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FINAL REPORT
PRELIMINARY DESIGN OF THE CRYOGENIC COOLED LIMB SCANNING INTERFEROMETER RADIOMETER (CLIR)

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FOR THE NATIONAL AERONAUTICS AND SPACE ADMINISTRATION GODDARD SPACE FLIGHT CENTER

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ILLUSTRATIONS

1.0 INTRODUCTION

This study was performed for the preliminary design of the cryogenic cooling system for the Cryogenic Cooled Limb Scanning Interferometer Radiometer (CLIR) instrument to be flown on the Atmospheric-Magnetospheric Physics Satellite (AMPS). This study was in response to the statement of work, generated by NASA Goddard Space Flight Center on Jan. 1978.

The studies were divided into two major tasks, (1) the first a top level trade study to determine the relative advantages of various cryogen systems and make a baseline selection and (2) to make more detailed analysis on the baseline system and to prepare a layout showing hardware detail of the system.

The top level trade studies were more extensive than initially expected due to the instrument requirement for cooling at three temperature levels as opposed to the two levels initially described for the instrument. Approximately twelve different combinations of cryogens were investigated.

The basic lifetime requirement for the instrument was 30 days, however, limited studies were also conducted for a follow-on mission requiring a one year life (UARS).

The top level trades led to the selection of a single stage supercritical helium baseline. This selection was made primarily based on advantages associated with safety, ground handling, and state of development.

2.0 INSTRUMENT COOLING REQUIREMENTS

The instrument cooling requirements evolved during the study. As the configuration of the instrument matured the heat rates changed, so that several study iterations were required. The instrument required cooling of different elements to three temperature levels.

The cooling loads and temperatures are presented in Fig. 2-1 which shows the heat rates associated with the various iterations throughout the program. The designation of the iteration number was adhered to throughout the study so the results could be sorted out. The baffle (or radiation shield) temperature varied somewhat during the study depending upon the wavelength under consideration, however, a value of 110°K was assumed at the end of the study.

The detailed break-down of the heat loads which were supplied by Honeywell Electro Optics Center is presented in Fig. 2-2 , for the first and second iteration. The third iteration was not utilized for any cooling system trades. A fourth and final iteration was also generated and is described in the section presenting the baseline design.

The mission lifetime, as shown in Fig. 2-1, was for 30 days.

FIG. 2-1 CLIR INSTRUMENT COOLING REQUIREMENTS

	MAXIMUM TEMPERATURE (K)	HEAT RATES,[a] AVERAGE (W)		
		ITERATION 1	ITERATION 2	ITERATION 3
DETECTORS	13	0.141	0.075	0.068
OPTICS	30	1.92	0.93	0.832
BAFFLES (APERTURE LOAD ONLY)	115–130	7.5	7.5	7.9 @ 100 K[b]

INSTRUMENT LIFETIME: 30 DAYS CONTINUOUS OPERATION.

(a) VALUES INCLUDE BOTH HEAT GENERATION AND PARASITIC.

(b) ITERATIONS NO. 1 AND NO. 2 ASSUME BAFFLE AT 70 K; NO. 3 AT 100 K.

FIG. 2-2 PRELIMINARY CLIR HEAT LOADS (SUPPLIED BY HRC)

	Heat loads (W): Iteration No. 1	Heat loads (W): Iteration No. 2
• OPTICAL BENCH (30K, T_{HOT} = 70K)		
A. SUPPORT	1.0	0.5
B. LEAD WIRES	0.08	0.08
C. RADIATION	0.30	0.30
D. I/F DRIVE MOTOR	0.01	0.01
E. CALIBRATION AND REFERENCE SOURCES		
(1) WHITE LIGHT	0.5	0.01 (LVDT)
(2) BLACK BODIES	0.03	0.03
TOTALS	1.92 W	0.93 W
• FOCAL PLANES (10K, T_h = 30K)		
INTERFEROMETER		
– BIAS	0.003	0.003
– SUPPORT	0.043	0.010 (EST.)
– LEAD WIRES	0.003	0.003
RADIOMETER		
– BIAS	0.027	0.027
– SUPPORT	0.043	0.010 (EST.)
– LEAD WIRES	0.022	0.022
TOTALS	0.141 W	0.075 W
• APERTURE LOAD (AVERAGE, 400 KM ALTITUDE)	7.5 W	7.5 W

3.0 SYSTEM SELECTION STUDIES

3.1 INTRODUCTION

Within the framework of the instrument requirements there are many combinations of cryogens which will provide the required temperatures to the detector, optical bench, and radiation baffle. Fig. 3-1 shows the latent heat of sublimation and operating temperature range for some of the more efficient cryogens. Also included is the heat of vaporization of helium and the instrument temperature requirements. For detector cooling to below 13°K. only two cryogens are candidates; solid hydrogen and helium in either superfluid, liquid or supercritical state. Solid neon will approach 13-14°K if the vapor pressure over it is maintained low enough (requiring very large vent lines) and no temperature gradient exists between the solid neon and detectors.

The optics can be cooled to 30°K by vapor cooling from either hydrogen or helium, or by thermal coupling to solid neon. For the baffle (radiation shield) many cryogen combinations are possible as shown.

Consideration of the available cryogens and instrument requirements led to the number of possible combinations shown in Fig. 3-2, which include one, two, and three stage systems.

Within this framework additional options are possible relating to whether the optics are cooled (thermally grounded) by the cryogen utilized for the detector or cooled separately either by vapor or by a solid cryogen stage.

Due to the large number of combinations which can be utilized, a first level trade study was performed based on the primary assumption that the parasitic heat loads to the cooler does not impact the cryogen requirement preliminary analyses·indicated that these cooler parasitic heat loads were lower than the required heat input to the cryogen which would produce sufficient vapor cooling of the optics or baffle.

FIG. 3-1 CANDIDATE CRYOGENS FOR CLIR

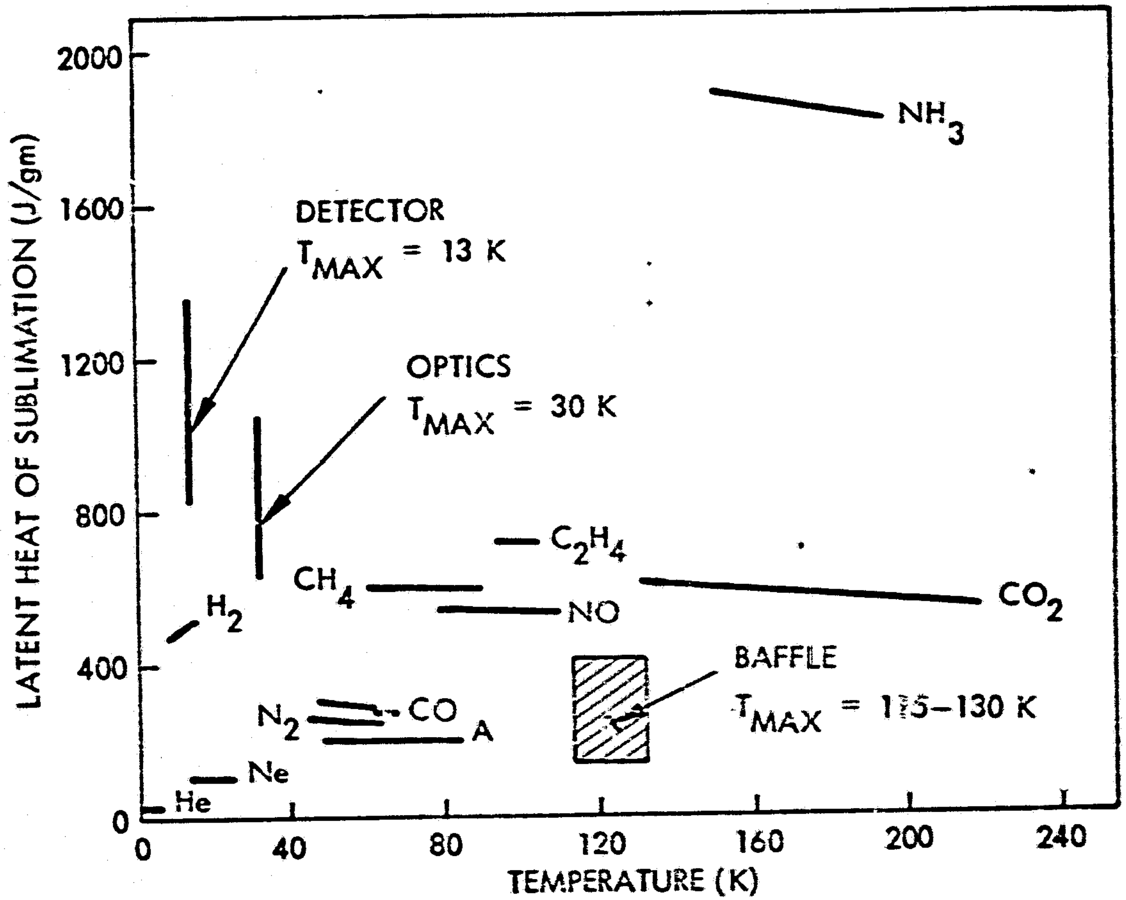


FIG. 3-2 CRYOGEN COMBINATIONS STUDIED FOR CLIR

DETECTOR	OPTICS	BAFFLE	
He	Ne	N_2	3-STAGE SYSTEM
He	He VAPOR	Ne	2-STAGE SYSTEM
He	He VAPOR	N_2	
He	He VAPOR	CH_4	
He	He VAPOR	CO_2	
H_2	H_2 VAPOR	Ne	
H_2	H_2 VAPOR	N_2	
H_2	H_2 VAPOR	CH_4	
H_2	H_2 VAPOR	CO_2	
H_2	H_2	N_2	
He	He VAPOR	He VAPOR	1-STAGE SYSTEMS
H_2	H_2 VAPOR	H_2 VAPOR	

This assumption was borne out by more detailed analysi at a later point in the program and allows the thermal analysis of the instrument and the cooler to be uncoupled. The principal tasks were, therefore, to determine the flow rate of cryogen necessary to meet instrument temperature requirements and determine the size and weight of the corresponding cooling system.

3.2 INITIAL TRADES

The preliminary values for the heat loads to the instrument due to various sources have been described in the previous section. Heat rates from iteration No. 2 were utilized to set up a thermal network for determination of the coolant flow rates.

These heat rates were used to calculate the thermal resistances between instrument temperature regions. The resistances were then assumed to be constant, that is, invariant with temperature. This assumption is adequate when the temperatures of the instrument stages in the analysis are close to the required values.

The thermal network along with the assumed resistance values and instrument heat generation is shown in Fig. 3-3.

In the analysis the optics and baffle temperatures are either determined by an energy balance when those systems are vapor cooled or fixed at the cryogen temperature for systems which are thermally grounded to the cryogen.

3.3 SINGLE STAGE STUDIES

The schematic of one version of the single stage system is shown in Fig. 3-4 for the case where the optics, as well as the baffle, is cooled by vent gas.

The results of the analysis for helium and solid hydrogen cooling are presented in Figs. 3-5 and 3-6. Fig. 3-5 shows the optics and baffle temperature as a function of the helium flow rate. The results show the flow rate requirement is set by cooling of the optics to $30^{\circ}K$ resulting in a baffle temperature of $100^{\circ}K$ (below the required temperature).

FIG. 3-3 THERMAL ANALYZER PROGRAM FOR VAPOR-COOLING EFFECTS

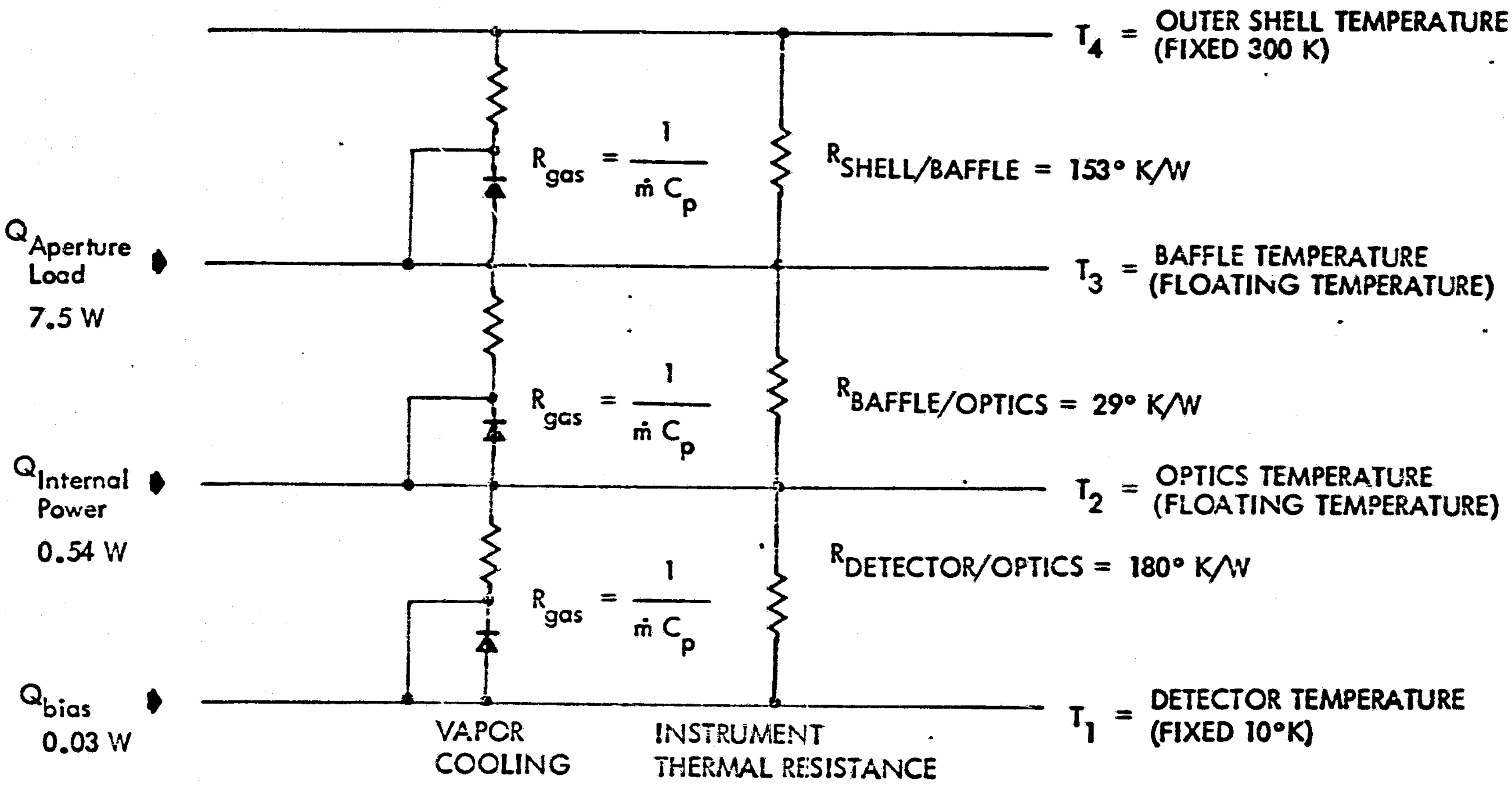


FIG. 3-4 CLIR SINGLE-STAGE COOLING SYSTEM

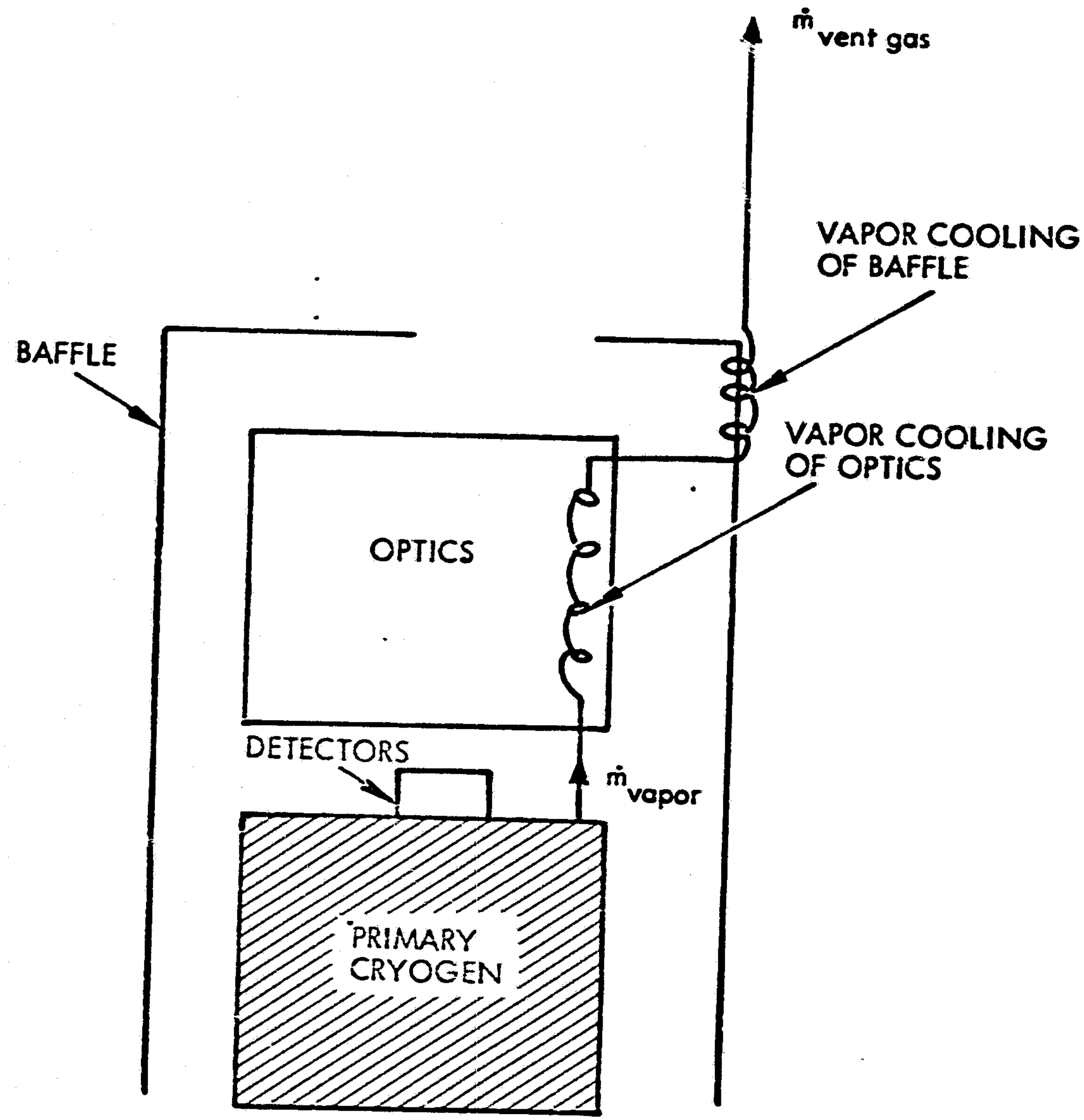


FIG. 3-5 EFFECT OF HELIUM FLOW RATE ON OPTICS AND BAFFLE TEMPERATURES

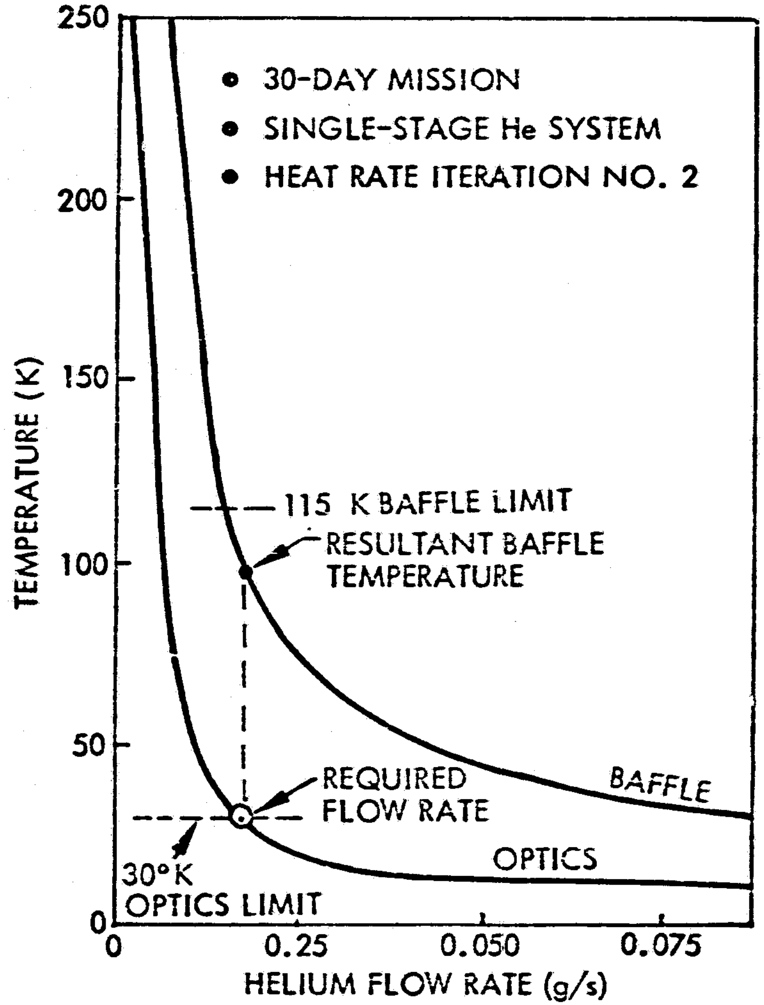


FIG. 3-6 EFFECT OF HYDROGEN FLOW RATE ON OPTICS AND BAFFLE TEMPERATURES

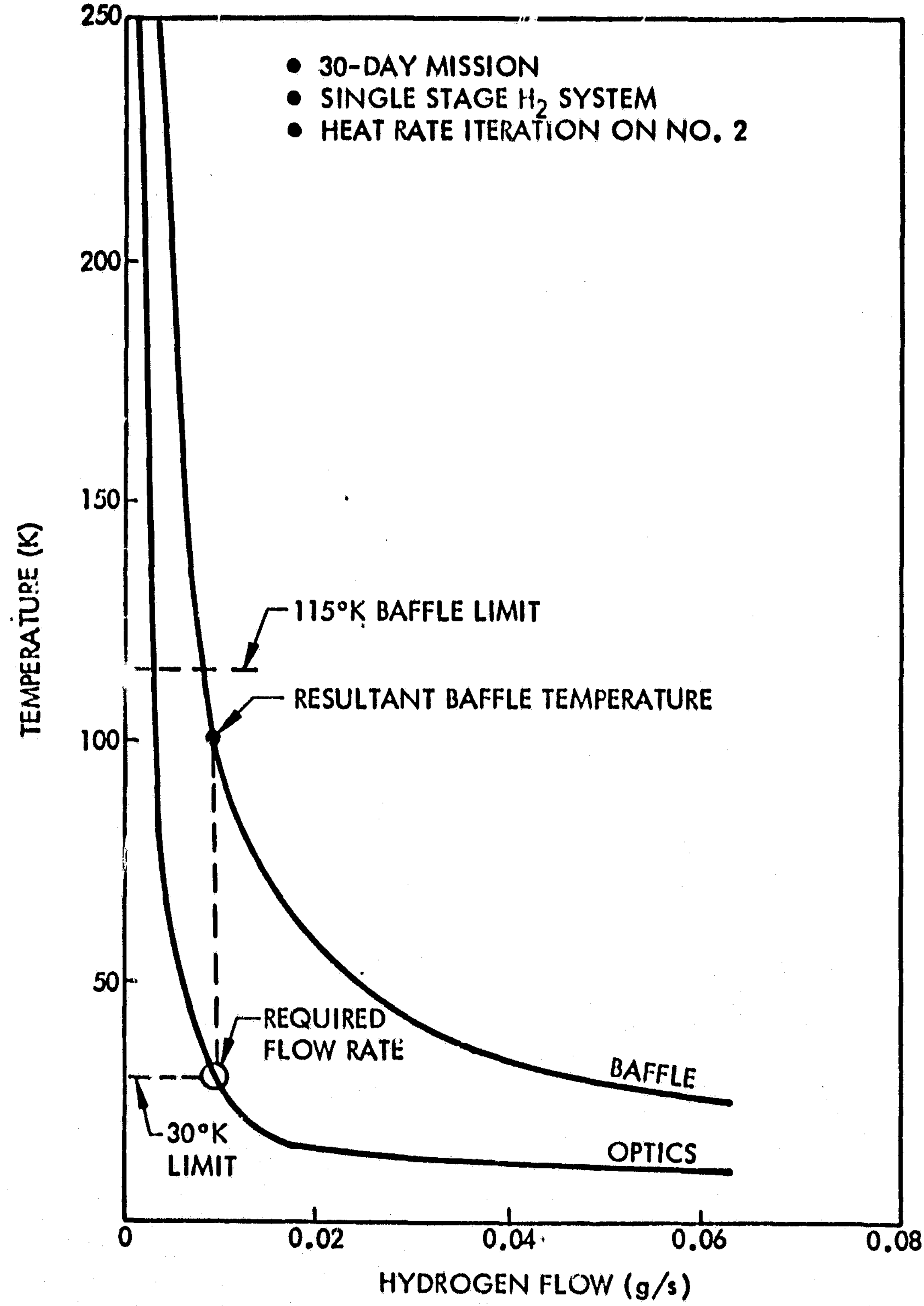


Fig. 3-6 shows the same parameters for solid hydrogen, again showing that the optics sets the flow rate requirement.

A second approach for a single stage solid hydrogen cooler was also studied and found to be much better than the approach just described. In this scheme both the optics and the detector are thermally grounded to the solid hydrogen and the vapor is utilized in cooling the baffle only. As will be seen, this approach leads to the most efficient system of all, in terms of weight and volume.

After the flow rates required to meet the instrument cooling requirements were established this set the quantity and volume of tankage required for the cryogen storage.

The system weights were determined by utilizing the ratio of total system weight to cryogen weight. These ratios were established for the two stage systems by utilization of an existing computer program which was developed in prior programs for trade studies. These ratios have been verified by actual weights for flight coolers which have been developed at LMSC.

For the single stage helium and hydrogen coolers, weight ratios were based on values obtained in various hardware programs.(1) The total weight to cryogen weight fractions used for the hydrogen and helium were 5 and 4, respectively. These values are quite conservative as the baseline study shows.

In calculating the system weights, an additional factor of conservatism of 20% was utilized to account for uncertainties in predicting heat rates. This led to a weight which was 20% greater than the calculated weight.

Figure 3-7 presents a comparison of the weights and volumes for the single state systems. The total system weight includes all elements of the cryogen system; that is the tankage, plumbing, insulation, support and vacuum shell.

The required heat input which is indicated shows the heat rate which is required to produce the gas flow rate necessary to meet instrument cooling requirements. If the parasitic heat load into the cryogen tank is at or below the indicated level, then no additional cryogen is required to compensate for tank heat leaks.

Calculations show that for the hydrogen cases the parasitic heat leaks are substantially below the required level, while for helium the values are about the same.

The comparison shows that the solid hydrogen system which cools both optics and detector to $10^{\circ}K$ is substantially lighter and smaller than the other systems, while also providing a stable $10^{\circ}K$ temperature for the optics.

3.4 DUAL STATE SYSTEM

The second system studied is shown in Fig. 3-8 which illustrates the dual stage system in which the detector and optical system operate at the same temperature and the baffle operates at a second temperature (two-temperature version).

In this approach use of helium as the primary coolant leads to excessive weights because the low heat of vaporization does not effectively deal with the high heat input from the optics.

FIG. 3-7 COMPARISON OF CRYOGENIC SYSTEMS FOR CLIR
I. SINGLE STAGE SYSTEMS

A. DETECTOR, OPTICS, AND BAFFLE ALL AT DIFFERENT TEMPERATURE

GENERAL FEATURES

- CRYOGEN FLOW RATE AND QUANTITY SET BY REQUIRED VAPOR COOLING OF OPTICS
- THIS FLOW RATE COOLS BAFFLE BELOW REQUIRED TEMPERATURE
- HEATER REQUIRED TO MAINTAIN/CONTROL VENT RATE

CRYOGEN	TEMPERATURES			CRYOGEN QUANTITY		TOTAL SYSTEM WT. KG (LBS)	REQD. HEAT INPUT (W)
	DET.	OPTICS	BAFFLE	V, L (FT)	M, KG, (LB)		
He	2-10	30	95	365(12.9)	46(101)	220(484)	0.36
Solid H_2	8-10	30	100	272(9.6)	25(54)	147(324)	4.26

B. DETECTOR AND OPTICS AT SAME (10°K) TEMPERATURE, BAFFLE AT SEPARATE TEMPERATURE

- FLOW RATE SET BY BAFFLE COOLING REQUIREMENT

SOLID H_2	8-10	8-10	115	181(6.4)	16(36)	98(216)	2.7

FIG. 3-8 CLIR DUAL-STAGE COOLING SYSTEM (TWO-TEMPERATURE VERSION)

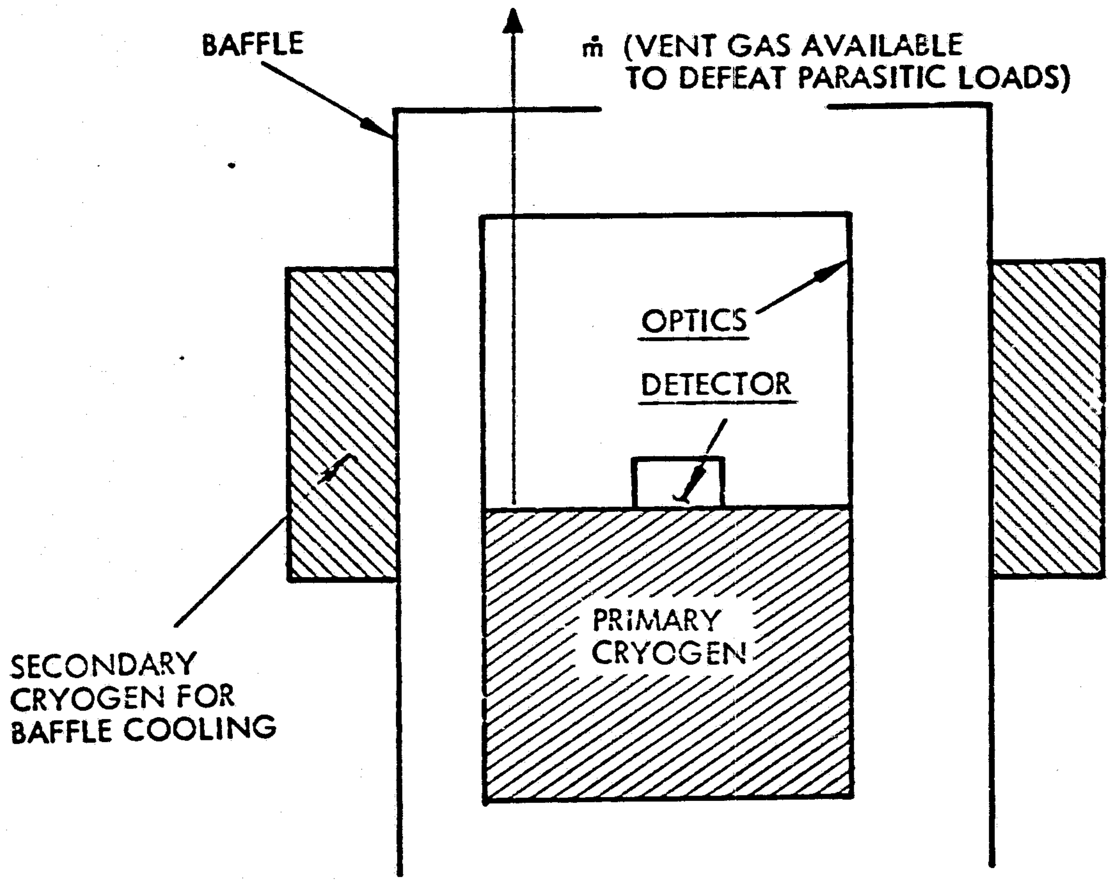


Fig. 3-9 summarizes the weight and volume for this approach using hydrogen as the primary coolant. The minimum weight system is achieved with a solid methane secondary which cools the baffle to 65°K, while the solid CO_2 secondary leads to minimum volume but slightly greater weight. The use of solid methane has been deemed questionable by the science team as a possible interference with the scientific objective, but its effects have not been quantified.

The solid CO_2 leads to baffle temperatures which are somewhat higher than the present instrument requirements. These two cryogen choices were included in the event that these uncertainties are resolved.

An alternate dual stage system in shown in Fig. 3-10. This dual stage approach utilizes the vent gas from the primary to cool the optics while the baffle is cooled by the secondary. In this approach the three areas of the instrument are cooled to different temperatures (three-temperature version).

Figs. 3-11 and 3-12 show the effect of vent gas flow rate on the optics temperature for various secondary cryogens for both a hydrogen and helium primary. The same thermal network shown in Fig. 3-3 is utilized with the baffle set at the temperature of the selected secondary cryogen, which is indicated on the figures.

FIG. 3-9 COMPARISON OF CRYOGENIC SYSTEMS FOR CLIR
II. DUAL STAGE SYSTEMS

B. TWO TEMPERATURE SYSTEMS

GENERAL FEATURES

- BOTH DETECTOR AND OPTICS COOLED TO SAME TEMPERATURE WITH PRIMARY CRYOGEN
- BAFFLE COOLED WITH SECONDARY CRYOGEN

CRYOGENS		TEMPERATURES		CRYOGEN QUANTITY				TOTAL SYSTEM WT. KG(LBS)
				PRIMARY		SECONDARY		
PRIMARY	SECONDARY	DETECTORS OPTICS	BAFFLE	V, L(FT^3)	M, KG(LBS)	V, L(FT^3)	M, KG (LBS)	
H_2	N_2	8-10	45	54(1.9)	5(11)	99(3.5)	96(211)	191 (420)
H_2	CH_4	8-10	65	91(3.2)	8.3(18.3)	76(2.7)	39(85)	128 (283)
H_2	CO_2	8-10	130	173(6.1)	16(35)	22(.77)	37(82)	158 (348)

NOTE: He UTILIZED AS THE PRIMARY IN THIS SYSTEM LEADS TO EXCESSIVE WEIGHT

FIG. 3-10 CLIR DUAL-STAGE COOLING SYSTEM (THREE-TEMPERATURE VERSION)

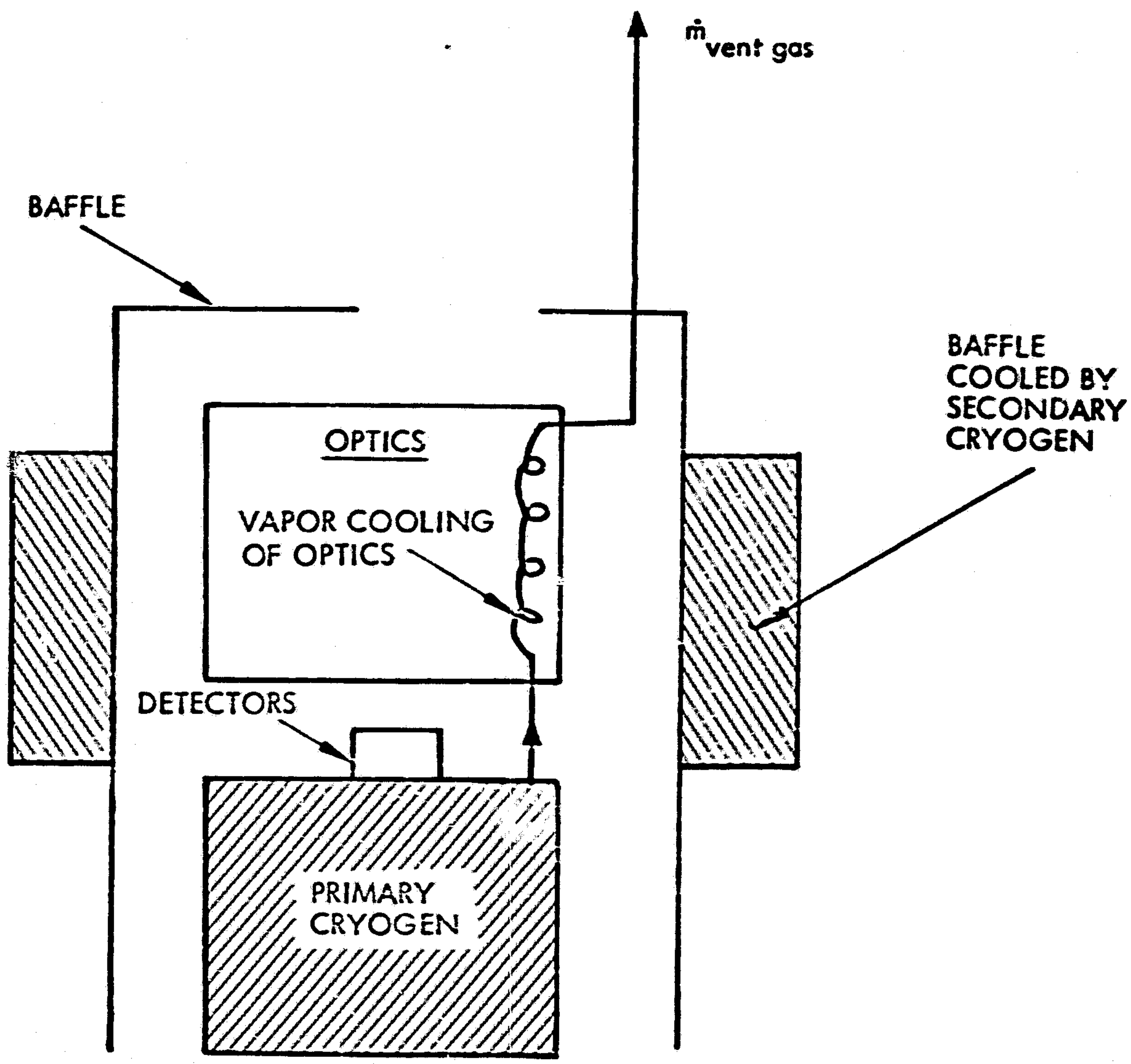


FIG. 3-11 EFFECT OF HYDROGEN FLOW RATE ON OPTICS TEMPERATURE

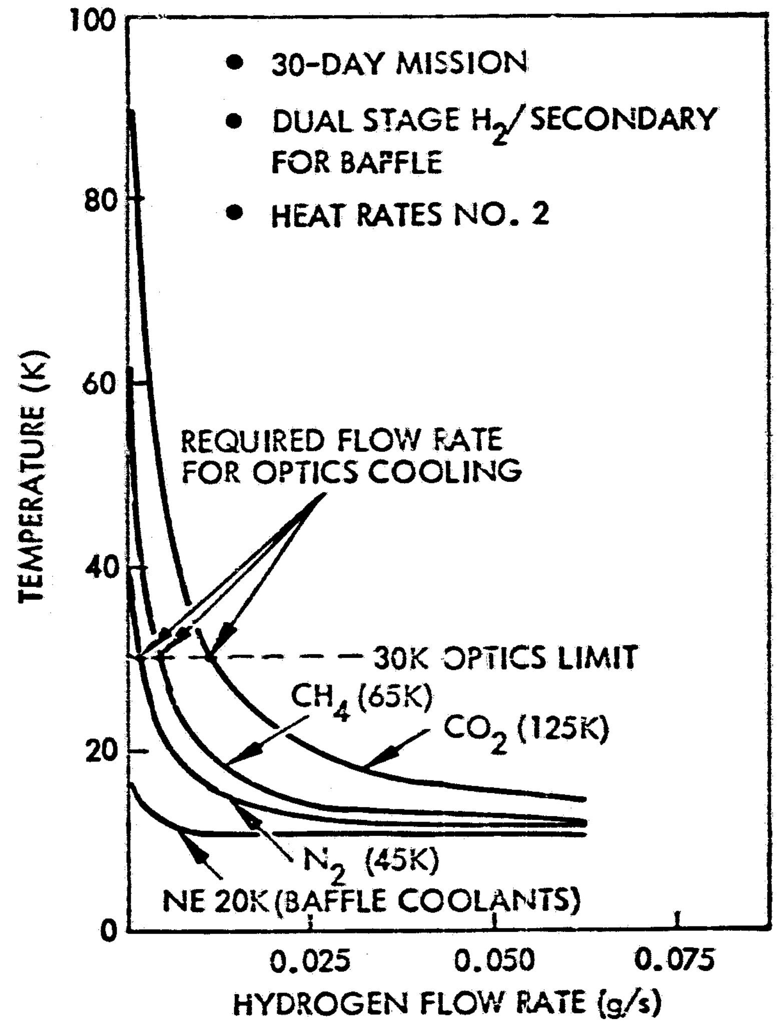
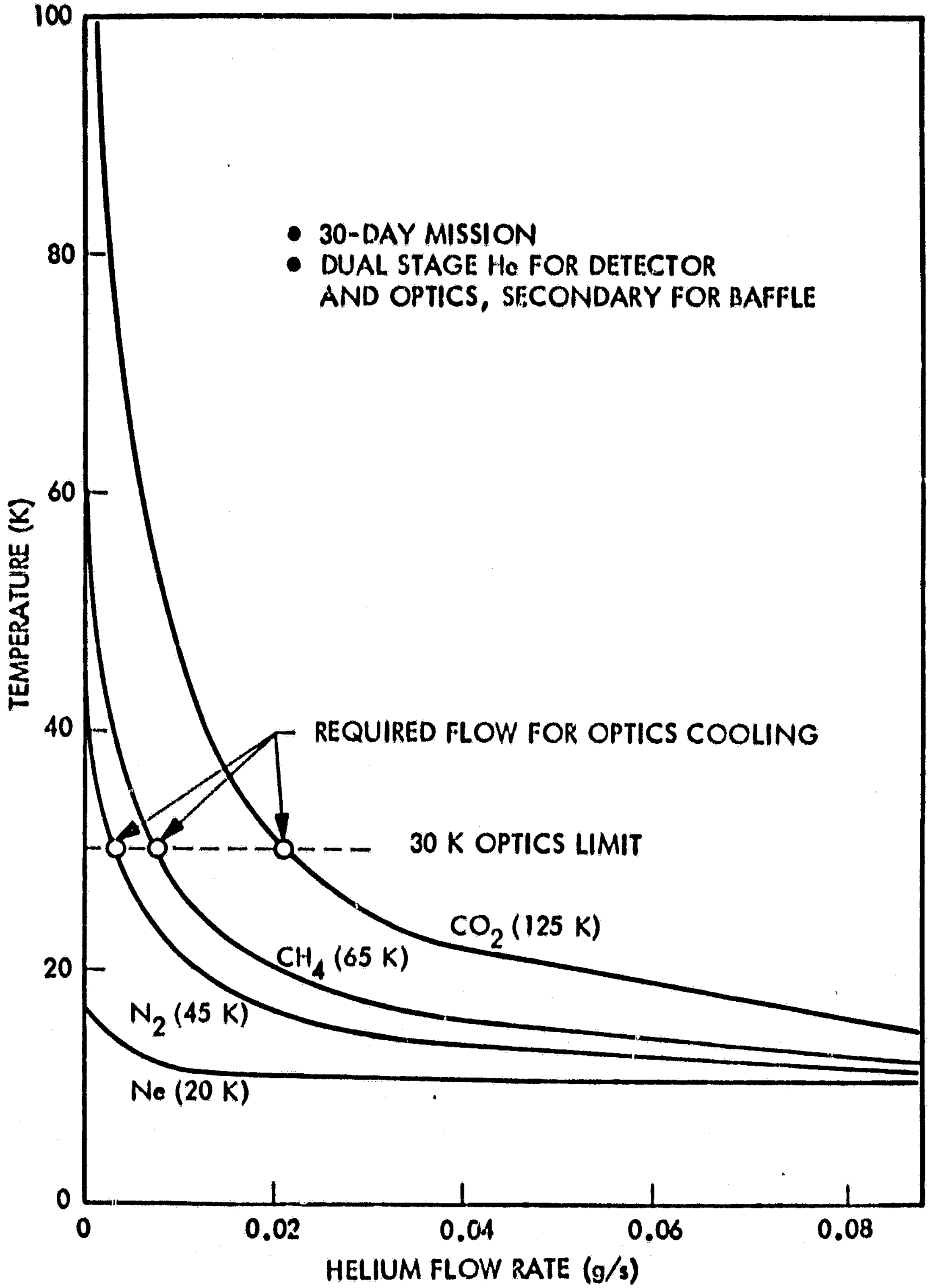


FIG. 3-12 EFFECT OF HELIUM FLOW RATE ON OPTICS TEMPERATURE



The resulting instrument temperatures, cryogen volumes and system weights are summaried in Fig. 3-13 for the various combinations.

The results show the use of solid methane as the secondary with hydrogen or helium leads to the lightest system while meeting instrument temperature requirements. If the use of methane is excluded then the systems utilizing solid nitrogen with helium or hydrogen results in lowest weights, while the use of solid neon results in excessive weights.

3.5 THREE STAGE SYSTEM

The last system considered consists of a separate cryogen for each of the three temperature requirements. A screening process resulted in the selection of a three stage helium/neon/nitrogen system for analysis. The characteristics of that system are presented in Fig. 3-14. The relatively high system weight combined with the additional complexity appear to eliminate this system as a candidate.

3.6 SYSTEM COMPARISON

The relative weights and volumes of the systems investigated are compared in Fig. 3-15.

From the above trade studies, the following results are highlighted:

- The single stage solid hydrogen system with optics cooled to $10^{o}K$ is the minimum weight system at 98 Kg.
- The three stage system $He/Ne/N_2$ leads to the minimum volume at 120 liters of cryogen.
- If helium is assumed for the primary, considering its inert non-flammable properties, the minimum weight system is He/CH_4 at 172 Kg followed by He/N_2 at 197 Kg which also has a small volume of 158L.
- The single stage He system is relatively large and heavy but has advantages in terms of ground handling, safety and state of development

FIG. 3-13 COMPARISON OF CRYOGENIC SYSTEMS FOR CLIR DUAL STAGE SYSTEMS / THREE TEMPERATURE VERSION

GENERAL FEATURES

- DETECTOR COOLED WITH PRIMARY CRYOGEN
- OPTICS COOLED WITH PRIMARY CRYOGEN VENT GAS
- BAFFLE COOLED WITH SECONDARY CRYOGEN

CRYOGENS		TEMPERATURES			CRYOGEN QUANTITY				TOTAL SYSTEM WT.
					PRIMARY		SECONDARY		
PRIMARY	SECONDARY	DET.	OPTICS	BAFFLE	V, L(FT^3)	M, KG(LBS)	V, L(FT^3)	M, KG(LBS)	KG (LBS)
He	Ne	2-10	15	20	42(1.5)	5.3(11.6)	161(5.7)	232(511)	449(989)
He	N_2	2-10	30	45	59(2.1)	7.6(16.6)	99(3.5)	96(211)	197(433)
He	CH_4	2-10	30	65	153(5.4)	19(41.8)	79(2.8)	39(87)	172(378)
He	CO_2	2-10	30	130	442(15.6)	55(122)	23(0.8)	37(82)	329(724)
H_2	Ne	8-10	15	20	2.5(.09)	.23(0.5)	161(5.7)	232(511)	419(922)
H_2	N_2	8-10	30	45	42(1.5)	3.9(8.6)	99(3.5)	96(211)	184(405)
H_2	CH_4	8-10	30	65	119(4.2)	11(23.8)	79(2.8)	39(87)	145(320)
H_2	CO_2	8-10	30	130	329(11.6)	29(64.8)	23(0.8)	37(82)	239(527)

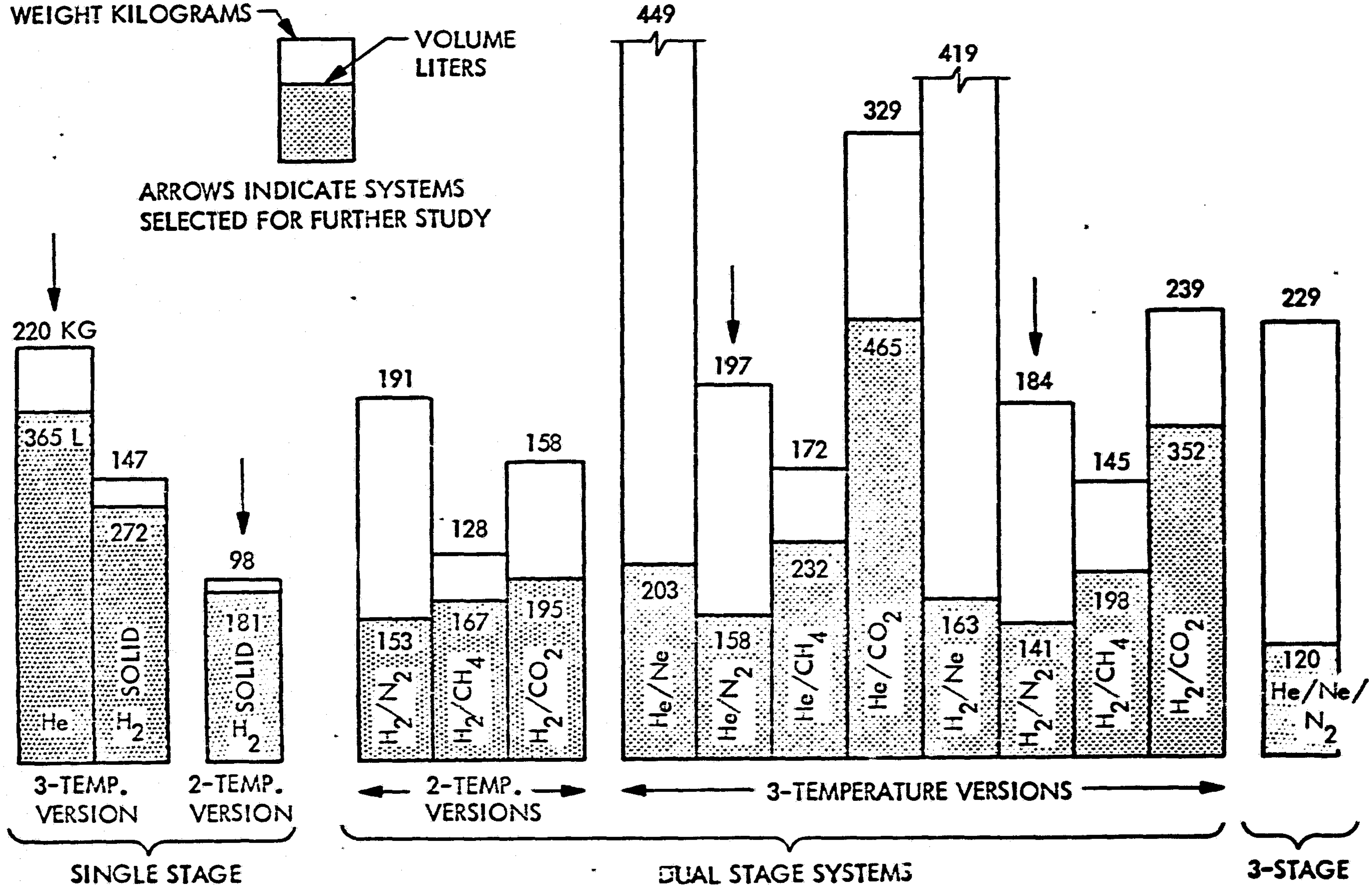
FIG. 3-14 COMPARISON OF CRYOGENIC SYSTEMS FOR CLIR
III. THREE CRYOGEN SYSTEMS

GENERAL FEATURES

- DETECTOR, OPTICS, AND BAFFLE EACH COOLED WITH INDIVIDUAL CRYOGEN

CRYOGENS			CRYOGEN WTS.			TOTAL SYSTEM	
PRIMARY	SECONDARY	TERTIARY	PR.	SEC.	TERT.	VOL.	WT.
He	Ne	N_2	8.6(19)	15(33)	96(211)	98(6.3)	229 (504)

FIG. 3-15 SUMMARY OF WEIGHT AND VOLUME OF VARIOUS SYSTEMS (HEAT RATE ITERATION NO. 2)



3.7 REFINED TRADE STUDIES

These considerations have led to a selection of some of these systems for further comparative analysis. Fig. 3-16 summarizes these systems and some of their considerations.

The analysis which follows incorporates heat rates which have evolved from firming up the instrument requirements, and also includes thermal coupling between the cooler and the instrument.

These new heat rates are shown in Fig. 3-17 for the three temperature regions and also for two optical bench temperatures. These heat rates and following analysis will be referred to as "Iteration 4".

The thermal net-work utilized is shown in Fig. 3-18. This net work includes thermal resistance values for both the instrument and the cooler. It is also assumed that the temperature zones corresponding to the optics and baffle extend around the cooler, so that we are incorporating vapor cooling of the cryogen tank also. In addition, a third vapor cooled shield which surrounds both the instrument and the cooler is assumed so that the sensible enthalpy of the vapor can be more efficiently used.

Within the framework of these assumptions the selected systems were analyzed and the results are shown in Fig. 3-19. The results show substantially higher weights and volumes than the previous comparison, a consequence of the higher instrument heat rates for this iteration. The relative weight and volume of the systems has not changed, that is the systems increased by a fairly constant percentage.

The effect of helium flow rate on instrument temperature is shown in Fig. 3-20. It can also be seen that the new heat rates have resulted in the baffle temperature setting the required flow rate whereas in the prior iteration (No. 2) the optics set the requirement. The resulting optics temperature is 24°K.

FIG. 3-16 SELECTED CRYOGEN SYSTEMS FOR FURTHER STUDY

SYSTEM	SYSTEM WT. (KG)	CRYOGEN VOLUME (ℓ)	COMMENTS
SINGLE-STAGE He	220	365	GROUND HANDLING RELATIVELY SIMPLE, TEMPERATURE CONTROL RELATIVELY COMPLEX
SINGLE-STAGE SOLID H_2	98	181	MINIMUM WEIGHT SYSTEM; DETECTOR AND OPTICS COOLED TO 10 K
DUAL-STAGE He/ SOLID N_2	197	158	LOW-WEIGHT, LOW-VOLUME SYSTEM

FIG. 3-17 CLIR INSTRUMENT HEAT LOAD BREAKDOWN (ITERATION 4)

	WATTS	
A. RADIATION SHIELD (100 K)		
1. SUPPORT	2.50	
2. MLI	1.86	
3. COND PENET	0.21	
4. DOOR PLATE	0.05	
5. EARTH LOAD (AVERAGE) (400 km ALT)	8.66	
	(30 K)	(45 K)
B. OPTICAL BENCH		
1. SUPPORT	0.5	0.41
2. WIRES	0.042	0.034
3. RADIATION (MLI)	0.074	0.072
4. EARTH LOAD (AVERAGE)		
a. PRIMARY MIRROR	0.020	0.020
b. APERTURE PLATE	0.178	0.178
	(7 K)	(7 K)
C. FOCAL PLANES		
1. I/F		
a. BIAS	0.001	0.001
b. SUPPORT	0.006	0.010
c. WIRES	0.0015	0.002
2. RADIOMETER		
a. BIAS	0.022	0.022
b. SUPPORT	0.006	0.010
c. WIRES	0.028	0.046

FIG. 3-18 THERMAL NETWORK FOR ITERATION NO. 4 STUDIES

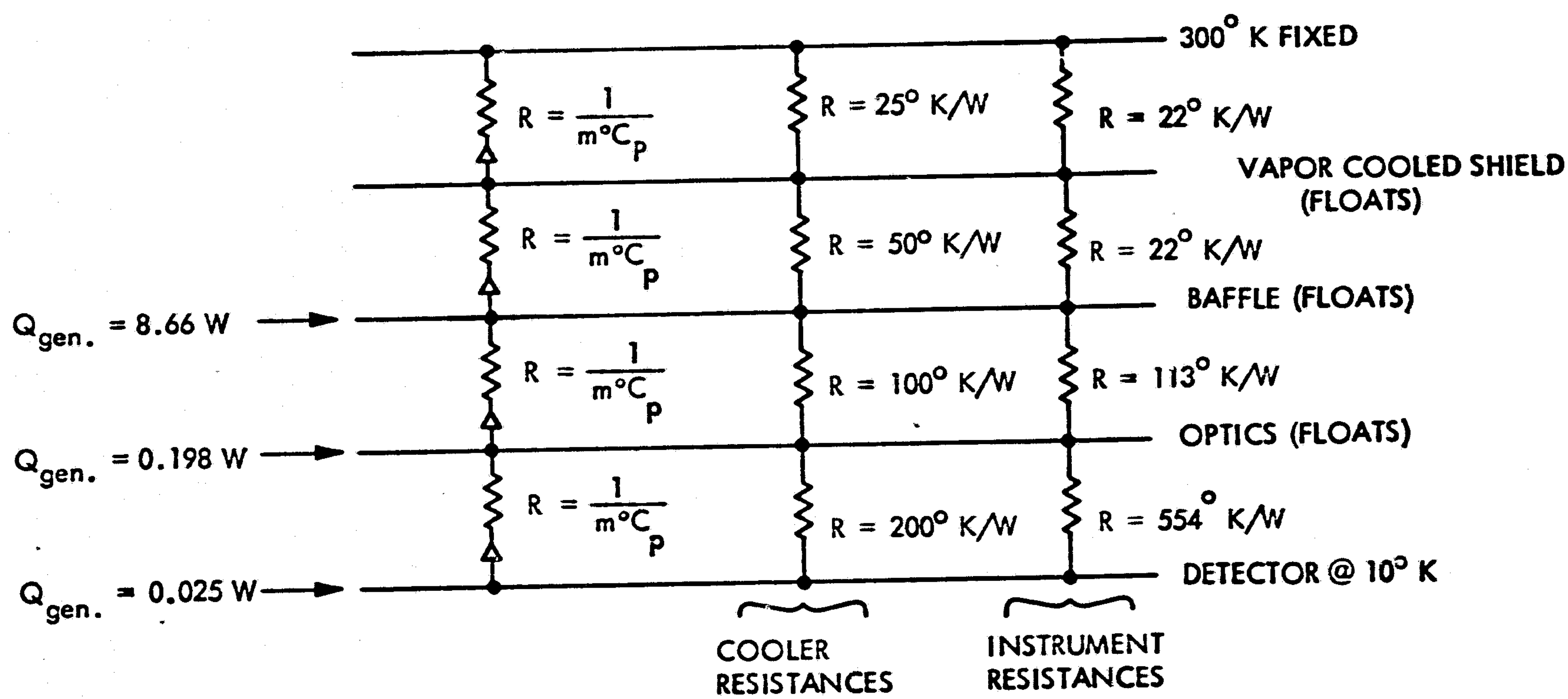
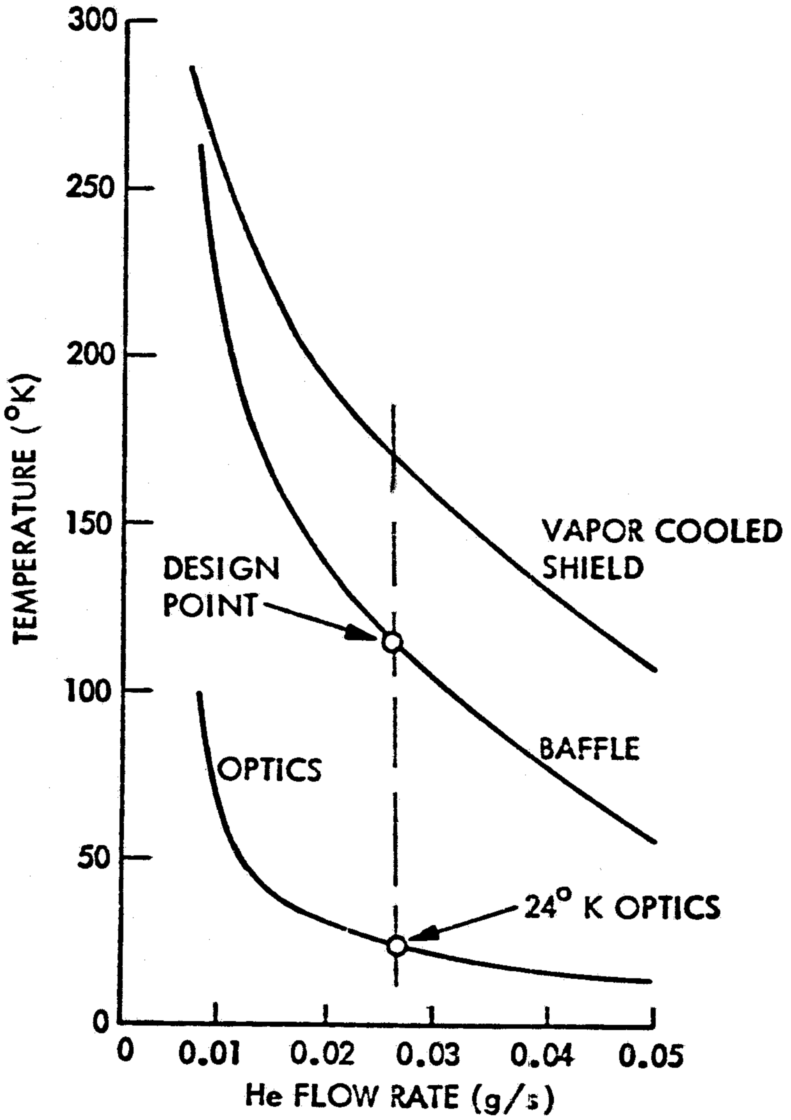


Fig. 3-19 CLIR CRYOGEN SYSTEM WEIGHTS FOR ITERATION NO. 4, 30 DAY MISSION

SYSTEM	REQUIRED CRYOGEN FLOW RATE gms/sec	PRIMARY CRYOGEN MASS, KG	SECONDARY CRYOGEN MASS, KG.	PRIMARY CRYOGEN VOLUME, L	SECONDARY CRYOGEN VOLUME, L.	SYSTEM WT, KG*
SINGLE STAGE He, $T_{baffle} = 115° K$ $T_{optics} = 24 K$	.0265 (0.21 lbs/hr) $Q_{avg.} = 0.54W$	69 (151 lbs)	NA	549 (19.4 Ft^3)	NA	330 (725 lbs)
SINGLE STAGE SOLID H_2, $T_{baffle} \cong 115° K$ $T_{optics} = 10° K$	.00883 (0.07 lbs/hr)	23 (50.4 lbs)	NA	254 (9 Ft^3)	NA	137 (302 lbs)
He/SOLID N_2 $T_{optics} = 30°K$, $T_{baffle} = 45° K$	0.003 (0.023 lbs/hr)	7.73 (17 lbs)	188 (414 lbs)	60 (2.1 Ft^3)	191 (6.8 Ft^3)	375 (827 lbs)

* SYSTEM WEIGHTS CONTAIN 20% MARGIN FOR UNCERTAINTY IN HEAT RATE PREDICTIONS

FIG. 3-20 CLIR SINGLE STAGE HELIUM SYSTEM
30-DAY MISSION ITERATION NO.4



3.8 UAR ONE YEAR MISSION

The 30-day mission is the baseline duration for the CLIR instrument. In addition to CLIR a follow-on mission which is under study is UARS which would be a free flyer mission with a one year duration. This mission was briefly studied primarily to determine the commonality between the cooling system for CLIR and UARS.

It was assumed in this analysis that the instrument was the same and resulted in the same heat rates. It was further assumed that the mission duty cycle was 10%.

The principal results from this study are shown in Fig. 3-21. The results again indicate that the solid hydrogen system which cools both detector and optics to 10°K is the minimum weight minimum volume system. The analysis utilized the same thermal network as the previous comparisons and results were based on average heat rates with 10% operation in which heat was generated for the appropriate elements and 90% operation in which no heat was generated. Fig. 3-22 shows the temperature of the baffle and the vapor cooled shield vs. flow rate for both the operating and non-operating conditions. During operation a flow rate of 0.009 gms/sec is required to provide adequate cooling of the baffle, while during non-operating periods (no heat generation, no aperture load) the parasitic heat load results in a flow rate of .0037 gms/sec and a temperature of 100°K for the optics and 170°K for the vapor cooled shield. This is fortunate, because it leads to a fairly small temperature adjustment when the instrument is turned on, with resulting small transient effects.

These studies for the 1 year UARS mission must be considered very preliminary, as the transient operation requires more study, and better thermal isolation techniques can probably be devised with a resultant reduction in system weight. In addition, the instrument configuration may change due to the pointing system requirements in combination with the free flyer operation. The results

FIG. 3-21 SYSTEM WEIGHTS FOR ONE YEAR UARS MISSION, HEAT RATE ITERATION NO. 4, 10% DUTY CYCLE

CRYOGENS		TEMPERATURES			CRYOGEN QUANTITY				SYSTEM WT.*
					PRIMARY		SECONDARY		
PRIMARY	SECONDARY	DET.	OPT.	BAFFLE	V,L	M, KG.	V,L	M, KG.	KG.
H_2	NONE	10	10	115	1303	118	NA	NA	709
He	NONE	10	24	115	2600	236	NA	NA	1136
He	N_2	10	30	45	1047	95	639	627	1589
H_2	N_2	10	10	45	891	22.7	639	627	1127

*CONTAINS 20% WEIGHT INCREASE FOR DESIGN MARGIN.

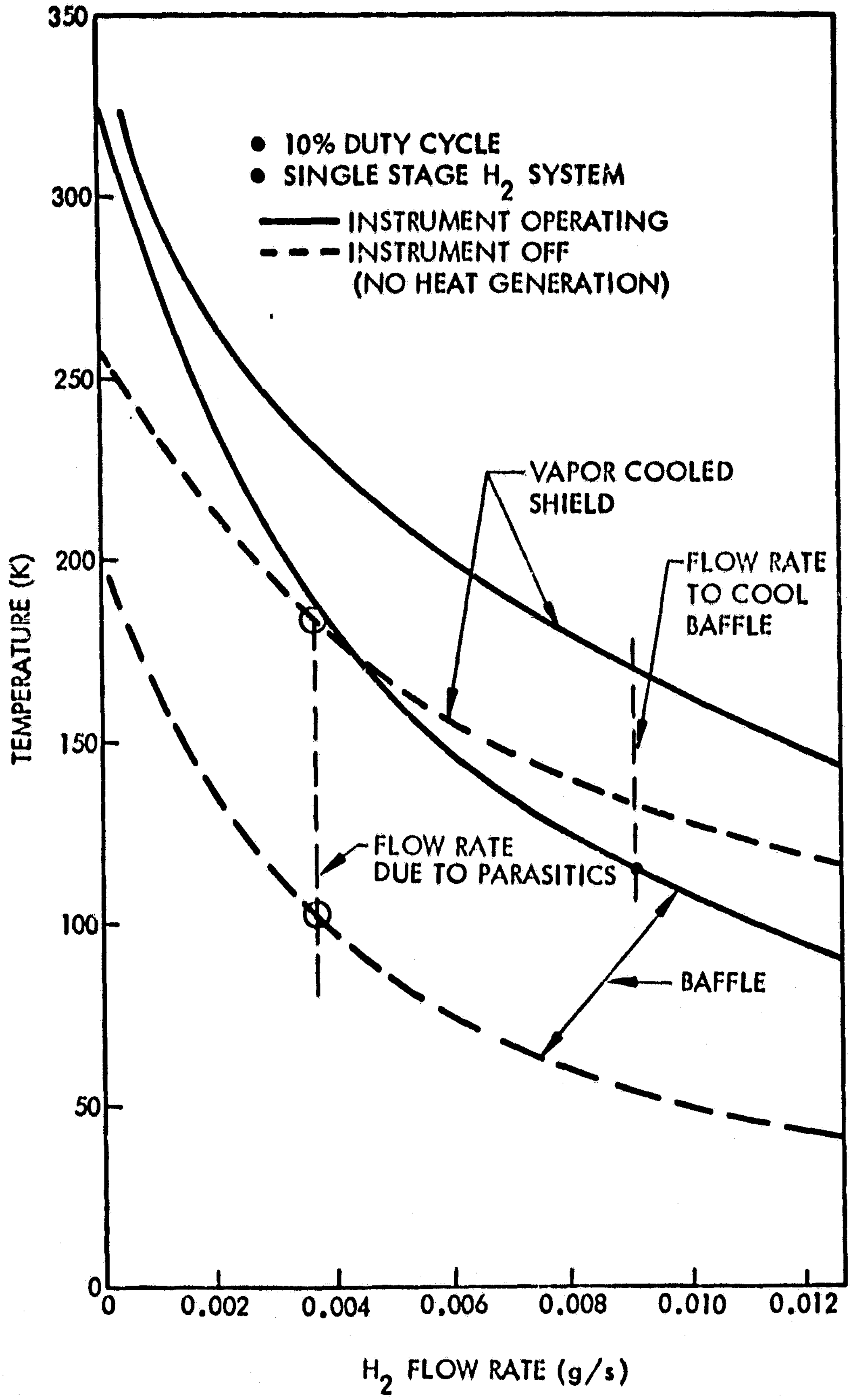


FIG. 3-22 EFFECT OF FLOW RATE ON INSTRUMENT TEMPERATURES FOR ONE-YEAR UARS MISSION

may be considered as a first estimate of size and weight, and it appears that a substantially larger cooling system could be required for UARS than CLIR. Solid hydrogen appears to be the most attractive candidate for this mission, and substantial weight reductions could be realized by utilizing the endothermic para to ortho reaction. This reaction and consequent heat absorption can be obtained by passing the hydrogen vent gas over the proper catalyst. This approach requires additional study to determine the associated weight reduction, and increased complexity.

3.9 BASELINE CRYOGEN SELECTION

A baseline selection was made in the program in order that a representative cryogenic system could be established in some detail. Fig. 3-23 summarizes the various considerations on which the selection was based for CLIR. Although several numerical rating approaches were made, they are not presented here because of their subjective nature.

Safety for the man rated vehicle is a paramount consideration, and had a strong influence in the selection. This was one of the primary considerations which led to the selection of helium for the baseline over the lighter and smaller hydrogen systems. The ground handling of the helium system was judged to be much simpler and safer than for the hydrogen system or for other cryogen combinations. In the operation of the helium system, it can be filled with normal boiling point liquid helium, while the other cryogens would require more elaborate operations and GSE to load and maintain on the ground.

Orbital reliability for the helium system was rated lower than for some of the other systems, since operation with supercritical helium would require an absolute pressure relief valve and a temperature sensor/heater feedback loop, items which are not required for operation of some of the totally passive solid cryogen systems.

Orbital temperature stability for those systems which thermally ground various temperature zones of the instrument to a solid cryogen stage have far better

FIG 3-23. CRITERIA FOR SELECTION OF CRYOGENIC SYSTEM FOR CLIR

1. SAFETY
2. GROUND HANDLING
3. ORBITAL RELIABILITY
4. DEVELOPMENT RISK
5. RELATIVE COST
6. ORBITAL TEMPERATURE STABILITY
7. CONTAMINATION RESISTANCE
8. WEIGHT
9. VOLUME
10. AFFECT ON INSTRUMENT DESIGN
11. TECHNOLOGY APPLICABLE TO UARS MISSION

temperature stability than those systems which utilize vapor cooling or supercritical helium for cooling. Orbital temperature stability of a few tenths of a degre has been demonstrated by solid cryogen coolers for as long as 7 months of orbital operation.[2]

Development risk must be considered to be relatively small for the supercritical helium system since several of these systems have been built and flown, at least for short duration flights. It is not felt, however, that the development risk associated with a solid hydrogen system is high, since many different cryogens have been loaded and operated in the solid state without problems which were unique to the particular cryogen selected.

Considerations of contamination resistance lead to the flexibility of the various systems to allow intermittent warm-up of optical elements to allow outgassing of contaminants, which may have cryopumped during operation. Systems utilizing vent gas cooling have the potential for stopping or bypassing the cooling gas flow for an intermittent period to allow warm-up of optical elements whereas the solid cryogens may not without thermal switching.

One system which appears to have a significant affect on instrument design is the solid hydrogen system in which the optics and detector are both thermally grounded to the hydrogen. This may result in somewhat simplified instrument design since the detectors may be firmly attached to the optical bench, allowing isothermal operation of these elements.

For the UARS mission it appears that solid hydrogen may be necessary to minimize system weights, therefore technology and experience developed on CLIR would be beneficial for UARS although it appears that the cooling systems would be substantially different in size and weight.

These considerations, with emphasis on safety and ground handling led to a selection of a single stage helium cooler for the baseline on CLIR. The following sections describes this baseline.

4.0 BASELINE SYSTEM

4.1 OPTIONS STUDIED

Several options were considered before the selection of the baseline parameters. The principal options and their results are discussed in the following sections.

4.1.1 Vapor Cooling

Because of the large specific heat of the helium vent gas, the manner in which it is utilized both in the instrument and to defeat parasitic heat loads in the helium dewar assembly is of paramount importance. Various approaches for utilizing the vent gas were investigated for the system. In the comparison of these systems the thermal net-work shown in Fig. 3-18 was utilized with various vapor cooled shield combinations. Fig. 4-1 presents the results of these studies.

The first case (No. 1) presents the arrangement assumed for the iteration No. 4 heat rates, in which it was assumed that both the cooler and instrument utilized vapor cooled shields at three locations. The optics and baffle cooling shields were assumed to extend over to the cooler where they surrounded the cooler and were used to defeat parasitic heat loads to the helium dewar. In addition, a third vapor cooled shield was assumed to surround both the instrument and the helium tank. The resulting instrument temperatures and the parasitic heat load to the helium are summarized in the table. This system leads to a fairly complex system because of the large number of shields.

The second case shows the same configuration except the shields from the optics and baffle were not utilized in the cooler section.

The results show an overall improvement in all features of the system. The instrument temperatures are lower, the parasitic heat load to the helium dewar is reduced and the system complexity is considerably reduced.

FIG. 4-1. EFFECT OF VAPOR COOLING CONFIGURATION

Case	No.	$\dot{m}$ Lbs./Hr	Instrument Temp, ^{o}K			Cooler Shield Temp, ^{o}K			Heat Load to Helium Dewar, W	Comments
			Optics	Baffle	Shield	Optics	Baffle	Shield		
Iteration #4 Heat Rates	1	0.21	24	110	168	24	115	174	0.54	Cooled shields from optics and baffle extended around cooler.
No vapor cooled shields from instrument	2	0.20	17	100	163	no	no	163	0.46	Elimination of optics and baffle shielding from cooler improves system.
Vapor cooled shield @ 1/3 MLI thickness	3	0.20	18	110	no	no	no	152	0.43	Removal of vapor cooled shield from instrument increases instrument temperature slightly.

The third case considered elimination of the vapor cooled shield in the instrument but retention of the single vapor cooled shield around the cooler. The results show a slight increase in the instrument temperatures, and somewhat lower helium dewar heat loads.

It was felt that the simplified instrument configuration was sufficient compensation for the slightly higher instrument temperatures and this vent shield configuration was selected for the baseline design.

Additional studies of the vapor cooling arrangement with a more detailed thermal model should be performed to substantiate these results.

4.1.2 Liquid or Supercritical Storage of Helium

The heat absorption per pound of helium is approximately the same for liquid helium, supercritical helium or superfluid helium. The primary differences are related to the storage pressure which affects system weight through tank pressure and the techniques of controlling the fluid in the space environment without loss of cooling efficiency. The storage of the helium in the supercritical state eliminates problems associated with phase separation since the supercritical helium exists as a single fluid phase.

In this study an extensive analysis of the fluid management of liquid helium has not been performed although techniques and systems have been developed which have been successfully used(4) and are being developed.(5)

To determine if a significant weight savings could be achieved with the lower pressure (1 atm) liquid helium system a weight comparison was made with the 60 psia supercritical system. The weight summary of these two systems is shown in Fig. 4-6. The comparison of the systems was based on the assumption of the same size tank volume for both configurations.

The results show that the liquid helium system is 58 Kg lighter than the supercritical system. This difference is due to the lighter tank weight and lower residual helium mass for the supercritical system.

4.1.3 Supercritical Pressure Selection

In selecting the operating pressure for the supercritical baseline system, the effect of pressure and ullage % on the following system parameters was considered.

- Ground hold
- Tankage weight
- Residual cryogen weight
- Proximity to subcritical operating regime

The ground hold period is defined as the time from fill of the tank with normal boiling point liquid helium to the desired tank volume to the time at which venting of the helium starts. Long hold times result from high tank pressures and large ullage volumes. The hold time varies for example from 50 hours for a 50 psia 10% ullage condition to 92 hours for a 70 psia 20% ullage.

The ground hold conditions must be further examined in terms of the instrument requirements since it may be necessary to cool the instrument prior to launch, and this would require vent gas cooling from the tank or auxilliary cooling.

The effect of operating pressure on tankage weight and residual helium weight favor minimum operating pressure and ullage %. In this case an optimum pressure does not exist, the minimum pressure results in minimum weight. A limited trade-off analysis indicates that the weight difference between a 50 psia 10% ullage condition (just above critical temperature) and a 70 psia 10% ullage condition leads to a weight difference of 45 lbs of system weight.

The consideration which limits the operating condition is to assure that supercritical conditions are maintained in the tank during all operating conditions in orbit or during launch so that liquid formation is prevented. In order to determine safe operating conditions to achieve this requires detailed analysis of the venting system with consideration of operating characteristics of the absolute pressure relief valve and temperature/heater feed-back loop with their associated repeatability and tolerances.

With these considerations in mind an operating pressure of 60 psia and a 10% ullage volume was selected.

4.1.4 Mission Abort Operations

Should an anomaly occur during flight (not payloaded concerned) that would necessitate an early return of the shuttle to earth, the question arises as to how the helium should be handled. Fig. 4-2 summarizes some of the possibilities which have been considered.

Probably the simplest and most desirable would be to return with the helium in its normal condition. The cryogen would continue venting, maintaining the internal tank pressure at 4.1×10^5 nt/m^2 (60 psia). The possibility does exist for a slight increase in the vent rate during the return due to an increased tank heating caused by random vibration oscillations but the effect should be minimal. At present, it is felt that return from orbit with a full measure of helium will not compromise any safety standards over those employed during the launch, and it appears to be the best approach.

Other approaches call for the cryogen to be dumped from the cooler prior to reentry. Controlled dumping of the cryogen can be accomplished by increasing the heat load to the dewar by auxiliary heating. With this technique, removal of 90% of the stored cryogen (m_o = 80 kg) can be accomplished by supplying 661 w-hr of energy (331 watts for 2 hrs or 66 watts for 10 hrs), and venting through the vapor cooled shield vent line. Some disadvantages do exist. First, the long length of plumbing ($\sim$ 16 meters) coupled with the high venting rate causes an increase in the pressure differential along the lines so as to raise the tank pressure to between 4.3×10^5 n/m^2 (62 psia) and 5.3×10^5 nt/m^2 (77 psia), the uncertainty being due to the uncertainties in the gas temperature.

Second, flow rates (11.1 gm/sec) may be too large for a single absolute pressure relief valve. This matter requires further study.

For rapid dumping, which is the least desirable method, an explosive pyrotechnic valve or solenoid can be installed on both the helium fill and vent line (for redundancy) to open the dewar to space. After opening the pyro valve, vapor

Fig. 4-2 MISSION ABORT CRYOGEN OPTIONS

Return Mode	Vent Rate	Comments
Return with cryogen stored in dewar at normal conditions	Normal rate (2.5×10^{-2} gm/sec)	- No special equipment required - Tank pressure at 60 psia (4.1×10^5 nt/m^2)
Controlled cryogen dump, without liquid formation	2.3-11.1 gm/sec for dump in 2-10 hours	- Requires 661 w-hr heater input to cryogen tank (331 w @ 2 hrs or 66 w @ 10 hrs). - Tank pressure to rise between 4.3×10^5 nt/m^2 (62 psia) and 5.3×10^5 nt/m^2 (77 psia)
Cryogen dump, initiated by opening valve directly to space	Highly variable	- Two phase liquid-vapor withdrawal will result

venting will take place until the pressure drops below 1.9×10^5 n/m^2 (27.5 psia) at which time two phase liquid/vapor withdrawal begins. If the tank pressure is allowed to drop to less than 680 nt/m^2 (0.1 atm) then over 90% of the cryogen will be removed. This approach is difficult to control and analyze and does not appear to have any advantage over the other approaches.

4.2 BASELINE DESCRIPTION

4.2.1 Summary

Fig. 4-3 summarizes some of the major thermal, mechanical, and operating considerations for the cooler system.

After an extensive parametric study into possible single, and dual stage, cryogen systems, supercritical helium was selected as the baseline coolant for this study. To maintain the cryogen in the supercritical state, the tank is maintained at a pressure of $1x10^5$ nt/m^2 (60 psia) during withdrawal. This is achieved by an absolute pressure relief valve located at the exit of the tank vent line. Before leaving the cooler, the cold vent gas is circulated through the Honeywell instrument to provide the necessary cooling to the detector focal planes, optical bench, and a radiation baffle as well as a vapor-cooled shield located around the helium tank to reduce the parasitic heat load into the cooler. The total loaded mass of the cooler is 197 Kg.

The design flow rate required to maintain the optical elements at 30^oK was found to be $2.5x10^{-2}$ gm/s (0.2 lb/hr). To provide this flow rate, auxiliary heating will have to be supplied to the dewar by a resistance heating element, located on the tank wall. Since the required heat input for a constant vent rate varies as a function of the cryogen volume remaining (density), a feedback control loop is required to provide the proper experiment temperature and provide the necessary heat input. The required heater power will vary between a minimum of 0.12 watts (with the tank at 6^oK) and a maximum of 1.10 watts (with the tank at 12^oK).

For initial fill, the tank will be filled to a 10% ullage with normal boiling point helium. The fill and vent lines are closed and the system is allowed to self-pressurize to $4.1x10^5$ nt/m^2 (60 psia) which will take approximately 68 hours. After venting begins, the expected steady state parasitic heat load to the dewar will be 415 mw. This fill procedure leads to a simple ground support system and simple handling when compared with filling with supercritical helium at high pressure.

Fig. 4-3

CLIR COOLING SYSTEM SUMMARY (SUPERCRITICAL HELIUM)

MECHANICAL

• OVERALL LENGTH (INCL. PLUMBING)	210 cm
• OVERALL DIAMETER (INCL. VACUUM SHELL)	95.5 cm (37.6 in.)
• CRYOGEN TANK VOLUME	701 liters (24.8 ft^3)
• DRY WEIGHT (INCL. 20% MARGIN)	117.1 kg (257.9 lb)
• USABLE HELIUM WEIGHT	68.1 kg (150.0 lb)
• RESIDUAL HELIUM WEIGHT	11.9 kg (26.3 lb)
• TOTAL LOADED WEIGHT	197.1 kg (432.2 lb)
• PRIMARY RESONANCE	28 Hz
• DESIGN SAFETY CRITERIA	
– SUPPORT TUBE	SURVIVE QUAL RANDOM WITH 3 σ PROBABILITY
– HELIUM TANK	SAFETY FACTOR = 4 (BURST)
– VACUUM SHELL	SAFETY FACTOR = 2 (BUCKLING)

THERMAL

- 6.4 cm SILK NET/DOUBLE ALUMINIZED MYLAR INSULATION
- SINGLE VAPOR COOLED SHIELD AT 117.6 K
- HEAT LOAD TO HELIUM 0.415 W

OPERATING CONDITIONS

- SUPERCRITICAL HELIUM CRYOGEN
- CONSTANT PRESSURE WITHDRAWAL AT 4.1×10^5 nt/m^2 (60 psia)
- REQUIRED FLOW RATE TO INSTRUMENT 2.5×10^{-2} gm/s (0.2 lb/hr)
- REQUIRED HEAT INPUT FOR WITHDRAWAL OF 2.5×10^{-2} gm/s HELIUM: 0.12 W AT 6°K; 1.10 W AT 12°K
- FILLING MODE
  - FILL TO 90% (MIN) WITH NBP HELIUM
  - SELF PRESSURIZE TO 4.1×10^5 nt/m^2 (60 psia)
- GROUND HOLD TIME (FROM END OF FILL TO START OF VENT) 68 hr

4.2.2 Cooler Construction

A layout showing hardware detail of the CLIR cooler is shown in Figure 4-4 .

Helium Tank

The helium tank is a 79.5 cm dia by 87.9 cm long 6061 T4 aluminum cylinder. Hemispherical domes are welded to each end, providing a 701 liter total volume. The wall thickness of the tank including the dome ends is 0.28 cm. For additional structural support along the cylindrical portion of the tank, 6 integrally machined 0.53 cm by 1.42 cm high ring stiffeners are provided internal to the tank.

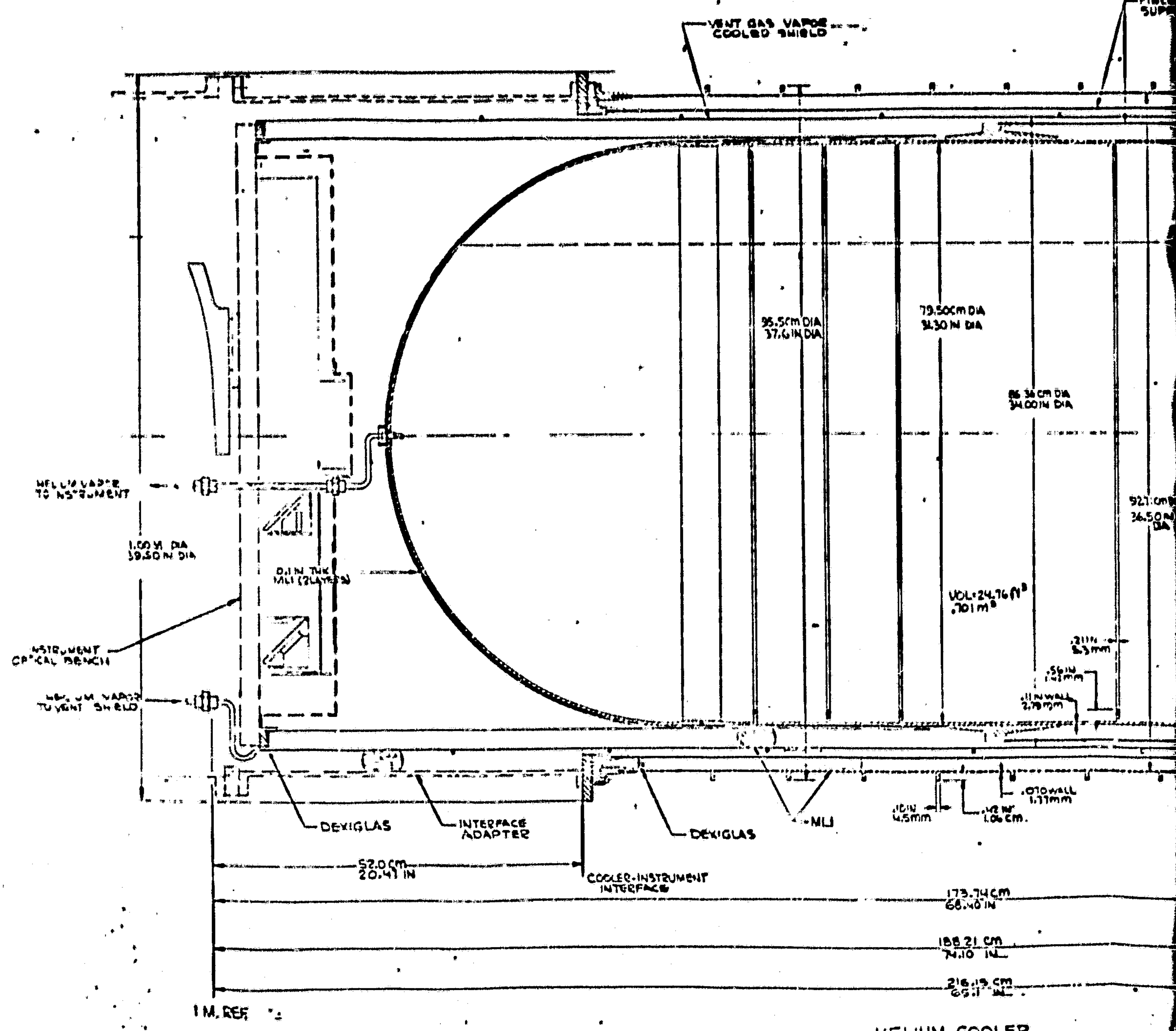
Support Tubes

The cooler assembly is held in place by a two-piece folded fiberglas tube assembly. The inner tube which connects to the center of the cooler is 84.4 cm diameter by 37.1 cm long and the outer tube which interfaces with the honeywell mounting plate is 89.1 cm diameter by 90.7 cm long. Each tube is 0.152 cm thick.

Vapor-Cooled Shield

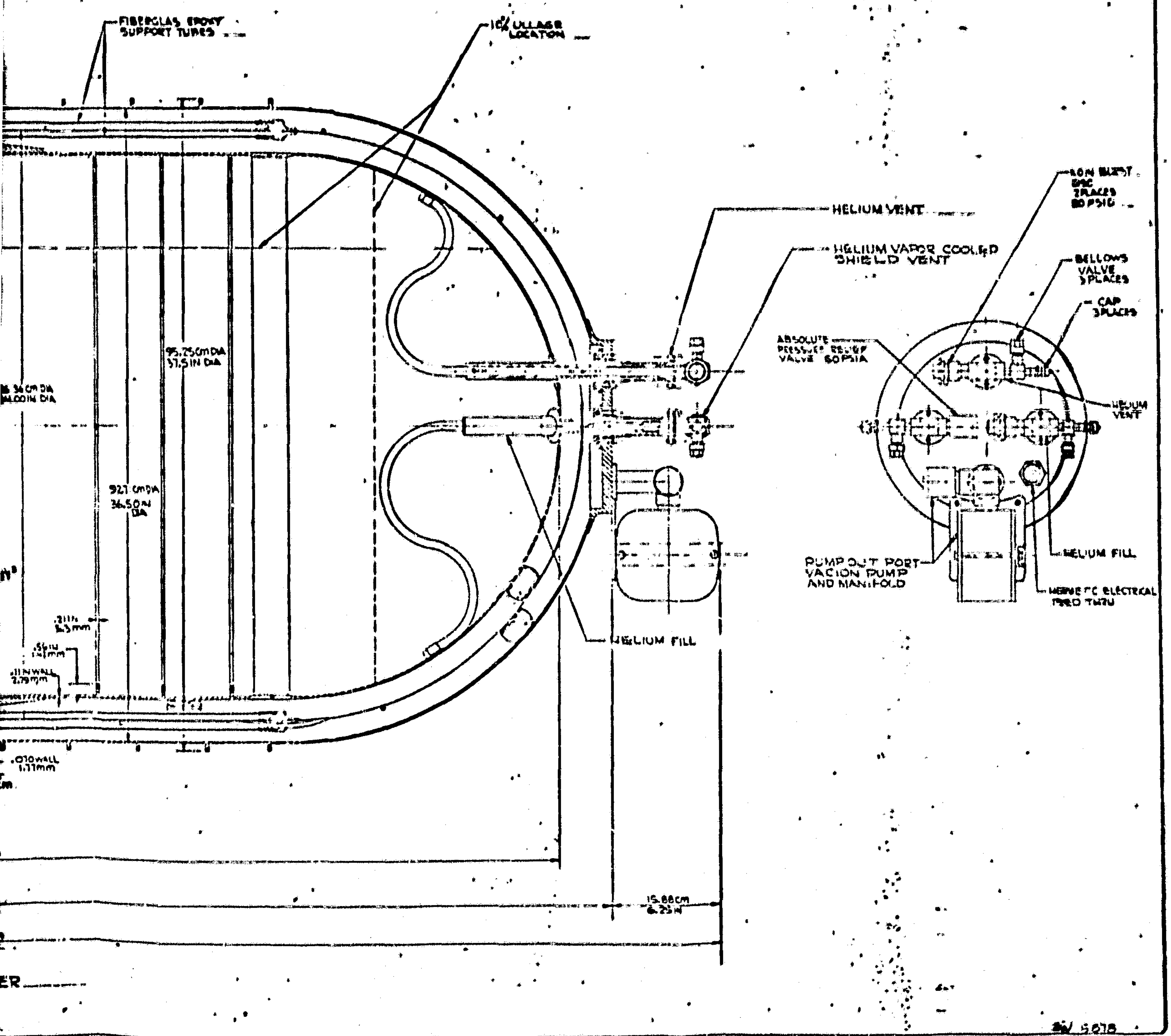
At the interface point between the two fiberglas tubes a single vapor cooled shield is thermally grounded. The shield which is located 3.1 cm from the helium dewar follows the contour of the dewar over the cylindrical and rear portion of the cooler. Around the front portion of the cooler, or that portion nearest the Honeywell experiment, the shield continues cylindrically out to a point near the experiment's optical bench. The plumbing interface with the vent shield is made by a Gamah fitting (or equivalent) in an area still to be defined.

FIG. 4-4 BASELINE COOL



FOLDOUT FRAME

ASELINE COOLER LAYOUT



FOLDOUT FRAME 2

Vacuum Shell

A vacuum shell surrounds the entire cooler assembly. The shell, which mounts to the Honeywell interface at the 1.52 meter reference point, is a 0.18 cm thick 6061-T6 aluminum shell reinforced by 9 each 0.45 cm by 1.06 cm high integrally machined external ring stiffeners. The shell includes a single hemispherical end dome upon which a 25.4 cm diameter by 0.79 cm thick access mounting flange is welded in place.

Insulation

In the region between the cooler front end and the extended vent gas shield, a thin 0.03 cm aluminum retainer is wrapped with silk net/double-aluminized mylar insulation out to a thickness of the vent gas shield. This retainer is mounted to the Honeywell optical bench, providing an environment temperature of approximately 30^{o}K to the front end of the cooler. The relative position of the retainer within the insulation wrap is selected to impose no additional net heat flow to either the cooler or to the optics bench. This requires only two layers of MLI to be wrapped about the front end of the dewar. The area outside the vent gas shield is wrapped out to the vacuum shell, as is the area between cooler and vacuum shell along both the cylindrical and back section of the cooler, making for a total insulation thickness of 6.4 cm about the entire cooler area.

Plumbing

Attached to the rear mounting flange are the helium fill, helium vent and vapor cooled shield vent line, as well as a vacuum space pumpout line and electrical feedthru. The pumpout line to the insulation space consists of a 2.5 cm diameter Cryolab valve to provide access to an auxiliary sorption pump system and an 8 ℓ/s vacion pump to ultimately maintain pressures in the insulation space at less than 10^{-5} torr. The electrical feedthru provides access to tank thermometry, several redundant strain gauges used to monitor the tank internal pressure, and a heater located on the tank to control the cryogen venting rate.

The helium fill and vent lines utilize a 1.3 cm diameter by 0.013 cm thick convoluted stainless steel tubing between the warm and cold end points. The dewar side of the tubing is brazed to an aluminum/stainless steel transition tube which protrudes into the cooler to provide a longer heat leak path. To further reduce the heat leak into the dewar, both lines are thermally grounded to the vapor cooled shield at an intermediary point. The fill and vent lines continue to the dewar side walls where they are clipped in place at a point which allows for filling to within a 5% ullage in both a horizontal or vertical tank position. On the outside of the tank, the lines are supported by a bayonet fitting capped by a 0.95 cm diameter Nupro metal bellows type access valve in parrallel with a 2.5 cm diameter, 5.5×10^5 nt/m^2 (80 lb/in^2) differential burst disk. This level is set above the 4.1×10^5 nt/m^2 (60 lb/in^2) absolute pressure of the tank to insure no venting of the cryogen occurs through these lines. Instead, venting of the cryogen is accomplished through the vapor flow vent line.

A schematic of the plumbing is shown in Figure 4-5. The optics and baffle temperatures are monitored and maintained constant through a cryogen tank heater feedback control loop to control helium vent rate. Supercritical helium is vented through the two detector focal planes maintaining temperatures to less than 13°K. The cold vapor continues through Honeywell supplied hardware, cooling the optics box to 30°K and a radiator baffle to 104°K. From the radiative baffle the cooled vapor is directed to the cooler vapor cooled shield which is driven to 117.6°K. The gas is then directed to a room temperature heat exchanger and finally exited from the cooler through a 4.1×10^5 nt/m^2 (60 lb/ft^2) absolute pressure relief valve which controls the pressure of the tank. In parallel with the pressure relief valve is a Nupro metal bellows-type valve which can be used to blow the tank down to one atmosphere pressure for refill operation on ground.

Weight

A weight summary of the main components required for the CLIR cooler is shown in Figure 4.6. A total cooler system weight of 197 Kg (434 lbs) is predicted for the supercritical helium cooler of which 80 Kg (176 lbs) or 40.6% is due to the helium cryogen. Also shown are the calculated weights if

FIG. 4-5 COOLER PLUMBING SCHEMATIC

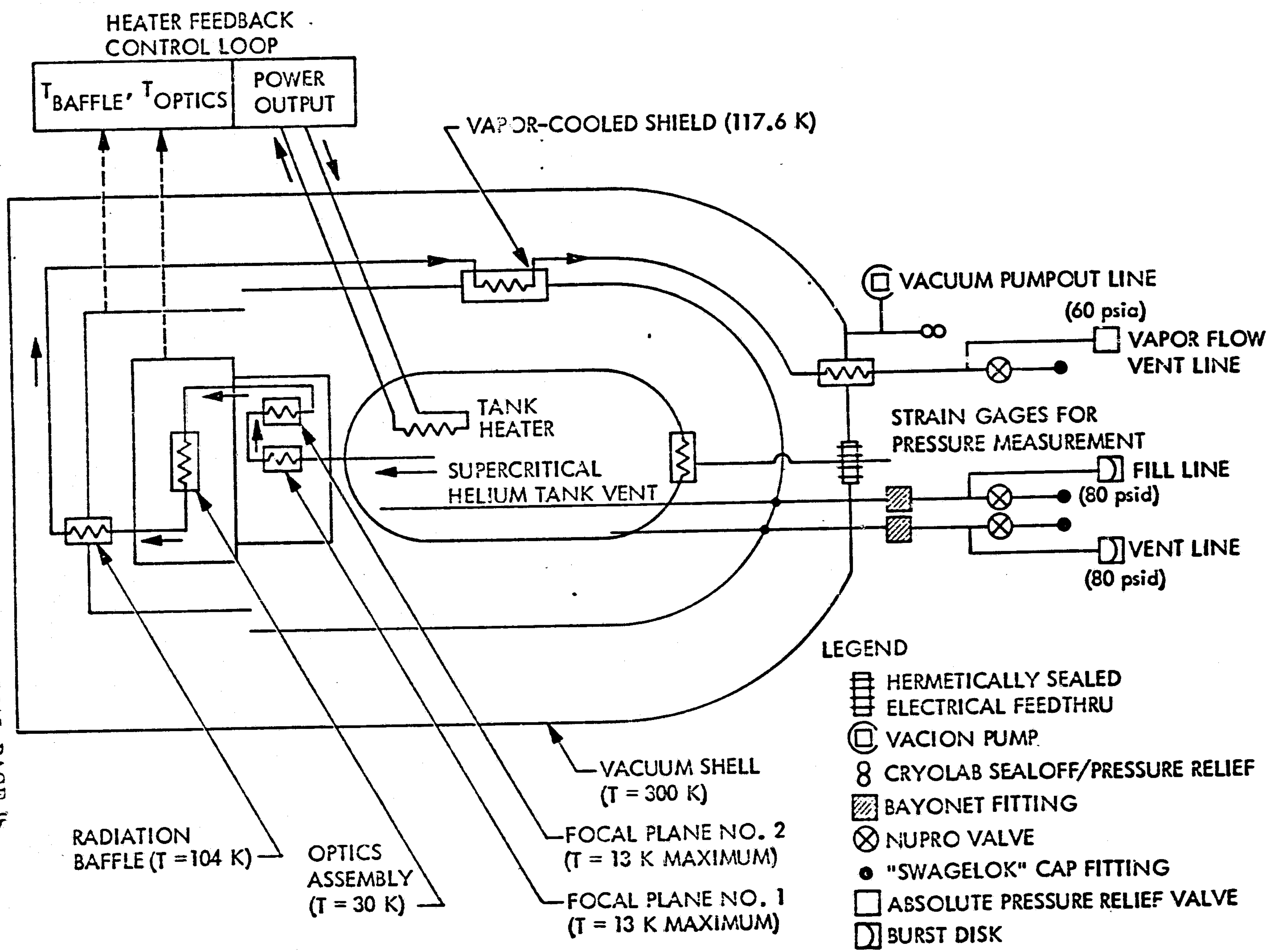


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Fig. 4-6 COOLER WEIGHT SUMMARY*

tank pressure ref feed

ITEM	SUPERCRITICAL HELIUM BASELINE (60 PSIA) KG	(LBS)	LIQUID HELIUM OPTION KG	(LBS)
PRIMARY CRYOGEN TANK	34.4	(75.8)	20.2	(44.6)
VAPOR COOLED SHIELD	9.4	(20.8)	9.4	(20.8)
VACUUM SHELL	30.2	(66.6)	30.2	(66.6)
SUPPORT TUBE	7.8	(17.1)	7.8	(17.1)
MOUNTING FLANGE	1.9	(4.1)	1.9	(4.1)
MLI	6.0	(13.3)	6.0	(13.3)
PLUMBING LINES	2.7	(6.0)	2.7	(6.0)
VAC ION PUMP 8 ℓ/s (INCL. MAGNETS)	4.1	(9.0)	4.1	(9.0)
MISC ASSEMBLY HARDWARE	1.0	(2.2)	1.0	(2.2)
TOTAL DRY WEIGHT	97.5	(214.9)	83.3	(183.7)
20% MARGIN	19.5	(43.0)	16.7	(36.7)
DRY WEIGHT WITH MARGIN	117.0	(257.9)	100.0	(220.4)
CRYOGEN WEIGHT	80.0	(176.3)	70.8	(156.1)
TOTAL LOADED WEIGHT	197.0	(434.2)	170.8	(376.5)
TANK CAPACITY 100% LOADED AT NBP	88.5 KG	194 LBS	88.5 KG	194 LBS

*ASSUMED SAME TANK VOLUME USED IN BOTH DESIGN CONFIGURATIONS liquid helium is selected instead of supercritical helium which was discussed in Section 4.1.

4.2.3 Thermal Analysis

The relationship between the required heat input to the supercritical helium dewar for a constant vent rate of 2.5×10^{-2} gm/s(0.2 lb/hr) at 4.1×10^{5} nt/m^2 (60 psia) and the percentage of helium remaining in the dewar is shown in Figure 4-7. To insure no more helium is vented than required, the parasitic heat load to the tank must be below the minimum required heat rate of 537 mw which occurs at the 65% filled condition.

The calculated parasitic heat loads to the helium tank (assuming no auxiliary resistance heating to the tank) from the support tubes, multilayer insulation system, and plumbing is 486 mw. When the tank is 65% filled this will yield a maximum helium gas vent rate of 2.3×10^{-2} gm/s (0.18 lb/hr), 10% less than that required to maintain the Honeywell experiment temperature within specifications. Auxiliary heating is required to bring the flow rate up to 2.5×10^{-2} gm/sec.

By fixing the flow rate at 2.5×10^{-2} gm/sec (0.2 lb/hr), the parasetic heat load drops to 415 mw because of a reduction in the vapor cooled shield temperature from 128.4°K to 117.6°K. The auxiliary heat required to maintain the flow rate constant at 2.5×10^{-2} gm/s (0.2 lb/hr) from the resistance heater during the course of the mission will then vary from a minimum of 124 mW at 6°K to 1.10 watt at 12°K.

A breakdown of the overall parasitic load showing the relative effect of support tube assembly, multilayer insulation system and plumbing system is shown in Figure 4-8 for the 2.52×10^{-2} gm/sec (0.2 lb/hr) flow condition. These heat rates do not represent the minimum that could be achieve by varying first the location of the vapor cooled shield within the insulation thickness and second, the location of the thermal grounding point along the fiberglas tube. The design selected was driven to a large extent by the envelope and fabrication constraints. In Figure 4-9 the sensitivity of the vapor cooled shield and MLI heat rate are shown as a function of the shield position within the insulation space. Improvement of only 10 mw can be expected by a change in the shield location from the present design point to the optimum location. It is expected that a more significant reduction would be realized with a change in the support tube grounding.

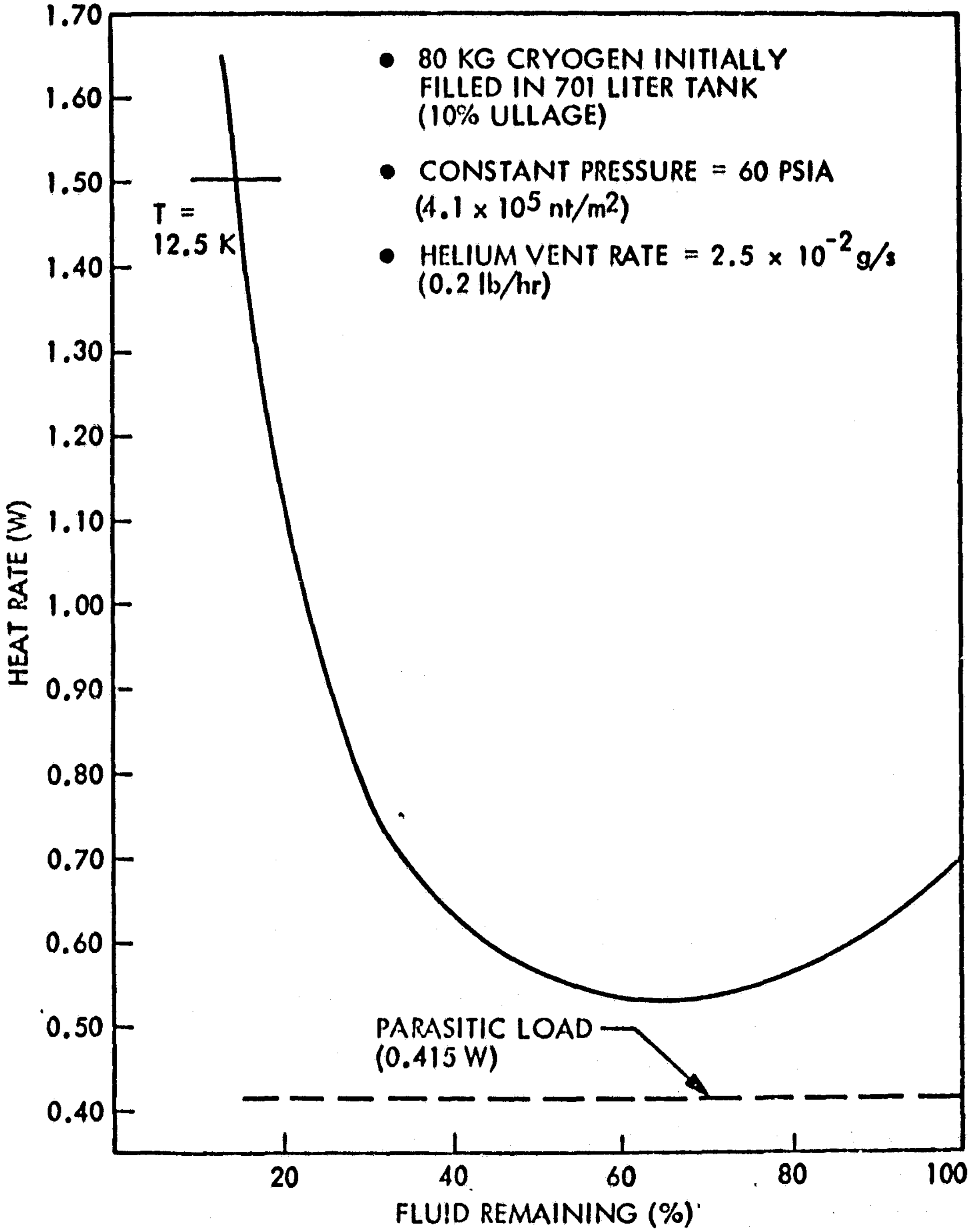
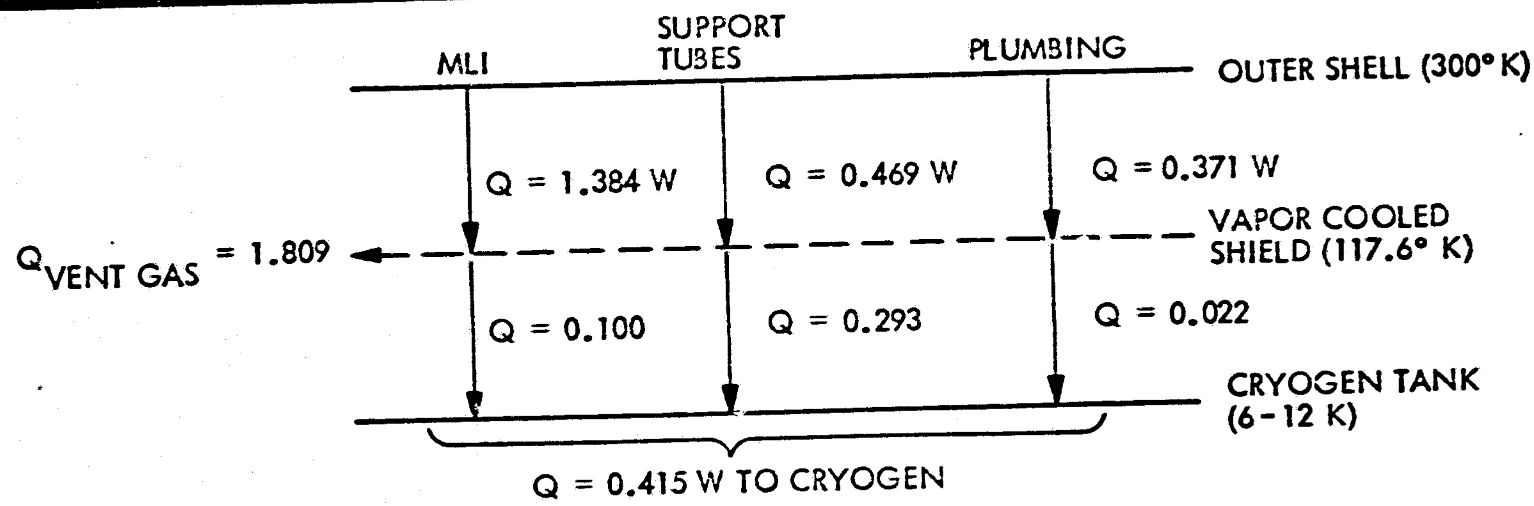


FIG. 4-7 REQUIRED HEAT INPUT FOR WITHDRAWAL OF SUPER-CRITICAL HELIUM AT CONSTANT PRESSURE

FIG. 4-8 SUMMARY OF PARASITIC HEAT LOADS TO HELIUM TANK



	CONFIGURATION	HEAT LOAD TO CRYOGEN (W)
SUPPORT TUBES	INNER TUBE: 84.4 CM DIA 37.1 CM LONG 0.152 CM THICK OUTER TUBE: 89.1 CM DIA 90.7 CM LONG 0.152 CM THICK	0.293
MULTILAYER INSULATION	OVERALL THICKNESS = 6.35 CM VAPOR COOLED SHIELD LOCATED 3.05 CM FROM CRYOGEN TANK SILK NET/DOUBLE-ALUMINIZED MYLAR $K = \begin{cases} 5.3 \times 10^{-7}\ W/CM - °K\ (117.6 - 300) \\ 5.8 \times 10^{-8}\ W/CM - °K\ (6 - 117.6) \end{cases}$	0.100
PLUMBING	STAINLESS STEEL 1.27 - CM DIAMETER 0.013 - CM WALL THICKNESS	0.022

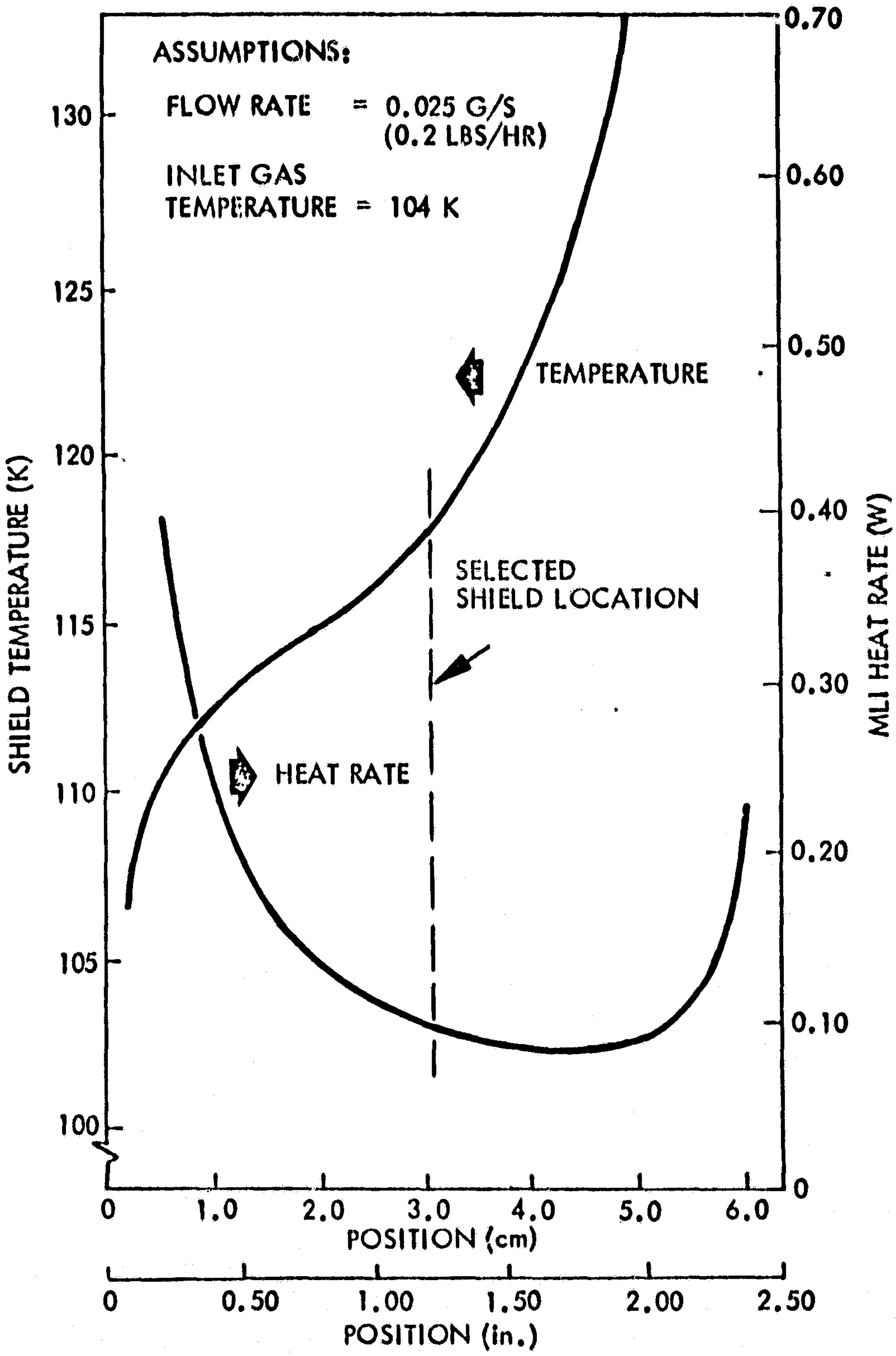


FIG. 4-9 EFFECT OF VAPOR-COOLED SHIELD LOCATION ON HEAT RATES AND TEMPERATURE

4.2.4 Structural Analysis

The following structural analyses were made for the He cooler:

- o Free vibration: modes
- o Sine vibration: stress and buckling
- o Random vibration: stress and buckling
- o Static acceleration: stress and buckling
- o Internal and external pressure: stress and buckling

The dynamic analyses were made for the entire cooler assembly (support tube and coolant tank), but only the support tube is reported on here, since dynamic stresses in the coolant tank are much smaller. The method of analysis was to employ the computer program BOSOR[(3)] for a small number of parametric variations of support tube length and thickness, and to use these computer solutions in conjunction with modified simple beam analysis to arrive at a viable design.

SUPPORT TUBE ANALYSIS

The static and dynamic environment which governs the design of the support tubes is defined by:

Ultimate static lateral acceleration: 4 g

Lateral acceptance vibration level:

Frequency, Hz	Level, g-pk
5-40	0.5
40-80	0.0125 x Hz
80-200	1.0

Random vibration acceptance level:

Frequency, Hz	W
20-130	6 db/Oct.
130-1000	0.18 g^2/Hz
1000-2000	-3db/Oct.

Level: 16.7 g-rms

The material in the support tube is glass fiber cloth in an epoxy matrix. The completed tube has the following material properties:

Shear modulus: G = 750000 psi

Shear modulus: G = 750000 psi

Elasticity modulus:	
axial:	$E_x = 2.14 \times 10^6$ psi
circumferential:	$E_y = 5.55 \times 10^6$ psi
Poisson's ratio:	$\nu_{xy} = 0.0956$
Cloth thickness per layer:	0.010 inch
Ultimate strength, axial direction:	26.4 KSI
Proportional limit, axial direction:	5.7 KSI
Ultimate strength, hoop direction:	119 KSI
Proportional limit, hoop direction:	72 KSI

The configuration is shown in Fig.4-10 . The weight of the tank with contents is as follows:[(a)]

Coolant:		150 lb
Skin:		60 lb
Rings:		7 lb
	Total	217 lb

The weight of the support tube is based on a mass density of 1.69×10^{-4} lb sec^2/in^4.

This analysis was performed for a single support tube, while the final design incorporate a dual or folded design. To a conservative first approximation these results can be applied for the folded tube configuration providing the same tube length is utilized. The total length of the folded tube design utilized is 50.3 in. and this design length is superimposed on the appropriate figures.

Typical results from the BOSOR Model analysis are shown in Figure 4-11. Figure 4-11 depicts the computer model with a 30 inch long 0.06 inch thick support tube, the following three figures, Figures 4-12, 4-13, and 4-14, show the modal deteriorations of the first three frequencies. The displacements are exaggerated in the figures; actually the structure does not puncture itself. From the subsequent analysis it was found that the first and to a lesser degree the second and third modes are the major contributors to stresses in the support tubes.

The first frequency is plotted as a function of support tube length and thickness in Figure 4-15. It is found that this frequency may be represented by

a) Final weight summary indicates the supported weight is 252 lbs excluding MLI and vapor cooled shield, which are not rigidly supported.

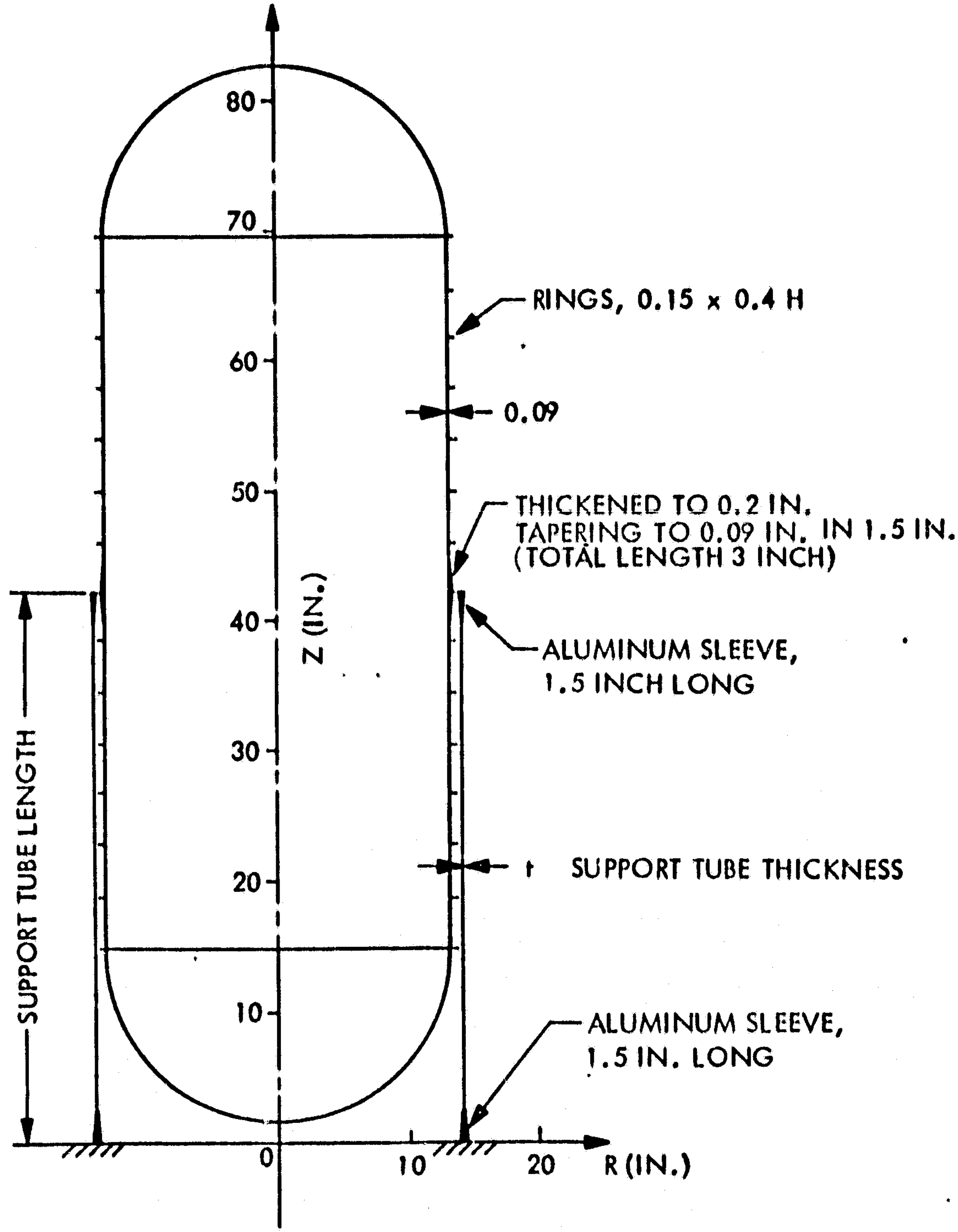


FIG. 4-10 STRUCTURAL MODEL

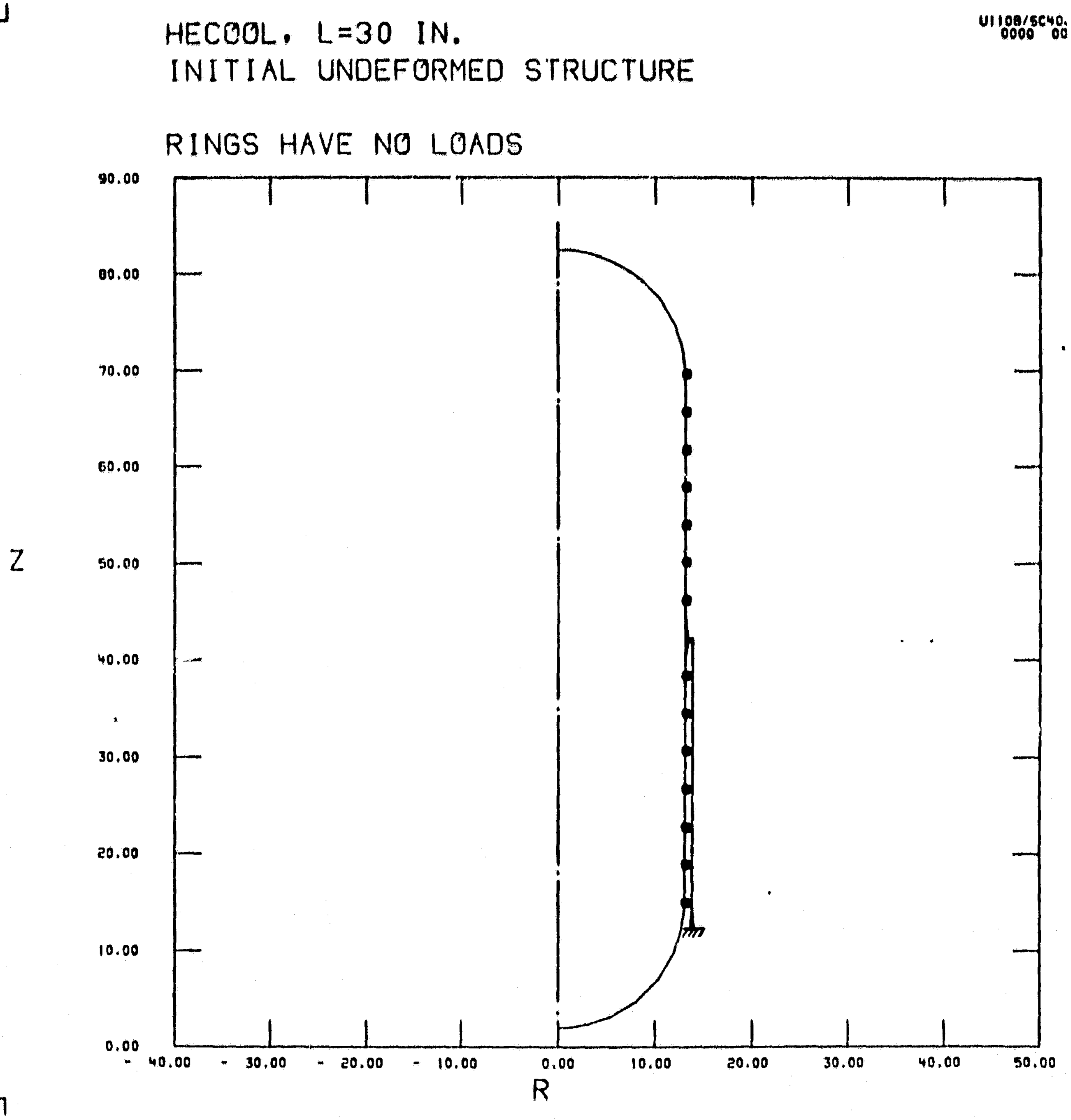


Fig. 4-11 Computer Model

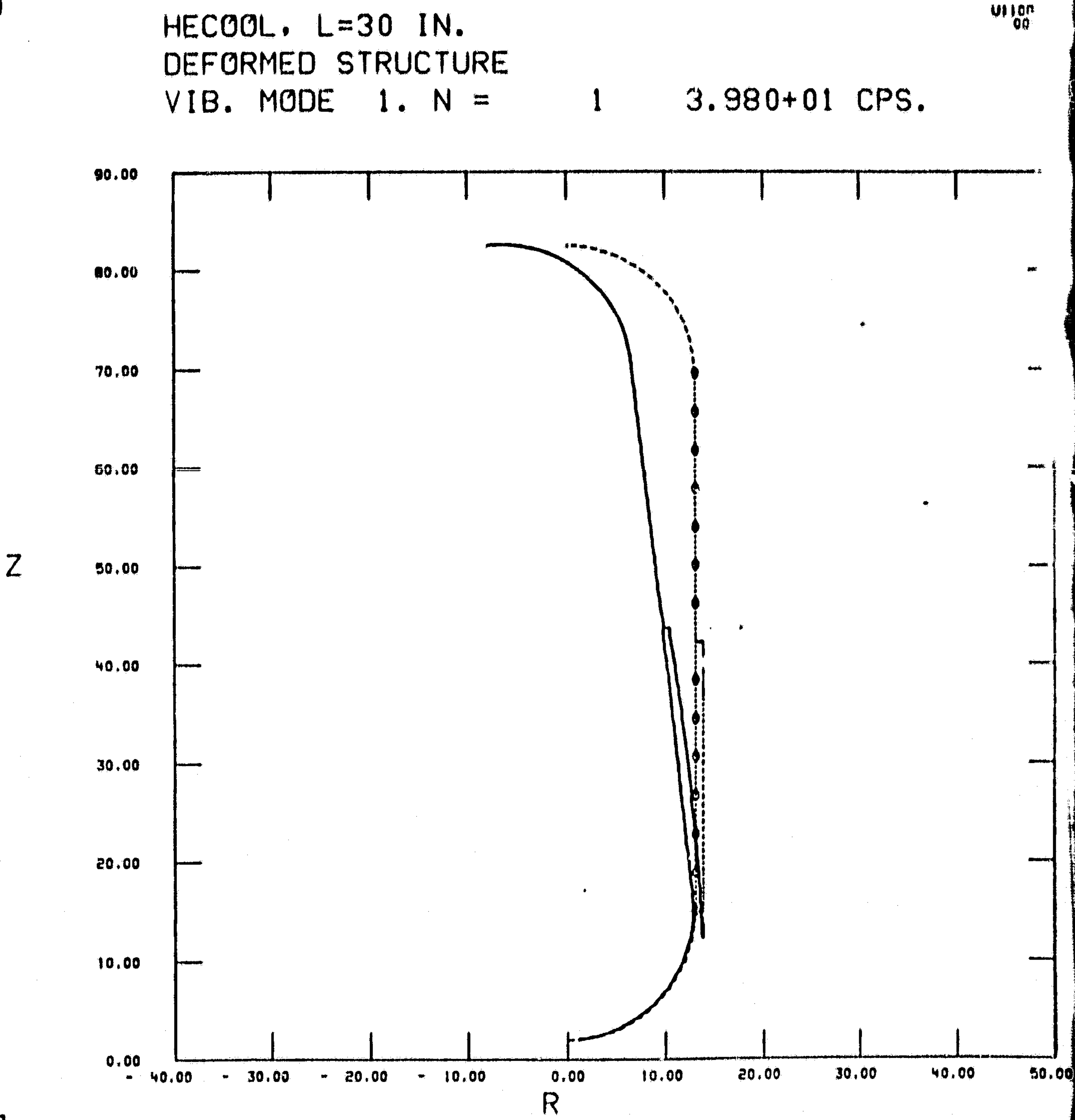


Fig. 4-12 Ist Mode, 39.80 Hz

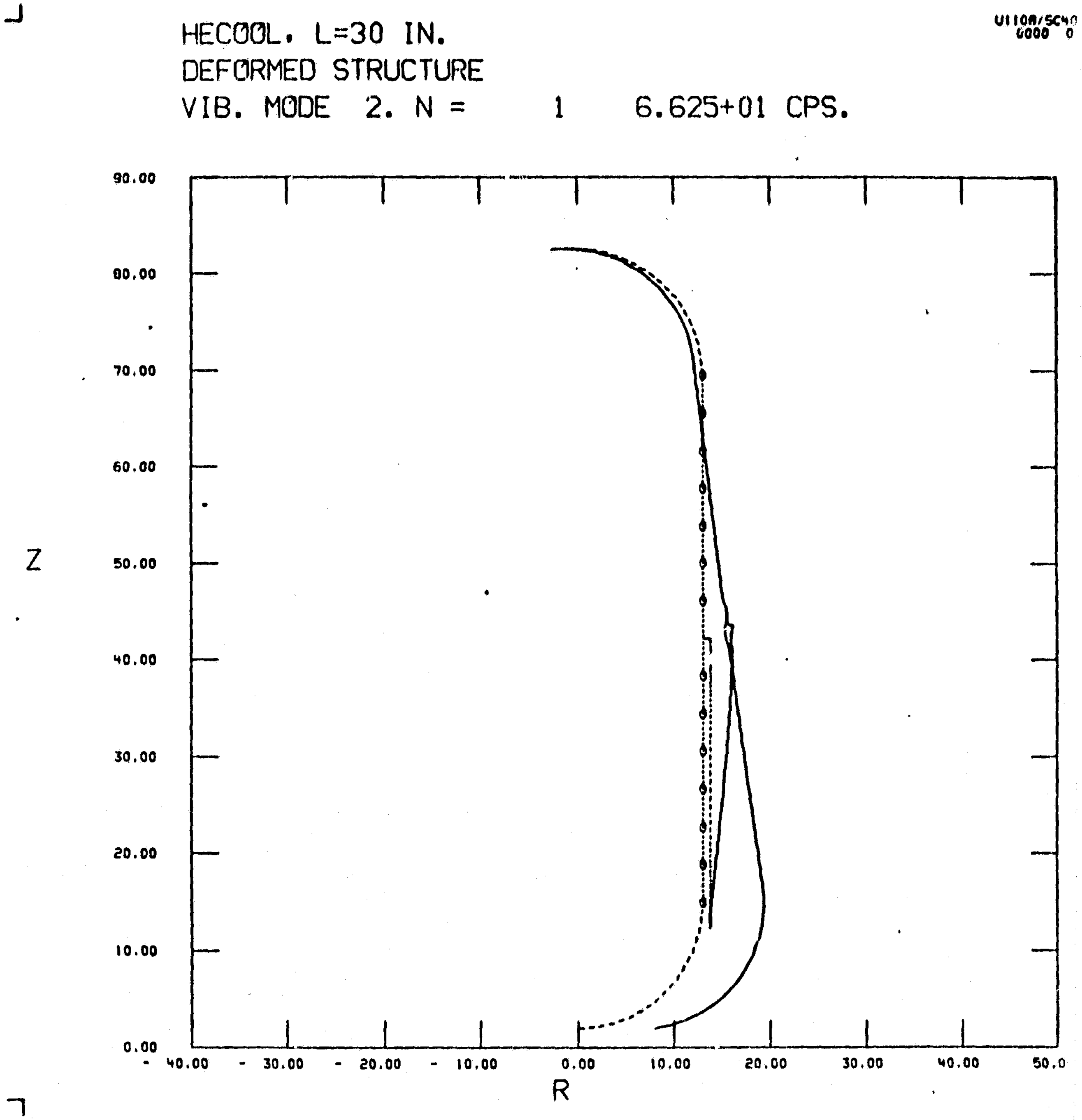


Fig. 4-13 2nd Mode, 66.25 Hz

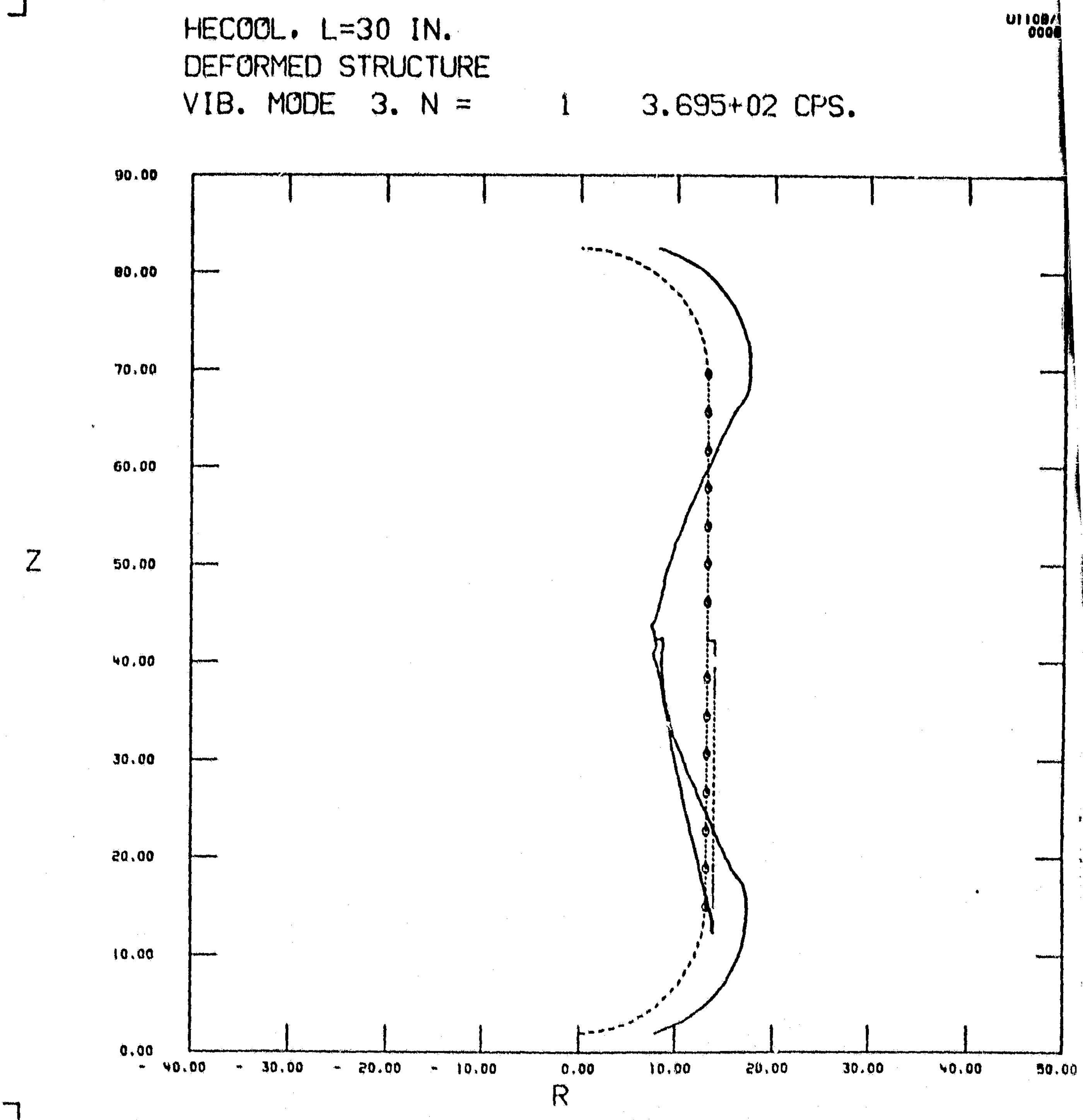
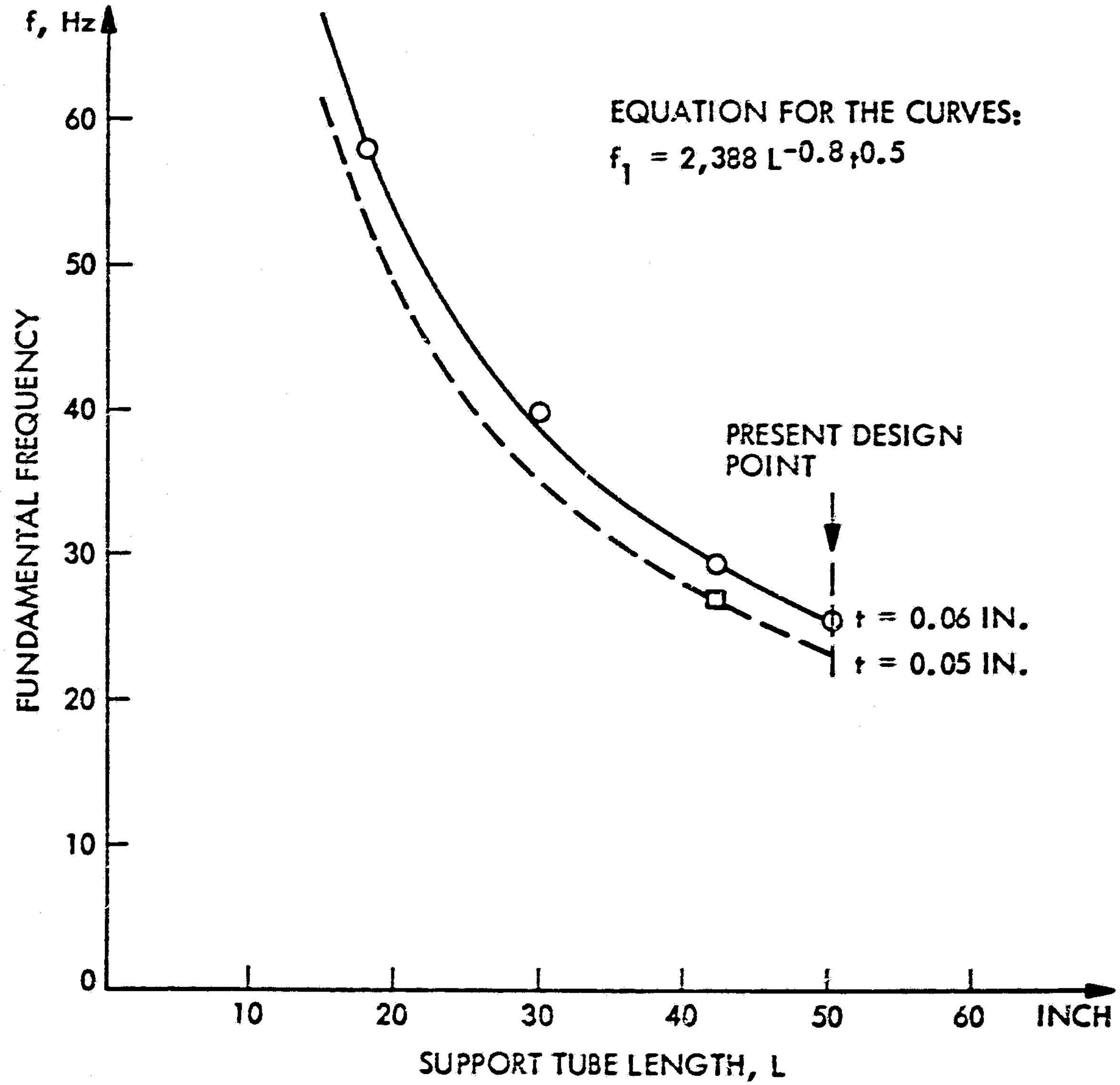


Fig. 4-14 3rd Mode, 369.5 Hz

FIG. 4-15 FIRST-MODE FREQUENCY VERSUS SUPPORT TUBE LENGTH



$$f_1 = \text{const} \times L^{-0.8}\, t^{0.5} \qquad (1)$$

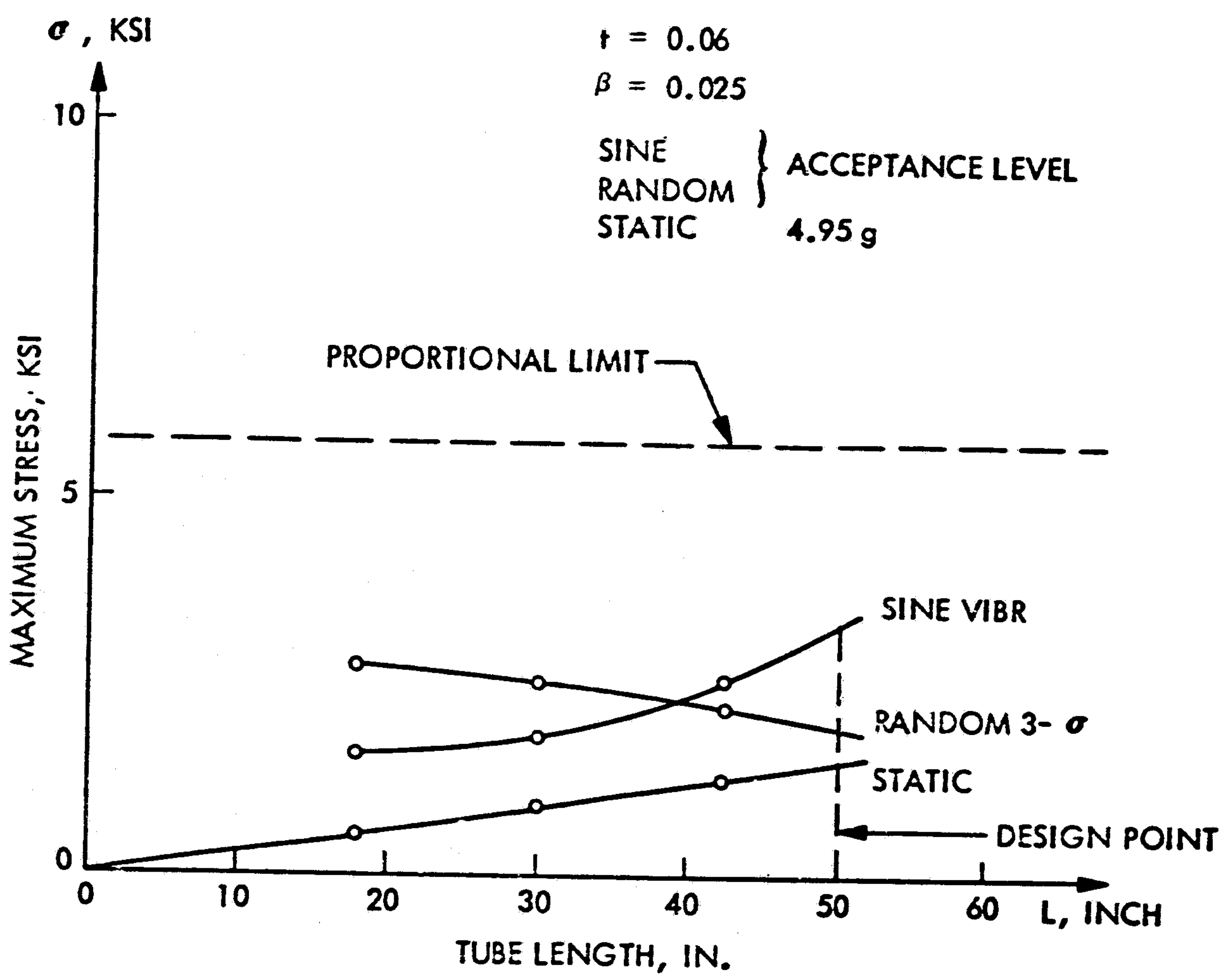
where L and t are the length and thickness of the tube. From elementary considerations one would have expected the length influence to be represented by $L^{-1.5}$, but the lower power in Eq. 1 no doubt is traceable to the distortion of the tank wall of the support tube attachment point, which is discernible in Figure 4-12, and to the very large inertia of the tank.

Stresses in the tube are quite small, as can be seen in Fig.4-16, but buckling is a critical factor in the design. Note that the stresses shown in Fig. 4-16 are based on the ultimate static condition, on the acceptance level for sine vibrations, and on the 3-σ (99.3%)* acceptance level for the random vibrations.

In the dynamic analysis a damping of β = 2.5% is assumed. The buckling allowable is based on the 99% level, i.e. there is one chance in a hundred that a tube randomly picked and not tested would fail at the allowable buckling load.

*i.e. 3 x Rms

FIG. 4-16 SUPPORT TUBE STRESSES VERSUS TUBE LENGTH



The main results of the support tube analysis are summarized in Figures 4-17 and 4-18, which show the required thickness for different configurations and design conditions. It is interesting to note that the required thickness is almost independent of length; this follows in part from the definition of the sine and random vibration environments.

No qualification level is defined, but for the sake of argument it was assumed that the qualification level is twice the acceptance level and four times the spectral density W for the random vibrations. Results from an analysis using this definition of the qualification level are given in Figure 4-18.

To summarize, based on the acceptance level the thickness required to meet the buckling requirement is about 0.060 inch for a damping of 2.5%, about 0.045 for a damping of 7%. Measured damping factors on prior coolers have been found to be in that range, the upper value corresponding to substantial damping from the MLI while the 2.5% represents the damping of the fiberglass only (very stiff structure). Additional analysis will be required to determine the appropriate value for this design. If the design is to be based on qualification levels as defined above, the thickness requirements become about 0.09 cm and 0.06 inch, respectively for β of 2.5% and 7%. The foregoing statements are true for tube lengths in the range 15-50 inch.

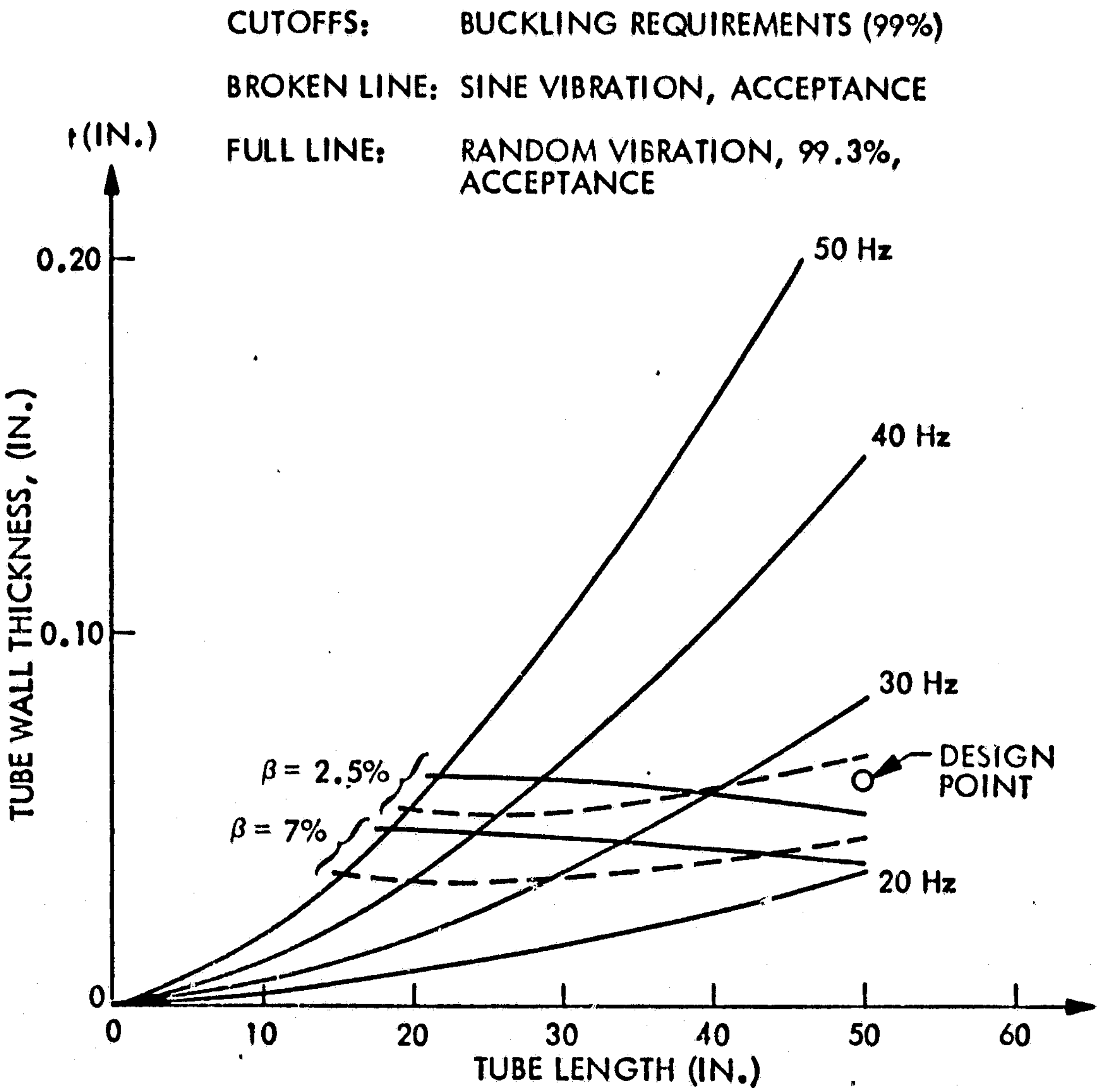


FIG. 4-17 SUPPORT TUBE THICKNESS-ACCEPTANCE LEVELS

ASSUMPTIONS:

SINE VIBRATION	$g_{QUAL} = 2 \times g_{ACCEPT}$
RANDOM VIBRATION	$W_{QUAL} = 4 \times W_{ACCEPT}$
CUTOFFS:	BUCKLING REQUIREMENTS (99%)
BROKEN LINE:	SINE VIBRATION, QUALIFICATION
FULL LINE:	RANDOM VIBRATION, 99.3% QUALIFICATION

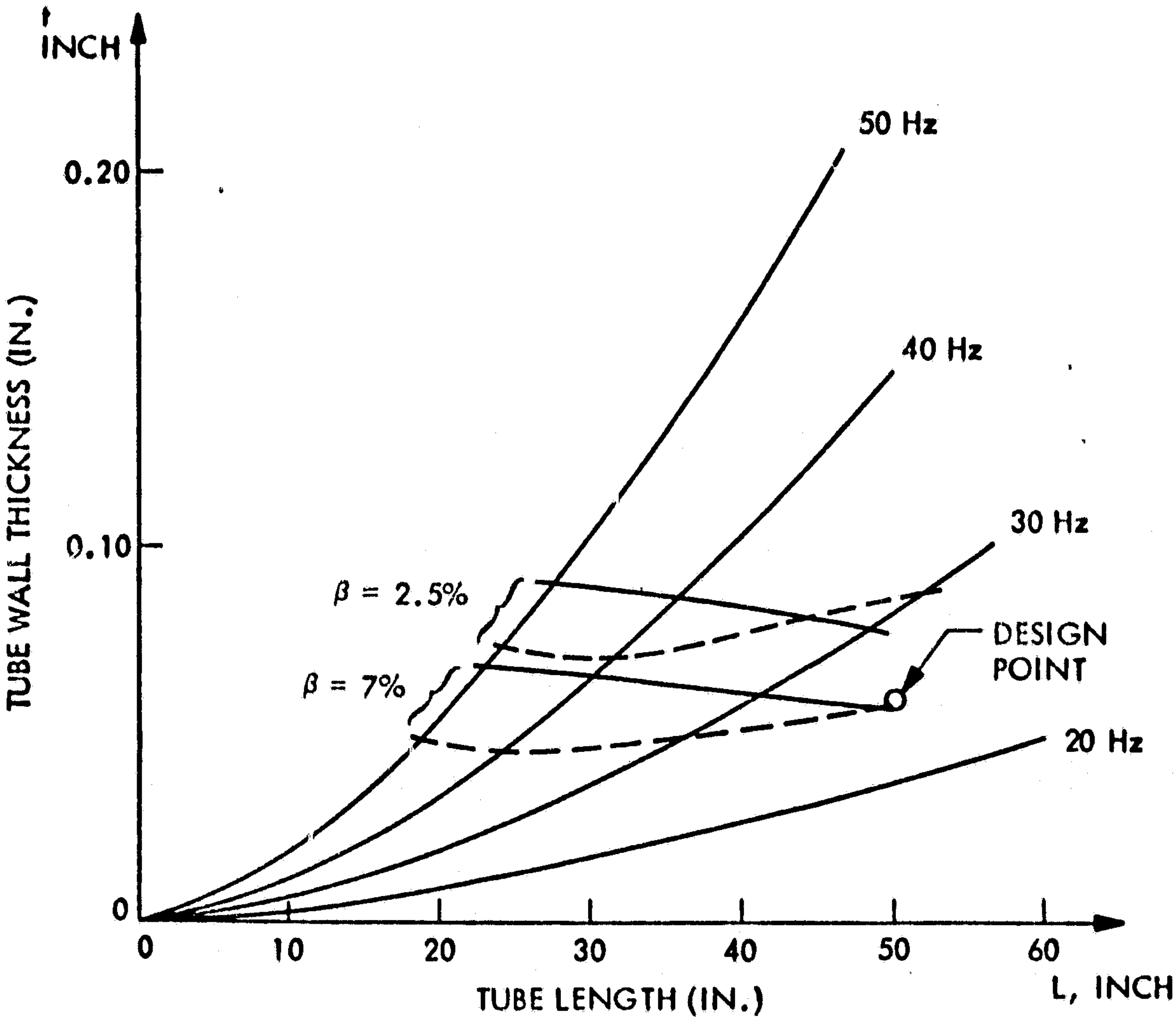


FIG. 4-18 SUPPORT TUBE THICKNESS – QUALIFICATION LEVELS

Shell Analysis

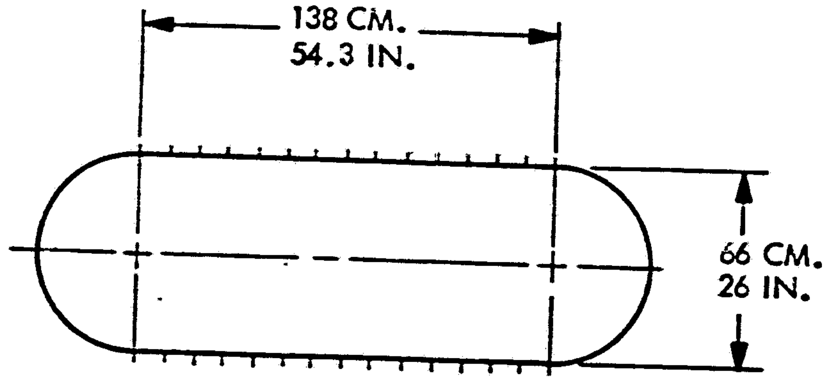
The helium tank was analyzed for both internal and external pressure in various combinations. Both ring stiffened and monocoque designs were investigated, as shown in Figure 4-19. Although a monocoque tank would be adequate, for internal pressure alone, the ring stiffened design is necessary for external pressure that would be present during the various leak checking operations that will be required. The tank will be constructed of 6061 aluminum. The tank geometry was fixed at a diameter of 66 cm and cylinder length of 138 cm with hemispherical domes for the parametric study. The final tank design dimensions were based on an extrapolation of this analysis for the ring stiffened configuration.

The vacuum shell design was based on a direct extension of these studies.

FIG. 4-19 HELIUM TANK STRUCTURAL ANALYSIS

6061-T4 ALUMINUM, F. S. = 4

	Alt.	t Inch	Number Rings 0.15 x 0.4 H	Pcrit psid	Int Press.	F. S.		Tank Wt, Lb
						Stress	Class Buckl	
RING-STIFFENED	1	0.09	15	-100	+50	4	6.7	67
	2	0.06	15	-55	+34.5	4	3.7	47
MONOCOQUE	3	0.15	0	-30	+50	6.7	2	99
	4	0.09	0	-8.5	+50	4	0.6	59



5.0 SUMMARY AND RECOMMENDATIONS

An extensive trade study was conducted to evaluate possible single stage, dual stage and triple stage cooler options for the CLIR mission. The two cases which independently minimize weight and volume are the single stage, 2-temperature solid hydrogen (98 kg) system and triple stage helium, neon and nitrogen systems (120ℓ), respectively.

For further studies, the single-stage, 2-temperature solid hydrogen system was grouped with three of the more promising candidate systems; the single stage, 3-temperature helium system, as well as the dual stage, 3-temperature helium/nitrogen and hydrogen/nitrogen systems. The three stage system was eliminated from further study because of the additional system complexity required for the marginal gains received over some of the other systems.

Using the smaller subset of options, more detailed system analyses were performed, incorporating refined experiment heat rates and cooler/experiment interaction. From an engineering standpoint, the single stage, 2-temperature solid hydrogen cooler still appeared to be the most desirable cooler selection for the CLIR program. However, safety considerations drove the selection to a heavier and larger single stage, three temperature supercritical helium system.

A baseline cooler concept was developed to allow for a more detailed thermal and structural analysis of a single stage, three temperature supercritical helium cooler. To satisfy the 30-day mission lifetime a 701 liter tank was required for an 80 Kg supply of helium. The total system mass is 197 Kg which includes cryogen, cryogen tanks, support tubes, insulation, vapor cooled shield, vacuum shell, and system plumbing.

The heat loads to the cryogen tank were determined for the support tube assembly, multilayer insulation system and plumbing lines. A total of 415 mw was computed, less than the 537 mw required to provide the 2.5×10^{-2} gm/sec vent gas flow rate required by the instrument.

Additional effort is still needed in the analysis of the baseline cooler to either improve on the present design or increase the confidence level in the computed heat loads. The effort should include:

- Thermal analysis of the insulation wrap - including edge effects
- Study of cooling of the support tube between the vapor cooled shield and ambient terminations
- Further design and analysis in the fill/vent plumbing hardware
- Additional study of the ground hold conditions in relation to instrument cooling prior to launch

6.0 REFERENCES

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3. Bushnell, D., "Stress, Stability, and Vibration of Complex Branched Shells of Revolution: Analysis and User's Manual for BOSOR 4," NASA Langley Research Center, NASA CR-2116.

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