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19 ABSTRACT (Continue on reverse if necessary and identify by block number)

The Naval Air Development Center is taking action to improve the reliability of aircraft flight control hydraulic actuators. Feedback and loop closure electronics are being mounted on the actuator, thereby eliminating numbers of electrical conductors between the actuator and the flight control computer. Methods for cooling the electronics are required to help assure reliability.

The Thermacore effort documented in this report investigated the feasibility of using heat pipe technology to thermally protect the flight critical actuator electronics. The cooling method investigated used a flexible heat pipe cold plate that interfaced with the integrally connected flexible hose to a nearby heat sink. The flexible section accommodates relative motion between the actuator and the heat sink.

The successful Phase I work program first established the design requirements; evaluated cold plate to actuator integration techniques; conducted analysis to select the best heat pipe material/working fluid system; and concluded with the design, fabrication and testing of a flexible heat pipe cold plate assembly that integrates with the SMART aileron actuator being developed by H. R. Textron. The cold plate transferred over 60 watts. This power exceeded the 44.5 watt requirement.

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FLEXIBLE HEAT PIPE COLD PLATE

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PROJECT SUMMARY

Navy aircraft use hydraulic actuators to position control surfaces on the wing, tail, rudder, etc. The feedback and position indication signals for one flight control actuator require dozens of electrical wires between the actuator and the flight control computer. It is desirable to reduce this wire count by incorporating loop closure and redundancy management electronics on or near the flight control actuator. Actuator case temperatures will exceed 100°C requiring the electronic components to be cooled since their failure rate increases exponentially with temperature. Therefore a highly reliable cooling method will be needed to operate flight critical electronics in this environment.

The Phase I effort demonstrated the feasibility of using heat pipe technology to thermally protect the flight actuator electronics. The electronics package is thermally isolated from the actuator by a heat pipe cooled cold plate. Heat entering the cold plate from the actuator and electronics is transferred to surrounding structure via an integrally connected flexible heat pipe. Figure 1 shows a photograph of one of the flexible cold plates fabricated in Phase I. Details of Figure 1 include the cold plate, flexible connection and finned attachment for coupling to surrounding aircraft structures. The flexible section accommodates relative motion between the actuator and the heat sink.

The Phase I demonstration effort was successful in all respects. It included identification of requirements, design, fabrication and testing of a heat pipe cold plate assembly that integrates with the SMART FA/18 type aileron servo/actuator being developed by H.R. Textron. The cold plate transferred the 44.5 watt requirement with less than a 5°C cold plate-to-condenser wall temperature drop. Output power as high as 60 watts was recorded. This power carrying capability exceeded the 44.5 watt requirement by a factor of 1.3.

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1.0 INTRODUCTION

Aircraft use hydraulic actuators to control the position of the rudder, stabilator, ailerons, trailing edge flaps, etc. The feedback and command signals for one flight control actuator are interfaced to flight control computers through a large number of conductors. The Naval Air Development Center in Warminster, Pennsylvania is taking action to reduce this wire count by developing "smart" actuators which employ electronics on or near the flight control actuator. This has advantages in improving performance, reducing wire weight, providing electromagnetic protection, and ease in actuator maintenance.

Actuator case temperatures will exceed 100°C requiring the electronic components to be cooled since their failure rate increases exponentially with temperature. Therefore, a highly reliable cooling method is needed to operate flight critical electronics in this environment. A highly effective means to achieve electronic component thermal control is through the use of heat pipe technology.

The heat pipe is a heat transfer element that makes use of two phase heat transfer to carry heat at low temperature loss from an input area where a fluid is evaporated, to an output area where the vapor is condensed, giving up its heat of vaporization. The heat pipe uses capillary pressures in a porous wick to return the condensed liquid to be re-evaporated, thus, making it passive requiring no external power for its operation. Heat pipe technology is versatile and is being developed to provide cooling for various electronic applications. A cut-a-way view of a basic heat pipe is seen in Figure 2.

Thermacore, Inc., Lancaster, Pennsylvania, is under a Phase I Small Business Innovative Research SBIR contract (Contract No. N62269-88-C-0210) to the Naval Air Development Center, Warminster, Pennsylvania, to investigate heat pipe technology as a reliable cooling method for this application. This work was carried out over a nine month period that began in February of 1988 and is documented in this report.

The specific objective of Phase I was to develop a flexible heat pipe cold plate system capable of cooling loop closure and redundancy management electronics mounted on or near flight control actuators. This objective was satisfied in every respect.

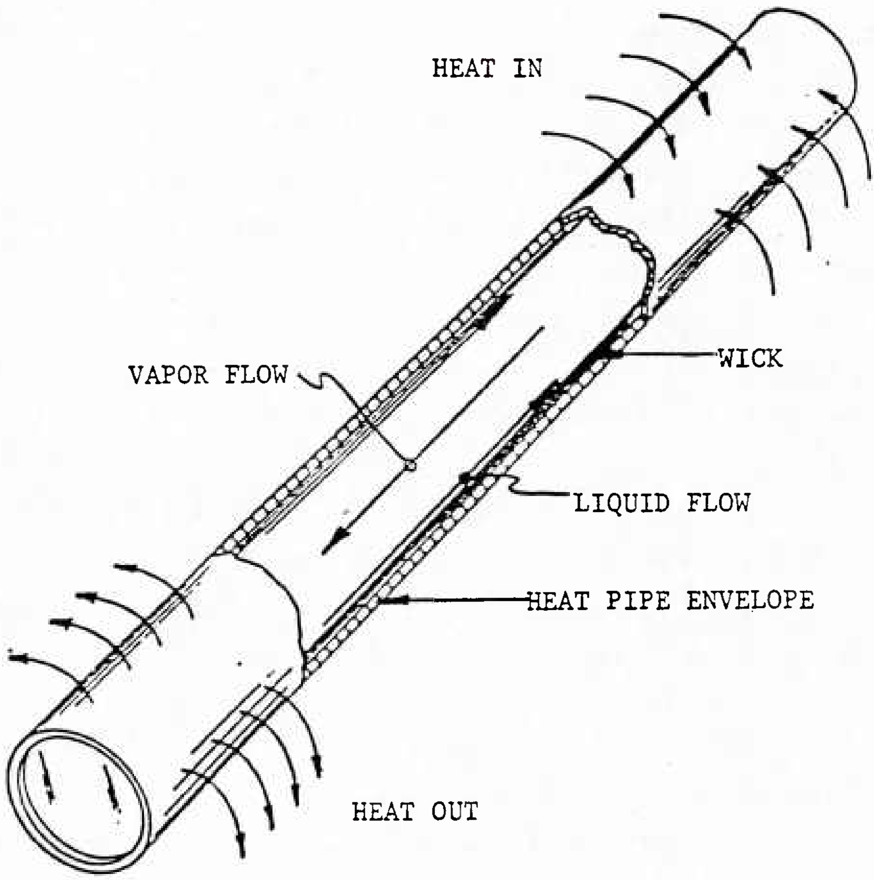


Figure 2. Basic heat pipe schematic

The Phase I program had four sequential technical tasks plus reporting. Each task is outlined below:

- Task 1 - Establish design requirements for the flexible heat pipe cold plate. Results are covered in Section 2.1
- Task 2 - Formulate alternative flexible heat pipe cold plate concepts capable of being integrated on or near the actuator electronics package. See Section 2.2.
- Task 3 - Evaluate each concept for its potential as a flexible heat pipe cold plate. See Section 2.2 and 2.3.
- Task 4 - Design, fabricate and test a proof-of-principle flexible heat pipe cold plate system. Design details are covered in section 2.4; test results are covered in Section 2.5.

The remainder of this report is divided into two sections as follows: Section 2.0 makes up the body of the report and it details the technical results of the program; Section 3.0 covers the specific conclusions and recommendations reached as a result of this effort.

2.0 TECHNICAL DISCUSSION

This section of the report details the results of the work conducted under this Phase I effort. The work effort began with the establishment of requirements; evaluated cold plate to actuator integration techniques; selected the best heat pipe materials; and concluded with the design, fabrication, and testing of a proof-of-principle flexible heat pipe cold plate. This section is sub-divided as follows:

2.1 Design Requirements
2.2 Concept Integration Techniques
2.3 Materials
2.4 Flexible Heat Pipe Design
2.5 Test Set-up and Results

2.1 DESIGN REQUIREMENTS

Thermacore met with cognizant personnel at the Naval Air Development Center and with H.R. Textron, manufacturers of aircraft hydraulic actuators and actuator electronics, to define requirements for the flexible heat pipe cold plate.

The established design requirements are outlined in Table 1. The geometry of the H.R. Textron SMART aileron actuator, shown in Figure 3, was used to form the basis for sizing and integration of the flexible heat pipe cold plate.

The flexible heat pipe was designed to transfer both the heat load generated by the electronics and the heat load imposed by the hydraulics. The total heat load generated by the electronics is 10 watts as specified by H.R. Textron. The heat load imposed on the electronics by the hydraulics was estimated as described below.

The temperature of the hydraulic actuators and heat sinks were measured by NADC personnel on an actual F/A-18 aircraft during flight. The temperatures are recorded in Figure 4.

These measurements show that aileron actuator body temperatures range from 77 to 93°C. The structure near the actuator to be used as the heat sink appears to range in temperature from 49°C to 54°C according to the data. Based on this temperature information, Thermacore prepared an analytical thermal model of the cold plate and aileron actuator installed in a typical aircraft.

TABLE 1. Design Requirements

PARAMETER	MAGNITUDE
ACTUATOR TYPE	AILERON
HEAT LOAD IMPOSED BY ELECTRONICS	10 WATTS
HEAT LOAD IMPOSED BY HYDRAULICS	34.5 WATTS**
ACTUATOR BODY TEMPERATURE	77 - 93°C
HEAT PIPE OPERATING TEMPERATURE RANGE	20 - 110°C
FLEXIBLE SECTION LENGTH	23 CM
HEAT PIPE ORIENTATION	GRAVITY AIDED
VIBRATION	RANDOM/SINUSOIDAL
ACCELERATION	9 g*
HEAT SINK	STRUCTURAL NEAR ACTUATOR
SINK TEMPERATURE	49 - 54°C
MATERIAL	COPPER, ALUMINUM, STAINLESS STEEL, OTHERS
WORKING FLUID	AMMONIA, ACETONE, METHANOL, FREON OR WATER
HEAT PIPE WEIGHT	MINIMUM

*SHORT TERM TRANSIENT IMPOSED BY AIRCRAFT MANEUVERING

**CALCULATIONAL RATIONALE FOUND IN APPENDIX A

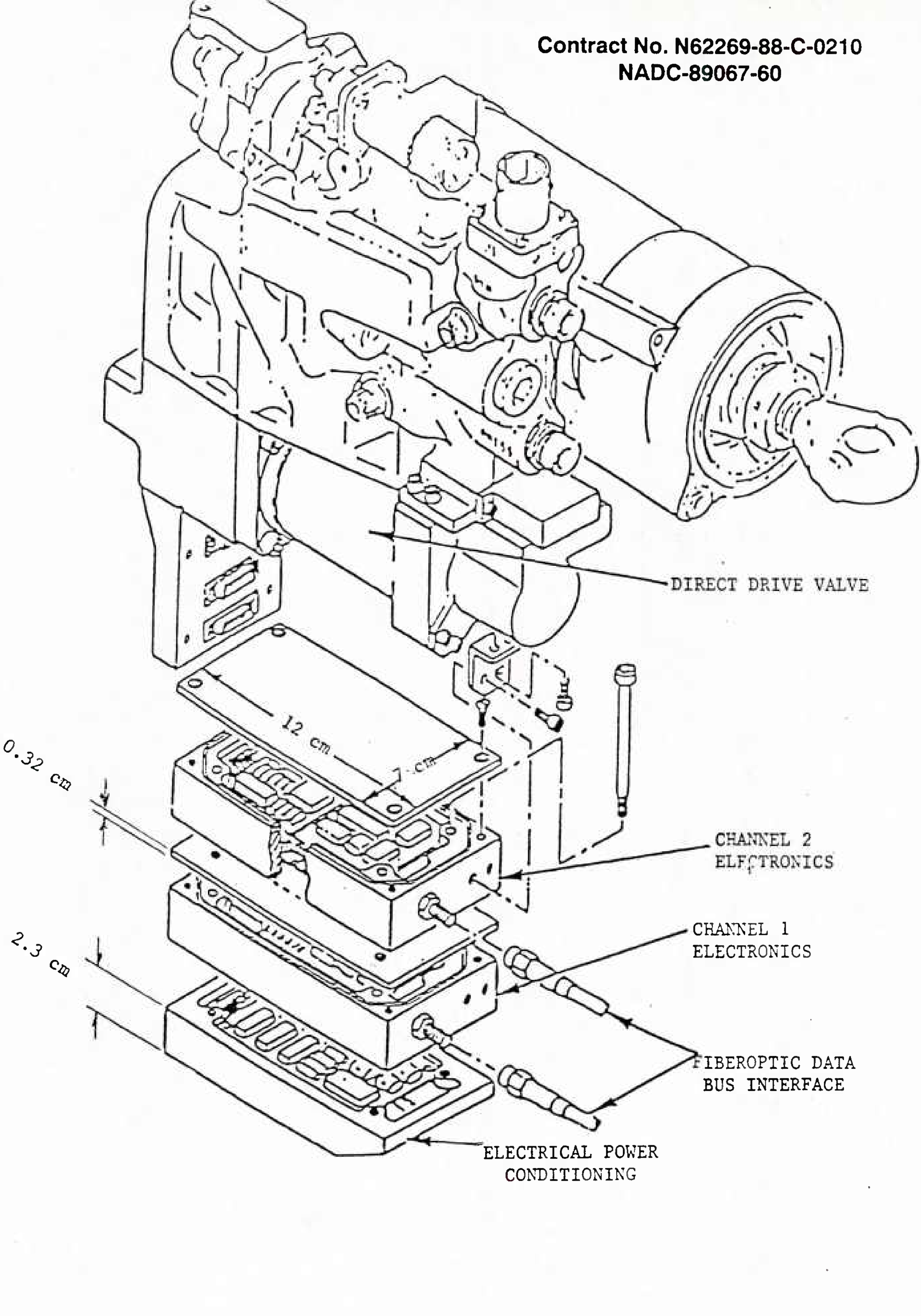


Figure 3. Smart FA/18 servoactuator

TRAILING EDGE FLAP	MAXIMUM TEMPERATURE ATTAINED (°C) RUN 1	MAXIMUM TEMPERATURE ATTAINED (°C) RUN 2	MAXIMUM TEMPERATURE ATTAINED (°C) RUN 3
Actuator Body	82	77	82
High Pressure Line	77	82	82
Structure Near Actuator	60	49	54
Bay "A" Structure	54, 54	49, 49	49, 49
Bay "B" Structure	54, 54	49, 49	49, 49
AILERON ACTUATOR			
Actuator Body	93	82	77
High Pressure Line	93	93	77
Structure Near Actuator	54	49	54
Fairing on Lower Wing	54	43	46
STABILATOR ACTUATOR			
Actuator Body	82	110	93
High Pressure Line	93	77	77
*Structure Near Actuator	82, 110	71, 71	65, 93
RUDDER ACTUATOR			
Actuator Body	93	93	82
High Pressure Line	77	82	82
Structure Near Actuator	54	49	49

NOTE: *TWO LOCATIONS TESTED

Figure 4. Measured F-18 actuator temperatures

The analysis was done to predict the total heat load to be transferred by the heat pipe cold plate. Results from this analysis are documented in Appendix A and are summarized below:

- The electronics package will generate 10 watts of thermal power.
- Eight watts will be transferred from the surrounding atmosphere through the electronics package insulation to the heat pipe.
- Heat will also enter the electronics package through the bolted attachment to the actuator. The use of a 0.64 cm thick Kapton polyamide insulator will hold this heat load to under 5 watts. Figure 5 illustrates the insulator location.
- The three electrical connectors and the four attachment bolts are estimated to transfer about 21.5 watts to the electronics package.

The total of the above heat loads to be transferred by the heat pipe cold plate is about 44.5 watts. This energy is transferred to surrounding aircraft structure where it is dissipated. Rough calculations show that the structure is capable of dissipating this heat during normal operation. Additional work is required in Phase II to provide assurance of adequate thermal capacity of the aircraft structure for normal and transient operating modes.

For the best heat pipe operation, the flexible heat pipe should be installed in the gravity assisted orientation if practicable. This is defined as the orientation where the heat sink condenser section is elevated slightly above the cold plate. In this orientation, the working fluid can flow back to the evaporator without requiring the wick as a capillary pump. This orientation will provide the most reliable operation. The flexible heat pipe to be demonstrated in the Phase II program will be designed to work either gravity assisted or against gravity where the cold plate would be elevated above the heat sink condenser.

The established design requirements were used to formulate alternative flexible heat pipe cold plate concepts. The results of this effort are discussed below.

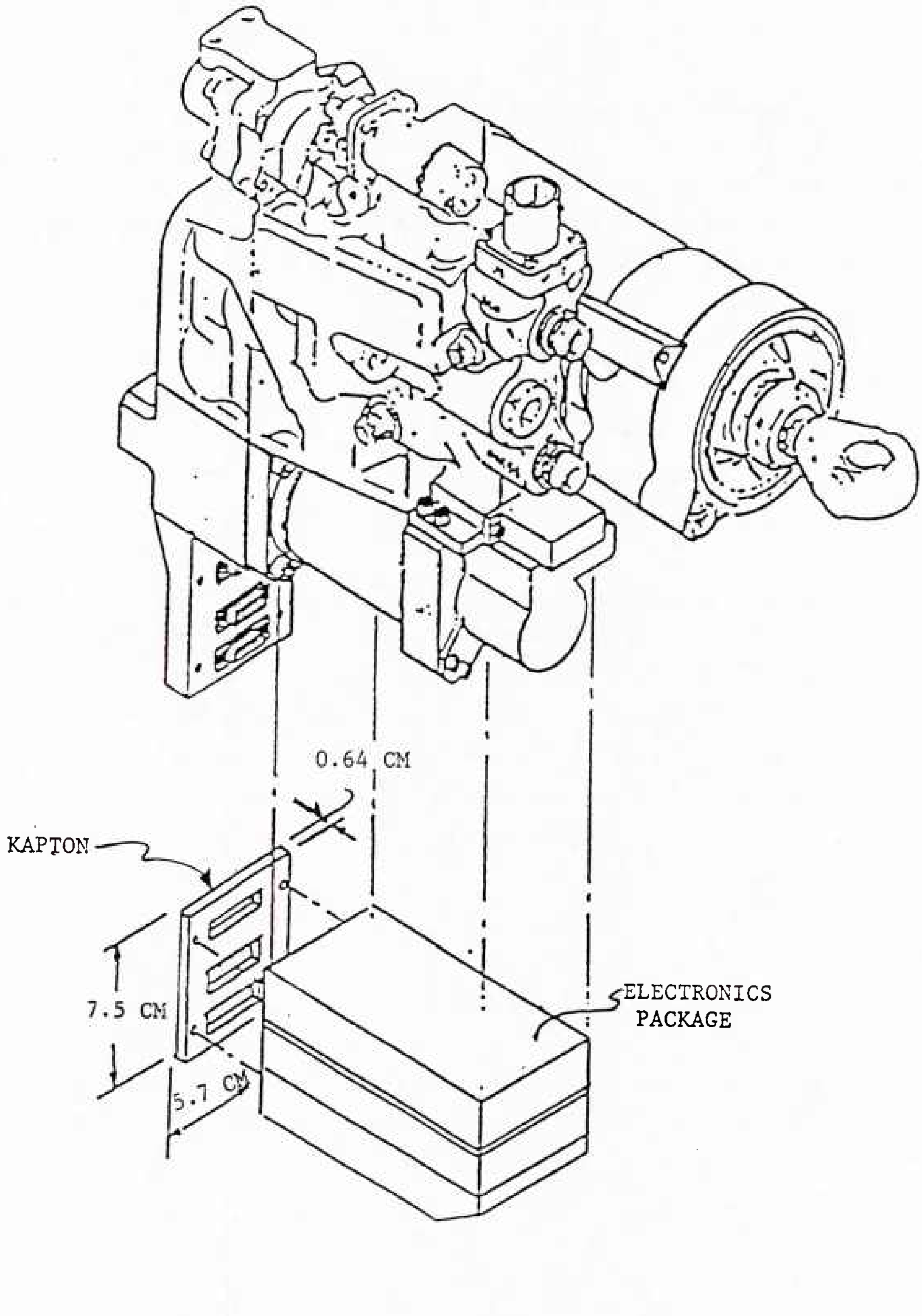


Figure 5. Insulating gasket between actuator and electronics

2.2 CONCEPT INTEGRATION TECHNIQUES

Flexible heat pipe cold plates can be designed to interface with the electronics package at different locations and orientations. Several integration techniques are described below:

2.2.1 Concept 1 - Mounted on Top Plate

The cold plate portion could be mounted on the top or bottom of the electronics package as illustrated in Figure 6. In this arrangement, heat is transferred from the cold plate to the heat sink through the flexible section leading from the side of the cold plate.

2.2.2 Concept 2 - Mounted on Sides

This cold plate could be mounted on either side of the electronics package as illustrated in Figure 7. The side that faces down in the wing would be the most desirable since gravity will assist the return of condensate from the condenser to the evaporator.

2.2.3 Concept 3 - Mounted Between Actuator and Electronics

The cold plate could be designed to sandwich between the actuator and the electronics package as illustrated in Figure 8. This concept would prevent conduction of heat from the actuator directly to the electronics. An insulating gasket type of material between the actuator and cold plate is required. This location is the most desirable, however, this concept would require that the heat pipe be inserted during assembly of the actuator.

Concept 2 was selected as the best arrangement for the Phase I proof-of-principle demonstration. This is the side mounted design which seems to integrate easily with the H.R. Textron electronics package. The flexible heat pipe cold plate can be supplied separately from the actuator which is a preferred packaging configuration. In this instance, installation of the heat pipe would best be done by the airframer. The cold plate and condenser section would be designed to be bolted to the structures after actuator installation.

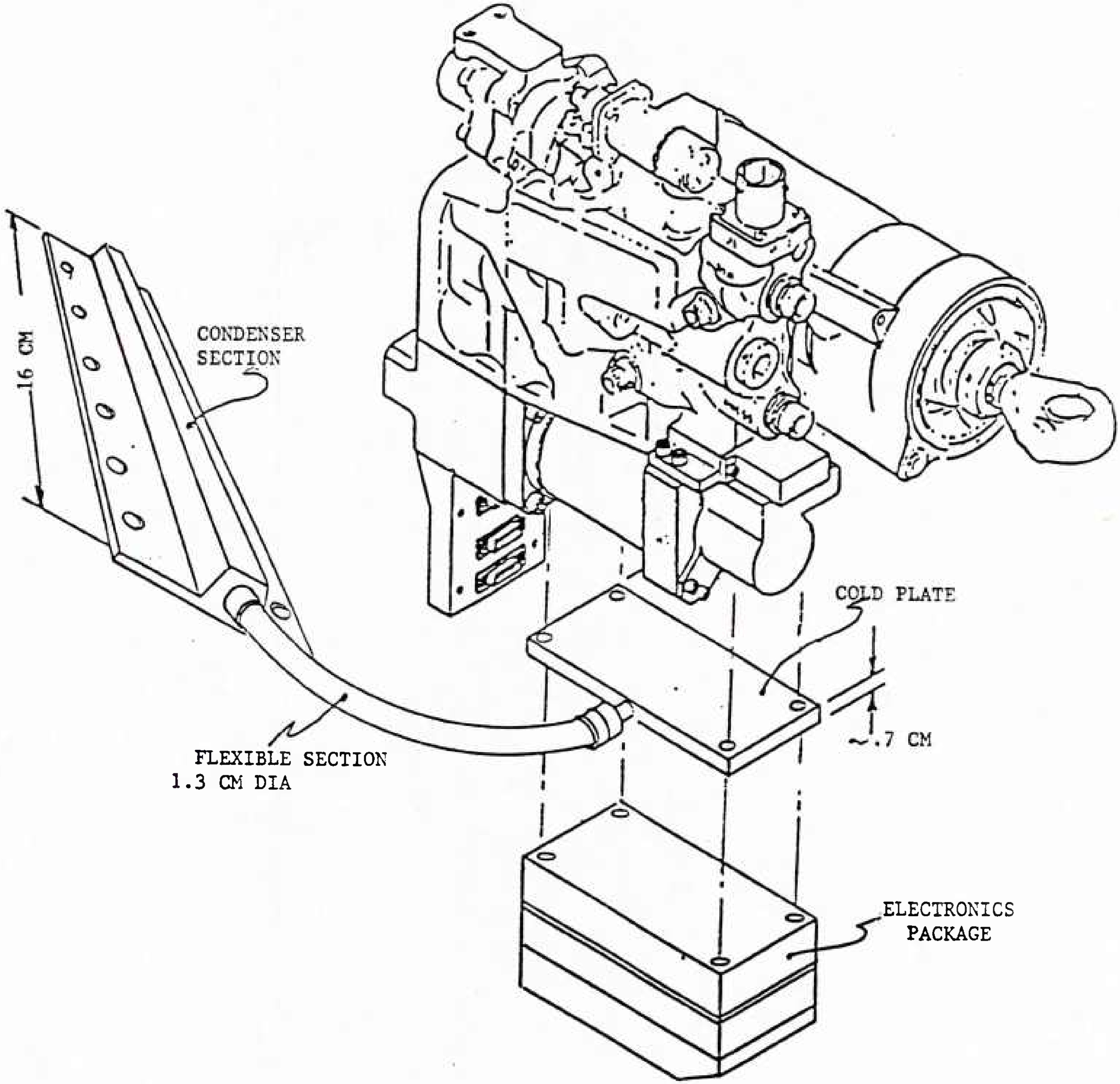


Figure 6. Cold plate mounted on top

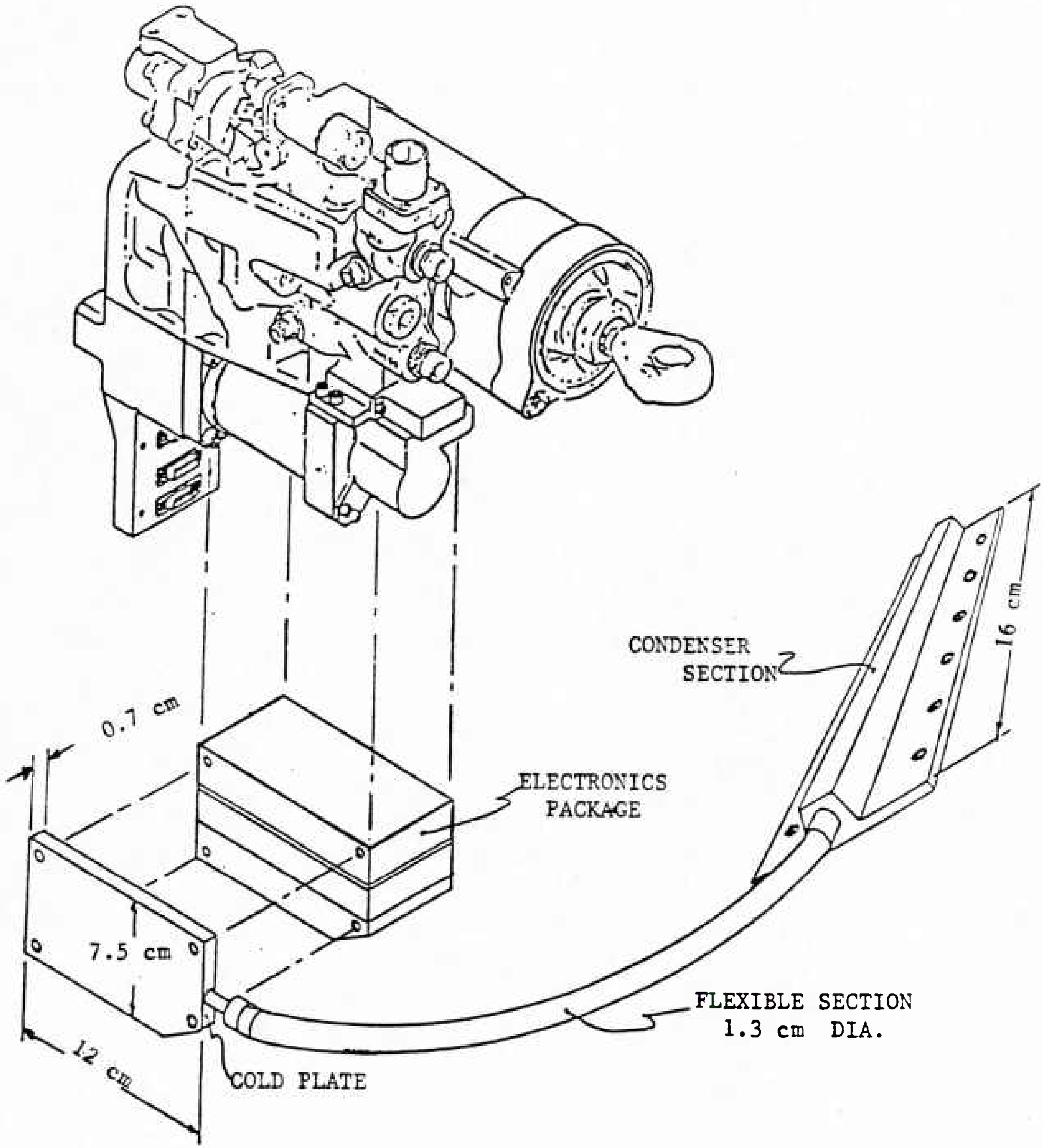


Figure 7. Cold plate mounted on the side

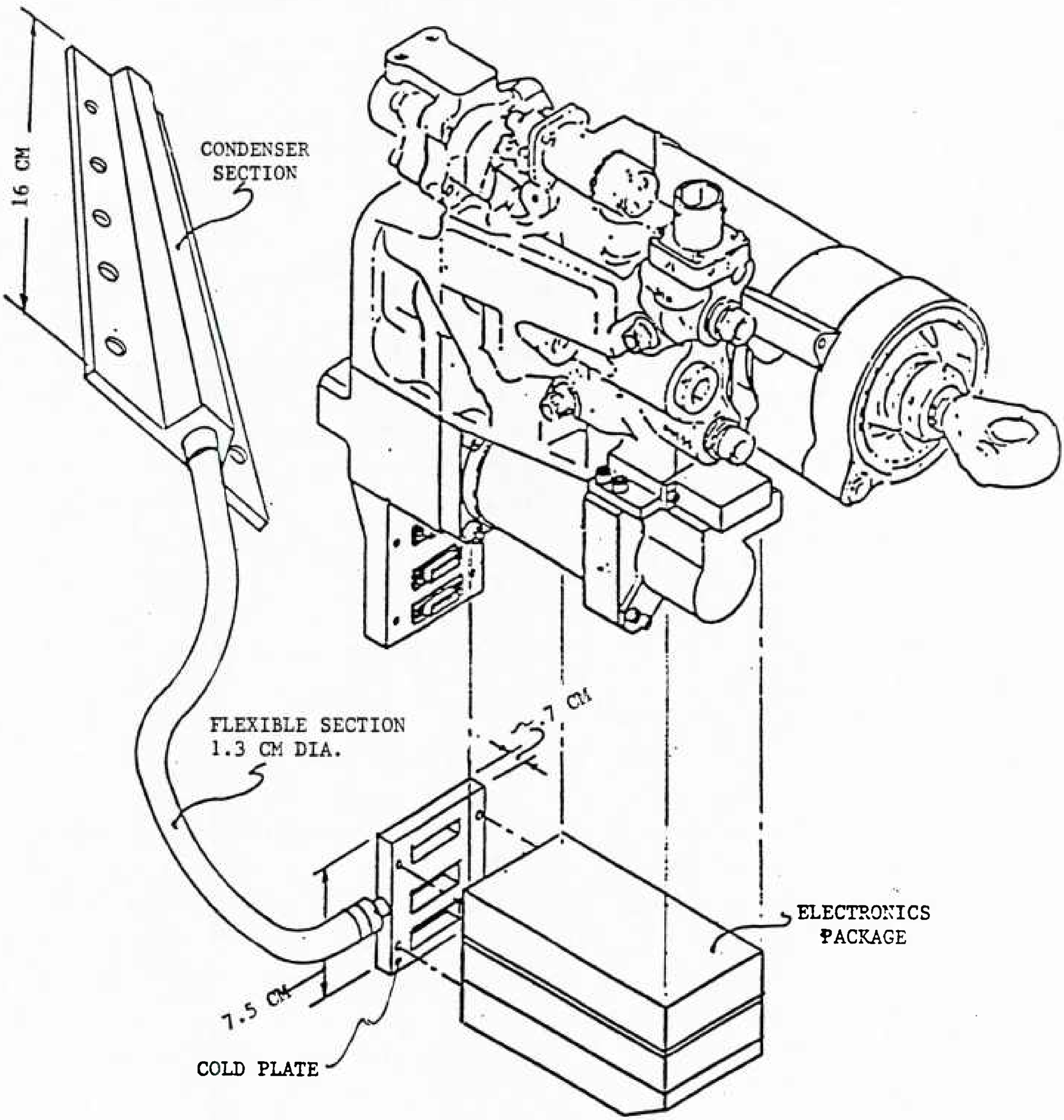


Figure 8. Cold plate between actuator and electronics

2.3 MATERIALS

The selected flexible heat pipe concept can be made from many different material/working fluid combinations. These combinations are listed below:

Material	Working Fluids
Aluminum	Ammonia or Acetone
Copper	Acetone, Methanol, Freon, or Water
Titanium	Water

Analysis showed that all these combinations would transfer the required thermal load. Since all material/working fluid systems will work, the selection criteria was based on internal vapor pressure and its effect on cold plate thickness. Because of the limited space available, it is important that the cold plate be as thin as possible.

The material/working fluid combinations that proved to be the best were copper/water or titanium/water. At the highest actuator temperature of 110°C, the differential pressure of water is only 5 psi. In addition, water has the highest power carrying capability of the fluids listed. Finally, water is non-toxic and non-flammable.

The other working fluids have much higher vapor pressures requiring thicker cold plate walls to contain the working fluid as shown in Table 2. Copper or titanium offer the thinnest cold plate design.

Titanium will provide a lighter, stronger wall material than copper; however, experience has shown that the sintered titanium powder wick material has trace amounts of iron. This iron reacts with water producing non-condensible gas in service which collect in the condenser and shuts down the heat pipe. High purity iron-free titanium powder is required to prevent this reaction. The work to identify a source of iron-free titanium powder is outside the scope of Phase I and will be done in Phase II.

Copper was selected as the wall material for the Phase I proof-of-principle heat pipe cold plate. This material is readily available and easy to fabricate.

The design of the copper/water flexible cold plate is covered in Section 2.4 below.

TABLE 2. Cold Plate Thickness As A Function Of Material/Working Fluid Combinations

Material	Working Fluid	Vapor Pressure Above atm @ 110°C	Cold Plate Thickness*
Aluminum	Ammonia	1082 psi	6.0 cm
Aluminum	Acetone	60 psi	1.7 cm
Copper	Acetone	60 psi	1.3 cm
Copper	Methanol	50 psi	1.3 cm
Copper	Freon 113	80 psi	1.5 cm
Copper	Water	5 psi	0.6 cm
Titanium	Water	5 psi	0.3 cm

*Calculational Rational found in Appendix B.

2.4 FLEXIBLE HEAT PIPE DESIGN

The overall design of the flexible heat pipe is shown in Figure 9; Figure 10 is an exploded view showing the internal design. The key features of the design are summarized in Table 3. The cold plate portion is approximately 13 cm long by 8 cm wide by 0.64 cm thick. The cold plate wick structure is shown in Figure 11. Capillary pumping is provided by sintered copper powder metal. The ten rectangular spaces are vapor flow passages. Between the vapor spaces are 0.15 cm wide intermediate sintered powder support posts. These posts provide multiple paths for liquid flow as well as providing a mechanical support structure between the top and bottom portion of the cold plate to withstand clamping loads and differential pressures. The wick is designed to distribute liquid to all areas of the cold plate to provide for uniform heat removal. A photograph of the wick cross-section is shown in Figure 12.

The ten vapor flow passages are manifolded together using the technique illustrated in Figure 13. This vapor manifold provides a smooth transition from the flat cold plate to the cylindrical flexible hose.

A copper expanded bellows was used for the flexible section. It is encased in a wire braided jacket for strength. In the final application, this bellows will undergo a number of small amplitude deflections as the actuator is stroked. Vendor data are currently not available for the fatigue life of this bellows when subjected to the cycle amplitude, internal pressure, and temperatures expected for this application. Extensive bellows fatigue testing is being planned in a Phase II effort.

The length of the flexible section was selected at 23 centimeters. This provided sufficient length for demonstration purposes. Heat pipe performance evaluations show that the flexible section can be made shorter or longer with little effect on heat pipe performance. This provides versatility for accommodating other actuator and heat sink installations.

This particular design used an artery to aid the return of working fluid from the condenser to the evaporator. An artery is used in a heat pipe as a low pressure drop means to return working fluid back to the evaporator cold plate. Thermacore has used solid wall flexible teflon tubing as the artery in other flexible heat pipes in the past, however,

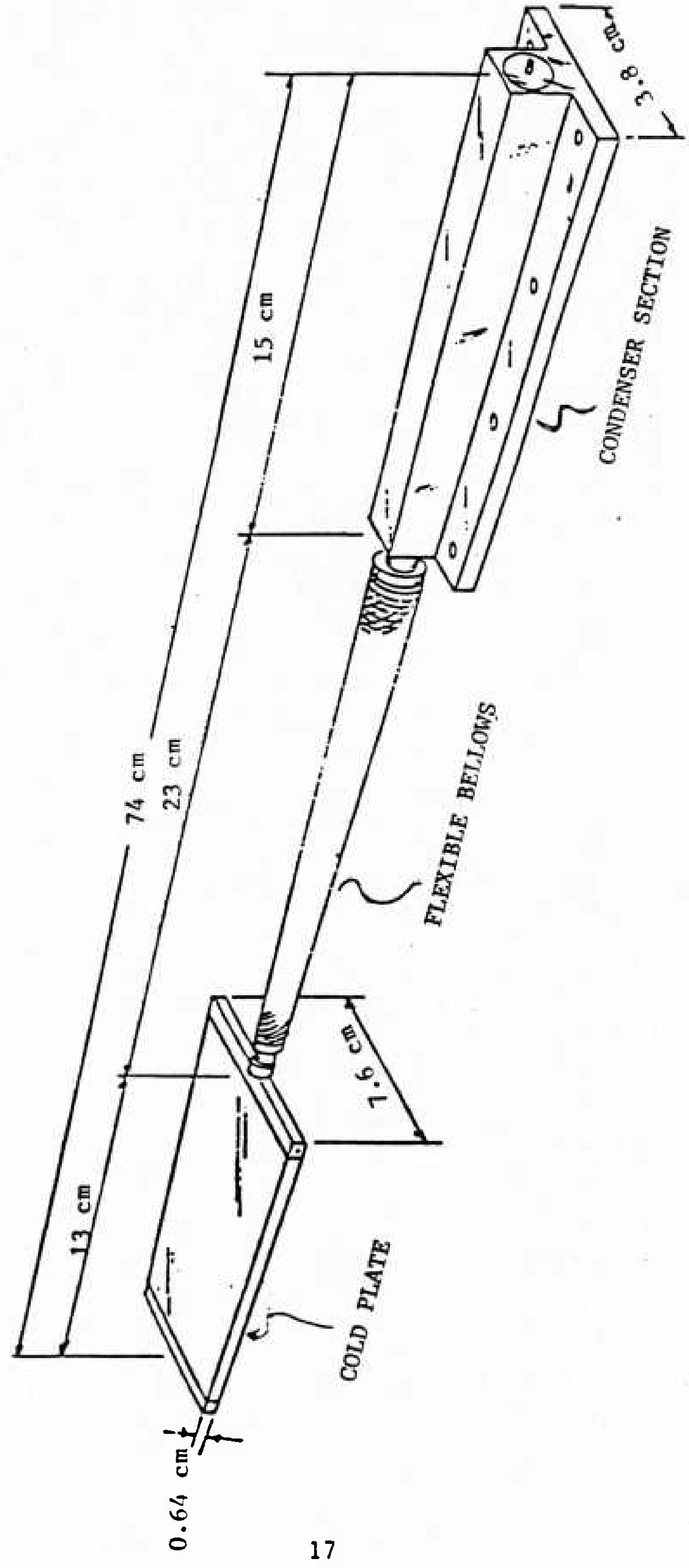


Figure 9. Flexible heat pipe cold plate design

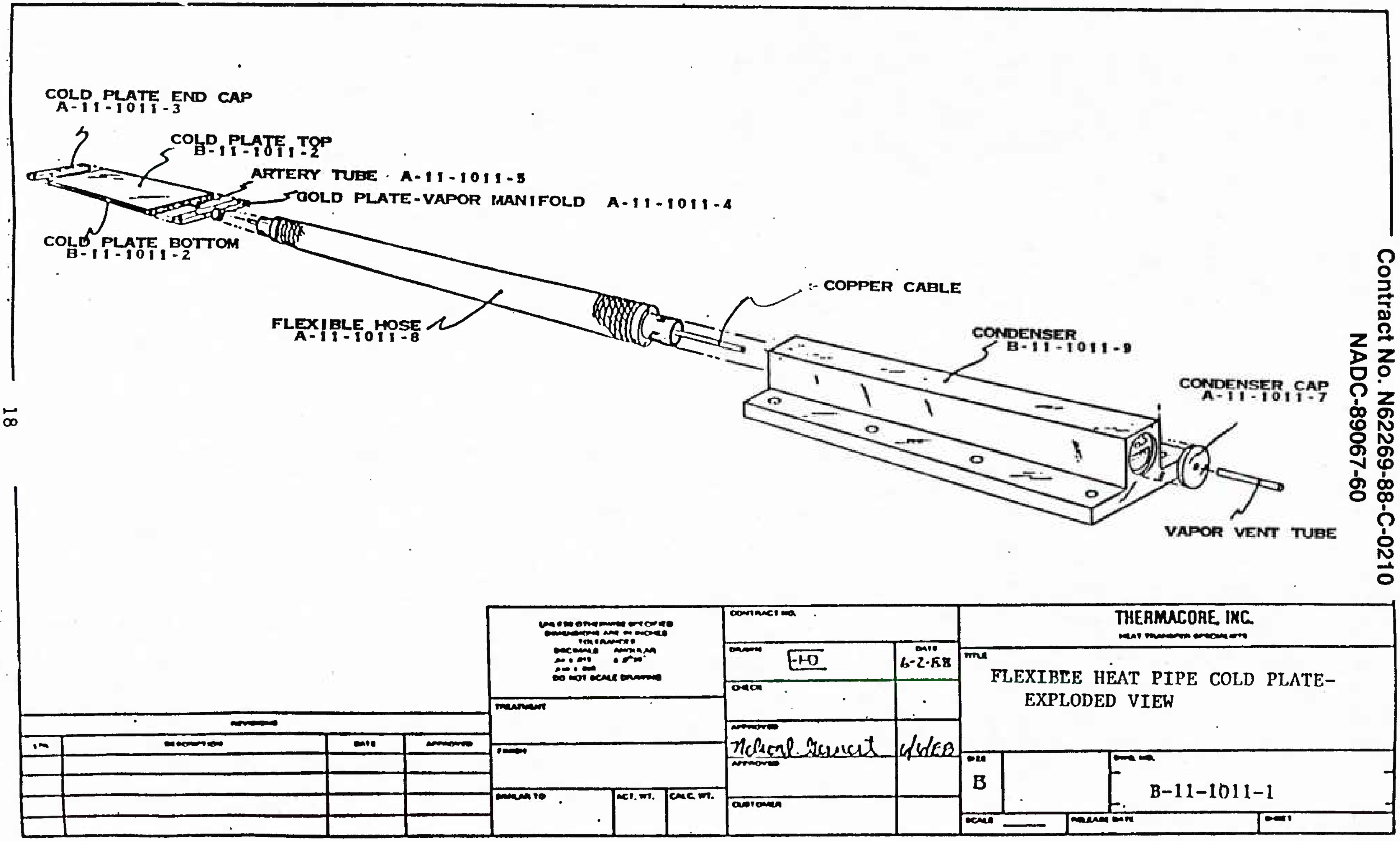
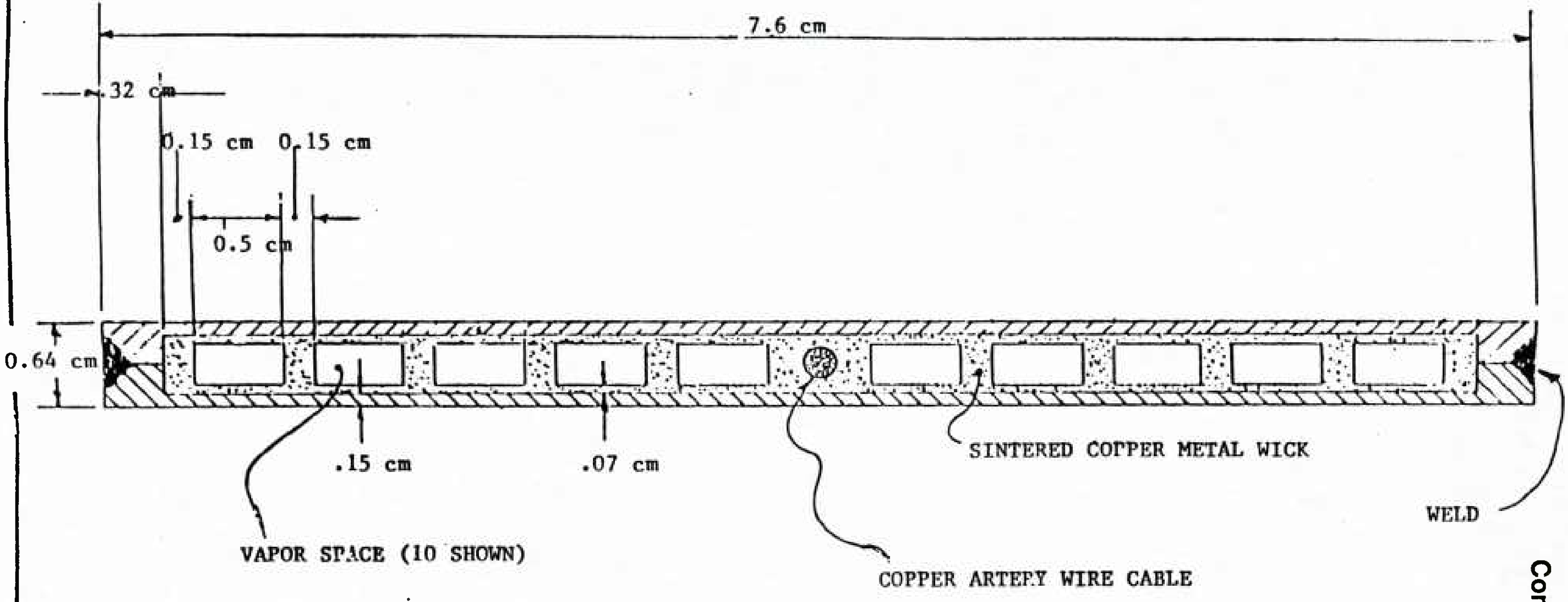


Figure 10. Exploded view - flexible heat pipe cold plate

TABLE 3. Phase I Flexible Heat Pipe Cold Plate Parameters

Parameters	Magnitude
Power (Watts)	44.5
Height of condenser above evaporator for rated power (cm)*	3.75
ΔT at rated power (°C)	<5
Operating Temperature Range (°C)	40 to 110°C
Fluid	Water
Fluid Charge (cc's)	20
Wall/Wick Materials	Copper
Weight	Cold plate 0.5 kg Flexible hose 0.2 kg Condenser 0.3 kg
Wick	
- Powder Mesh	-100, +150
- Permeability (cm^2)	2.5×10^{-7}
- Pore Size (cm)	0.003
- Thickness (cm)	0.07
Artery	
- Type	Braided Cable
- Vendor	New England Electric Wire Corp.
- Part Number	NE805-44B
- No. wires	805
- Wire size (mils)	1.97
Bellows	
- Length (cm)	23
- Diameter (cm)	1.27
- Vendor	Hydroflex Corp., Kansas
- Part Number	HF-233
- Maximum Rated Pressure (psi)	1750
Construction	
- Evaporator	Welded
- Bellows	Brazed
- End Caps	Brazed

*No requirement for minmum height specified in Phase I.



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TREATMENT	DRAWN EHD	DATE 6-1-88	TITLE COLD PLATE-WICK STRUCTURE	
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Figure 11. Cold plate wick structure

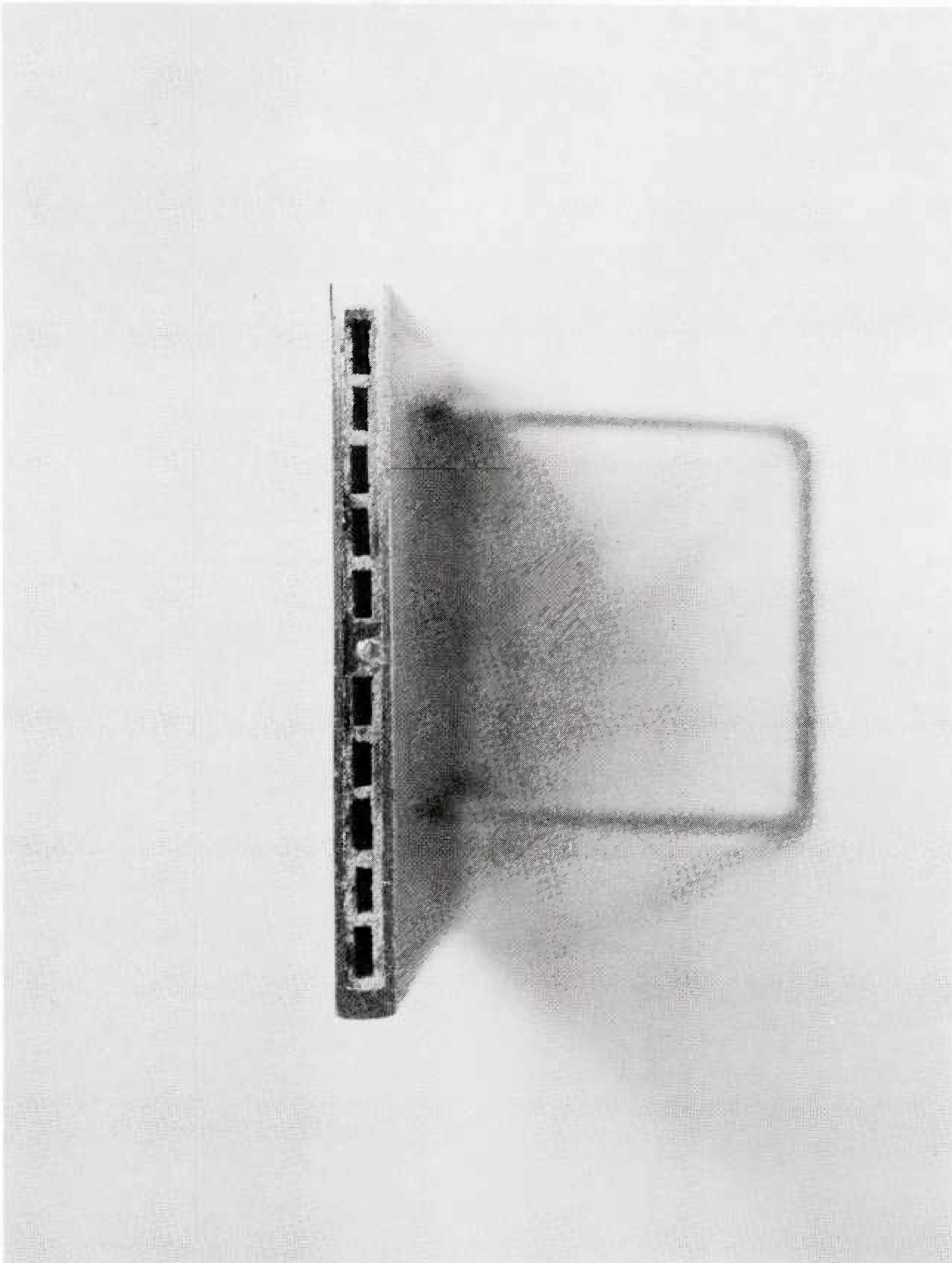

Figure 12. Photograph of evaporator wick structure

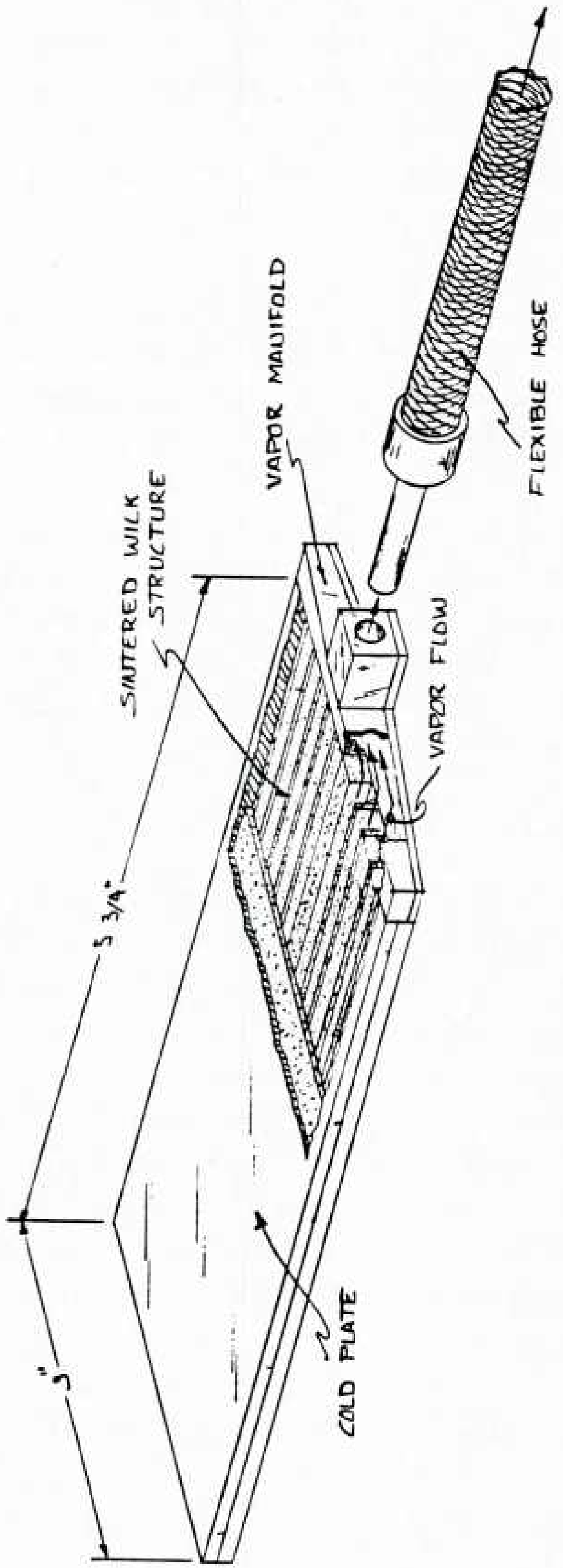

Figure 13. Cold plate-to-flexible hose vapor manifolding technique

there is a concern about damage occurring from the freezing of the working fluid inside the tubing. As a result, an alternate flexible artery material was evaluated. This alternate material is a bundle of copper braided wire. Liquid is returned through the capillary space between the wires. It is anticipated that this artery material will provide freeze protection by accommodating for the expansion of water after freezing. Thermacore is currently investigating the patent possibilities of this artery material.

The braided cable artery selected for the Phase I prototype is not considered optimum. Subsequent work shows that flexible arteries are available which could provide better performance. The effort required to select a more effective cable artery design will be covered in a Phase II effort.

The condenser section is 4 cm wide by 15 cm long and is designed to be bolted to a heat sink. The internal wick structure details are shown in Figure 14. Sintered copper powder is used to collect condensate from the condenser wall and transfer it to the braided cable artery embedded in the sintered powder. A photograph of the condenser wick and artery cable is shown in Figure 15.

2.5 TEST SET-UP AND RESULTS

The test set-up is shown schematically in Figure 16; Figure 17 is a photograph of the test set-up. The heat pipe was mounted on an optical bench for accurately determining its performance as a function of tilt angle. Heat input to the cold plate was achieved using two "Minco" thermofoil electrical resistance heaters glued to the top and bottom of the cold plate. Each heater provided approximately equal power to the cold plate; no attempt was made to proportion the power to represent heat imposed from the electronics and the hydraulics on one side of the cold plate and heat from compartment air on the other side. This proportioning was not considered necessary for this Phase I effort since cold plate power is limited by the capability of the artery. The effects of properly proportioning power on cold plate performance will be evaluated in Phase II. The heat pipe was cooled using a water cooled calorimeter that was clamped to the condenser section.

Heat pipe temperature was monitored using eleven type J thermocouples; four were attached to the cold plate, two were attached to

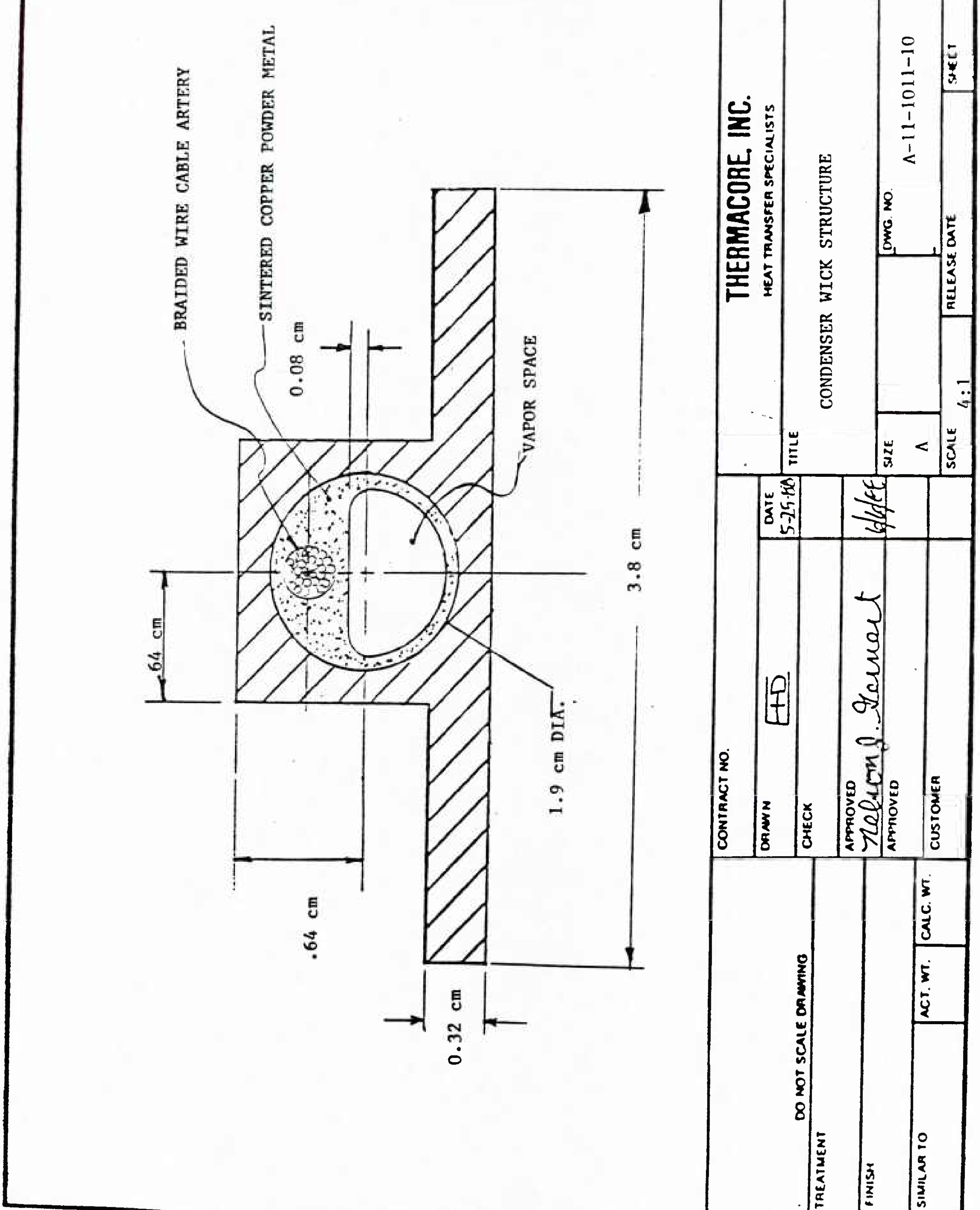

Figure 14. Cross-section of condenser section

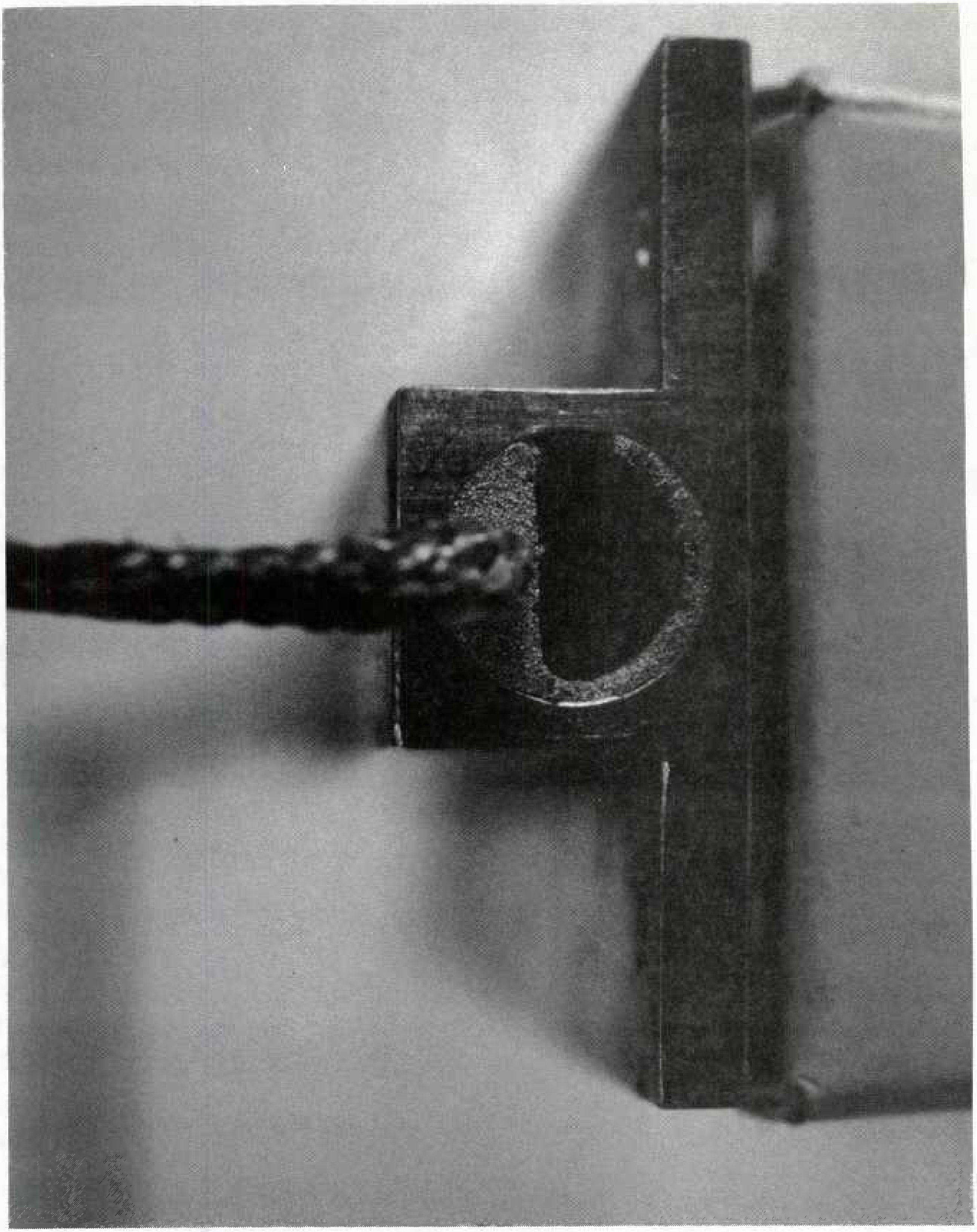

Figure 15. Photograph of condenser wick structure and artery cable

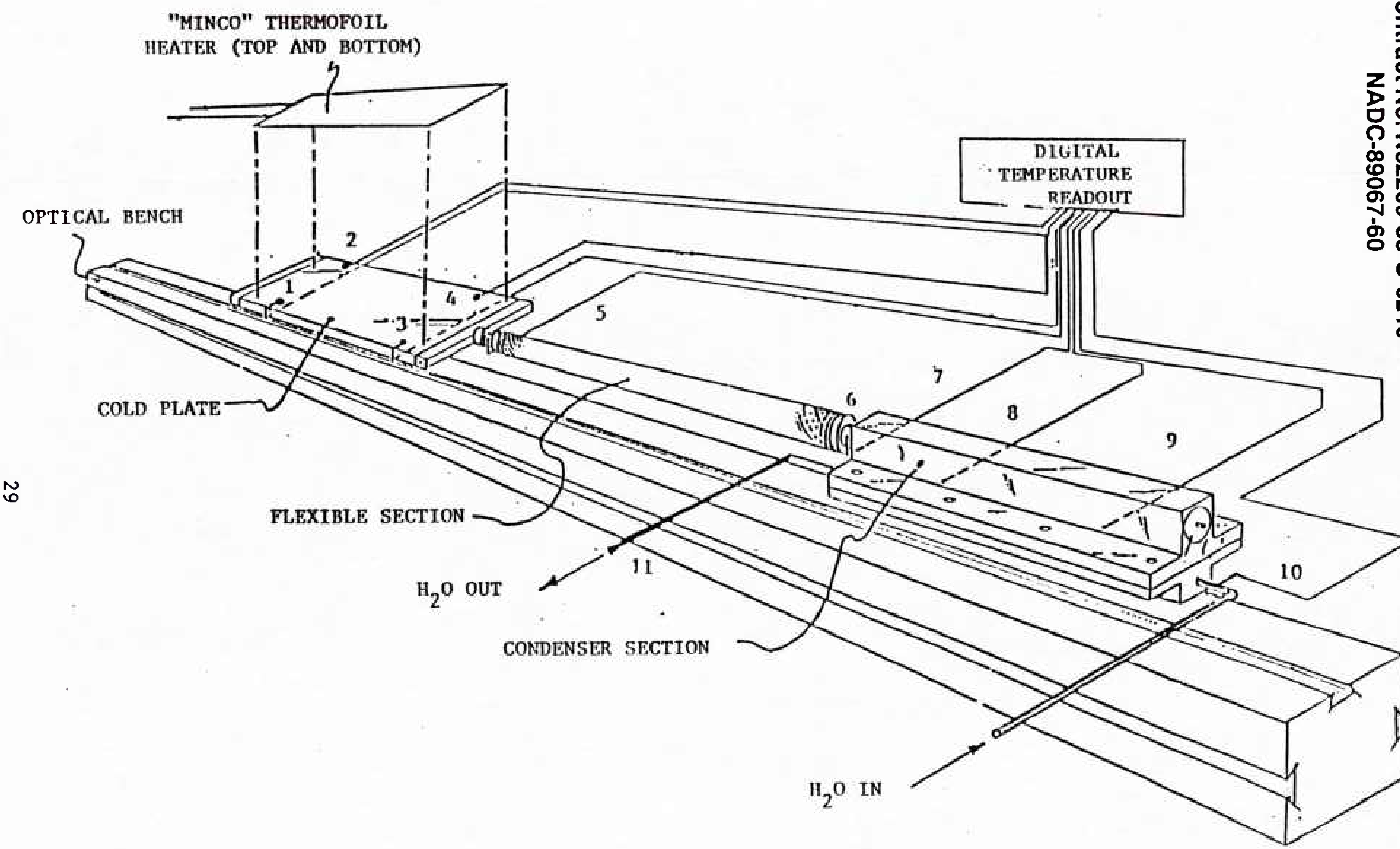

Figure 16. Test set-up for flexible heat pipe cold plate

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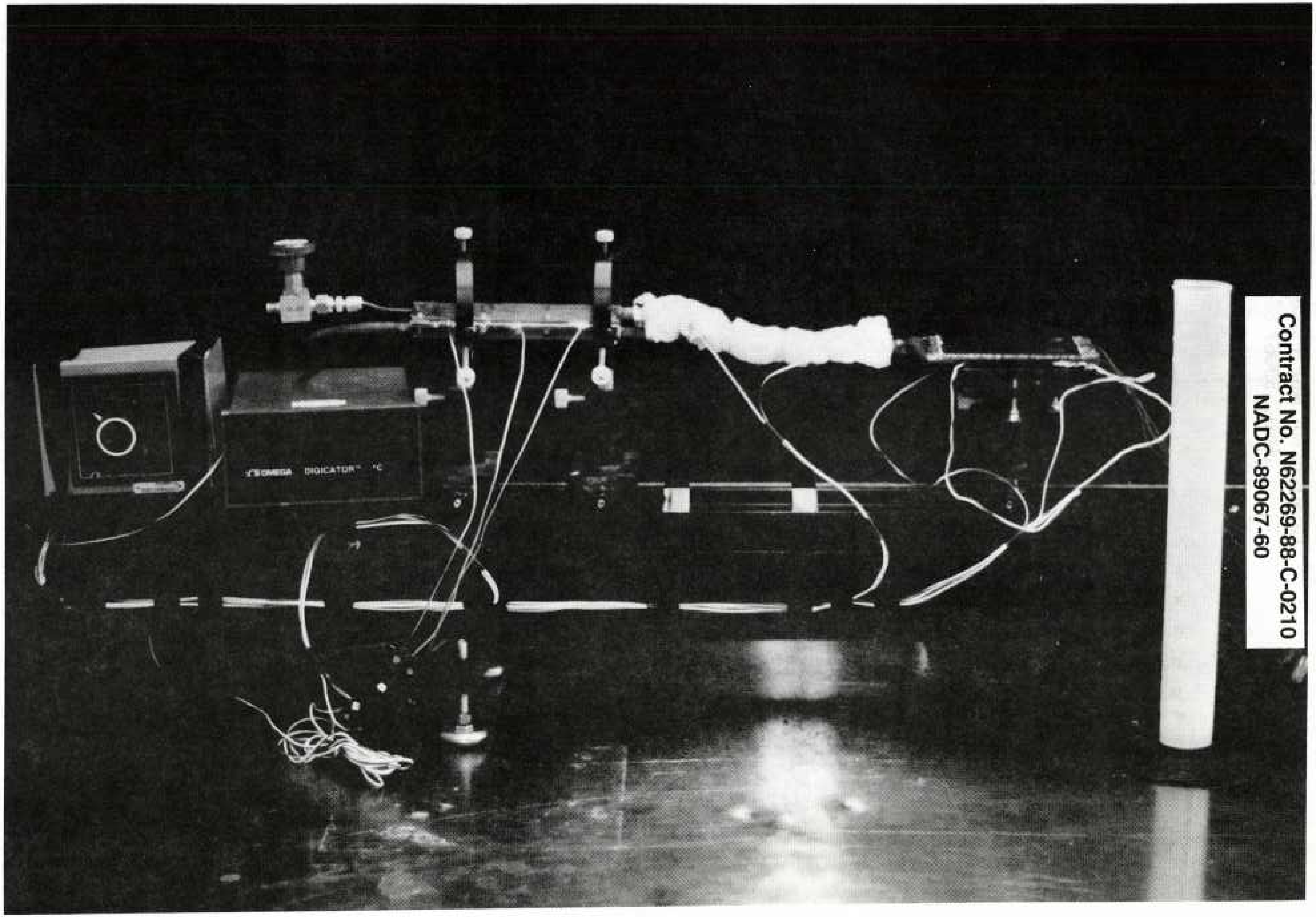


Figure 17. Photograph of test set-up

the flexible hose, three were attached to the condenser, and two measured water temperature at the inlet and outlet. Coolant flow rate was determined by using a stop watch and a graduated cylinder. Flow rate and coolant line delta-T information were used to determine output power. During testing, the heat pipe was insulated to limit heat losses.

The heat pipe cold plate was tested to determine its power carrying capability. The test results are shown plotted in Figure 18, as a function of maximum power versus vertical height of the condenser above the evaporator. The cold plate transported the 44.5 watt project goal with a cold plate wall-to-condenser wall temperature drop of less than 5°C. Output power as high as 60 watts was recorded. This performance exceeds the 44.5 watt requirement by a factor of 1.3.

Perhaps a design with less power margin could be made even smaller and lighter than the prototype fabricated in Phase I. The effort to reduce the size and weight of the cold plate will be conducted in a Phase II effort.

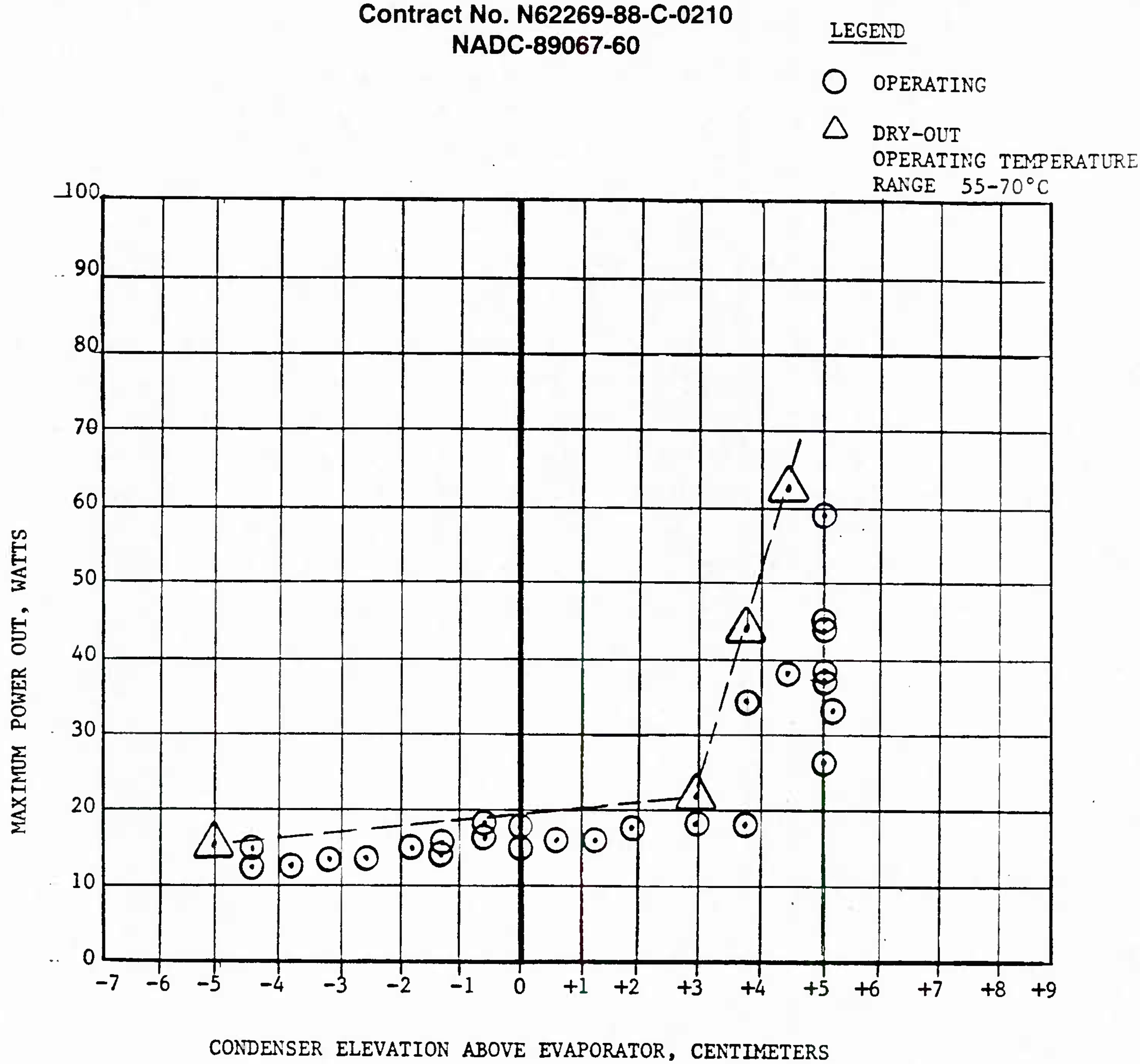


Figure 18. Cold plate measured power out vs. condenser elevation above evaporator

3.0 CONCLUSIONS AND RECOMMENDATIONS

This Phase I effort demonstrated the feasibility of using flexible heat pipe technology to provide a means to cool flight control actuator electronics. The specific conclusions and recommendations regarding key items of this program are summarized below:

1. For best heat pipe operation, the flexible heat pipe should be installed in the gravity assisted orientation if practicable. This is defined as the orientation with the heat sink condenser section elevated slightly above the cold plate. In this orientation, the working fluid can flow back to the evaporator without being capillary pumped. This orientation will provide the most reliable operation.

2. The length of the flexible section used in the demonstration heat pipe was selected at 23 centimeters. In the gravity aided orientation, the length of the flexible section length can be made shorter or longer with little effect on heat pipe performance. This provides versatility to reach the various heat sinks depending upon the actuator location.

3. The flexible bellows will undergo a number of small amplitude deflections as the actuator is stroked. Vendor data are currently not available for the fatigue life of these bellows when subjected to the cycle amplitude, internal pressure, and temperature expected for this application. Bellows fatigue testing is recommended.

4. The flexible heat pipe used a flexible copper cable to aid the return of working fluid from the condenser to the evaporator. Liquid is returned through the capillary space between the wires. This artery material is anticipated to provide freeze protection and to vent any non-condensible gas that may be trapped in its interior. The braided cable selected for the Phase I effort is not considered optimum. Work to select a more effective cable artery design is recommended.

5. A copper/water material system was used in the demonstration heat pipe. The resulting system weight will most likely be excessive for practical aircraft application. A significantly lighter system is possible if titanium or aluminum is used.

   In this Phase I effort, aluminum was not used because it becomes annealed and weakens after sintering; however, within the past year on IR&D funding, Thermacore developed a technique to sinter aluminum powder without annealing the aluminum pressure boundary. This technique uses plastic powder as a binder for the aluminum powder. The adhesive properties of the plastic bonds the particles together. In a Phase II effort, aluminum should again be investigated for use in this application. The use of aluminum has the potential to halve the weight of the Phase I cold plate.

6. Heat from the actuator electronics is transferred to surrounding aircraft structure where it is dissipated. Rough calculations done in Phase I help show that the structure is able to dissipate this heat during normal operation. Additional work is required to provide better assurance of adequate thermal capacity of the aircraft structure for normal and transient operating modes. This should be done using an analytical model verified through test. The cold plate heat load could be simulated by an I^2R heater attached to actual heat sink structure.

APPENDIX A

THERMAL ANALYSIS

APPENDIX A

A.1 Heat Load Imposed By Hydraulics

The maximum thermal operating conditions of the aileron actuator and the surrounding environment were established using the measured temperatures listed in Figure A1. These temperatures were taken from an F/A-18 during flight and were provided to Thermacore by NADC personnel (J. Koper and S. Donley).

According to the aileron temperature data, maximum actuator body temperatures can range from 77 to 93°C. The structure near the actuator to be used as the heat sink appears to range in temperature from 49°C to 54°C.

Thermacore prepared an analytical model of the heat transfer paths seen by the electronics when installed in a typical aircraft. The actuator body and the structure were each seen as an infinite heat source and sink, respectively. To reduce the amount of heat to be dissipated, the electronics will have to be insulated from the atmosphere and the actuator. The analysis assumed that a 0.64 cm thick layer of glass insulation would be used to encase the electronics package. This insulation has a thermal conductivity of 0.036 W/m·K. In addition, it was assumed that the heat pipe will maintain the electronics temperature within a few degrees of the sink temperature (49-54°C).

Under these conditions, an estimated maximum of 8 watts will be transferred by natural convection from the surrounding atmosphere through the insulation. Most of the heat to be dissipated, comes through the bolted interface between the electronics and the actuator. As much as 750 watts would need to be transferred if this interface was left non-insulated.

This heat load will be reduced to 5 watts using an insulation gasket made of 0.64 cm thick kapton polyamide film as shown in Figure A2. There is an additional 18.5 watts that would come through the three electrical connections. Four bolts are used to hold the electronics package in place. These bolts contribute approximately three watts. Based on this analysis, the total amount of heat to be transferred by the flexible heat pipe cold plate is 44.5 watts. The thermal analysis is presented below.

	MAXIMUM TEMPERATURE ATTAINED (°C) RUN 1	MAXIMUM TEMPERATURE ATTAINED (°C) RUN 2	MAXIMUM TEMPERATURE ATTAINED (°C) RUN 3
AILERON ACTUATOR			
Actuator Body	93	82	77
High Pressure Line	93	93	77
Structure Near Actuator	54	49	54

Figure A1. Measured aileron F-18 actuator temperatures

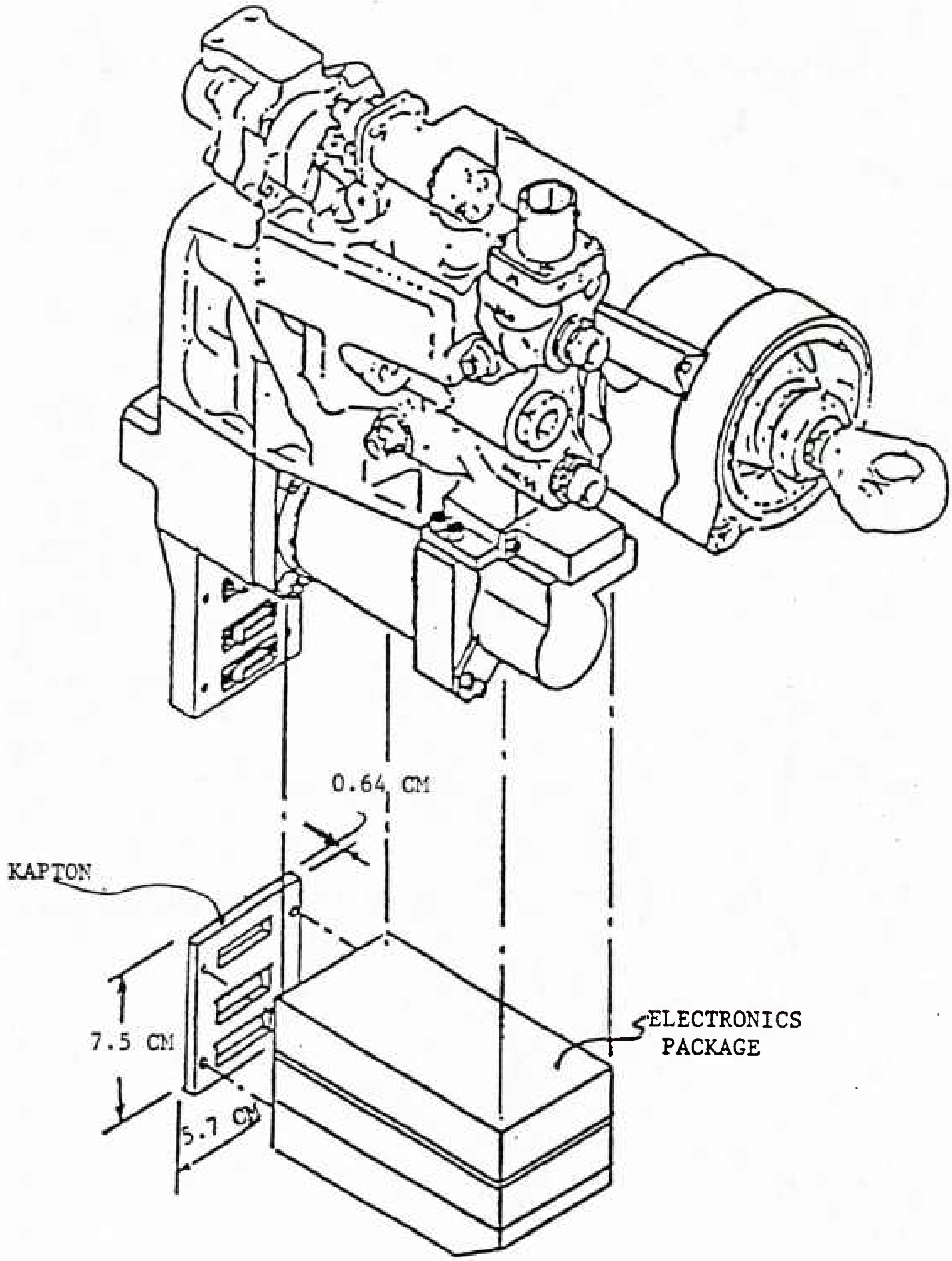


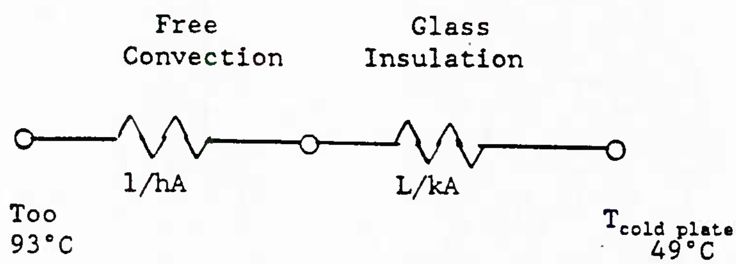
Figure A2. Insulating gasket between actuator and electronics

Estimation of Heat Load Into the Electronics from the Environment

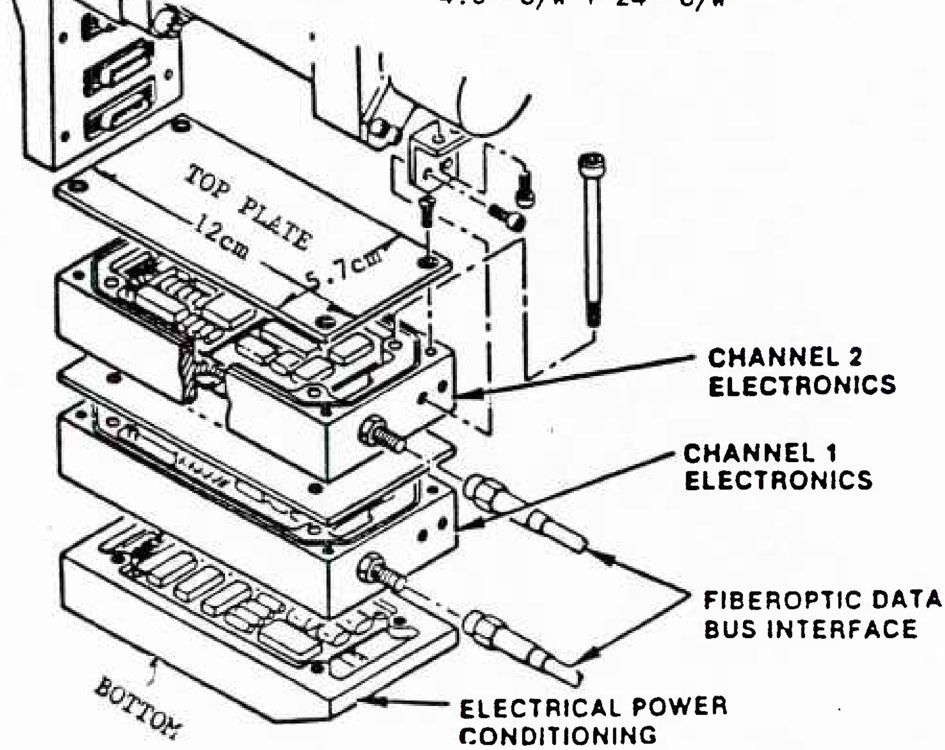
1. Top of Electronics at Plate (see figure below):

$$Q = hA\,\Delta T + kA\,\Delta T/L$$

Where,
$h = 30\ W/m^2{}^\circ C$
(see next page for free convection information)
$L = 0.006\ m$
$k = 0.036\ W/m\text{-}K$
$A = 0.007\ m^2$



$$= \frac{44^\circ C}{4.8\ ^\circ C/W + 24\ ^\circ C/W} = \underline{1.5\ W^*}$$



*The heat into the bottom will also be 1.5 W.

TABLE 15.1 APPROXIMATE VALUES OF THE CONVECTIVE HEAT-TRANSFER COEFFICIENT

Mechanism	h, Btu/hr ft^2 °F	h, W/(m$^2 \cdot$ K)
Free convection, air	1–10	5–50
Forced convection, air	5–50	25–250
Forced convection, water	50–3000	250–15 000
Boiling water	500–5000	2500–25 000
Condensing water vapor	1000–20 000	5000–100 000

Table from: "Fundamentals of Momentum, Heat and Mass Transfer" J.Welty, C. Wicks, R. Wilson, pp. 232, Copyright 1976. John Wiley & Sons, Inc.

2. Conduction Through Bolted Metal-to-Metal Interface:

$Q = \frac{kA\ \Delta T}{l}$ where, k/l = interface thermal resistance = 0.4 $W/cm^2 {}^\circ C$ (see next page)

$A = 43\ cm^2$

$= 0.4 \frac{W}{cm^2 {}^\circ C} \quad A\ (T_{hyd} - T_{elec})$

$= 17.2\ W/^\circ C\ (93^\circ C - 49^\circ C)$

$= 757\ W$

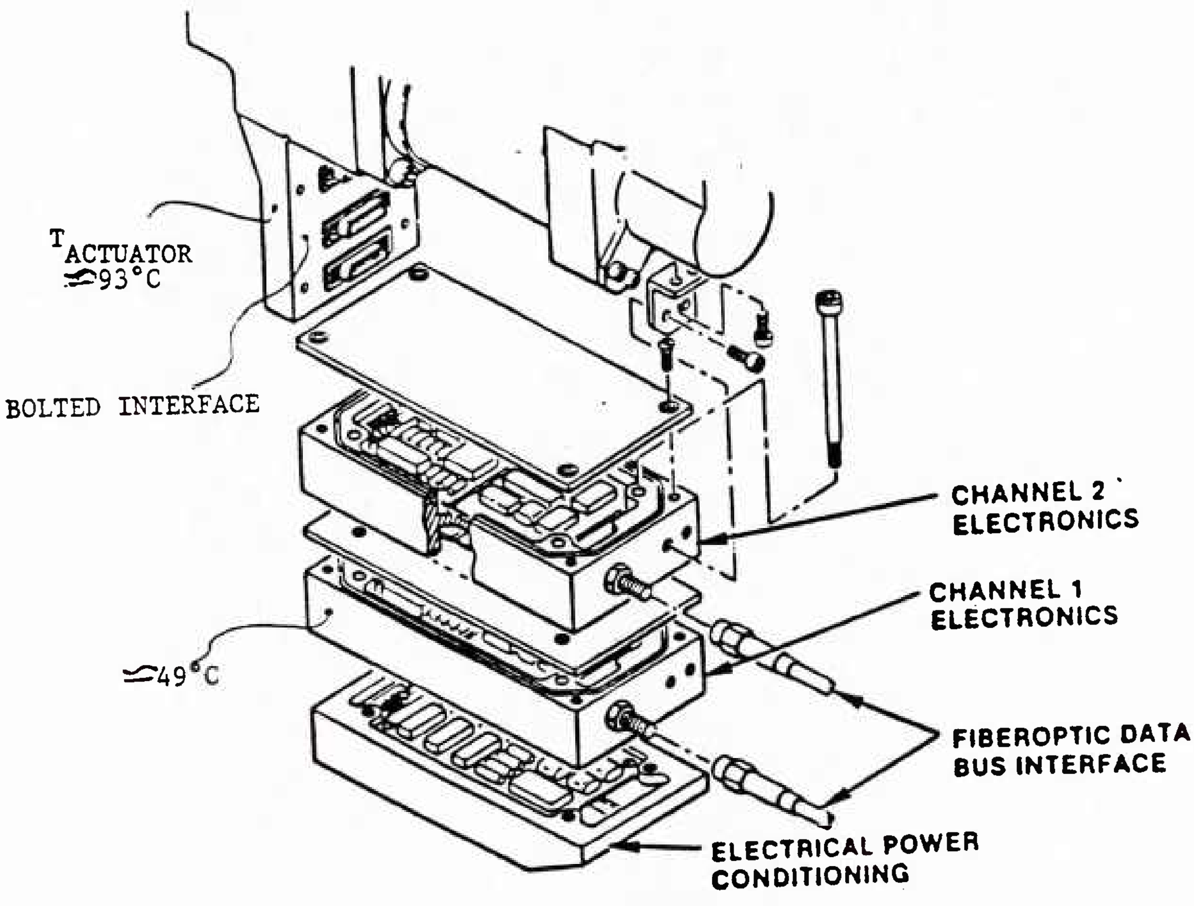


Table 3.2 Variation of thermal resistance for aluminum-aluminum interface (10 μm surface roughness) under 10^5 N/m^2 contact pressure with different interfacial fluids [1]

FLUID	THERMAL RESISTANCE, $R \times 10^4$ ($m^2 \cdot K/W$)
Air	2.75
Helium	1.05
Hydrogen	0.720
Silicone oil	0.525
Glycerine	0.265

Table taken from: "Fundamentals of Heat Transfer" Frank P. Incropera, David P. DeWitt, Copyright 1981, John Wiley & Sons, Inc., p. 72.

3. Conduction through interface with Kapton insulator:
(see next page for Kapton information)

$$Q = \frac{kA\ \Delta T}{l} = \frac{(0.163\ W/m^{\circ}C)\ (0.0043\ m^2)\ (93^{\circ}C - 49^{\circ}C)}{6.4 \times 10^{-3} m}$$

$$= 5 \text{ watts}$$

4. Heat into the side:

$$Q_{side} = hA\ \Delta T \quad + \quad kA\ \Delta T/L$$

$$Q_{side} = \frac{T\ oo \quad - \quad T_{cp}}{(1/hA) + (L/kA)}$$

Where,
$Too = 93^{\circ}C$
$T_{cp} = 49^{\circ}C$
$h = 30\ W/m^2\text{-}K$
$k = 0.036\ W/m^2\text{-}K$
$A = 0.009\ m^2$
$L = 0.006\ m$

$$Q_{side} = 2 \text{ watts}$$

KAPTON® Type H Film

25 μm (1 mil)

Contract No. N62269-88-C-0210
NADC-89067-60

PHYSICAL PROPERTIES

PHYSICAL PROPERTIES	TYPICAL VALUES 78K (−195°C)	296K (23°C)	473K (200°C)	TEST METHOD
Ultimate Tensile (MD) Strength, MPa (psi)	241 (35,000)	172 (25,000)	117 (17,000)	ASTM D-882-64T
Yield Point (MD) at 3%, MPa (psi)		69 (10,000)	41 (6,000)	ASTM D-882-64T
Stress to Produce (MD) 5% Elongation, MPa (psi)		90 (13,000)	59 (8,500)	ASTM D-882-64T ASTM D-882-64T
Ultimate Elongation (MD)%	2	70	90	ASTM D-882-64T
Tensile Modulus, GPa (MD) (psi)	3.5 (510,000)	3.0 (430,000)	1.86 (260,000)	ASTM D-882-64T
Impact Strength, J/mm (kg-cm)		23 (6)		Du Pont Pneumatic Impact Test
Folding Endurance MIT		10,000 cycles		ASTM D-2176-63T
Tear Strength—Propagating (Elmendorf), g		8		ASTM D-1922-61T
Tear Strength—Initial (Graves), g (g/mil)		510 (510)		ASTM D-1004-61
Density, g/cm³		1.42		ASTM D-1505-63T
Coefficient of Friction Kinetic (Film-to-Film)		.42		ASTM D-1894-63
Refractive Index (Becke Line)		1.78		Encyclopaedic Dictionary of Physics, Vol. 1
Poisson's Ratio		.34		Ave. 3 Samples Elongated at 5%, 7%, 10%

MD—Machine Direction

THERMAL PROPERTIES

THERMAL PROPERTIES	TYPICAL VALUES	TEST CONDITION	TEST METHOD
Melting Point	NONE		
Zero Strength Temperature	1088K (815°C)	.14 MPa (20 psi) load for 5 seconds	Du Pont Hot Bar Test
Coefficient of Linear Expansion	2.0×10^{-5} m/m/K (2.0×10^{-5} in/in/°C)	259 to 311K (−14°C to 38°C)	ASTM D-696-44
Coefficient of Thermal Conductivity, W/m·K $\left(\frac{(cal)(cm)}{(cm^2)(sec)(°C)}\right)$	0.155 (3.72×10^{-4}) 0.163 (3.89×10^{-4}) 0.178 (4.26×10^{-4}) 0.189 (4.51×10^{-4})	296K (23°C) 348K (75°C) 473K (200°C) 573K (300°C)	Model TC-1000 Twin Heatmaster Comparative Tester
Specific Heat	1.09 (.261)	J/g·K (cal/g/°C)	Differential Calorimetry
Flammability	94 VTM-0		UL-94 (1-24-80)
Shrinkage	(See chart on Page 7)		
Heat Sealability	Not Heat Sealable		
Limiting Oxygen Index	100H-38		ASTM D-2863-74
Smoke Generation	100H − DM = less than 1	NBS Smoke Chamber	NFPA-258 procedures
Glass Transition Temperature (Tg)	A second order transition occurs in KAPTON between 633K (360°C) and 683K (410°C). This is assumed to be the glass transition temperature. Different measurement techniques produce different results within the above temperature range.		

5. Heat Load Imposed by Electrical Connection and Four Mounting Bolts

Purpose: To estimate the amount of excess heat coming through the electrical connections and attachments bolts.

Description: Connector information is listed on the next page. Two 32 pin and one 25 pin connectors are used. Wire is #26AWG with teflon insulation. Four mounting bolts are used to hold the electronics in position. These bolts are ¼"-20 stainless steel.

Analysis:

Pin Connectors: Two 31 pin
One 25 pin

Estimation of heat load coming through wires.

Equivalent single wire:

$$\text{X - sec. Area} = \frac{\pi D^2}{4} \times (31 + 31 + 25)$$

$$= \frac{\pi (0.0159")^2}{4} \times 87 \text{ wires}$$

$$= 0.0173 \text{ in}^2 \text{ or } 1.12 \times 10^{-5} \text{m}^2$$

$$Q = \frac{kA\,\Delta T}{l}$$

where, l = cold plate thk + Kapton Insulator = 0.25" + 0.25" = 0.127 m

$$= \frac{(391\text{ W/m°C})\,(1.12 \times 10^{-5}\text{m}^2)\,(93°\text{C} - 49°\text{C})}{0.0127\text{ m}}$$

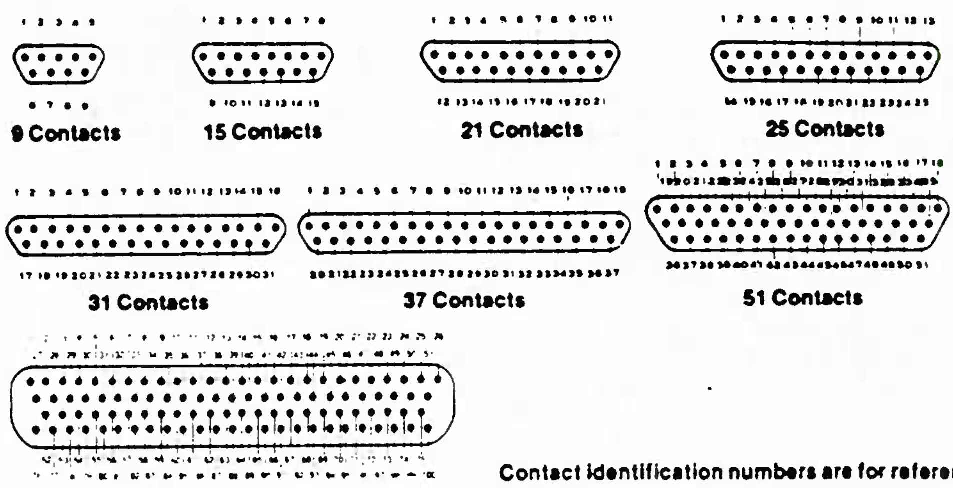
$$= \underline{15 \text{ watts}}$$

where, k_{copper} = 391 W/m°C

Contact Arrangements

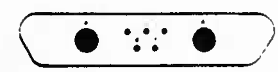
Face view of pin insert — use reverse order for socket side

Standard

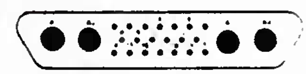


9 Contacts · 15 Contacts · 21 Contacts · 25 Contacts · 31 Contacts · 37 Contacts · 51 Contacts · 100 Contacts

Coaxial/Power



5 Micro Contacts
2 Coax or 2 Power



20 Micro Contacts
4 Coax or 4 Power
(Not MS)

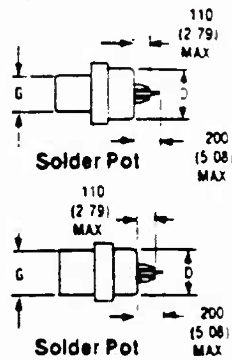
Contact identification numbers are for reference only and do not appear on insulator or connector body.

Shell Dimensions (conforms to MIL-C-83513)

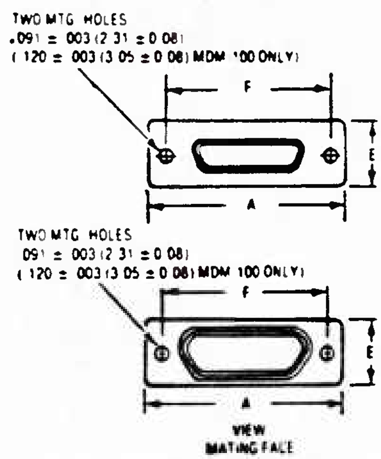
MDM

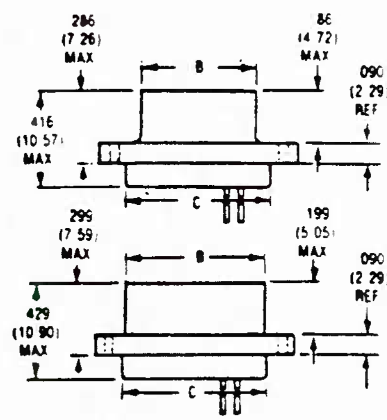
Plug

Receptacle



Solder Pot



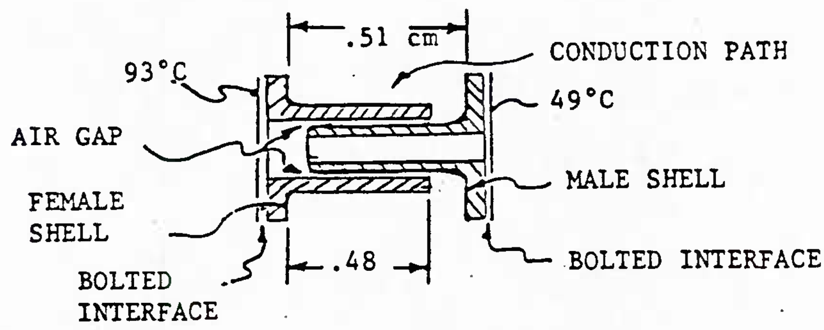


Part Number by Shell Size	A Max.	B Max.	C Max.	D Max.	E Max.	F ± .005 (0.13)	G Max.	Average Weights** oz (gm) ± 5%
MDM-9P*	.785 (19.94)	.333 (8.46)	.400 (10.16)	.270 (6.86)	.308 (7.82)	.565 (14.35)	.184 (4.65)	.063 (1.79)
MDM-9S*	.785 (19.94)	.401 (10.16)	.400 (10.16)	.270 (6.86)	.308 (7.82)	.565 (14.35)	.252 (6.38)	.062 (1.77)
MDM-15P*	.935 (23.75)	.483 (12.27)	.550 (13.97)	.270 (6.86)	.308 (7.82)	.715 (18.16)	.184 (4.65)	.084 (2.39)
MDM-15S*	.935 (23.75)	.551 (13.97)	.550 (13.97)	.270 (6.86)	.308 (7.82)	.715 (18.16)	.252 (6.38)	.083 (2.37)
MDM-21P*	1.085 (27.56)	.633 (16.08)	.700 (17.78)	.270 (6.86)	.308 (7.82)	.865 (21.97)	.184 (4.65)	.105 (2.99)
MDM-21S*	1.085 (27.56)	.701 (17.78)	.700 (17.78)	.270 (6.86)	.308 (7.82)	.865 (21.97)	.252 (6.38)	.104 (2.97)
MDM-25P*	1.185 (30.10)	.733 (18.62)	.800 (20.32)	.270 (6.86)	.308 (7.82)	.965 (24.51)	.184 (4.65)	.119 (3.39)
MDM-25S*	1.185 (30.10)	.801 (20.32)	.800 (20.32)	.270 (6.86)	.308 (7.82)	.965 (24.51)	.252 (6.38)	.118 (3.36)
MDM-31P*	1.335 (33.91)	.883 (22.43)	.950 (24.13)	.270 (6.86)	.308 (7.82)	1.115 (28.32)	.184 (4.65)	.140 (3.99)
MDM-31S*	1.335 (33.91)	.951 (24.13)	.950 (24.13)	.270 (6.86)	.308 (7.82)	1.115 (28.32)	.252 (6.38)	.139 (3.96)
MDM-37P*	1.485 (37.72)	1.033 (26.24)	1.100 (27.94)	.270 (6.86)	.308 (7.82)	1.265 (32.13)	.184 (4.65)	.161 (4.59)
MDM-37S*	1.485 (37.72)	1.101 (27.94)	1.100 (27.94)	.270 (6.86)	.308 (7.82)	1.265 (32.13)	.252 (6.38)	.160 (4.56)
MDM-51P*	1.435 (36.45)	.983 (24.97)	1.050 (26.67)	.310 (7.87)	.351 (8.92)	1.215 (30.86)	.227 (5.74)	.193 (5.50)
MDM-51S*	1.435 (36.45)	1.051 (26.67)	1.050 (26.67)	.310 (7.87)	.351 (8.92)	1.215 (30.86)	.295 (7.47)	.188 (5.35)
MDM-100P*	2.170 (55.25)	1.383 (35.23)	1.442 (36.97)	.360 (9.14)	.394 (10.16)	1.800 (45.72)	.270 (6.86)	.500 (14.30)
MDM-100S*	2.170 (55.25)	1.515 (38.48)	1.442 (36.97)	.360 (9.14)	.394 (10.16)	1.800 (45.72)	.394 (10.16)	1.040 (29.50)

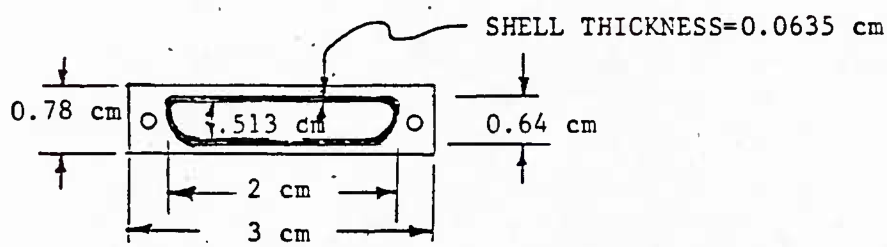
*Add lead type and length, see How to Order
**Weight given is 1/2", uninsulated, solid, #25 AWG gold plated copper pigtails

■ Estimation of Heat Load Across Connector Shells.

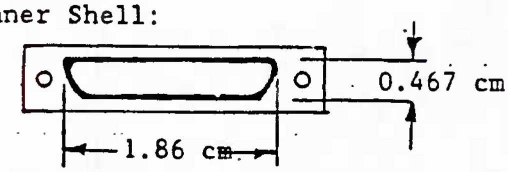
<u>Cross-sectional view of connector assembly</u>:

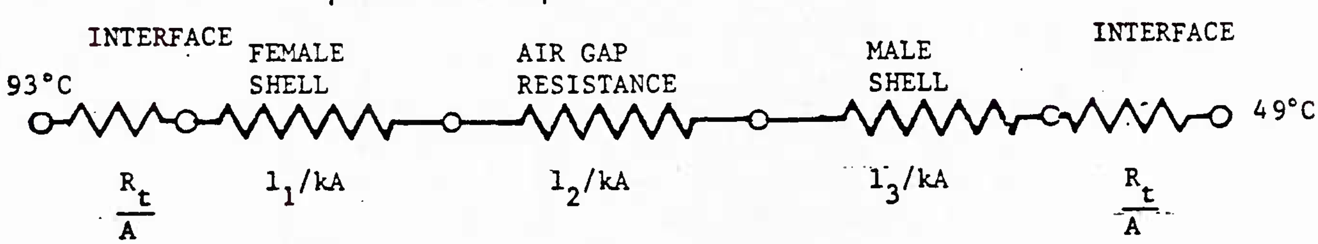


Outer Shell of 25 Pin Connector:



Inner Shell:





where,

$$A_{interface} = (0.78 \text{ cm} \times 3 \text{ cm}) - (0.64 \text{ cm} \times 2 \text{ cm}) = 1.06 \text{ cm}^2$$

$$A_{female\ shell} = (2 \text{ cm} + 0.64 \text{ cm}) \times 2 \times 0.0635 \text{ cm} = .33 \text{ cm}^2$$

$$A_{air\ gap} = (2 \text{ cm} + 0.64 \text{ cm}) \times 2 \times 0.4 \text{ cm} = 2.1 \text{ cm}^2$$

$$A_{male\ shell} = (1.86 \text{ cm} + 0.467 \text{ cm}) \times 2 \times ..0635 \text{ cm} = 2.9 \text{ cm}^2$$

$$A_{interface} = (0.78 \text{ cm} \times 3 \text{ cm}) - (0.467 \text{ cm} \times 1.86 \text{ cm}) = 1.47 \text{ cm}^2$$

$R_t = 2.75 \times 10^{-4}\ m^2 \cdot {}^\circ C/W$

$l_1 = 0.48\ cm/2$

$l_2 = (0.513\ cm - 0.467\ cm)/2 = 0.023\ cm$

$l_3 = 0.46\ cm/2$

$k_A = 180\ W/m^\circ C$

$Al = 30 \times 10^{-3}\ W/m^\circ C$ @ $T = (93^\circ C + 49^\circ C)/2$

$$R_{tot} = \frac{R_t}{A} + \frac{l_1}{kA} + \frac{l_2}{kA} + \frac{l_3}{kA} + \frac{R_t}{A}$$

$$= \frac{2.75 \times 10^{-4}\ m^2 {}^\circ C/W}{1.06 \times 10^{-4}\ m^2} + \frac{2.3 \times 10^{-3}\ m}{(180\ W/m^\circ C)(3.3 \times 10^{-5}\ m^2)}$$

$$+ \frac{2.3 \times 10^{-4}\ m}{(30 \times 10^{-3} W/m^\circ C)(2.1 \times 10^{-4}\ m^2)} + \frac{2.3 \times 10^{-3}\ m}{(180\ W/m^\circ C)(2.9 \times 10^{-5}\ m^2)}$$

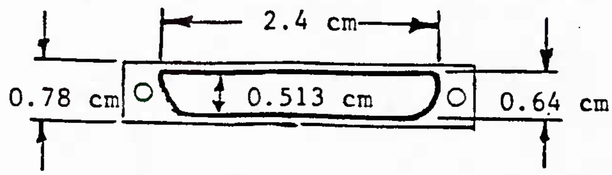
$$+ \frac{2.75 \times 10^{-4}\ m^2 {}^\circ C/W}{1.47 \times 10^{-4}\ m^2} = 42^\circ C/W$$

Heat Load Through Shell, Q_{25}:

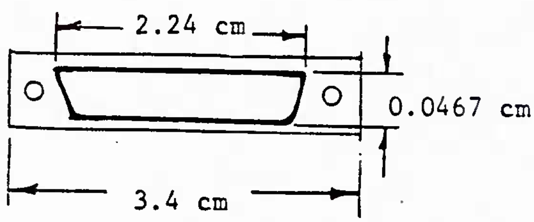
$$Q_{25} = \frac{\Delta T}{R_{tot}} = \frac{93^\circ C - 49^\circ C}{42^\circ C/W} = 1.05\ W$$

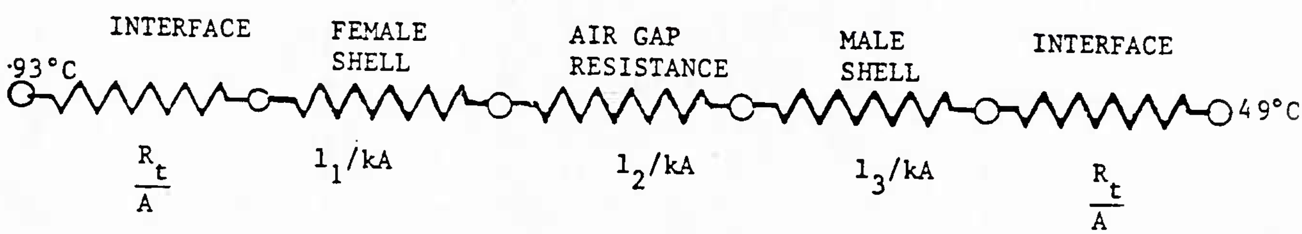
<u>Shell of 31 Pin Connector</u>:

Outer Shell:



Inner Shell:





Area Calculation:

$A_{interface}$ = (3.4 cm x 0.78 cm)-(2.4 cm x 0.64 cm) = 1.12 cm^2

$A_{female\ shell}$ = (2.4 cm + 0.64 cm) x 2 x 0.0635 cm = 0.386 cm^2

$A_{air\ gap}$ = (2.4 cm + 0.64 cm) x 2 x 0.4 cm = 2.43 cm^2

$A_{male\ shell}$ = (2.24 cm + 0.467 cm) x 2 x 0.0635 cm = 0.34 cm^2

$A_{interface}$ = (0.78 cm x 3.4 cm)-(0.467 cm x 2.24 cm) = 1.6 cm^2

where,

$R_t = 2.75 \times 10^{-4}\ m^2{}^\circ C/W$

$l_1 = 0.48\ cm/2 = 0.24\ cm$

$l_2 = (0.513\ cm - 0.467\ cm)/2 = 0.023\ cm$

$l_3 = 0.46\ cm/2 = 0.23\ cm$

<u>Total Resistance</u>, R_{tot}:

$$R_{tot} = \frac{R_t}{A} + \frac{l_1}{kA} + \frac{l_2}{kA} + \frac{l_3}{kA} + \frac{R_t}{A}$$

$$= \frac{2.75 \times 10^{-4}\ m^2{}^\circ C/W}{1.12 \times 10^{-4}\ m^2} + \frac{2.4 \times 10^{-3}\ m}{(180\ W/m^\circ C)(3.86 \times 10^{-5}\ m^2)}$$

$$+ \frac{2.3 \times 10^{-4}\ m}{(30 \times 10^{-3} W/m^\circ C)(2.43 \times 10^{-4}\ m^2)} + \frac{2.3 \times 10^{-3}\ m}{(180\ W/m^\circ C)(3.4 \times 10^{-5}\ m^2)}$$

$$+ \frac{2.75 \times 10^{-4}\ m^2{}^\circ C/W}{1.6 \times 10^{-4}\ m^2} = 36^\circ C/W$$

<u>Heat Transfer Through Shell</u>, Q_{31}:

$$Q_{31} = \frac{\Delta T}{R_{tot}} = \frac{93^\circ C - 49^\circ C}{36^\circ C/W} = 1.2\ W$$

Total Heat Transfer Through Shells:

$$Q_{total\ shell} = Q_{25} + Q_{31} + Q_{31}$$
$$= 1.05W + 1.2W + 1.2W = 3.45\ W$$

Mounting Bolts:

$$Q_{bolts} = \frac{kA\ \Delta T}{l} = \frac{(13.4\ W/m^{\circ}C)\ (1.78 \times 10^{-5}\ m^2)\ (93^{\circ}C - 49^{\circ}C)}{0.0127m}$$

$$= 0.8 \text{ watts}$$

where,

$$A = \frac{\pi\ D^2}{4} = \frac{\pi\ (0.005m^2)}{4}$$

l = cold plate thk + Insulator
= 0.25" + 0.25" = 0.0127m

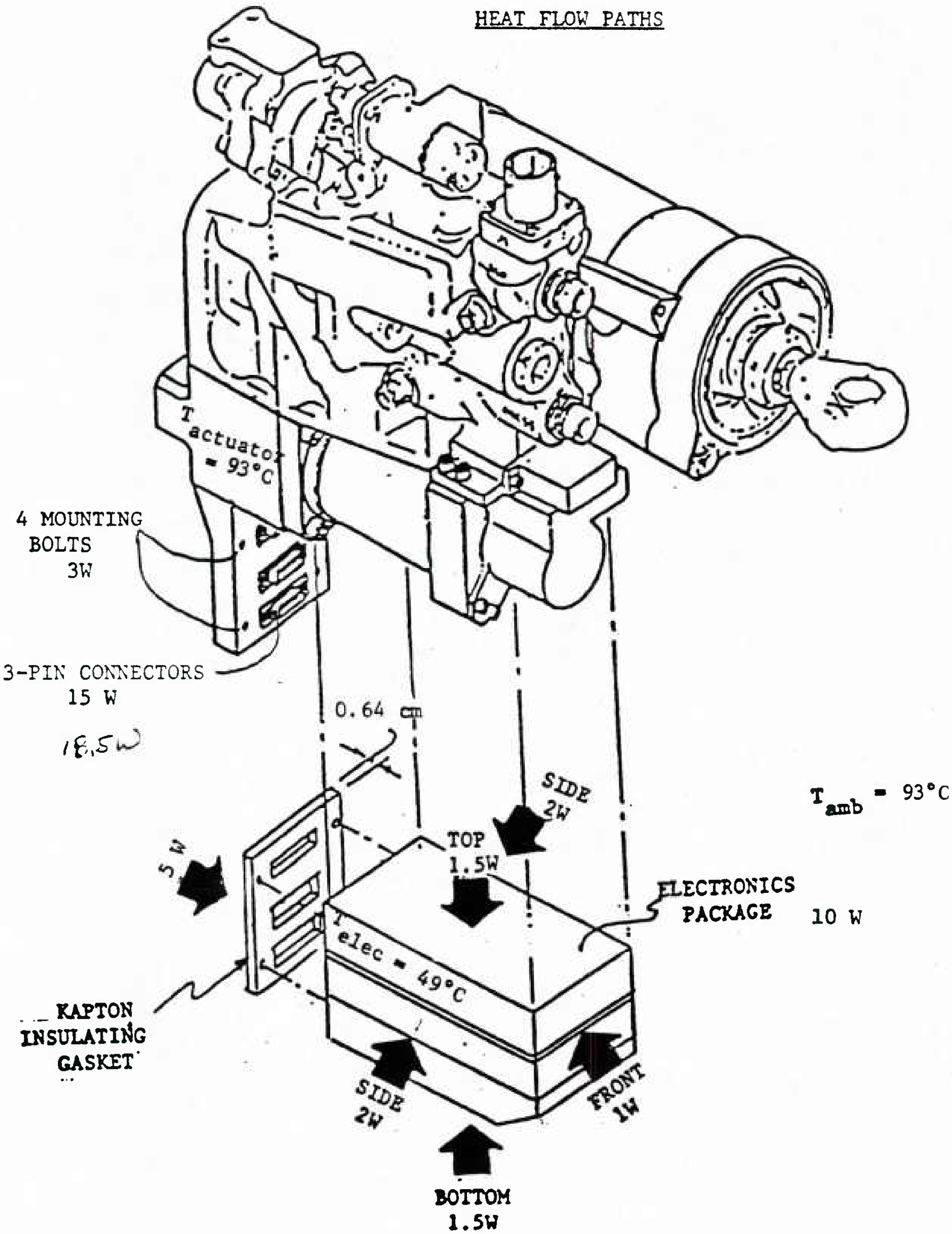
$k_{s.s.}$ = 13.4 W/m°C

Q_{tot} = Qx # of bolts = 0.8 W x 4 bolts = 3.2 watts

The heat load imposed on the electronics by the hydraulics are summarized below:

10 W	- Generated by electronics
3 W	- Into top and bottom
5 W	- Actuator to Electronics interface
4 W	- Into sides
1 W	- Into front
15 W	- 3 pin connectors
3.5 W	- Shells of electrical connectors
3 W	- 4 Mounting Bolts
44.5 W	- Total heat load to be dissipated by cold plate (heat flow paths are illustrated on the next page)

HEAT FLOW PATHS



APPENDIX B

COLD PLATE STRESS ANALYSIS

Purpose: To determine the necessary cold plate thickness to contain the working fluid. The material/working fluid combinations evaluated are listed in Table B1.

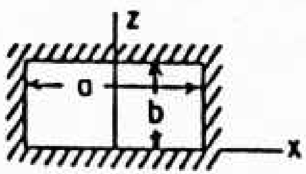
Description: The stress equation used in this analysis is shown in Figure B1. All edges are fixed, and the pressure is uniform over the entire plate.

Results: The results of this analysis are summarized in Table B2. It is evident that copper and titanium offer the thinnest cold plate.

TABLE B1. Compatible Heat Pipe Systems

Working Fluid	Materials
Ammonia	Aluminum
Acetone	Aluminum, Copper
Methanol	Copper
Freon	Copper
Water	Copper, Titanium

Rectangular plate, all edges fixed



8a. Uniform over entire plate

(At center of long edge) $\quad \text{Max } \sigma = \dfrac{-\beta_1 q b^2}{t^2}$

(At center) $\quad \sigma = \dfrac{\beta_2 q b^2}{t^2} \quad$ and $\quad \max y = \dfrac{\alpha q b^4}{E t^3}$

a/b	1.0	1.2	1.4	1.6	1.8	2.0	∞
β_1	0.3078	0.3834	0.4356	0.4680	0.4872	0.4974	0.5000
β_2	0.1386	0.1794	0.2094	0.2286	0.2406	0.2472	0.2500
α	0.0138	0.0188	0.0226	0.0251	0.0267	0.0277	0.0284

"Formulas for Stress and Strain" 5th ed. Raymond J. Roark and Warren C. Young, pp. 392, Copyright McGraw Hill Inc, 1982

Figure B1. Rectangular Plate Stress Equation

TABLE B2. **Plate Thickness as a Function of Material/Working Fluid Combinations**

MATERIAL	WORKING FLUID	VAPOR PRESSURE @ 110°C	YIELD STRESS, PSI	COLD PLATE THICKNESS
3003 aluminum	ammonia	1082 psi	6,000	6 cm
3003 aluminum	acetone	60 psi	6,000	1.7 cm
101 Copper	acetone	60 psi	10,000	1.3 cm
101 Copper	methanol	50 psi	10,000	1.3 cm
101 Copper	Freon 13	80 psi	10,000	1.5 cm
101 Copper	water	5 psi	10,000	0.6 cm
Titanium	water	5 psi	25,000	0.3 cm

Analysis Section

Copper/Water System:

$$\text{MAX} \quad \sigma_y = \frac{-\beta_1 P b^2}{t^2}$$

WHERE,

A = 4.75"

b = 3"

P = 5 PSI @ 110°C

σ_y = CDA101 Cu = 10,000 PSI

SOLVE FOR t,

$$t = \sqrt{-\frac{.4680\ (5\ \text{PSI})(3")^2}{.6\ (10{,}000\ \text{PSI})}}$$

$$= \quad 0.060"$$

Cold Plate Thickness, T

T = Top Plate + Bottom Plate + Wick Structure (common to each)

= 0.060" + 0.060" + 0.125"

= 0.245" or 0.6 cm

- This same calculation was conducted on each material/fluid combination listed. The results are summarized in Table B2.

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