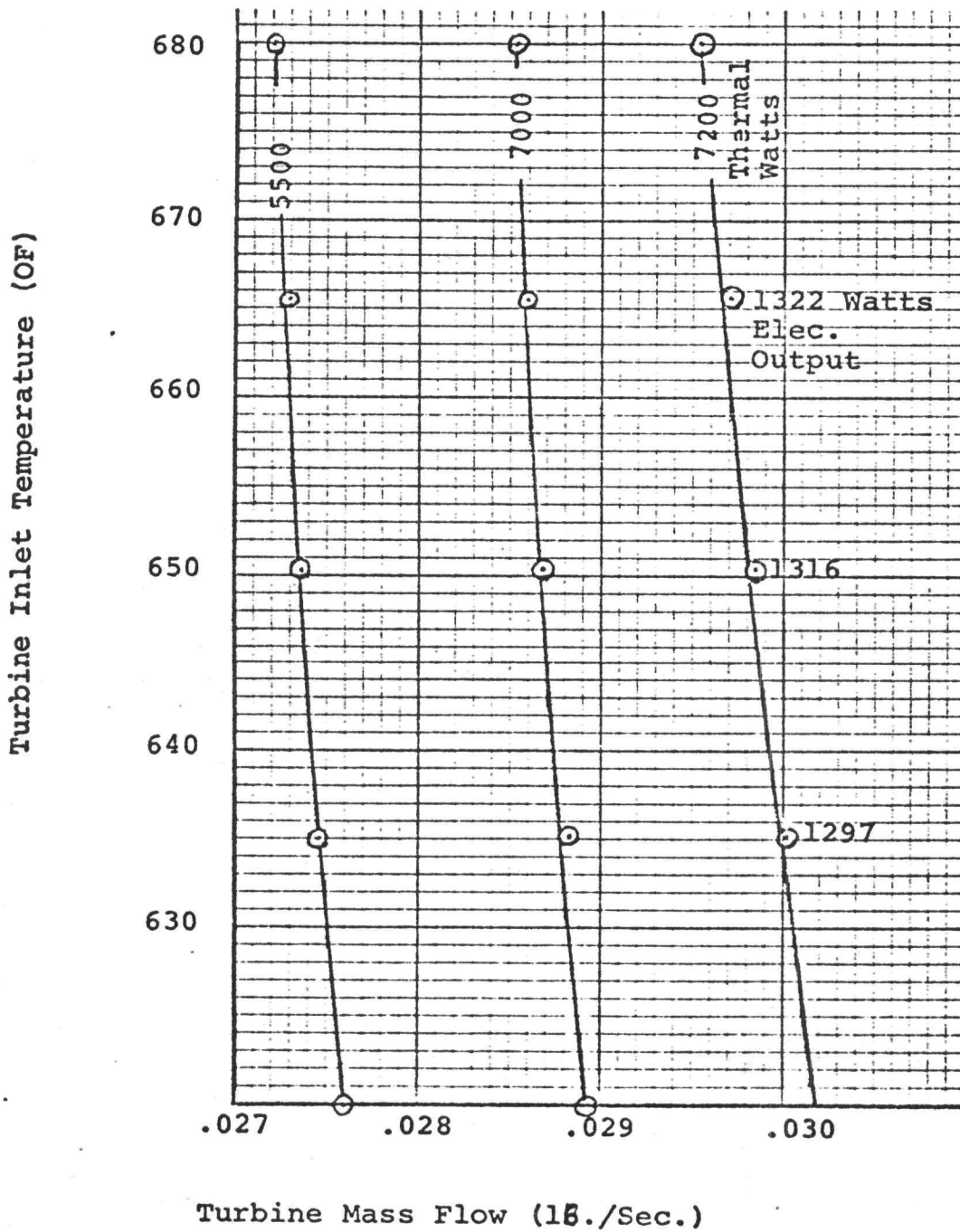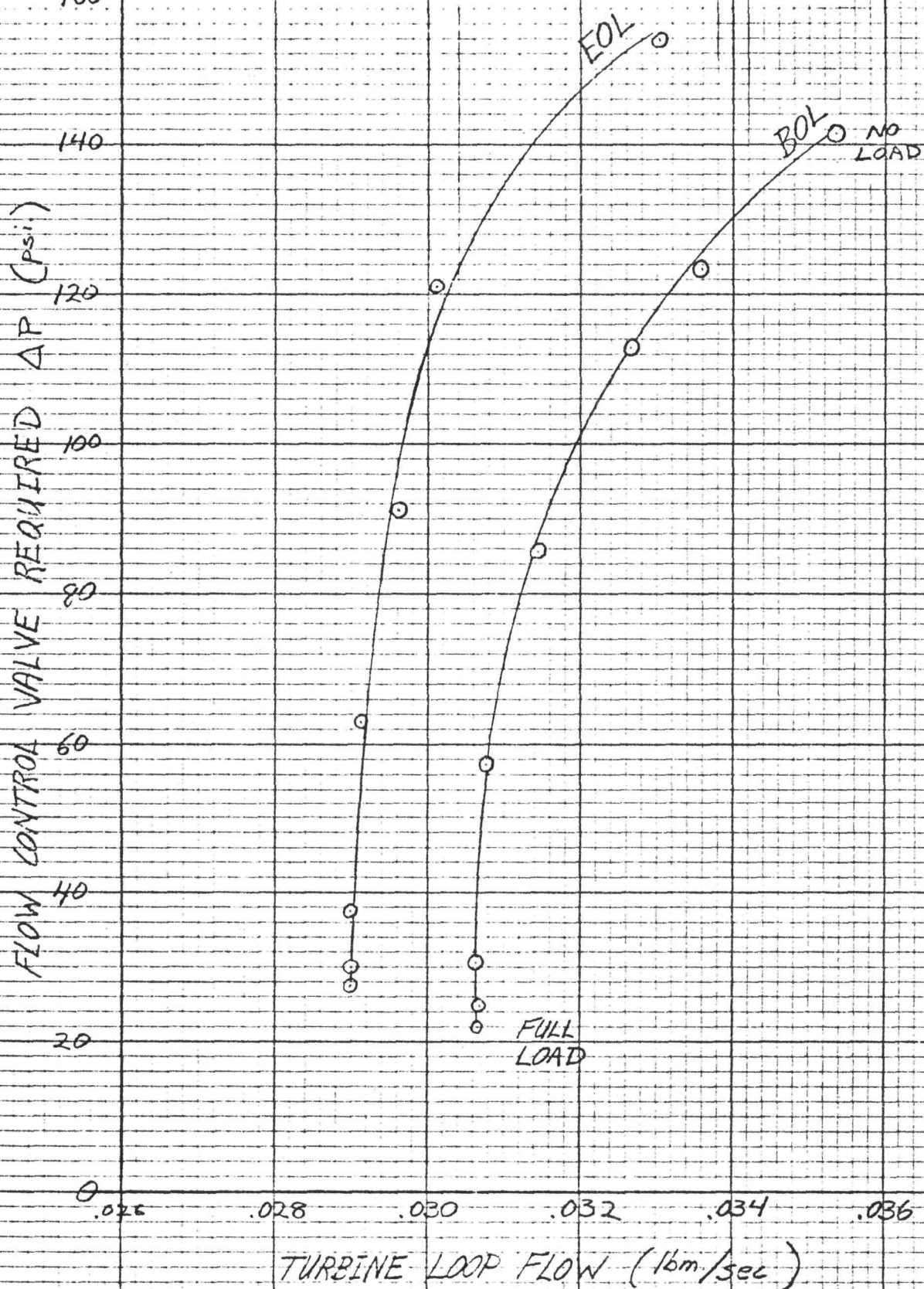at Constant Speed

Turbine Mass Flow (lb./Sec.)

FIGURE 4.3 (Rev A.)
KIPS TURBINE LOOP FLOW CONTROL
RANGE WITH VARIABLE SPEED

FLOW CONTROL VALVE REQUIRED ΔP (psi)

TURBINE LOOP FLOW (lbm/sec)

EOL

BOL   NO LOAD

FULL LOAD

4.1.2   Tradeoffs:

With the valve requirements established, several valve candi-
dates were defined and the tradeoff matrix procedure used to
select the valve configuration. The completed matrix is shown
on Fig. 4.4.

4.1.2.1   Valve Candidates

4.1.2.1.1 Configuration #1 Fluidic Diverter with Capillary Temperature
Sensor

The fluidic diverter valve with capillary temperature sensor
consists of a fluidic diverter flow control valve with tem-
perature biasing accomplished by means of a capillary tem-
perature sensor. Figs. 4.5 & 4.6 are schematic representa-
tions of the valve. The fluidic diverter section of the valve
consists of three stages of fluidic amplifiers. The temperature
sensor is a segment of capillary tube placed on the boiler out-
let tubing. The flow resistance of a capillary is a function
of viscosity which is, in turn, a function of temperature.
Thus, if turbine inlet temperature is constant, the flow re-
sistance of the capillary is constant and a change in turbine
inlet temperature causes a change in the flow resistance of the
capillary.

Assuming a constant turbine inlet temperature, an increase
in pump outlet pressure causes the flow through the flow
resistances in the control legs of the first stage amplifier
to change. Because of the differing resistance characteristic
of the flow resistances, the result of the increase in inlet
pressure is the deflection of the output stream toward the
output leg of the amplifier which is the "downward" leg on
the schematics. This deflection in turn causes the output
stream of the second stage amplifier to be deflected upward
and, finally, the output stream of the third stage to be
deflected downward. The downward deflection of the third
stage results in a pressure recovery change in the output
leg feeding the turbine loop that effectively cancels the
increase in pump outlet pressure.

Temperature biasing is accomplished at the capillary tempera-
ture sensor. As temperature increases the flow resistance
decreases causing the output stream of the first stage
amplifier to be deflected upward. The output stream of the
third stage amplifier is deflected upward and the pressure
recovery and flow into the turbine loop increased.

4.1.2.1.2   Configuration #2 Bimetallic Washer Valve

The bimetallic washer valve consists of a single metering
orifice with flow area proportional to temperature. Figure 4.7
is a schematic representation of the valve. The metering
orifice is a spool valve. The temperature sensor consists
of a stack of bimetallic washers placed in the boiler outlet
vapor flow stream. The bimetallic washers expand and contract
with temperature thereby, stroking the spool valve and meter-
ing flow to the boiler.

SELECTION MATRIX

KIPS FLOW CONTROL VALVE

CHARACTERISTIC	Merit Factor	Fluidic Capillary		Bimetallic Washer		Single Stage Thermal Bulb		Dual Funct. Thermal Bulb		*RECUNDANT* Dual Funct. Thermal Bulb	
		Value Rating	Factor Rating								
SAFETY + RELIABILITY	10	8	80	4	40	10	100	8	80	10	100
PERFORMANCE	9	6	54	6	36	8	72	10	90	10	90
RISK	8	6	48		.	5	40	10	80	10	80
WEIGHT	7	9	63			10	70	9	63	8	56
PRODUCIBILITY	6	5	30			10	60	9	54	10	60
DESIGN/OPERATIONAL FLEXIBILITY	5	3	15			10	25	10	50	10	50
TOTAL SCORE			290				367		417		436

FIGURE 4.4

R_1 = Over Pressure Bias Resistor

P_S = Supply Pressure

P_S

P_S

P_S

P_S

Flow To Turbine Loop

Bypass Flow

Vent Collector

R_2

P_S

P_S

Temperature Sensor Variable Resistance

To Pump Inlet

Check Valve

Hydraulic Resistance
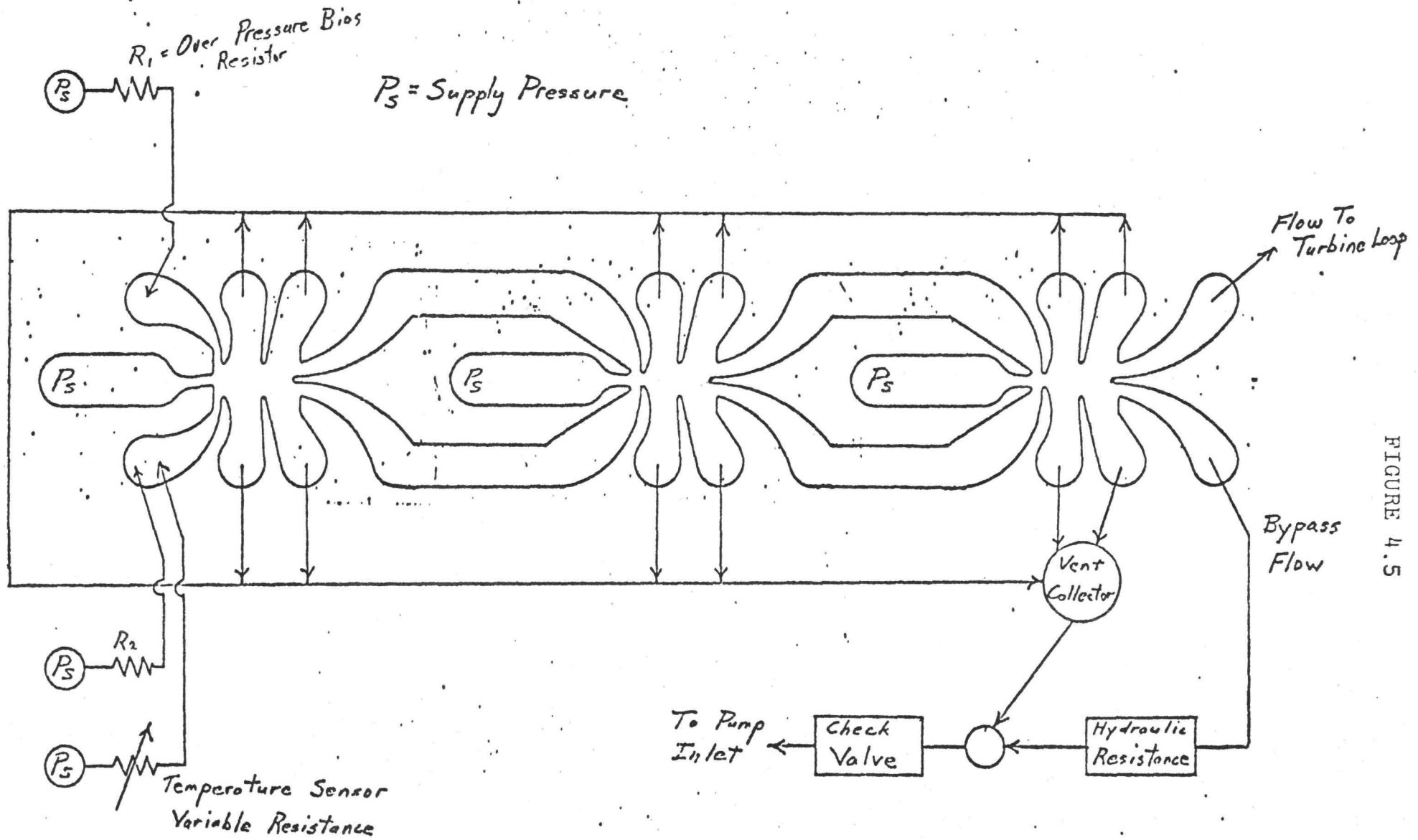
FIGURE 4.5

TYPICAL FLUIDIC LAMINATION

OVER PRESSURE
BIAS RESISTOR

3-STAGE DIVERTOR

PUMP

TURBINE

REGEN.   BOILER

RETURN
Hydraulic
Resistance

VENT   VENT

TEMPERATURE
SENSOR

JET COND.
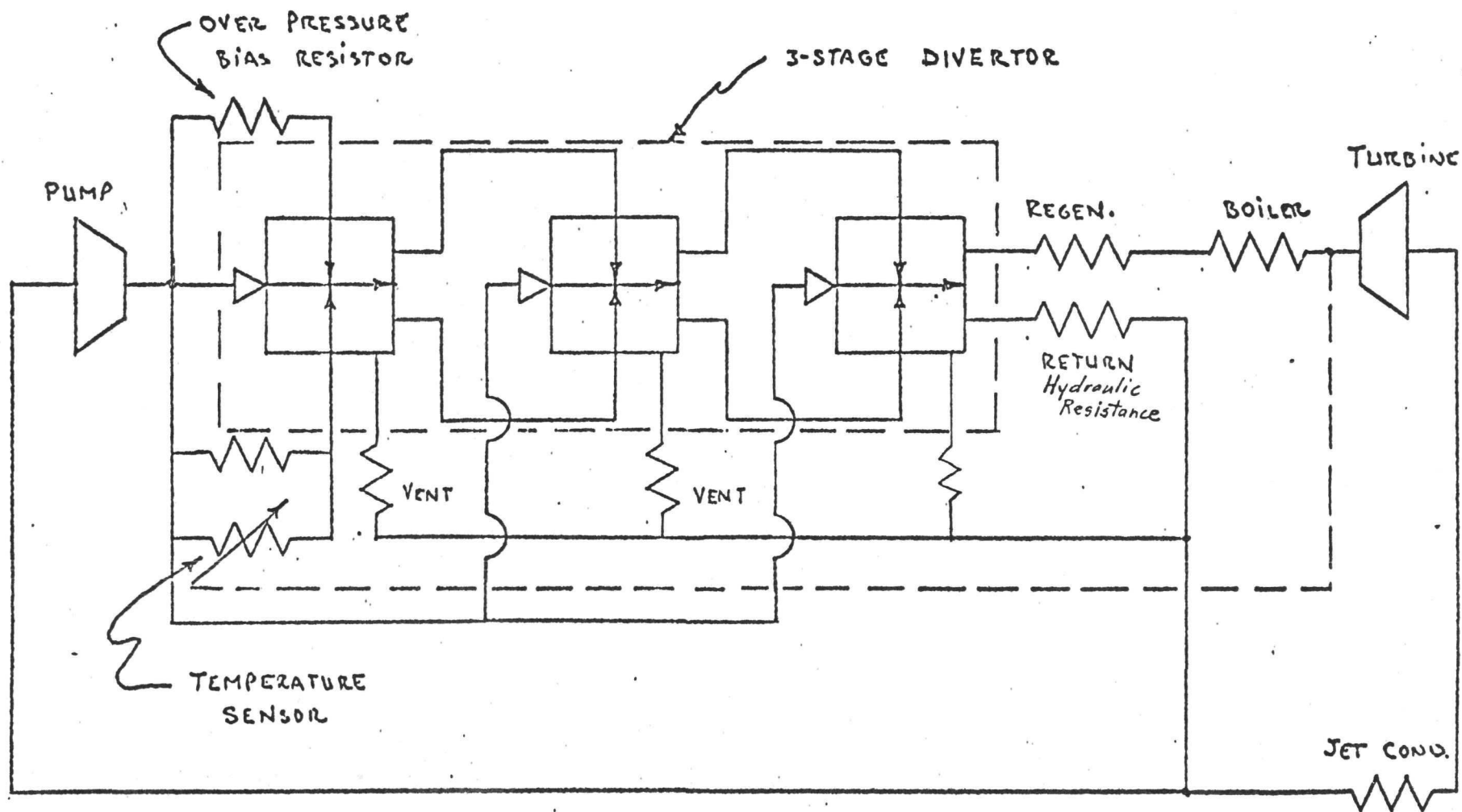
FIGURE 4.6

FLUIDIC TURBINE TEMPERATURE CONTROL
SCHEMATIC

To Turbine

Boiler

Disc Pack

From Regenerator
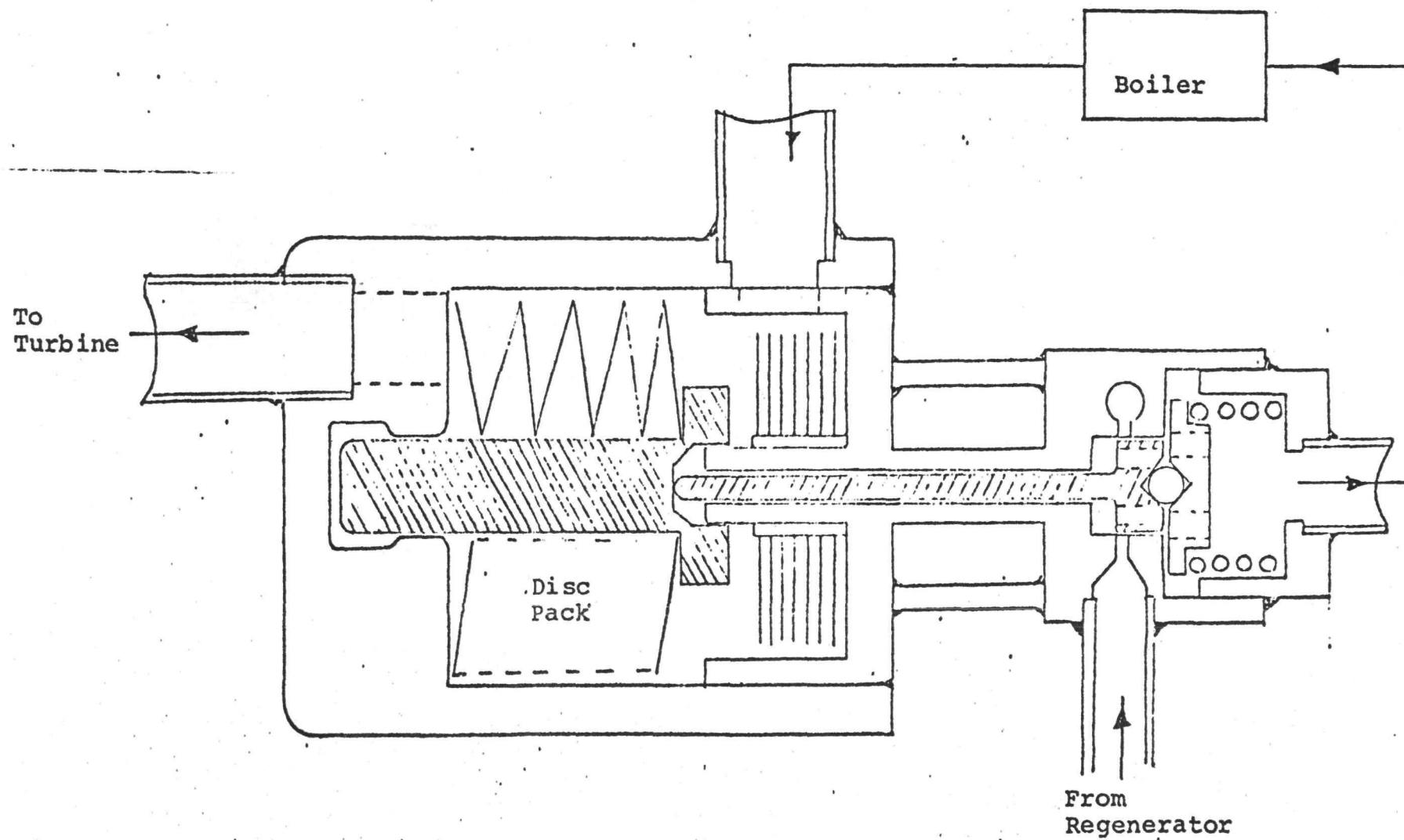
KIPS BIMETALLIC FLOW CONTROL VALVE

FIGURE 4.7

Because of the metering location in the boiler inlet and the temperature sensing location in the boiler outlet a leak path around the boiler is introduced. It was intended that this small leakage flow would be vaporized by heat transferred from the boiler outlet flow to the finned seal area of the valve.

Since the pressure drop across the metering area is not controlled, the spool valve must be capable of an area change sufficient to yield the throttling pressure drops required during non speed governed operation. The resulting area change requirement is, therefore, the larger than would be required if the pressure drop across the metering area were controlled. Since the temperature range is fixed, the change in area per degree must be larger than would be necessary if the pressure drop were controlled. As a result the flow change resulting from a given temperature change is relatively large.

.1.2.1.3   Configuration #3 Single Stage Mechanical Valve

The single stage mechanical valve is identical in concept to the bimetallic washer valve. Figure 4.8 is a schematic representation of the valve. The valve consists of a single metering location with area proportional to temperature. The metering orifice is a flapper valve; the temperature sensor is a liquid filled bulb with a welded bellows actuator assembly. The temperature bulb is a helical section of capillary tube brazed in place around the boiler out tube. As the boiler out temperature changes the liquid in the temperature bulb expands and contracts. The changes in volume cause the bellows actuator to change its length and, in doing so, change the metering area.

The single stage mechanical valve improved on some of the shortcomings of the bimetallic washer valve. The leak path around the boiler was eliminated; as was the sliding fit at the spool valve stem.

.1.2.1.4   Configuration #4 Two Stage Mechanical Valve

The two stage mechanical valve adds a throttling pressure drop regulator to the single stage mechanical valve. Figure 4.9 is a schematic representation of the two stage valve. The pressure drop regulator senses pressure drop across the temperature controlled flapper valve area. The regulator throttles flow as a function of the sensed pressure drop. At a constant turbine inlet temperature the flow area at the temperature controlled flapper valve is constant. An increase in flow across the flapper valve area would increase the pressure drop across this area. The pressure drop regulator would respond to this increase in pressure drop by further throttling flow through the valve by decreasing flow area at another metering location. It is evident that the pressure drop regulator must exhibit some "droop" (i.e., some increase in pressure drop across the temperature controlled flapper valve is necessary before the pressure regulator decreases its flow area). The valve characteristics have been chosen in such a manner that the droop characteristic will have no objectionable impact.

FIGURE 4.8

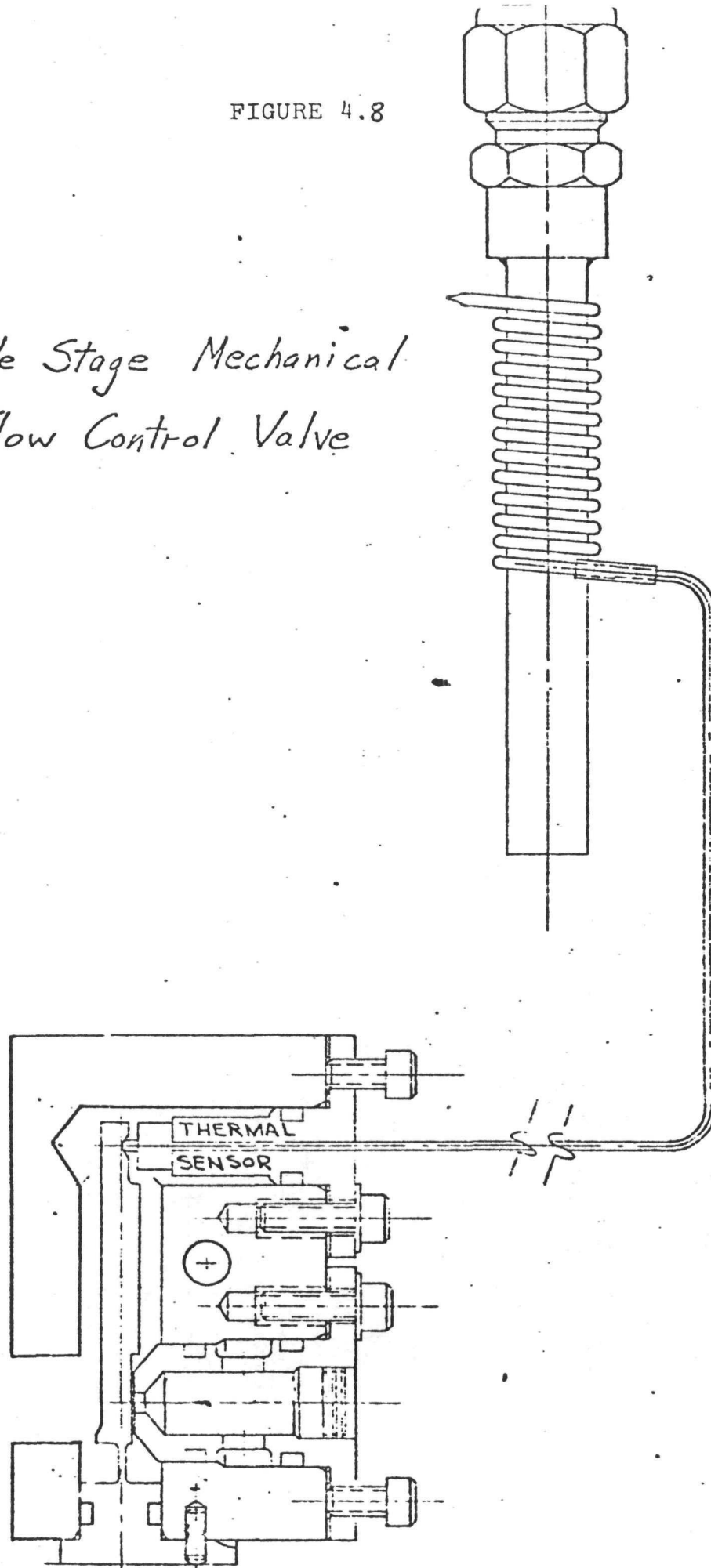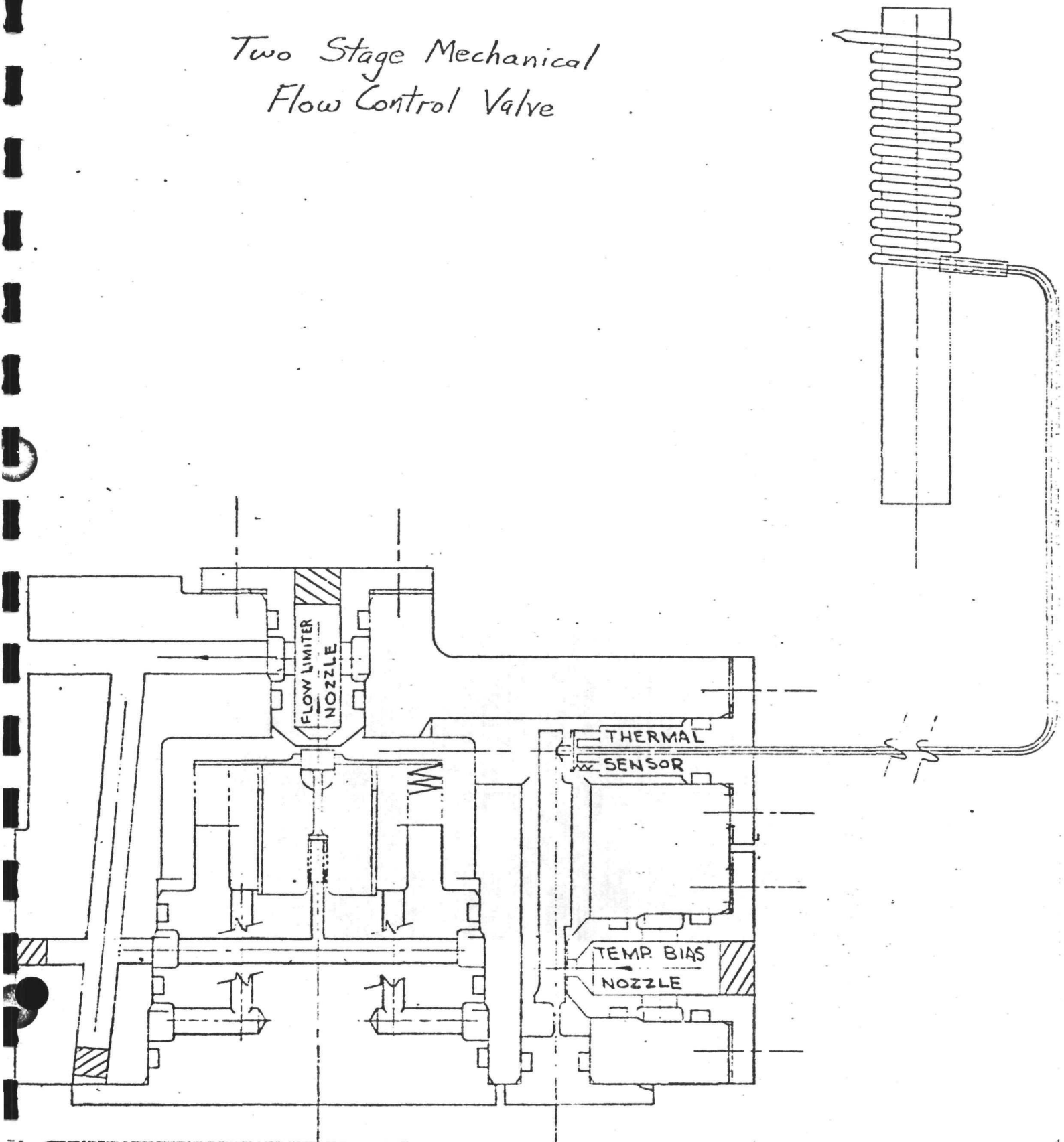Single Stage Mechanical
Flow Control Valve

THERMAL
SENSOR

FIGURE 4.9

Two Stage Mechanical
Flow Control Valve

With a controlled pressure drop, flow through the valve is
essentially dependent on only the flow area of the temperature
controlled flapper valve. This stage of the valve functions
as described for Configuration #3.

Since the pressure drop across the temperature controlled
metering area is controlled the temperature controlled
metering area need only vary as required to change flow over
the range as defined by Figure 4.3. Since the acceptable
temperature range is fixed, the two stage valve requires a
smaller area change at the temperature controlled metering
area, than does the single stage valve.

The single stage and two stage valves also react differentialy
during speed transients. The two stage valve will tend to
hold flow constant until a flow change is called for by the
temperature sensor. The single stage valve will allow flow
to increase or decrease with speed until a temperature change
is sensed. For the recovery from overspeed transient, for
instance, the two stage valve will hold flow constant at its
high value until the unit cools and turbine inlet temperature
calls for a flow decrease. The single stage valve, on the
other hand, would immediately decrease flow as speed decreased.
Since the regenerator stores heat and since the heat released
by the isotope does not change, turbine inlet temperature would
increase; the valve, responding to increased temperature, would
then increase flow to cool the unit. Thus the temperature
transients experienced with the two valves would be quite
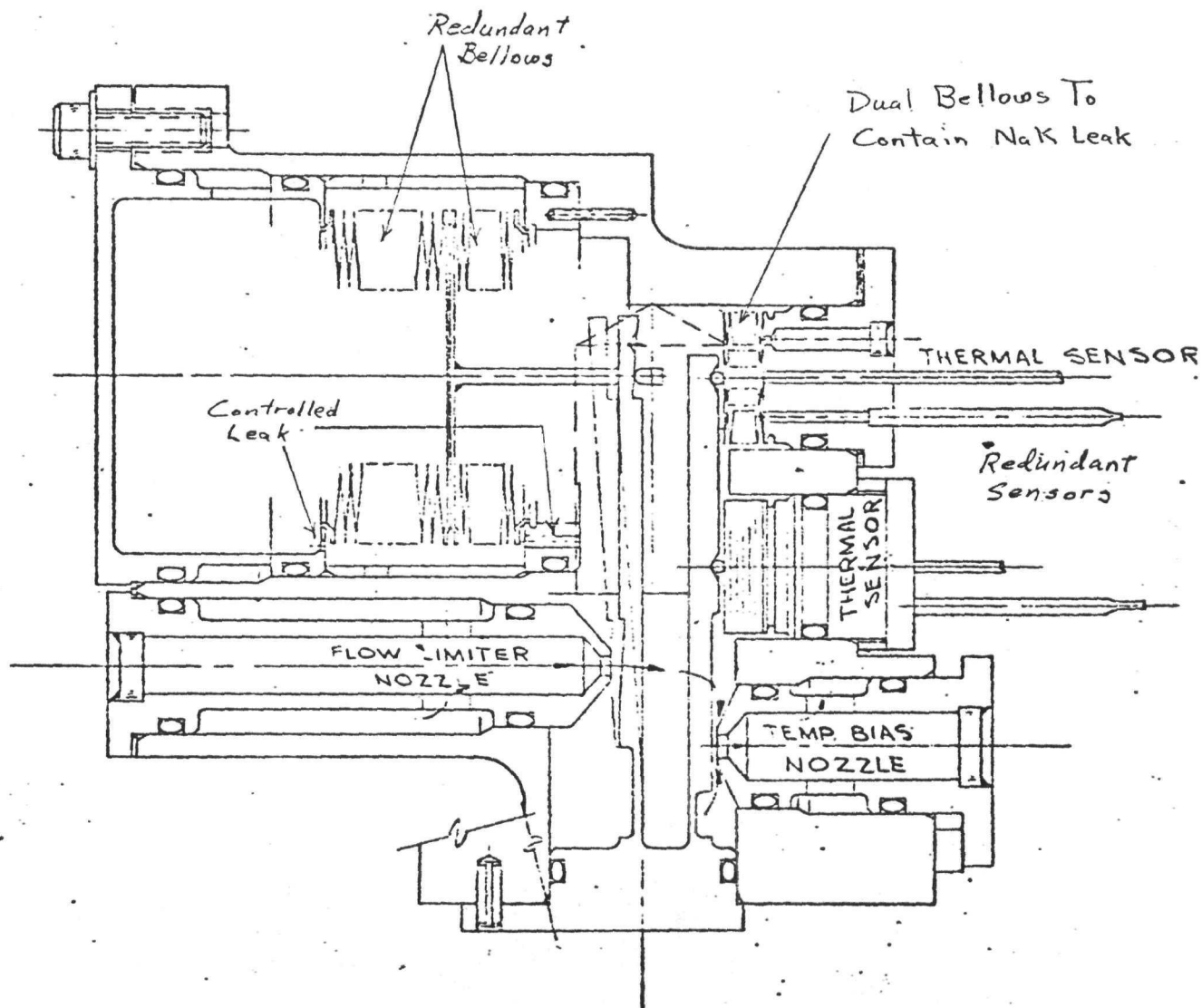different.

The two stage valve represents an improvement over the single
stage valve. The flow change per degree temperature change has
been reduced and a more desirable transient behavior attained.

4.1.2.1.5   Configuration #5 Two Stage Mechanical Valve with Redundancy

Functionally the two stage mechanical valve with redundancy
is identical to the two stage mechanical valve described
above. Redundant components have been added to enhance system
reliability. The pressure regulator bellows have been replaced
by a flapper valve assembly utilizing redundant bellows. Figure
4.10 is a schematic representation of the valve. Redundant
temperature sensors with containment bellows are utilized. The
purpose of the containment bellows is to ensure that if an
actuator bellows were to leak, the temperature sensing fluid
would not be released into the system.

The two stage mechanical valve with redundancy represents an
improvement of the two stage mechanical valve. The addition
of redundant components eliminates some potential single
point failure modes.

Redundant Bellows

Dual Bellows To Contain NaK Leak

THERMAL SENSOR

Controlled Leak

Redundant Sensors

THERMAL SENSOR

FLOW LIMITER NOZZLE

TEMP. BIAS NOZZLE

KIPS GDS SYSTEM FLOW CONTROL VALVE
FIGURE 4.10

4.1.2.2   Reliability

.1.2.2.1   Configuration # 1 Fluidic Diverter with Capillary Temperature
           Sensor

1. The unit has no moving parts.

2. A potential problem of fouling in capillary temperature
   sensor, (I.D. = .010), exists.

3. Potential for capillary fouling to become positive feed-
   back process, i.e., fouling restricts capillary reducing
   flow through capillary, valve response is to reduce flow
   thereby raising turbine inlet temperature and hastening
   fouling in capillary.

4. Localized cooler area in boiler exit tube may collect
   contaminants and restrict flow.

5. Potential susceptability to launch environment an unknown.

6. Limited reliability data for similar components.

7. Bypass line required a check valve.

8. Extra plumbing required for bypass and vent lines.

.1.2.2.2   Configuration #2 Bimetallic Washer Valve

1. Sliding fits in several locations result in possible fail-
   ure modes due to sticking. One sliding fit location is
   also location of high probability of contamination accumu-
   lation. Sliding fits also potentially generate wear parti-
   cles.

2. Low force margins are available to overcome sticking.

3. Relatively large number of parts, complex.

4. Leakage path around boiler.

5. Force balance design is susceptible to launch environment
   caused problems. Mass of parts makes it difficult to de-
   sign to have high resonant frequencies. Spool is loose
   piece located by pressure force.

.1.2.2.3   Configuration #3 Single Stage Mechanical Valve

1. Bellows uses small portion of available stroke. Margin of
   safety of two used in working stress diagram.

2. Bellows can be burned in to eliminate "infant mortality"
   failures.

3. Leaf spring working stresses small in comparison to allow-
   able.

4. Moving parts limited to flexures.

5. Launch environment susceptibility analysis is straight-
   forward.

.1.2.2.4   Configuration #4 Two Stage Mechanical Valve

1. Bellows uses small portion of available stroke. Margin
   of safety of two used in working stress diagrams.

2. Bellows can be burned in to eliminate infant mortality
   problems.

A 3. Leaf spring working stresses small in comparison to allowable.

4. Moving parts limited to flexures.

5. Launch environment susceptibility is standard analysis problem.

6. Mass of parts susceptible to launch environment problems can be made small to raise resonant frequencies.

7. This configuration is same as Configuration #3 with a pressure regulator stage added.

4.1.2.2.5   Configuration #5 Two Stage Mechanical Valve with Redundancy

1. Leaf spring working stresses small in comparison to allowable.

2. Pressure sensor bellows stresses small in comparison to allowables.

3. One pressure sensor bellows can fail without harming system operation.

4. Temperature sensors incorporate containment bellows to prevent leakage of temperature sensor fluid into system in event of an actuator bellows leak.

5. One temperature sensor can fail without harming system operation.

6. Moving parts limited to flexures.

7. Bellows can be "burned in" to eliminate "infant mortality" failures.

8. This configuration is functionally identical to configuration #4.

4.1.2.3   Performance

4.1.2.3.1   Configuration #1 Fluidic Diverter with Capillary Temperature Sensor

1. High ΔP is required for valve operation.

2. Bypass design increases required pump power by 10-20%.

3. Present amplifier gain is not sufficient to compensate for low temperature sensor gain resulting in unacceptable temperature control. Increasing amplifier gain may result in noise problem.

4. Heat shunt at capillary sensor.

5. Flow controller design concept gives lower temperature loop gain.

6. Contamination susceptible at capillary.

7. Launch environment susceptibility an unknown.

8. Poor analytical predictability.

9. Poor data base for similar components.

10. Overspeed recovery characteristic an unknown.

4.1.2.3.2   Configuration #2 Bimetallic Washer Valve

1. Very susceptible to contamination.

2. Very susceptible to launch environment.

3. Tolerant to small leakage.

4.1.2.3.3   Configuration #3 Single Stage Mechanical Valve

1. Single stage valve results in higher gain.

2. Can operate at low ΔP.

3. Launch environment performance amenable to analysis.

4. Small heat loss through liquid filled bulb.

5. Good analytical tools available.

6. Can supply required flow during overspeed recovery.

7. Transient begins in undesirable direction during over-
   speed and overspeed recovery.

4.1.2.3.4   Configuration #4 Two Stage Mechanical Valve

1. Flow controller design concept gives lower temperature
   loop gain.

2. Can operate at low ΔP.

3. Small heat loss through temperature sensor.

4. Good analytical tools available.

5. Can supply required flow during overspeed recovery.

6. Transients begin in desirable direction during overspeed
   and overspeed recovery.

4.1.2.3.5   Configuration #5 Two Stage Mechanical Valve with Redundancy

1. The performance of this valve is identical to Configu-
   ration #4.

4.1.2.4   Risk

4.1.2.4.1   Configuration #1 Fluidic Diverter with Capillary Temperature Sensor

1. Relatively new technology.

2. Cannot predict performance analytically.

3. Significant additional development cost.

4. Problems:

    a. low gain temperature sensor.

    b. low amplifier gain.

    c. Sundstrand has no production experience with this type of product.

    d. Not calibratable.

5. Problems unresolved by GDS.

    a. Not calibratable.

6. Problems requiring resolution by rig/lab testing.

    a. Temperature sensor development.

    b. Amplifier bias resistor design.

    c. Amplifier internal pressure drops.

7. Schedule effects - any of above problems.

8. Limited data available.

4.1.2.4.2   Configuration #2 Bimetallic Washer

This valve can be dropped from consideration at this point due to low ratings in high weighted categories.

4.1.2.4.3   Configuration #3 Single Stage Mechanical Valve

1. Analytical models are generally available but require knowledge of characteristics that is only available through test. Rig test will supply this information.

2. If required gain is sufficient to cause stability problem, no fix is readily available. Consequence of stability problem is extremely severe.

4.1.2.4.4   Configuration #4 Two Stage Mechanical Valve

1. Analytical models are readily available but require knowledge of characteristics which is only available through test. Rig test will supply this information.

2. Poppet instability could cause problem at pressure sensor. Valve should be overcompensated however.

.1.2.4.5 Configuration #5 Two Stage Mechanical Valve with Redundancy

1. The risk of this valve is the same as Configuration #4.

4.1.2.5 Producibility

.1.2.5.1 Configuration #1 Fluidic with Capillary Temperature Sensor

1. New product to Sundstrand but vendors with experience are available.

2. Complex shapes lead to inspection difficulty.

.1.2.5.2 Configuration #2 Bimetallic Washer Valve.

4.1.2.5.3 Configuration #3 Single Stage Mechanical Valve.

1. NaK filled bulb requires fill technique that prevents oxidation of the NaK. This is standard with NaK.

.1.2.5.4 Configuration #4 Two Stage Mechanical Valve

1. NaK filled bulb requires fill technique that prevents oxidation of the NaK. This is standard with NaK.

2. Pressure motor bellows assembly is difficult to inspect, except by leak check, at the assembly level.

4.1.2.5.5 Configuration #5 Two Stage Mechanical Valve with Redundancy

1. NaK filled bulb requires fill technique that prevents oxidation of the NaK. This is standard with NaK.

4.1.2.6 Design and Operational Flexibility

.1.2.6.1 Configuration #1 Fluidic with Capillary Temperature Sensor

1. The unit is not calibratable.

.1.2.6.2 Configuration #2 Bimetallic Washer Valve.

.1.2.6.3 Configuration #3 Single Stage Mechanical Valve.

1. The unit is calibratable.

.1.2.6.4 Configuration #4 Two Stage Mechanical Valve.

1. The unit is calibratable.

.1.2.6.5 Configuration #5 Two Stage Mechanical Valve with Redundancy.

1. The unit is calibratable.

4.1.3   Selected Design, Analysis

4.1.3.1   General

The general design procedure is to select three extreme
design points and force the valve envelope to pass through
these points at the extreme allowable temperature conditions.
The points selected are shown on Figure 4.11. It was decided
that the valve should operate down to a throttling pressure
drop of 10 psi and to split this total into nominal pressure
drops of 5 psi across each metering area.

4.1.3.2   Temperature Bias Function

Select Point 1 from Figure 4.11 and apply the following
flow formula.

$$(1) \qquad Q = C_D A_1 \sqrt{\frac{2\Delta P}{\rho}}$$

Assume the recommended handbook value for the valve configu-
ration for C_D. Q, ΔP, and ρ are known quantities. Therefore,
A_1 can be computed. It is known that the valve must decrease
flow with decreasing temperature and therefore A_1 is the area
required at 670°F. Repeating the procedure at point 2
yields A_2, the area required at 637°F.

The metering area for the flapper valve configuration is the
curtain area extending above the flapper nozzle edges to the
flapper. The area is given by:

$$A = \pi \, dh \qquad \text{where:} \qquad \begin{aligned} A &= \text{area} \\ d &= \text{nozzle diameter} \\ h &= \text{gap} \end{aligned}$$

A rule of the thumb for flapper design is to make the null
gap a maximum of 1/16 the nozzle diameter. Since the maxi-
mum area is known the flapper nozzle diameter can be selected.

Given a nozzle diameter and the two required areas, the flapper
gap as a function of temperature can be immediately defined.
A flapper arm and leaf spring can then be selected and a
stress analysis (Appendix A) carried out to ensure that the
leaf spring is lightly stressed and to define the required
temperature sensor gradient.

The temperature sensor design is quite straight forward.
The fluid in the bulb expands with temperature as does the
bulb itself. The incremental expansion of the fluid is
used to expand a bellows of known effective area (the area
of a piston that would displace an equivalent volume of
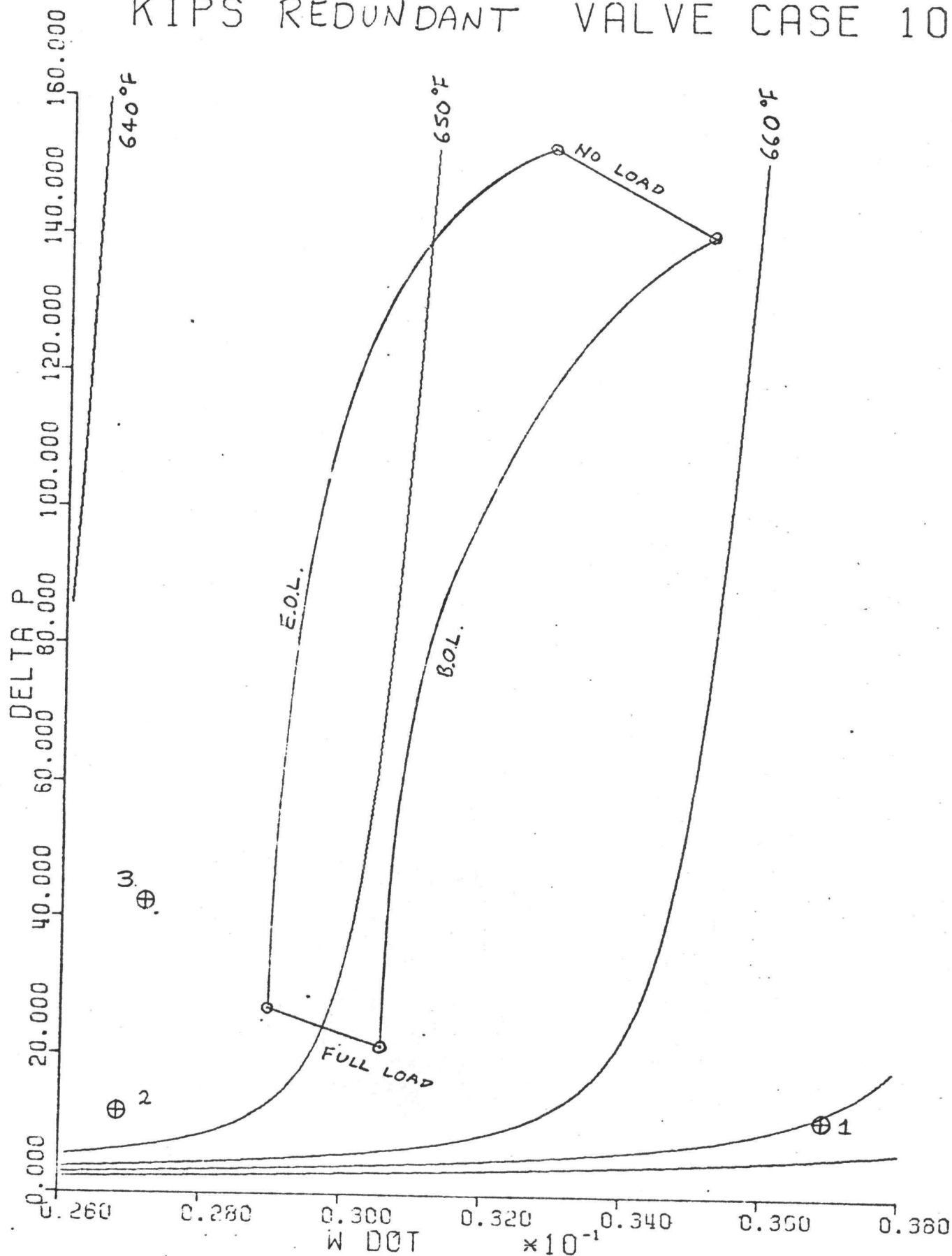fluid as does a bellows for an equal stroke is defined as
effective area).

FIGURE 4.11

$$\therefore \quad \text{T.S.Grad.} = \frac{1}{A_E} \left\{ \left(\frac{\pi d^2}{4} \right) \left(1 \right) \left(B_{fluid} - B_{tube} \right) \right\}$$

where T.S.Grad. = Temp Sensor Gradient $\frac{in.}{°F}$

A_E = Bellows Effective Area

d = bulb inside diameter

B_{fluid} = Volumetric Expansion Coefficient of Bulb Fluid

B_{tube} = Volumetric Expansion Coefficient of Bulb Material

1 = Bulb length

Several fluids were considered for temperature sensor fluids. Dowtherm A was first considered because of its resistance to thermal breakdown, and because Dowtherm A is the system working fluid. Closed capsule test data on the thermal breakdown was reviewed and it was concluded that the amount of gaseous products that could be expected to form over a seven year period was sufficient to produce an unacceptable calibration shift in the unit. There was insufficient data on the solubilities of the gases to ensure that they would remain dissolved. Since Dowtherm A is at least as stable as any alternative organic fluid considered, metals were considered next.

Mercury was the first metal considered. The problem encountered with mercury was embrittlement of the bellows material. The next metal considered was sodium. Sodium is commonly used in stainless steel containment in nuclear reactors and heat exchangers. Since Na has been extensively used much data is available on its compatability with stainless steel. The problem with sodium for the KIPS application is its high freezing point of 208°F. It is likely that a sodium temperature sensor would freeze during storage and might cause problems if melted non-uniformly. As an alternative to sodium, the eutectic NaK was considered. The freezing point of NaK is approximately 10°F and therefore the potential freezing problem should be obviated. The question of corrosion was investigated; the results of the investigation are presented in the Appendix.

4.1.3.3   Pressure Regulator Function

The pressure regulator was sized using Figure 4.11 points 2
and 3.   Since points 2 and 3 are on the same temperature
line, the difference between the two points must result
primarily from a position change of the pressure regulator.
Because turbine inlet temperature is the same at both points,
the temperature bias metering area is the same at both
points.   The area of the temperature bias metering area was
calculated previously and designated A_2.   At point 2 the
pressure drop of the pressure regulator is 5 psi and, since
the pressure drop across the temperature bias metering area
is also 5 psi, the area at the pressure regulator must,
therefore, also be A_2.   At point 3 the pressure drop across
the temperature bias area, A_2, will have increased because of
the flow increase from point 2 to point 3.   This pressure
drop is calculated using Eq. (1).   Since the total pressure
drop is known the pressure drop at the pressure regulator and
finally the area of the pressure regulator can also be cal-
culated.   Thus a schedule of pressure regulator area as a
function of pressure drop across the temperature bias area
has been defined.

Since the pressure regulator flow area at a 5 psi drop across
the temperature bias area is a maximum, the nozzle diameter
can be based on this area.   With the nozzle diameter selected,
the pressure regulator area schedule can be converted to a
flapper gap schedule.

The flapper nozzle location is adjustable by shimming.   The
flapper position is determined by the force balance on the
flapper arm.   Thus the flapper gap can be set to a reference
location and by proper selection of the leaf spring and
bellows area the desired flapper gap as a function of
temperature bias nozzle pressure drop can be attained.
Appendix A presents the stress analysis procedure utilized
to design the flex pivot leaf spring.

4.1.3.4   Valve Design Program

The logic described in Sections 4.1.3.3 and 4.1.3.4
was incorporated into a valve design computer program
and several possible valve selections run. The program
was also used to examine the sensitivity of the design
to changes in characteristics as would be expected to
result from normal manufacturing variations.

4.1.3.5   Valve Design Results

Using the three selected design points, hand calculations
were first carried out to define a starting valve con-
figuration. The program was then used to perform a
variation of parameters study to refine the valve design.
Figure 4.13 presents a table showing the valve character-
istics selected. Figure 4.11 presents the expected
valve performance. The valve pressure drops as a function
of flow through the valve is presented for several
different turbine inlet temperatures. The three design
points originally selected and used for the original
hand calculations are shown for reference only. The
valve can be calibrated to adjust the position of the
family of constant temperature lines along the flow axis.

Figure 4.11 also shows the system operating lines for
beginning of life and end of life isotope heat input
rates. During a speed transient flow will initially
follow a constant temperature line. As turbine inlet
temperature changes, the valve will change the constant
temperature line it operates on. The valve will continue
to adjust its operating line until the new steady state
point is reached. If, for example, turbine inlet tempera-
ture increases, the valve will shift to a higher flow
line thereby eliminating most of the temperature increase.

4.1.3.6   Further Analysis

Further analytical effort is required for the valve.
Dynamic analysis of the valve will be carried out to
A   verify the design's integrity with respect to acceler-
tion, vibration, and shock loading. Bellows stress
analysis will be supplied by the bellows vendor.

4.1.4   Similarity to Flight System

Figures 4.10 and 4.12 are sketches of the GDS and Flight
System two stage mechanical valve with redundancy. The
units are interchangeable functionally. The GDS unit is
configured in such a way as to allow the unit to be dis-
assembled, if necessary. The flight unit will be a welded
assembly with the welding accomplished after the unit is
calibrated and tested. The functioning parts (e.g., the
bellows, flapper arms, temperature sensor bulbs, tempera-
ture sensor actuators) will be identical for the two
valves. The GDS valve fittings that the functioning parts
are mounted on are altered in such a way as to allow
disassembly. Because the GDS unit is a bolted assembly,
the GDS unit should be somewhat heavier than the flight
unit. In general, the differences between the units are
limited to those caused by the anticipated need to dis-
assemble the GDS unit.

4.1.5   Test Plan

The two stage mechanical valve concept will be rig tested
on the KIPS Dowtherm test loop. The valve utilized for
this proof of principle testing will be the two stage
mechanical valve. The test loop will be used to define
the flow and throttling pressure drop envelope for the
desired temperature range. The valve will be calibrated
to produce the required flows over the given temperature
range. The valve design procedure, of necessity, assumes
values of discharge coefficients. The actual value of
parameters like discharge coefficients is unlikely to
be exactly as assumed. The calibration procedure is
used to compensate for differences between these actual
and assumed parameters. The capability of the valve to
produce the flows required for multi power level opera-
tion can also be investigated by determining the envelope
of flows and pressure drops that can be attained from a
given valve over the required temperature range for various
calibration conditions.

The two stage valve will then be altered to simulate the
single stage valve. This alteration requires that the
two nozzles be replaced. The flow limiter nozzle will
be replaced with a nozzle designed so that the motion of
the pressure sensor bellows will not result in throttling.
The temperature bias nozzle will be replaced with one
sized for the required flows and pressure drops.

The valve will then be tested to determine its flow,
throttling pressure drop envelope. The valve will then
be calibrated to determine the envelope of flows and
pressure drops that can be attained from a given valve
over the required temperature range for various cali-
bration conditions.

Redundant
Bellows

Dual Bellows to
Contain Nak Leak

Controlled
Leak

Redundant
Sensors
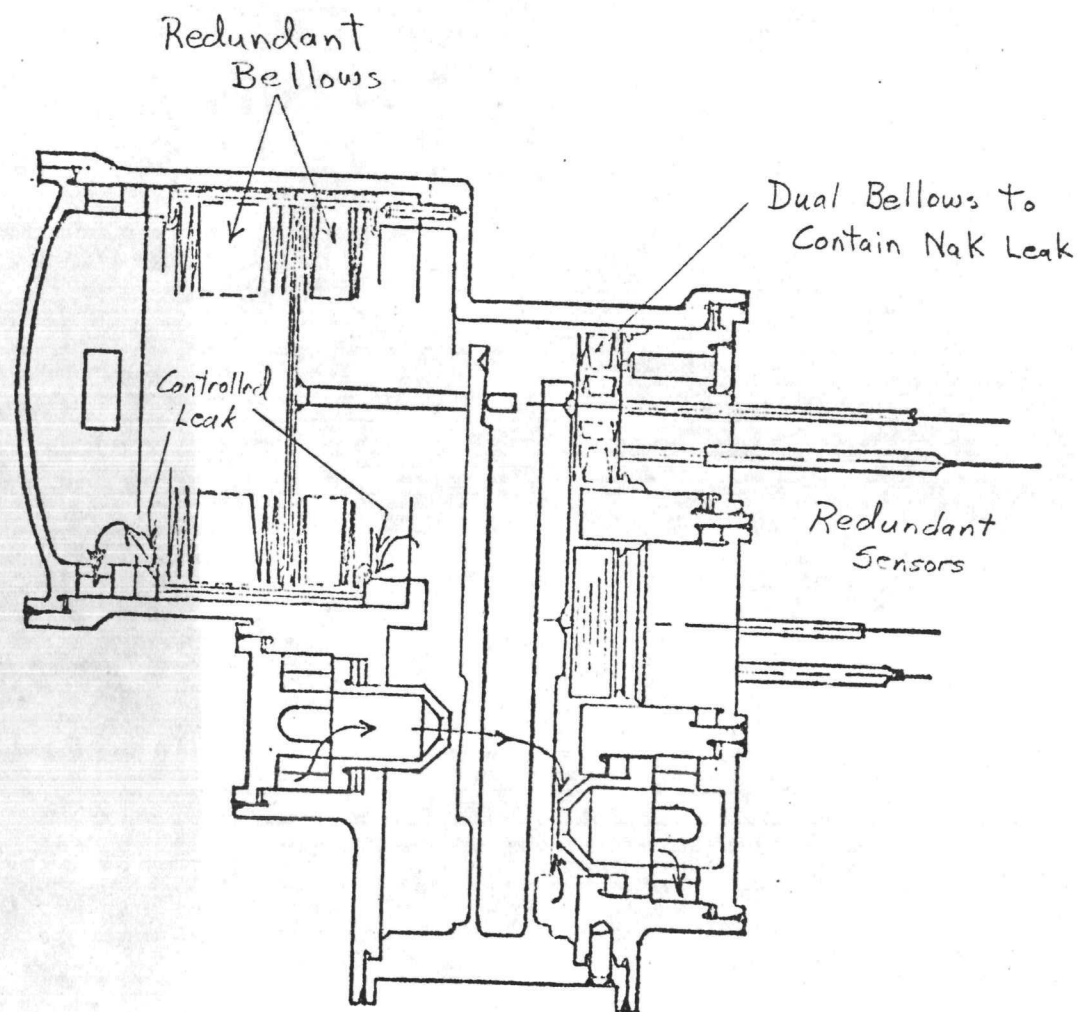
KIPS FLIGHT SYSTEM FLOW CONTROL VALVE
FIGURE 4.12

L.W. Gustafson
11-18-76

Fig. 4.13

Valve Characteristics

Temperature Bias Nozzle Diameter =   .125"

Pressure Regulator Nozzle Diameter = .250"

Temperature Bias Nozzle Gap

T (°F)	Gap (in.)
637	.0061
670	.0082

Pressure Regulator Nozzle Gap

Total ΔP	Gap (in.)
0	.0180
125	.0010

Bellows Area = 1.59 in.^{2}

Pressure Regulator Leaf Spring Rate = 53.7 lbf/in.

4.1.6 Reliability

An objective of the valve design was to eliminate single point failures to the maximum extent practical and to demonstrate the reliability of those components which retained possible single point failures by life testing.

The valve incorporates redundant thermal sensors. Either of the thermal sensors may fail without affecting the operation of the valve. The thermal sensors incorporate containment bellows in order to prevent leakage of the thermal sensor fluid into the system in the event of an actuator bellows failure. Helium leak test ports have been incorporated into the thermal sensor so that the containment bellows can be verified to be leak tight. The space between the actuator and containment bellows will be evacuated and cleaned so that a failure of the containment bellows will not release contaminants into the system, and to prevent the NaK from becoming contaminated in the event of an actuator bellows failure.

The pressure regulator section of the valve incorporates redundant bellows. Either of the bellows may fail with only minor consequence. Figure 4-18 illustrates the flow shift resulting from a failed bellows. Failure of a bellows results in a loss of the bellows spring effect and "short circuiting" one of the calibrated leaks. Loss of the bellows spring rate has a small effect on flow because the bellows operates near its free length and has a small spring rate. The shorting of calibrated leak produces a small increase in flow. The failed bellows case is illustrated in Fig. 4-18. The flow change as a result of a 10% reduction in flex pivot spring rate is also illustrated in Fig. 4-18. If the reduction in spring rate exists at assembly the effect would be removed by calibration.

Single point failures still exist in the flapper valve flex pivots. Five flex pivots will be tested for 2000 lifetimes each. The objective of the test is to demonstrate .9997 reliability for the flex pivots. The flex pivot design utilized margins of safety as specified in the Reliability Volume (Volume 4) para. 2.2.4.5 to achieve a failure probability of (10^{-9}).

4.1.7 Quality Control, Cleaning, Critical Characteristics

1. Pressure Sensor Bellows and Pressure Regulator Flapper Arm

The flapper arm flex pivot spring and the two bellows springs are in parallel when the valve is operating. The flex pivot spring will be controlled by machining the flex pivot in a two stage process. The first stage will machine the thickness and measure to determine the thickness of the pivot within the required tolerance band. Then with the thickness known, machine the pivot to within the width tolerance specified for the known thickness. The table of widths and thicknesses is presented in the flapper stress analysis in the Appendix. A variation of 10% can be accepted.

The spring rates of the bellows are not critical since the bellows operates near its free length.

The effective area of the bellows will be experimentally determined by the bellows vendor. A variation of ± 10% will be accepted.

The bellows assembly is configured so as to allow access to both sides of the bellows for cleaning. The bellows cleaning procedure is as yet undetermined. However, a bellows cleaning procedure used by Sundstrand in the past is presented in the Appendix. The bellows vendor will be asked to supply Sundstrand with a manufacturing outline defining the bellows assembly processing. Definition of special procedures utilized by the vendor will be requested. Further definition of manufacturing processing awaits the selection of a vendor.

2. Thermal Sensor

The thermal sensor gradient (i.e. change in length per °F) will be experimentally determined by the thermal sensor vendor. A variation of ± 10% is acceptable. The thermal sensor bellows, fitting, and capillary tubing will all be cleaned as piece parts prior to assembly. After assembly the helical bulb, transition capillary, and actuator bellows will be leak checked by helium mass spectrometer to 10^{-8} scc/sec. The containment bellows will be leak checked to 10^{-8} scc/sec. Prior to filling with NaK, the helical bulb, transition capillary, and actuator bellows will be vacuum baked at a pressure of 10^{-6} torr or smaller and a temperature of 500°F or greater for a period of 48 hr minimum. The area between the actuator bellows and the containment bellows will have the same vacuum baking process applied. The thermal sensor vendor will be requested to supply a manufacturing outline and definition of special procedures.

3. Shimming

The valve is calibrated by shimming. Shimming adjustments may require smaller changes in shim size than are available in standard shim stock. Locations where these small shimming adjustments are required are one of the thermal sensors and the temperature bias nozzle. At other shimming locations standard shim stock is adequate. Where small shimming adjustments need to be made shims will be fabricated and machined as required to attain required valve performance.

4. Parallelism

Since the pressure regulator nozzle gap can go to a minimum value of approximately .001, parallelism between the nozzle face and flapper will be controlled by drilling and pinning the flapper assembly and valve housing at assembly.

KIPS REDUNDANT VALVE CASE 10
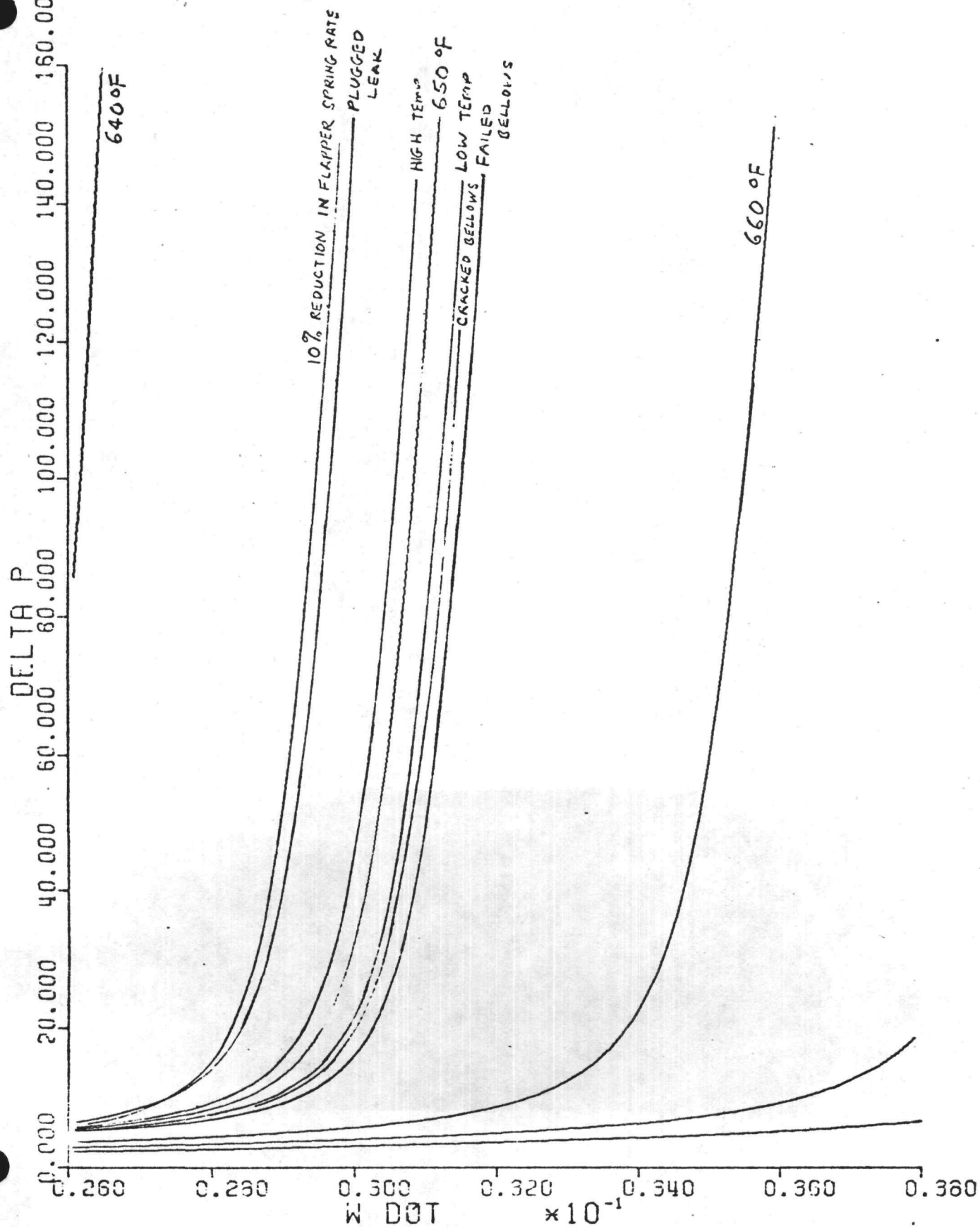
EFFECT OF FAILURE MODES

FIGURE 4.18

APPENDIX B

CORROSION OF THERMAL SENSOR MATERIALS BY NaK

Summary:

Corrosion of thermal sensor materials by NaK should not present problems. Clean NaK will be introduced into the thermal sensor and complete reduction of the oxide film on the helical temperature bulb surface and the transition capillary tube surface will not liberate sufficient oxygen to cause corrosion problems over a five year period. The temperature of the bellows actuator is sufficiently small to prevent reduction of the oxide film on the bellows.

History:

Consideration of NaK as a heat transfer fluid in liquid metal cooled nuclear reactor systems began in 1945. NaK was favored over Na because the lower freezing point of the NaK prevented potential freezing problems that might occur with Na. The lower freezing point of NaK was its sole advantage however. In other respects Na is a superior heat transfer fluid to NaK. A considerable amount of research was done on NaK; but, over the years, emphasis has shifted to Na. Na lines can be trace heated to prevent freezing thereby eliminating NaK's sole advantage and making Na the preferred heat transfer fluid.

NaK has also found application as a heat transfer fluid in industrial heat exchangers. Several of these applications are described in "Industrial Application of NaK Heat Transfer System," K.R. Barker, MSA Research Corporation. Of the systems described, the longest period of operation was eleven years at the time the report was prepared (March 1970). This report also discusses the design, fabrication, and cleaning of industrial NaK systems. The cleaning procedures discussed are no more elaborate than wire brushing and degreasing.

Discussion:

The thermal sensor may be divided into two sections on the basis of temperature. The "cold" section is made up of the bellows actuator. The bellows actuator is immersed in a liquid at a maximum temperature of approximately 220°F. The "hot" section is made up of the temperature bulb at a temperature of approximately 650°F. The transition capillary will be assumed to be in the hot section.

1. Hot Section

The rate of corrosion of stainless steel in NaK is known to be dependent on temperature, speed at which the NaK flows over the stainless steel, and concentration of oxygen in the NaK. One source of oxygen in the thermal sensor NaK is oxygen in the NaK used to fill the unit. The oxygen content of the NaK used to fill the thermal sensor will be monitored by sampling from the fill line and analyzing the sample. The other source of oxygen in the NaK fill is oxygen released by reducing the oxide film from the inside surface of the capillary tubes. Discussion with W. Reuther, Argonne National Lab., indicates that stainless steel may be conservatively assumed to have a 30 Å coating of Fe_2O_3. Complete reduction of the Fe_2O_3 was assumed to take place and the oxygen concentration resulting from the reduction calculated.

For the .063 in. i.d. capillary temperature bulb:

1. Volume of oxide $= \frac{\Pi}{4} (.063^2 - (.063 - 30(10^{-8})(\frac{1}{2.54}))^2) 30$

$$= 3.51 (10^{-7}) \text{ in}^3$$

$$= 5.75 (10^{-6}) \text{ cm.}^3$$

2. Weight of oxide $= (5.75(10^{-6}) \text{ cm.}^3) (5.12 \frac{g}{cm^3}) = 2.94 (10^{-5}) \text{ g.}$

3. Weight of oxygen $= (2.94(10^{-5})) (.3006) = 8.84 (10^{-6}) \text{ g.}$

4. Weight of NaK $= \frac{\Pi}{4} (.063)^2 (30) (2.54)^3 (.847) = 1.298 \text{ g.}$

5. Concentration $= \frac{8.84 (10^{-6})}{1.298} = 6.81 (10^{-6})$

$$= 6.81 \text{ ppm.}$$

For the transition capillary a similar calculation yielded an increase in oxygen content resulting from the total reduction of the oxide film of 12 ppm. The drawing requirement for oxygen concentration in the NaK used to fill the thermal sensor is less than 25 ppm. Therefor the maximum total concentration of oxygen in the hot section is 37 ppm.

Recent corrosion data for NaK do not seem to be available. However, D.L. Smith, Argonne National Lab. advised that, since Na and NaK are simlar with respect to corrosion of stainless steel, corrosion data available on Na, be utilized. Corrosion data are not not available for the exact conditions that are present in the thermal sensor. Corrosion data have generally been collected at higher temperatures than are present in the thermal sensor. The reason the data have been collected at higher temperatures is to simulate the conditions seen in service and to produce measurable corrosion rates. The minimum temperature for which data were available was approximately 700°F. The thermal sensor is an essentially zero flow speed situation but the smallest flow speed data was for .5 m/sec. Thus the estimated corrosion rate for the thermal sensor should be conservative because of the required assumed temperature of 700°F and speed of .5 m/sec. Given conditions of 700°F, .5 m/sec., and 37 ppm oxygen; Figure 1 can be entered and a corrosion rate of .16 u/yr. read. Converting to inches the corrosion rate becomes $6.118 \ (10^{-6})$ in./yr. If this rate is increased by an order of magnitude to allow for location to location variation in material properties, it becomes $6.118 \ (10^{-5})$ in./yr. If a seven year period is assumed the total corrosion becomes .0004 inches. This is much smaller than the wall thickness manufacturing tolerance. Assuming the entire bulb loses .0004" wall thickness, the other dimensions at the worst case tolerance stack up levels, and utilizing the required margins for (10^{-9}) probability of failure, the maximum allowable pressure in the bulb was calculated to be 1900 psi. Since this allowable pressure is much larger than the maximum bulb pressure of 300 psi, the reliability of the sensor should not be affected.

2. Cold Section

Discussion with W. Reuther, Argonne National Lab. on the topic of reduction of the oxide film on stainless steel indicated that the rate with which the reduction would take place is greatly dependent on temperature. Reference to the Liquid Metals Handbook confirmed this conclusion. There apparently has been a fair amount of research on this topic. The reason for the research is that the oxide layer increases the resistance to heat flow. Since the primary use of the liquid metals was in heat exchangers, it was highly desirable to remove the oxide layer. A recommended method for removing the oxide was to heat the liquid metal to a temperature where it would rapidly remove the oxide The objective of the research was to determine what temperature was required. Thus although there is research on the topic of interest, there is not much useful information because the objective of the research was to determine a temperature at which the reaction would take place rapidly not a temperature below which the reaction would take place extremely slowly.

Tests carried out at Argonne by W. Reuther did lend some information, however. A stainless steel mesh was immersed in NaK 260°C after 113 hr no wetting had occurred. At 304°C wetting of the stainless steel mesh was completed in less than 1 hr. The Arrhenius rate equation is known to model similar reactions and was therefore examined with respect to the available data. Reference to Skoog and West *Analytical Chemistry* yielded activation energy data indicating that the activation energy for the reactions that might take place is in the range of 3 to 3.5 electron volts. A value of three was assumed since it is conservative. With the value of actuation energy it was then possible to test the Arrhenius rate equation at the two data points.

Arrhenius Rate Equation $\quad A = A_o \exp - \left(\dfrac{Ea}{KT}\right)$

where A = measure of activity or reaction rate

Ea = activation energy

$K = 3.29 \ (10^{-4}) \ \dfrac{Cal}{°K}$, Stefan Boltzman Constant

T = temperature (°K)

A_o = constant

$$\frac{A\ 304}{A\ 260} = \frac{A_o \exp - \dfrac{(3ev)\ (3.83(10^{-20})\frac{cal}{ev}}{(3.29\ (10^{-24})\ \frac{Cal}{°K})\ (304 + 273)}}{A_o \exp - \dfrac{(3)\ (3.83\ (10^{-20}))}{3.29\ (10^{-24})\ (260 + 273)}}$$

$= 147.7$

Therefor if the Arrhenius rate equation holds the reaction should take place 148 times as fast at 304°C as it does at 260°C. Since this agrees quite closely with the test results, the Arrhenius equation was used to estimate the relative speed of the reaction at 104°C (220°F).

$\dfrac{A\ 260}{A\ 104} = 5.99\ (10^{11})$

Therefor the reaction should take place 5.99 (10^{11}) times faster at 260°C than it would at 104°C. Since no wetting occurred in 113 hr at 260°C, no wetting should occur in 6.76 (10^{13}) hr. at 104°C. This is a period of time $(1.1 (10^{9})$ longer than a seven year mission. Thus the estimate can be in error by a factor of a billion with no ill effect.

It is important to note that the proceding calculation is concerned with only reduction of the oxide film. Corrosion of the stainless steel will take place only after the oxide film is removed.

3. Welds

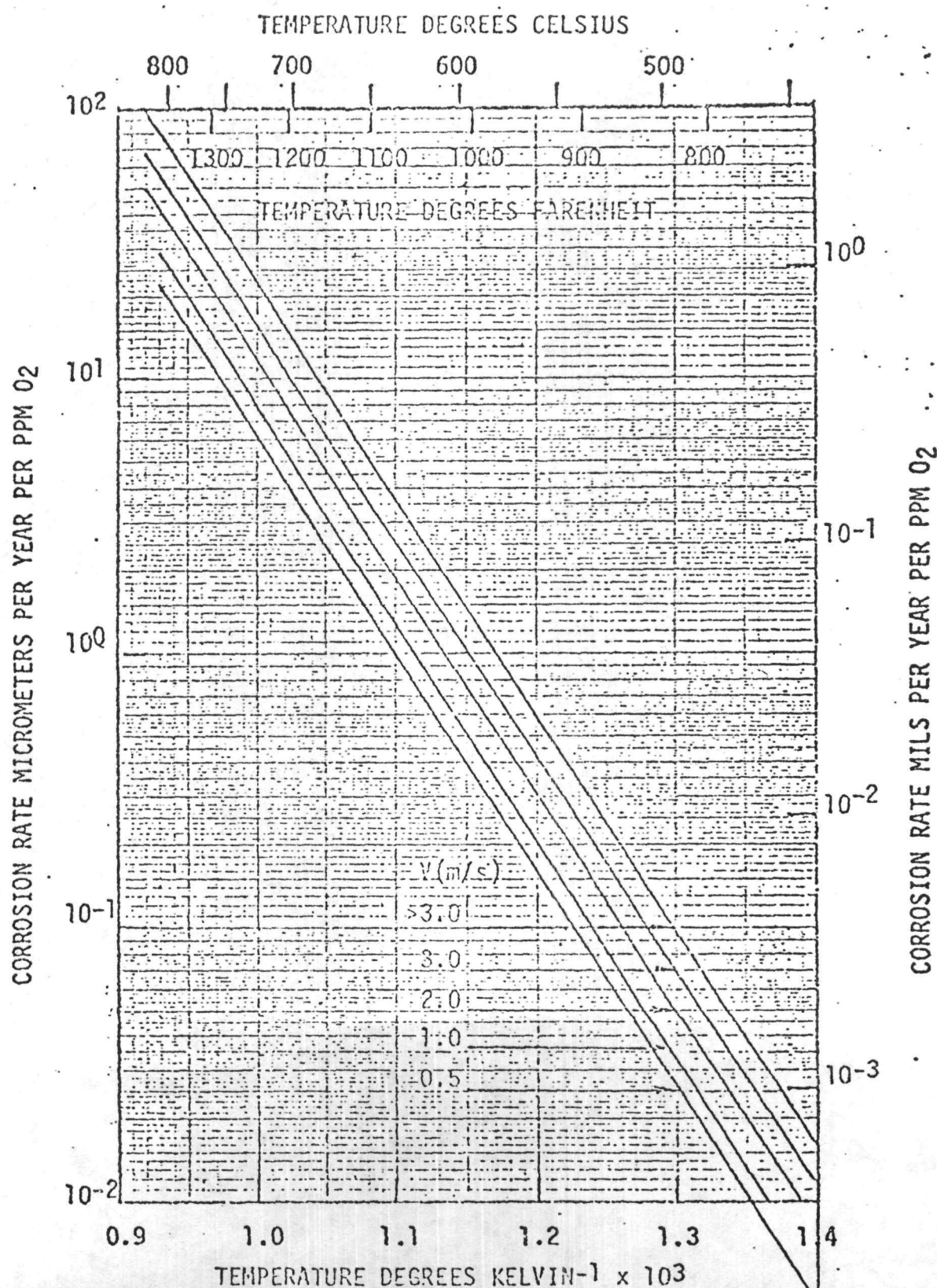The Liquid-Metals Handbook indicates that welds are as resistant to attack by NaK as are the base metals.

TEMPERATURE DEGREES CELSIUS

CORROSION IN LIQUID SODIUM

Applicable Product Forms - All Forms and Heat Treatments

Notes: (1) Read Volume II before using these curves
(2) Multiply value from curve by oxygen level (VWED) to obtain corrosion rate
(3) For high dT/dL conditions i.e., core components, multiply value from curve by 2.

Fig. 1

DESIGN
SECTION 4 - 316 SS ANNEALED
GROUP 1 - HIGH ALLOY STEELS
STRUCTURAL MATERIALS
PART
REVISION : 7-22-76
PREVIOUS REVISION: