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DRY-ICE, LIQUID-PULSE-PUMP, PORTABLE COOLING SYSTEM

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DEPARTMENT OF THE NAVY
SUPPLY SYSTEMS COMMAND

NAVY CLOTHING AND TEXTILE RESEARCH FACILITY
NATICK, MASSACHUSETTS

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The Navy Clothing and Textile Research Facility (NCTRF) has developed a bench-model, dry-ice, liquid-pulse-pump, portable cooling system which has a potential total cooling capacity of approximately 880 kcal (6.77 kg. dry ice) and weighs 9.56 kg. uncharged with dry ice. The system is designed for use with commercial liquid cooling garments to provide conduction cooling to personnel subjected to heat stress. (U)

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A unique feature of the system is the utilization of the carbon dioxide gas, given off in the sublimation of the dry ice, when cooling the heat exchange liquid, to drive the pump transporting the heat exchange liquid between the dry ice cooler and liquid cooling garment. (U)

The system, which is contained in a pack case, consists basically of an integral dry ice canister and heat exchanger assembly, a gas-operated diaphragm pump, and suitable plumbing, control, safety, and quick-connection hardware for proper operation of the system. (U)

Performance tests of the system have shown that both heat transfer rates greater than 240 kcal/hr. and adequate pump operation pressures and flow rates are available until more than 90% of the initial dry ice charge has been expended. (U)

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DRY-ICE, LIQUID-PULSE-PUMP, PORTABLE COOLING SYSTEM

INTRODUCTION

The Navy Clothing and Textile Research Facility (NCTRF) has developed a bench-model, dry-ice, liquid-pulse-pump, portable cooling system for use with commercially available liquid cooling garments to provide conduction cooling to personnel subjected to heat stress.

The system has potential application in many service situations where heat stress to personnel is known to occur. For instance, crash-crew firefighters can be debilitated by the heat when dressed in their turnout clothing during runway standby operations in warm climates. In the event an emergency fire condition occurred during this time, their ability to perform effectively would be questionable. Explosive Ordnance Disposal (EOD) personnel are required to perform multifunctional duties under all climatic conditions and for many applications are dressed in impermeable protective clothing. The combination of impermeable clothing, work effort, and warm or hot climatic conditions would obviously limit effective work time if no active body cooling system is used. Engine- and boiler-room ship personnel required to function normally and perform special repair missions in hot-humid locations in machinery spaces are prime heat stress candidates.

The bench-model dry-ice system, as currently sized, can provide a cooling capacity of over 800 kcal if the density of the dry ice is over 1.45 g/cc, but capacities achieved to date range between 534 and 588 kcal (density of dry ice 0.90 to 0.98 g/cc.). These lower capacity values would be sufficient to neutralize the body heat produced by individuals engaged in light to moderate work for up to 90 minutes.[1] Available dry ice volume is 4590 cc.

A unique feature of the system is the employment of the carbon dioxide gas given off by the dry ice as it sublimes, when cooling the heat transfer liquid and from ambient heating, to power the pump which circulates the liquid between the dry ice heat exchanger and liquid cooling garment. No electrical power source is required with the system.

This report describes the concept, features, and operation of the system and provides performance data on the system heat transfer and pump characteristics.

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1. Newbury, L. H., Physiology of Heat Regulation, W. B. Saunders, Philadelphia, 1949, p. 446.

SYSTEM

Concept

Many life support systems developed in the past utilized refrigerants such as wet ice to provide personnel cooling and as a result required an electrical energy source to drive a pump or fan which circulated a cooling fluid between the man being cooled and the heat exchanger of the system. The concept for this cooling system was to use a refrigerant which could also provide the power necessary to drive the heat transfer fluid circulator. The refrigerant to be employed had to have better specific cooling capacity than wet ice. It also required a heat sink temperature significantly lower than that produced by melting wet ice so that high heat transfer rates could be more easily accommodated. Finally, the refrigerant had to be relatively safe to use and handle.

Dry ice has many of these desirable features. It provides a specific cooling capacity when it sublimates at atmospheric pressure of 136 kcal/kg and 130 kcal/kg at 3 atmospheres, which is significantly better than can be derived from melting wet ice (80 kcal/kg.). At 1 atmosphere dry ice sublimes at -79°C and at 3 atmospheres -59°C, providing a much lower heat sink temperature than wet ice (0°C). Dry ice, like wet ice, is nontoxic and noncombustible.

A useful amount of carbon dioxide gas is given off when dry ice sublimes to operate a liquid circulating device. At 1 atmosphere 1 kilogram of dry ice produces 519 liters of gas and at 3 atmospheres 173 liters of gas. Figure 1 shows the amount of CO_2 gas available from the sublimation of the dry ice at pressures of 2 and 3 atmospheres for different metabolic rates. Assuming the volumetric efficiency of the circulator (i.e., the ratio of the liquid volume delivered to the gas volume used at pumping pressure) is only 50%, a flow rate of 86 LPH at 3 atmospheres is possible at metabolic rates as low as 130 kcal/hr. (sedentary activity).

Because of the low temperatures at which the dry ice sublimes, a 70% by weight aqueous solution of methyl alcohol was selected for the heat transfer liquid. At this concentration the solution from data extracted from reference 2 has the following estimated properties:

Freezing Point (°C)	-104.0
Density at 20°C (g/cc)	0.873
Specific Heat at 20°C (kcal/kg °C)	0.756

2. Geiringer, P. L., Handbook of Heat Transfer Media, Reinhold Publishing, New York, 1962, pp. 115-118.

Using dry ice creates some design problems with regard to obtaining good heat transfer between the dry ice and the heat transfer liquid and controlling the temperature of the heat transfer liquid supply to the liquid cooling garment at a level which will provide a comfortable situation for the individual wearing the garment.

Because dry ice is a solid and the CO_2 gas given off when it sublimes has a low thermal conductivity, good heat transfer between the dry ice and the heat-transfer fluid through a heat exchange device will occur only where the solid is in good thermal contact with the heat exchanger walls. Initially a container filled with dry ice will show excellent heat transfer characteristics since the dry ice is in intimate contact with the inside container walls, but as it sublimes contact will diminish and the dry ice will become engulfed by the low conductivity CO_2 gas. Eventually, a useful amount of heat transfer can take place only at the bottom of the container, where there is always some contact between the dry ice and the container. Suitable heat transfer performance will be attained only until most of the dry ice charge is sublimed if the heat exchange device is closely coupled thermally to the bottom and the sides of the dry-ice container. The low temperature at which the dry ice sublimes allows for practical heat-transfer rates even when the contact surface area between the dry ice and the heat-exchange device is limited to the bottom of the container because of the large temperature differential between the dry ice and the heat transfer liquid (typically 80°C).

To ascertain the heat-transfer performance to be expected some initial tests were conducted with a cylindrical aluminum container for storing the dry ice with a 10.8 cm. diameter X 46 cm. length X 0.318 cm. wall thickness. These tests were conducted in four different ways with water as the heat transfer liquid.

The test modes were:

1) Cylinder was closely wrapped circumferentially with thin neoprene tubing and heat transfer liquid was passed through the tubing.

2) A scroll made of copper tubing was located at the inside bottom surface of the cylinder and the heat transfer liquid was passed through the scroll.

3) Same as "2" except a 100 cc aqueous solution of methyl alcohol was placed in the container.

4) Condition "1" and "3" combined with the two heat exchangers connected in series.

For these tests the dry ice was crushed manually from commercial cakes and packed into the container and allowed to sublimate at normal atmospheric pressure. The bottom and top surfaces of the cylinder were insulated with 1.9 cm of polyurethane foam. Water flow rates were at least 50 LPH and were increased as necessary to prevent the water from freezing. Inlet water temperatures were between 20 and 30°C. The inlet and outlet water temperatures for the scroll and neoprene tubing exchangers were measured with copper constantan thermocouples, and water flow rates were measured with a graduated cylinder and stop watch. The rate at which the dry ice sublimed was also established by cylinder weight loss-time measurements over the length of each test. Figures 2 through 4 show the test results.

With only the circumferential heat exchanger (Figure 2), activity equivalent to light work could be supported only until 40% of the dry ice charge was expended. When only the bottom heat exchanger was used (Figure 3), moderate to light work activity could be supported until more than 82% of the available dry ice charge was expended. Figure 3 also shows that, by adding a small amount of an aqueous solution of methyl alcohol to the dry ice container, thermal coupling to the bottom heat exchanger was improved significantly. Under this condition heavy work-type activity could be supported until over 90% of the available dry ice charge was utilized. When both heat exchangers were coupled in series and the aqueous methyl alcohol solution was added to the canister (Figure 4), heat exchange rates exceeded those necessary to cool a man performing heavy work until 95% of the available dry ice charge was expended. Thus it was demonstrated that good heat exchange was possible until most of the dry ice charge was sublimed if a bottom heat exchanger with good thermal coupling to the dry ice was employed.

For the system to be useful, the inlet liquid temperature to the liquid cooling garment must be controlled at a level acceptable for human comfort, 0 to 25°C, depending upon activity level.[3] Consequently, the liquid circuit would have to be designed to permit only a fractional amount of the liquid returning from the liquid cooling garment to pass through the heat exchanger because of the low dry-ice, heat-sink temperature. The bulk of the liquid supply would simply be circulated between the pump and the liquid cooling garment. The amount passed through the heat exchanger would then be mixed with this liquid near the pump inlet. In addition the amount of liquid passed through the heat exchanger would have to be varied through a flow control valve to accommodate different activity levels.

When these factors were taken into account, it was expected that a workable dry-ice, pulse-pump, portable, liquid-cooling system could be demonstrated.

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3. Buchberg, H., and Harrah, C.B., "Conduction Cooling of the Human Body - A Biothermal Analysis," Thermal Problems in Biotechnology, The American Society of Mechanical Engineers, New York, 1968, pp. 82-95.

Description

The bench model system consists of an integral dry ice storage and heat exchanger canister, a CO_2 gas-operated liquid circulating pump, hardware necessary to direct and control the flow of heat exchange liquid between the heat exchanger and the liquid cooling garment, quick couplers to attach or separate either or both of the liquid cooling garment and dry ice canister from the pack, and several safety devices to limit the system operating pressure. The system is pictured in Figure 5.

For purposes of establishing configuration and packaging of components to facilitate evaluation, the case containing the system components was fabricated from aluminum angle and sheet stock. A cylindrical aluminum housing fabricated for holding the dry ice container was insulated with polyethylene vinyl acetate foam (λ = .004 watts/cm °C) to minimize heat gain from the ambient. The insulation was 1 cm. thick circumferentially and 5 cm. thick at the bottom of the housing.

The dry ice container, which can easily be placed in or removed from this housing, is attached to the system liquid and CO_2 gas circuits through quick couplers on the case. Any suitable liquid cooling garment is also attached to the system liquid circuit by other similar quick couplers on the case. A manual flow control valve on the side of the case varies the amount of heat transfer liquid passing through the heat exchanger. The case also accommodates a harness assembly and has a carry handle so that it can be mounted as a backpack or carried in suitcase fashion. Most of these arrangements can be seen in Figure 5.

Figure 6 shows the integrated dry-ice-storage and heat-exchange canister. The canister which is double-walled throughout has a plug and retaining ring two-piece cover. The canister is constructed of welded aluminum. The dry-ice-storage section has a cylindrical cavity with an internal diameter of 11.4 cm. and a length of 45cm. The space between the inner and outer walls is used for the heat exchange section. This section is constructed from aluminum fin stock material which is baffled to insure liquid entering the heat-exchange-section circuits down one side, across the bottom, and up and out the other side. The inlet-and-outlet liquid-circuit quick couplers are visible in Figure 6. The plug cover is equipped with CO_2 gas-outlet quick coupler, a relief valve set to operate at 3 ATM, and a safety valve set to burst at 11 ATM. The canister has a proof pressure rating of 21 ATM. The plug cover uses an "O" ring to seal the dry-ice-canister top opening. A threaded retaining ring is used to hold the plug cover in place during operation. When this ring is partially unscrewed, it allows the seal to disengage permitting any residual gas pressure in the canister to be relieved.

Figure 7 shows the CO_2 gas-operated liquid-pulse pump used with the working model. It is a diaphragm-type pump, which was finally selected after bellows- and bladder-type pulse pumps were evaluated, because it is compact and can operate at relatively high operating pressures (3 ATM). The pump has a single diaphragm and it requires the liquid circuit to be statically pressurized to operate properly. In operation CO_2 gas entering one side of the diaphragm in the pump gas cavity discharges liquid from the pump liquid cavity as the diaphragm is expanded. Check valves on the inlet-and-outlet pump liquid connections establish the flow direction. The gas exhaust valve seal is part of the diaphragm assembly and does not open the exhaust until the diaphragm is fully expanded. The exhaust is closed by the relaxation of the diaphragm as liquid reenters the pump liquid cavity during the return cycle. To limit the amount of CO_2 gas lost before the exhaust valve closes, a normally closed spring opposed-diaphragm-controlled inlet valve is employed. Pressure in the pump gas cavity acting on the inlet valve diaphragm working against the spring keeps the valve open during the pump cycle. When the pressure in the pump gas cavity is reduced substantially as the exhaust valve opens, the spring force overcomes the gas pressure on the inlet valve diaphragm and the gas valve inlet closes. When the exhaust valve recloses during the liquid return cycle, a small controlled gas leak by the inlet valve reapplies pressure to the inlet valve diaphragm, reopening the inlet valve.

Table 1 gives the physical characteristics of the major system components.

Table 1. Physical Characteristics of Major System Components

Component	Size (cm.)		Weight (kg.) Dry	Weight (kg.) Charged with Liquid
Integral Dry-Ice and Liquid-Heat-Exchanger Canister	Nom. Dia.	- 12.7	3.78	4.02
	Max. Dia.	- 14.3		
	Length	- 48.2		
Diaphragm Pump	Max. Dia.	- 10.2	0.55	0.61
	Max. Depth	- 5.1		
Pack Case	Max. Length	- 61.1	3.11	3.11
	Min. Length	- 53.5		
	Width	- 30.5		
	Depth	- 15.2		
Pack Case Equipped with Components, Hardware, and Harness Assembly			8.95	9.56

Operation

Figure 8 depicts the cooling system schematically. The operation is as follows:

1. The auxiliary connection (15) is used to prime the system with liquid, pressurize the system, and drain the system.

2. The bleed valve (10) is used in conjunction with the auxiliary connection (15) to control the liquid level in the accumulator (11) and to drain system.

3. The accumulator and liquid reservoir (11) reduces system pressure fluctuations and overcomes loss in system static pressure that may occur because of small leaks that happen when the canister (1) and the liquid cooling garment (14) are coupled to the pack.

4. The integral dry ice and liquid heat exchanger canister (1) is disconnected from the pack liquid and gas quick couplers (5) and removed from the insulated housing (2) to charge the canister (1) with dry ice. The gas quick coupler (5) on the canister (1) plug cover is opened to the atmosphere when the cover is being secured. After the cover is secured, the gas quick coupler (5) is closed and pressure in the canister (1) is allowed to build up until the pressure relief valve (3) on the canister (1) plug cover opens. The canister (1) is then reconnected to the pack.

5. The pressure relief valve (3) in the pack is a backup safety for the pressure relief valve (3) on the canister (1) plug cover. Both are set to open at 3 ATM.

6. The safety valve (4) will relieve canister (1) gas pressure in the event the two pressure relief valves (3) fail. The safety valve (4) is set to open at 11 ATM.

7. The pressure regulator (6) is used to maintain a constant gas pressure input to the pump (13). The pressure regulator (6) is normally set to operate at 2.67 ATM.

8. The gas pressure to the pump (13) initiates liquid circulation to the liquid cooling garment (14). The liquid return is directed back to the pack (13) through the orifice (12) and via the heat exchanger through the flow control valve (7) and the orifice (8).

9. The check valve (9) prevents the back flow of liquid from the heat exchanger.

10. The orifices (8) and (12) and the flow control valve (7) can be sized to provide a limited or extended range of possible heat exchange rates.

11. The orifice (8) permits sufficient liquid to pass through the heat exchanger to maintain the pump (13) flow when the flow control valve (7) is fully closed.

EVALUATION OF MODEL SYSTEM

Procedure

Tests were conducted on the model system to determine its heat-transfer and pump-performance characteristics. The test setup is shown in Figure 9. The pack flow-control valve was fixed at a particular opening for the duration of each test, the dry ice canister contained approximately 25 cc. of aqueous methyl alcohol solution, and the dry ice was hand-crushed from commercial cakes.

The liquid flowmeter was a fixed orifice coupled to a 0-to-2.5 cm. mercury draft gage calibrated for the aqueous-methy- alcohol heat-transfer liquid solution used in the test. The mercury flow gage had an equivalent full-scale flow rate of 100 LPH. The gas flowmeter was capable of handling flow rates to 370 LPH. A scale with a range of 15 to 25 kg. capable of measuring weight loss to 1 gram was used to monitor dry ice weight changes.

All pressures were monitored with bourdon-tube-type pressure gages having various ranges from 3 to 7.5 ATM. The temperature measurements were obtained with copper constantan thermocouples connected to a temperature recorder having a full scale range of -45 to +95°C. Flow rates through the heat exchanger were calculated from total flow rate and temperature measurement data. The liquid-to-liquid heat exchanger was used to simulate varying heat loads on the model pack. The flow control valve was regulated to different settings to achieve different heat exchange rates.

From the heat transfer test data, the heat dissipated by the dry-ice heat exchanger, the percentage of dry ice used, and pump gas consumption were calculated as a function of time. The pack operating pressures and temperatures were monitored directly from the instrumentation.

Test Results

Heat Transfer Performance. Table 2 and Figures 10 and 11 show some of the heat transfer results obtained with the system. Table 2 gives the average heat-exchange performance results. These results were obtained for average heat-exchanger flow rates of 11 to 45 LPH. Total average system flow rates ranged from 84 to 100 LPH and average dry-ice charge weights ranged from 4.11 to 4.52 kg. (densities 0.9 to 0.98 g/cc.). Total test times were 60 to 100 minutes depending upon the rate at which the dry ice was sublimed. Pack outlet temperatures averaged between 14 and 21°C for the average heat-transfer rates of 548 and 263 kcal/hr., respectively. Total cooling capacity based on dry-ice charge weights ranged from 534 to 588 kcal.

Table 2. Average Heat-Exchange Performance Results for Bench Model System

AVG Flow Rate (LPH)		Dry Ice Charge Weight (kg)	Total Cooling Capacity (kcal)	Total Test Time (min.)	AVG Temp (°C)			AVG Heat Transfer Rate (kcal/hr)	No. Tests
					Heat Exchanger				
Heat Exchanger	Total				Inlet	Outlet	Pack Outlet		
11	100	4.42	574	100	25	-12	21	263	2
19	99	4.11	534	60	25	- 5	18	424	1
28	84	4.51	586	90	24	- 3	15	461	5
45	96	4.52	588	60	19	- 2	14	548	2

Figure 10 shows the heat-transfer rate versus time characteristic for heat-exchanger flow rates ranging from 11 to 45 LPH and the relationship between the amount of dry ice sublimated versus time for these same heat-exchanger flow conditions. Rates of heat transfer increased directly with heat-exchanger flow rate. At the lowest rate, 11 LPH, activity equivalent to light work was supportable and, at 19 LPH, light to moderate work activity was supportable until over 75% of the initial dry-ice charge was sublimed. At the higher flow rates of 28 and 45 LPH, activity greater than moderate work was supportable until over 60% of the dry ice charge was sublimed. At these higher flow rates, activities greater than light work were supportable until over 90% of the initial dry-ice charge was used.

Figure 11 shows the heat transfer performance of the system at what was finally established as the maximum heat-exchanger flow setting (Average 28 LPH). The curves represent the average results from 5 tests. Under this condition activities greater than moderate work can be supported until over 60% of the dry ice charge is used and light to moderate work activities can be supported until over 90% of the dry ice charge is used. Pack outlet temperatures ranged from 8 to 14°C during that period when activities greater than moderate work could be supported. Eventually, the pack-outlet temperature increased to 25°C when nearly all the dry ice charge was spent (98%). The liquid flow rate was relatively steady (79 to 87 LPH) until 98% of the initial dry-ice charge was consumed.

Pump Performance. Figure 12 shows the pump performance characteristics operating within the system. With an input gas pressure of 2.7-to-2.8 ATM and a system static pressure of 1.51 ATM, flow rates greater than 80 LPH are possible at a differential pressure of 0.7 ATM or more, until 98% of the initial dry ice charge is spent. The volumetric efficiency of the pump was between 44 and 51% at this condition.

Discussion

System Operation. Most of the components used in the model pack worked effectively during performance testing. There was never any instance in which any of the safety devices failed to function properly. The only troublesome problem was the securing of the retaining ring of the plug cover on the canister because of frost buildup on the canister threads after the canister was filled with dry ice. This occurred only if one delayed too long in putting on the plug cover and retaining ring. The employment of a cover arrangement with another securing means that would overcome this problem would improve the operation.

For all tests the dry ice was obtained from commercial cakes. Large chunks were hand-chipped from the cake, placed in a flexible bag, crushed with a mallet and placed in the canister. Typical dry-ice densities achieved in this manner were 0.9 to 0.98 g/cc. The original cakes had densities of approximately 1.44 to 1.52 g/cc. Increasing the compaction of the ice to approach more nearly its original density would substantially increase the cooling capacity of the system (860 to 907 kcal) over that achieved by the means employed (531 to 585 kcal). However, the simple means employed does achieve a useful total capacity.

A commercially available hand-operated aspirator pump equipped with a pressure gage at its output was used with the system for filling and draining the heat transfer liquid and pressurizing the system to the desired level. The use of this approach to perform these functions worked well.

Performance Results. The performance tests on the model pack have demonstrated that the system can dissipate heat loads greater than those expected from light-work-type activity until over 90% of the dry ice charge is spent and support work activities equivalent to moderate work until more than 60% of the dry ice charge is spent. The tests also showed that relatively steady (79 to 87 LPH) heat-transfer liquid flow rates against a substantial head pressure (2.2 ATM) are achievable until most (98%) of the initial dry ice charge is used. The pump continues to provide adequate heat-transfer liquid flow rates beyond the time that useful heat-transfer rates can be obtained.

The heat transfer rates that can be obtained with the system at any flow control valve setting can be changed by resizing the orifice ((12) in Figure 11). If the system is to be used for a range of relatively low work rate conditions, this orifice can be increased in size. For a high range of anticipated work rates, the orifice can be decreased in size.

CONCLUSIONS

1. Tests of a bench model system have demonstrated that a dry-ice, liquid-pulse-pump portable cooling system that derives the energy for powering the heat-transfer liquid circulator from the refrigerant source has practical application. The system can dissipate heat loads greater than those created by personnel performing light to moderate work activities, and can provide useful system pack liquid outlet temperatures and flow rates.

2. The system is small and light enough (9.56 kg without the dry-ice charge and 14.0 kg with a typical dry-ice charge) to provide reasonable portability.

3. The system cooling capacity of 534 to 588 kcal. (dry ice density 0.9 to 0.98 g/cc) will support useful work activities (light to moderate) for 90 minutes. However, use of dry ice at higher densities (commercial cakes have typical densities of 1.44 to 1.52 g/cc) would increase cooling capacity to 860 to 907 kcal without increasing system size. These capacity levels would support light to moderate work activity for 145 minutes.

APPENDIX A. Illustrations

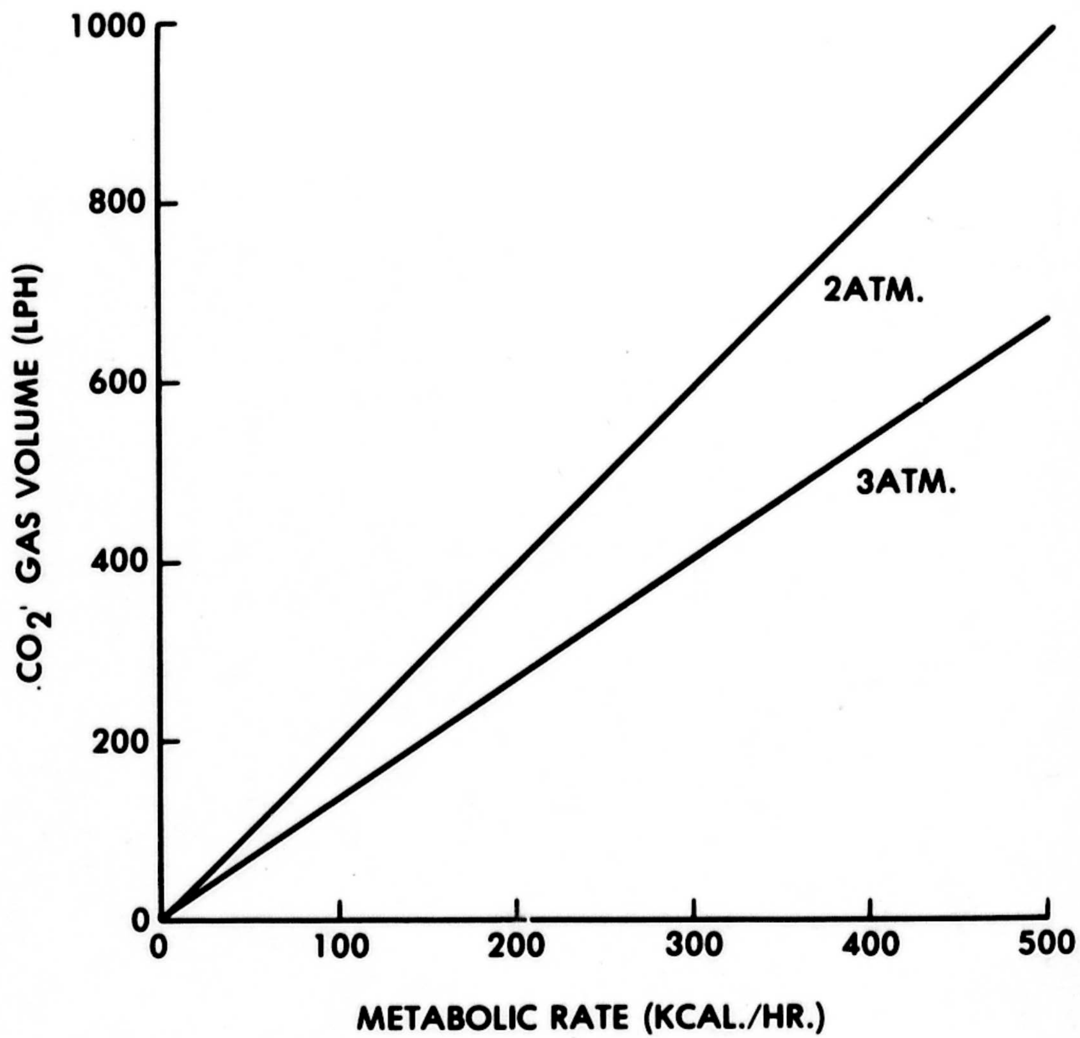


FIG. 1 CO_2 GAS AVAILABLE FOR OPERATION OF HEAT TRANSFER FLUID CIRCULATOR AT DIFFERENT WORKING PRESSURES AND METABOLIC RATES

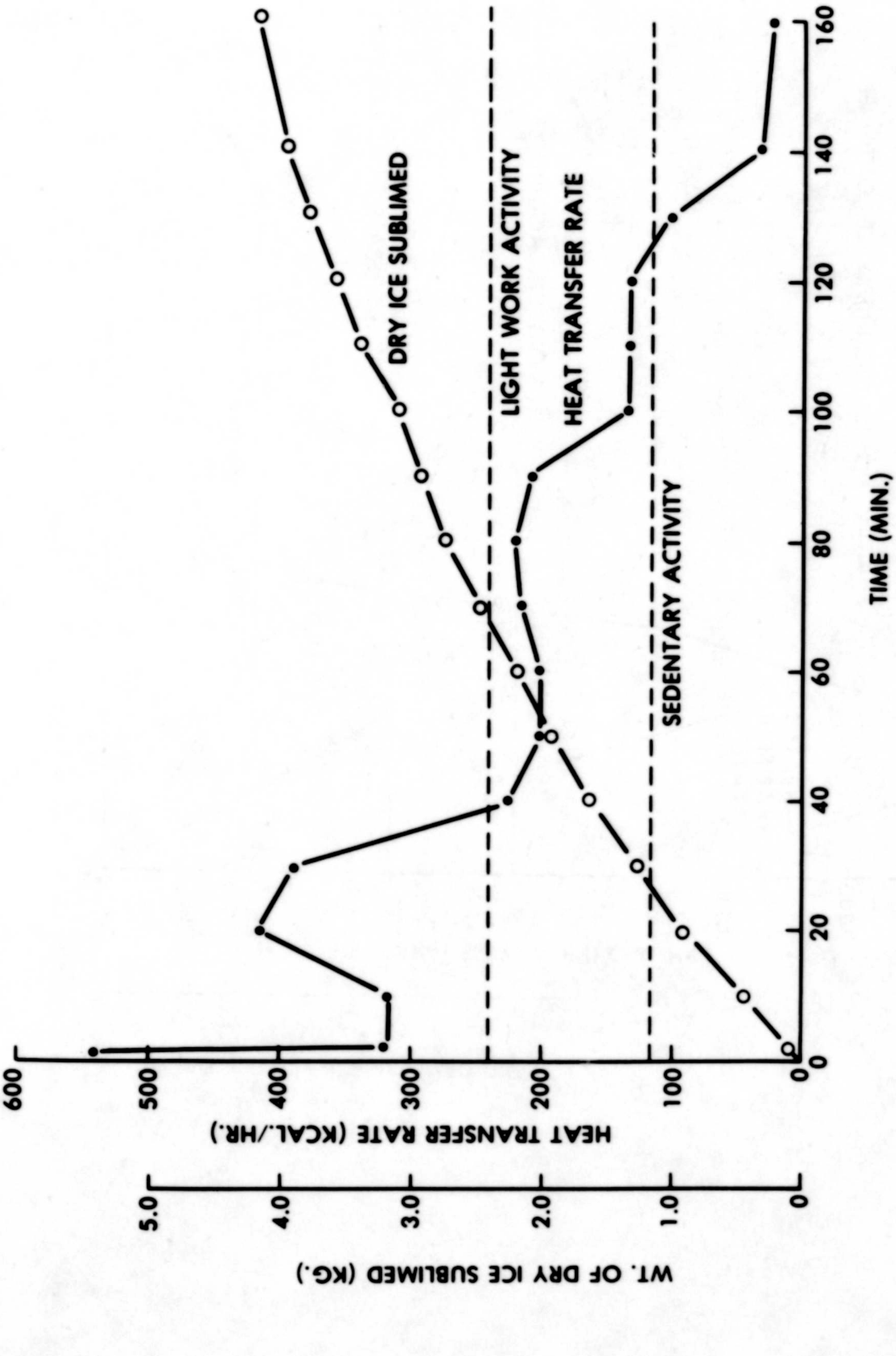


FIG. 2 HEAT TRANSFER RATES WITH CIRCUMFERENTIAL HEAT EXCHANGER

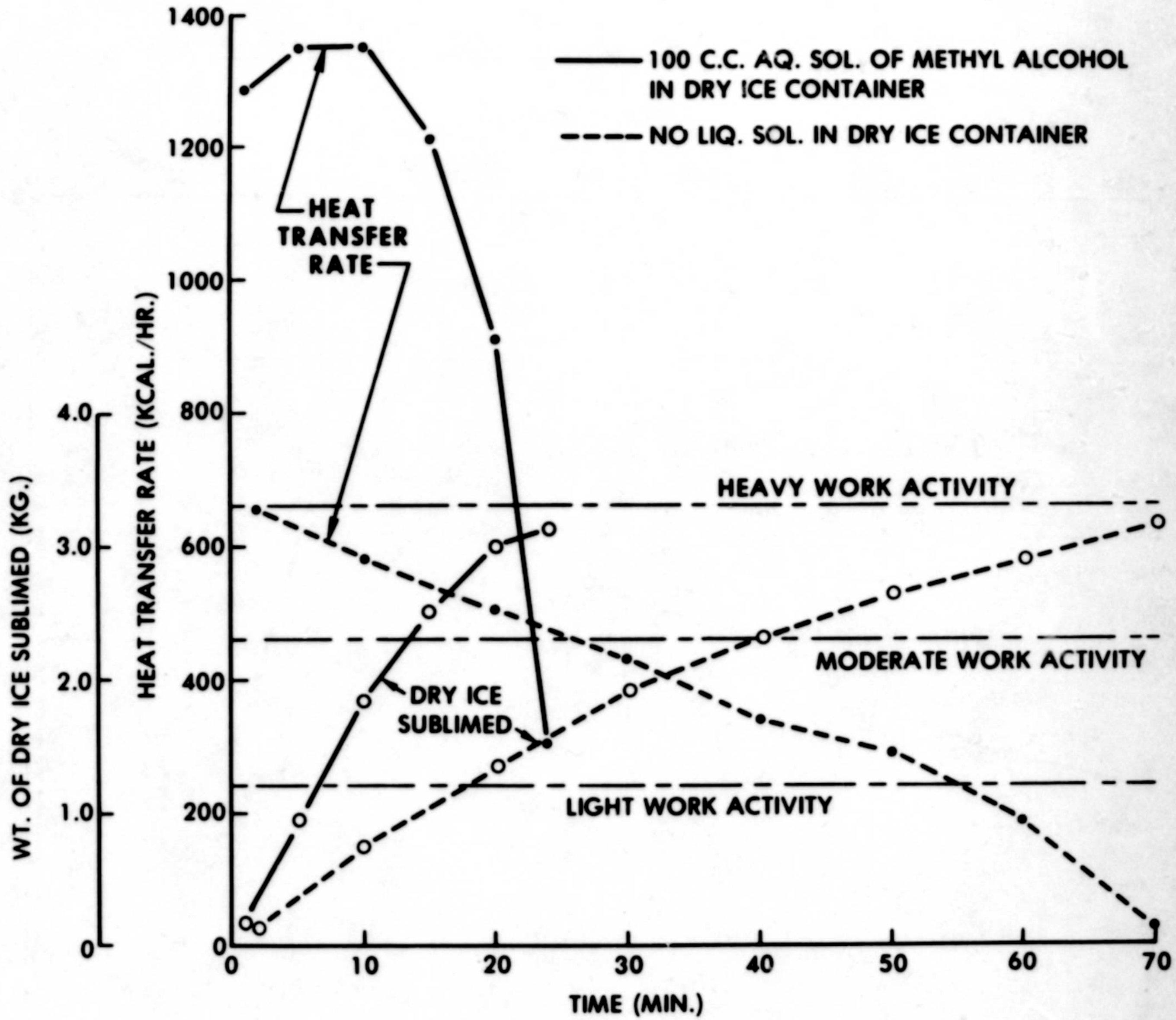


FIG. 3 HEAT TRANSFER RATES WITH BOTTOM HEAT EXCHANGER

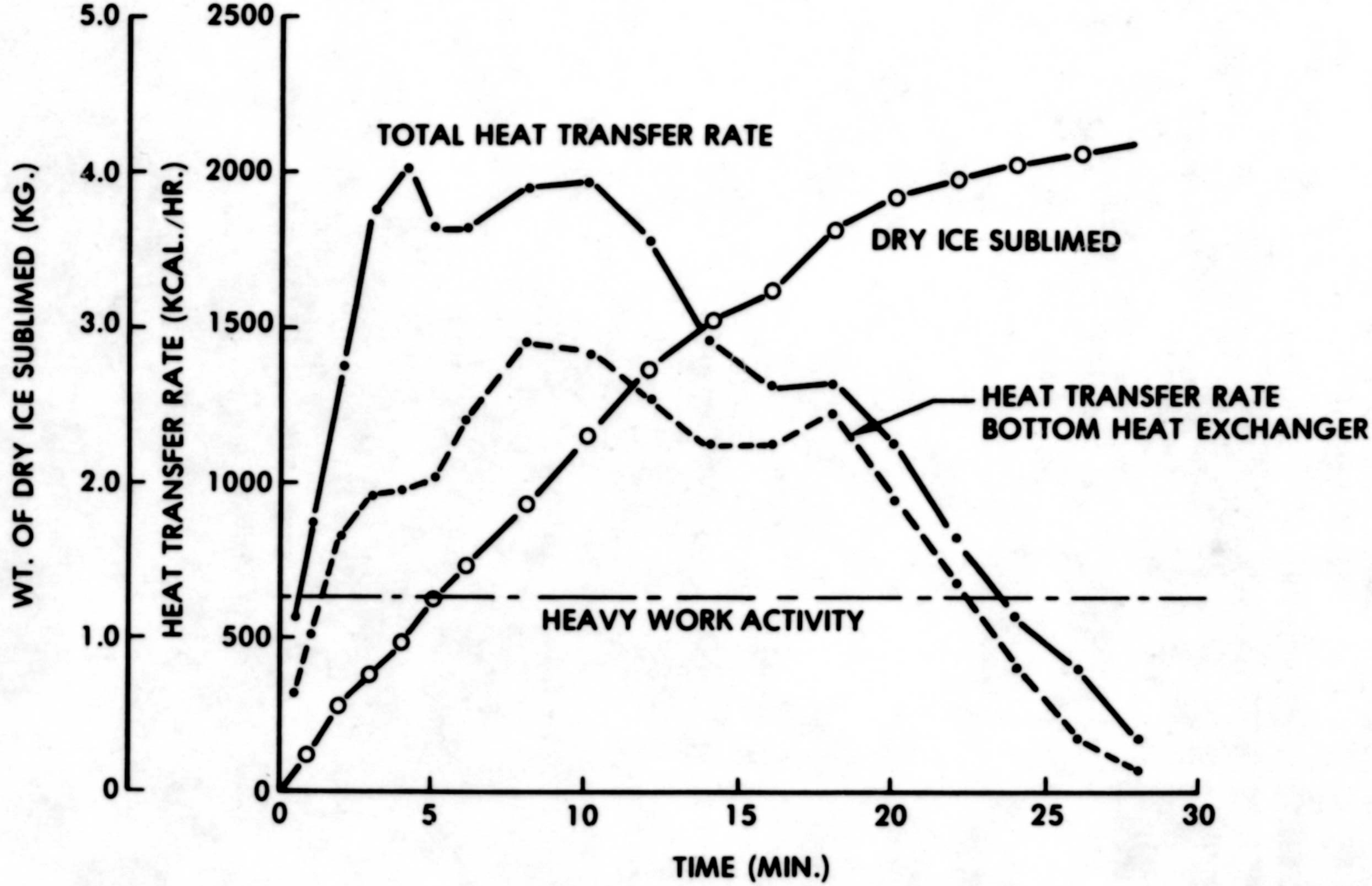


FIG. 4 HEAT TRANSFER RATES WITH SERIES COMBINATION OF CIRCUMFERENTIAL AND BOTTOM HEAT EXCHANGERS WITH 100C.C. AQ. SOL. OF METHYL ALCOHOL IN DRY ICE CONTAINER

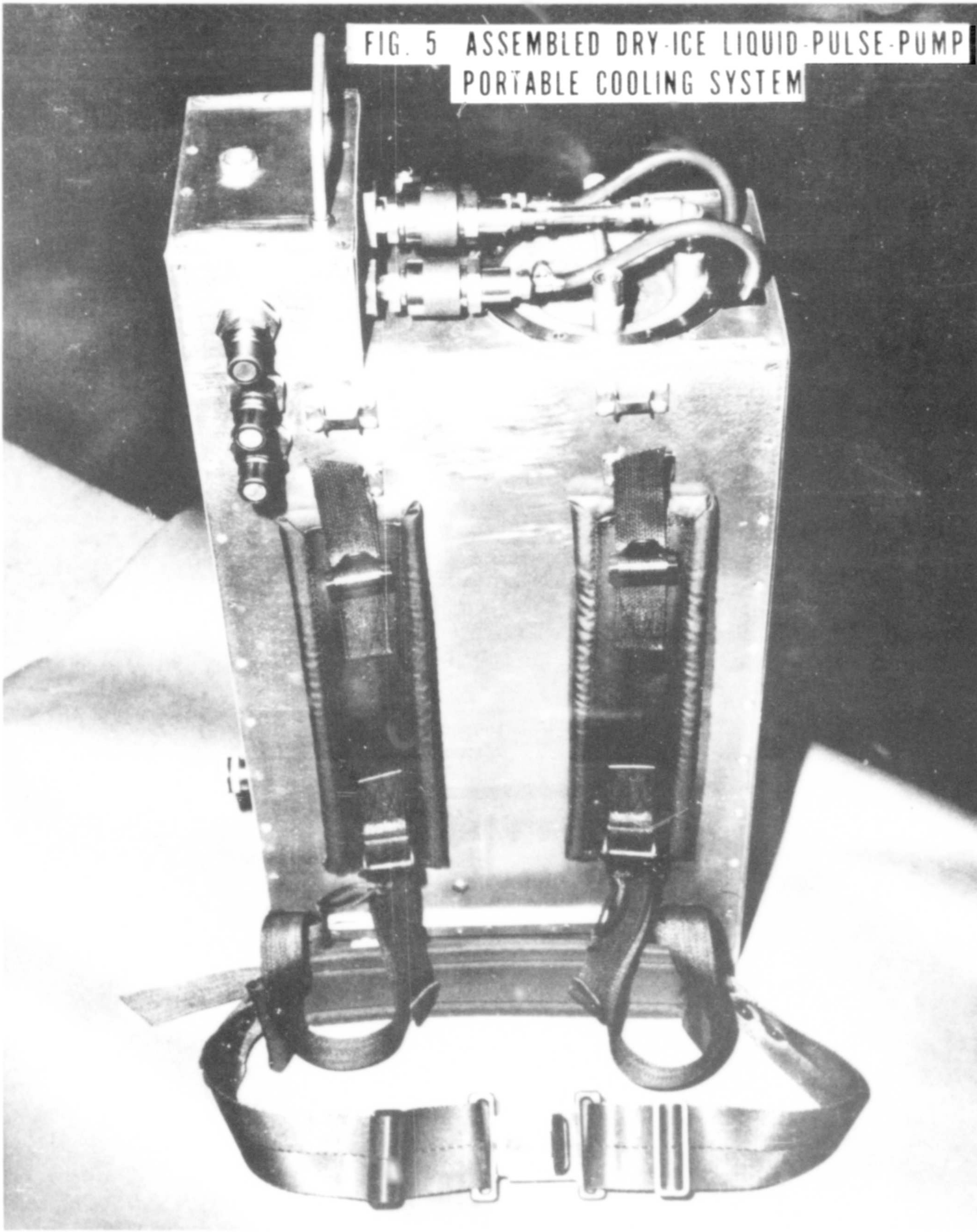


FIG. 5 ASSEMBLED DRY-ICE LIQUID-PULSE-PUMP PORTABLE COOLING SYSTEM

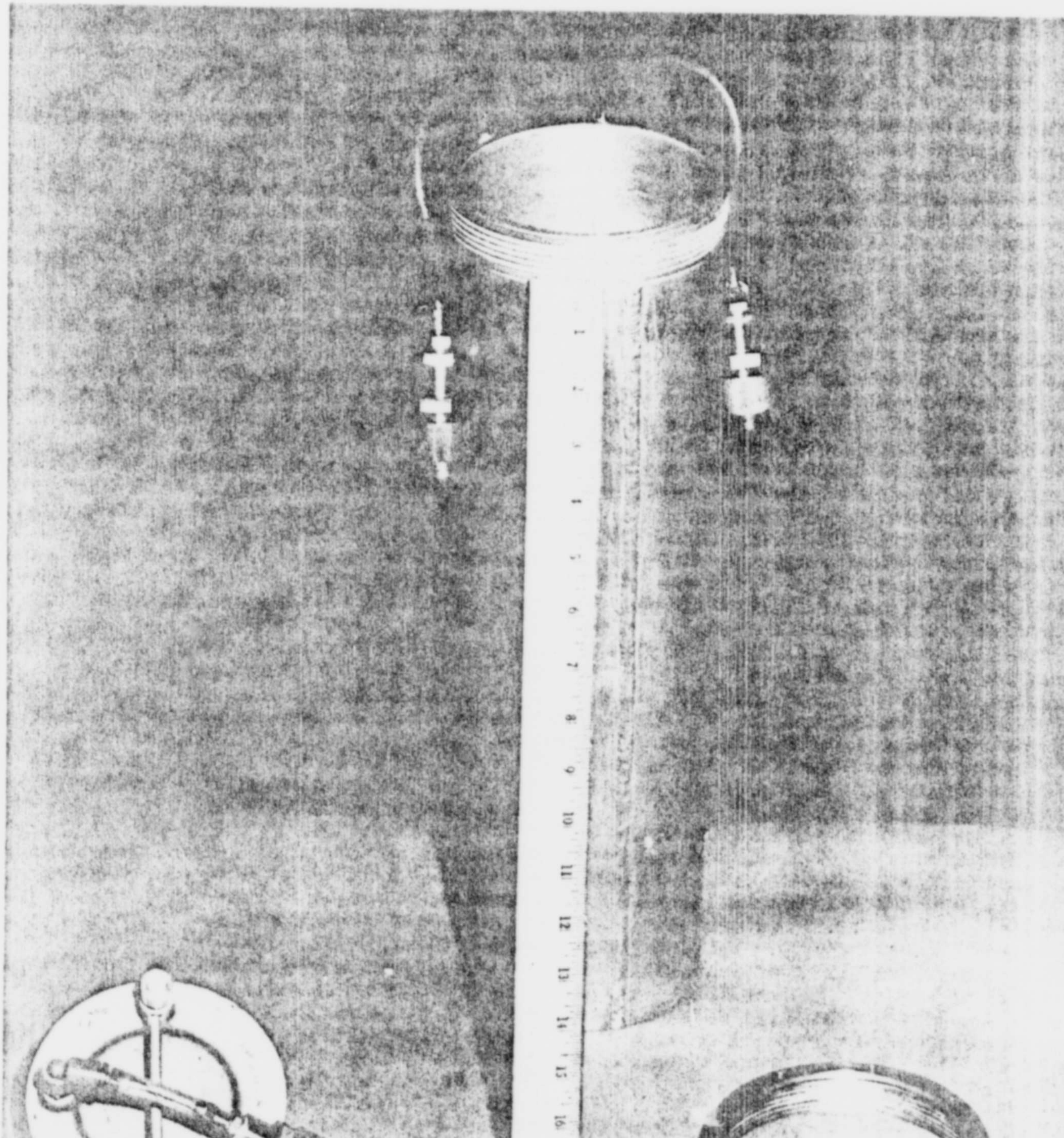


FIG. 6 INTEGRATED DRY-ICE STORAGE AND HEAT EXCHANGER CANISTER

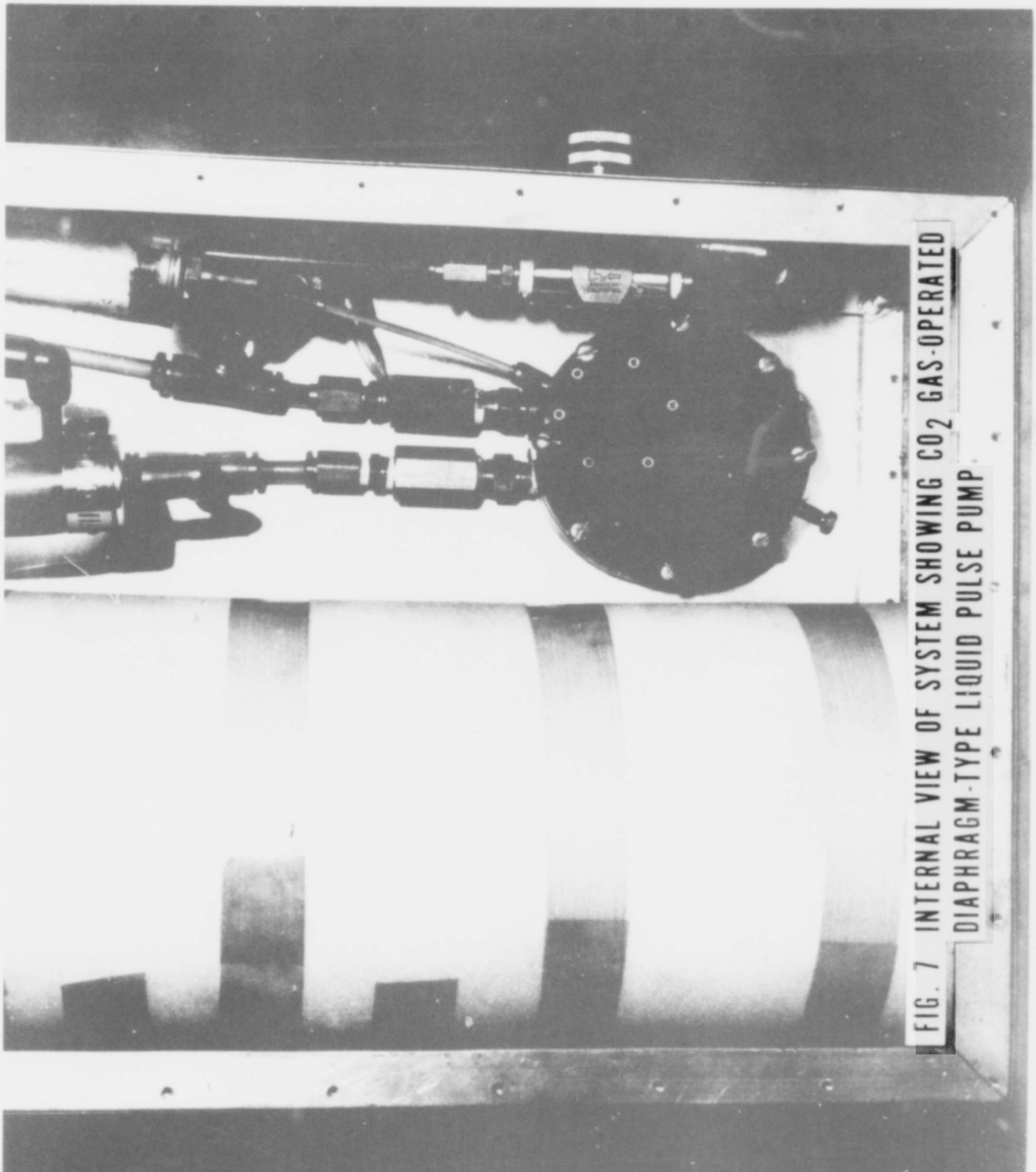
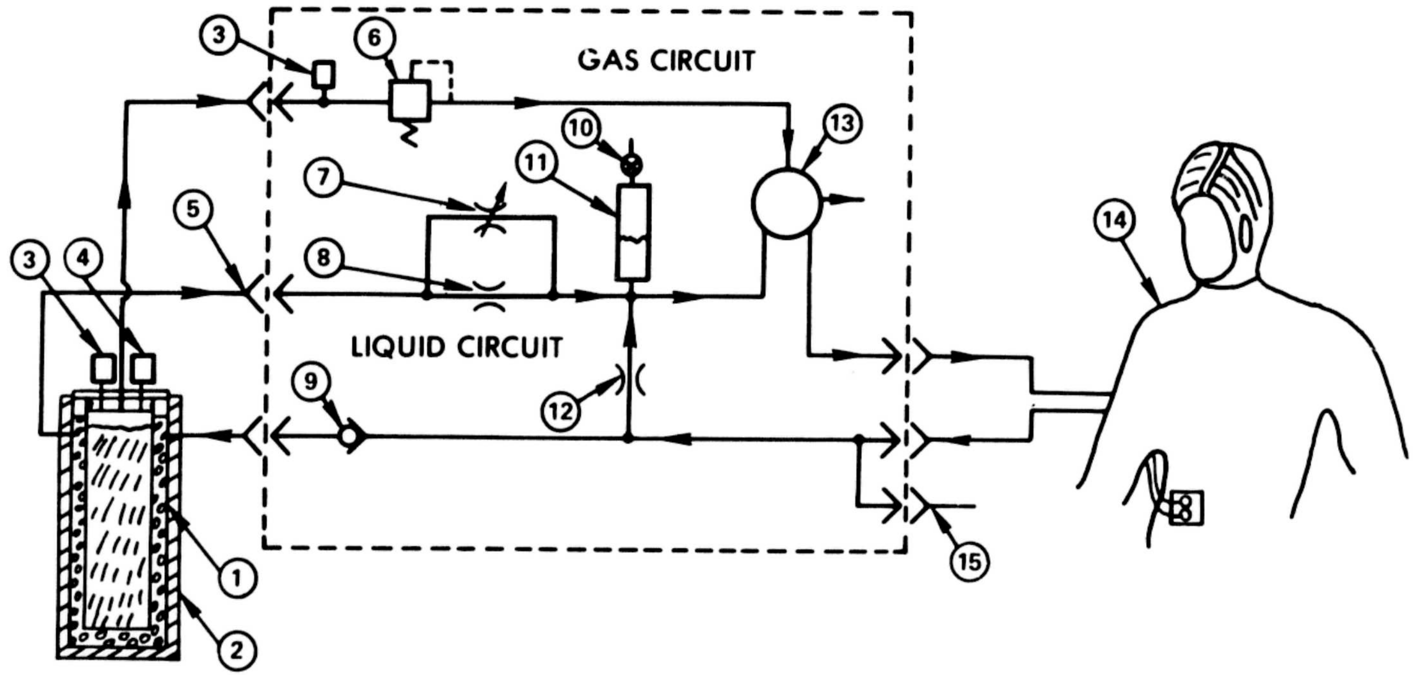


FIG. 7 INTERNAL VIEW OF SYSTEM SHOWING CO_2 GAS-OPERATED DIAPHRAGM-TYPE LIQUID PULSE PUMP



(1) INTEGRAL DRY ICE & LIQ. HEAT EXCH. CAN.
(2) INSULATED HOUSING
(3) PRESSURE RELIEF VALVE
(4) SAFETY VALVE
(5) QUICK COUPLER TYP.
(6) PRESSURE REGULATOR
(7) FLOW CONTROL VALVE
(8) ORIFICE
(9) CHECK VALVE
(10) BLEED VALVE
(11) ACCUMULATOR & LIQ. RESERVOIR
(12) ORIFICE
(13) CO_2 GAS OPER. PUMP
(14) LIQ. COOLING GARMENT
(15) AUXILIARY CONNECTION

FIG. 8 DRY ICE-LIQUID PULSE PUMP COOLING SYSTEM

PRESSURES

GAS

P1 CANISTER

P2 PUMP INLET

LIQUID

P3 PUMP DISCHARGE

P4 PUMP RETURN

TEMPERATURES

T1 PACK OUTLET

T2 WATER BATH

T3 PACK INLET

T4 CANISTER INLET

T5 CANISTER OUTLET

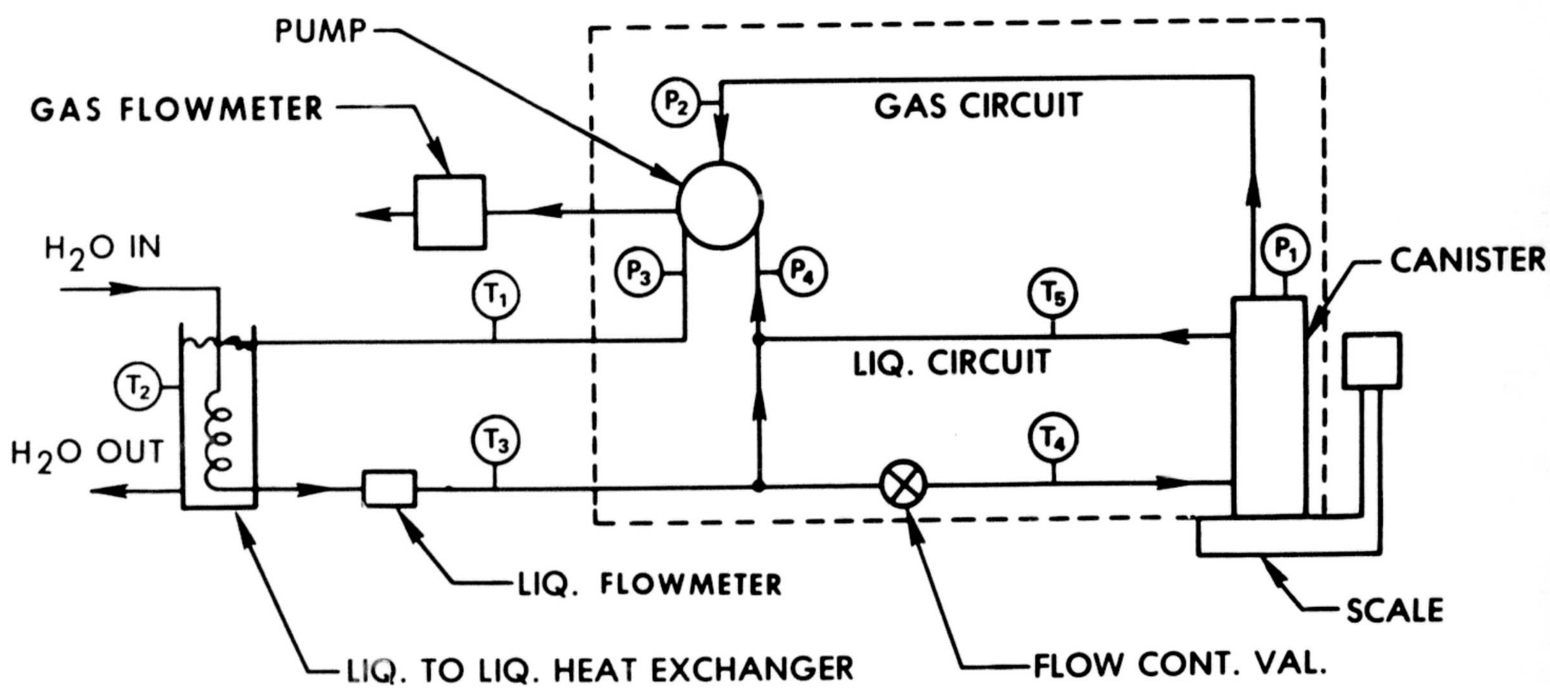


FIG. 9 SETUP FOR HEAT TRANSFER PERFORMANCE TESTS

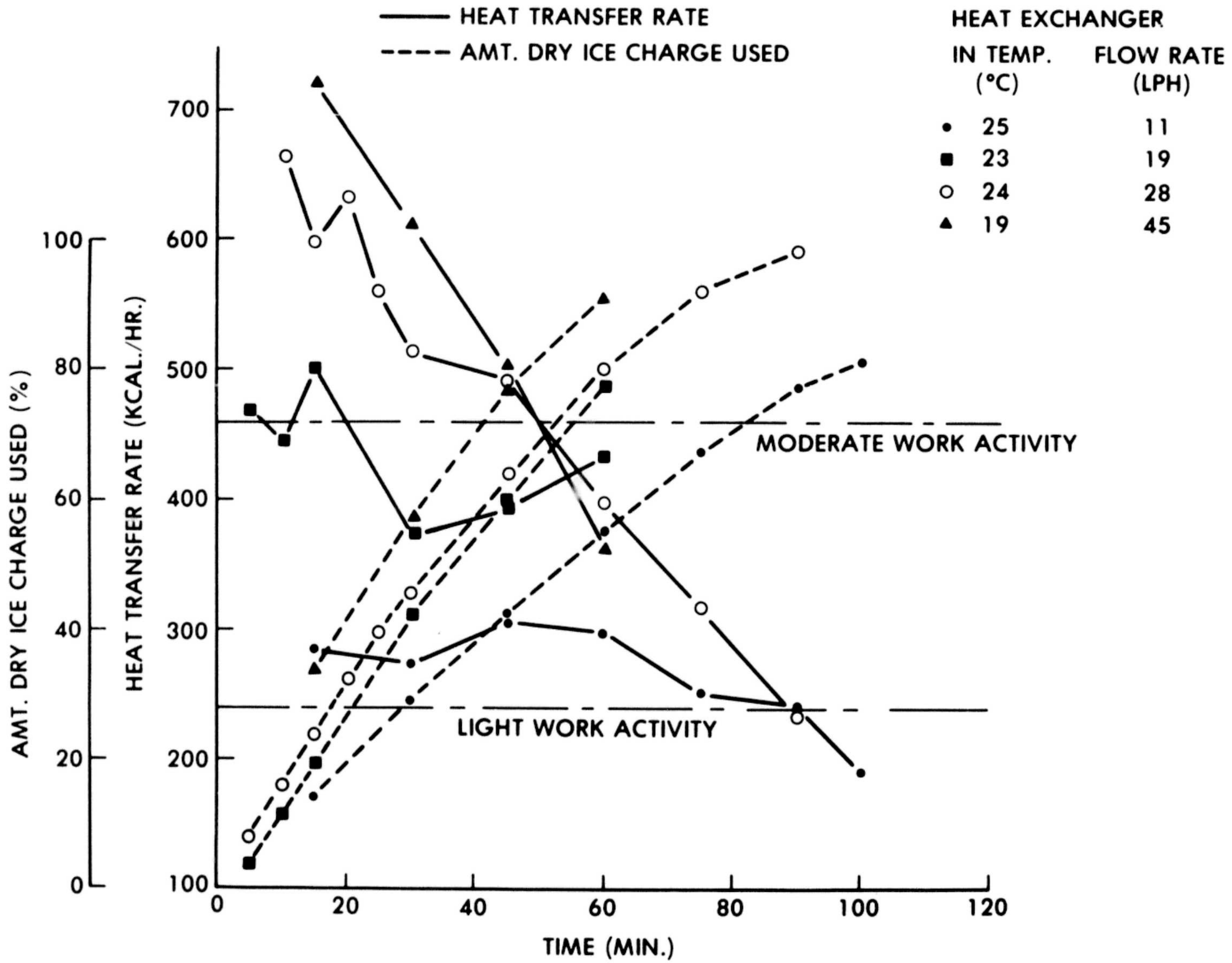


FIG. 10 HEAT EXCHANGER PERFORMANCE AT DIFFERENT FLOW RATES

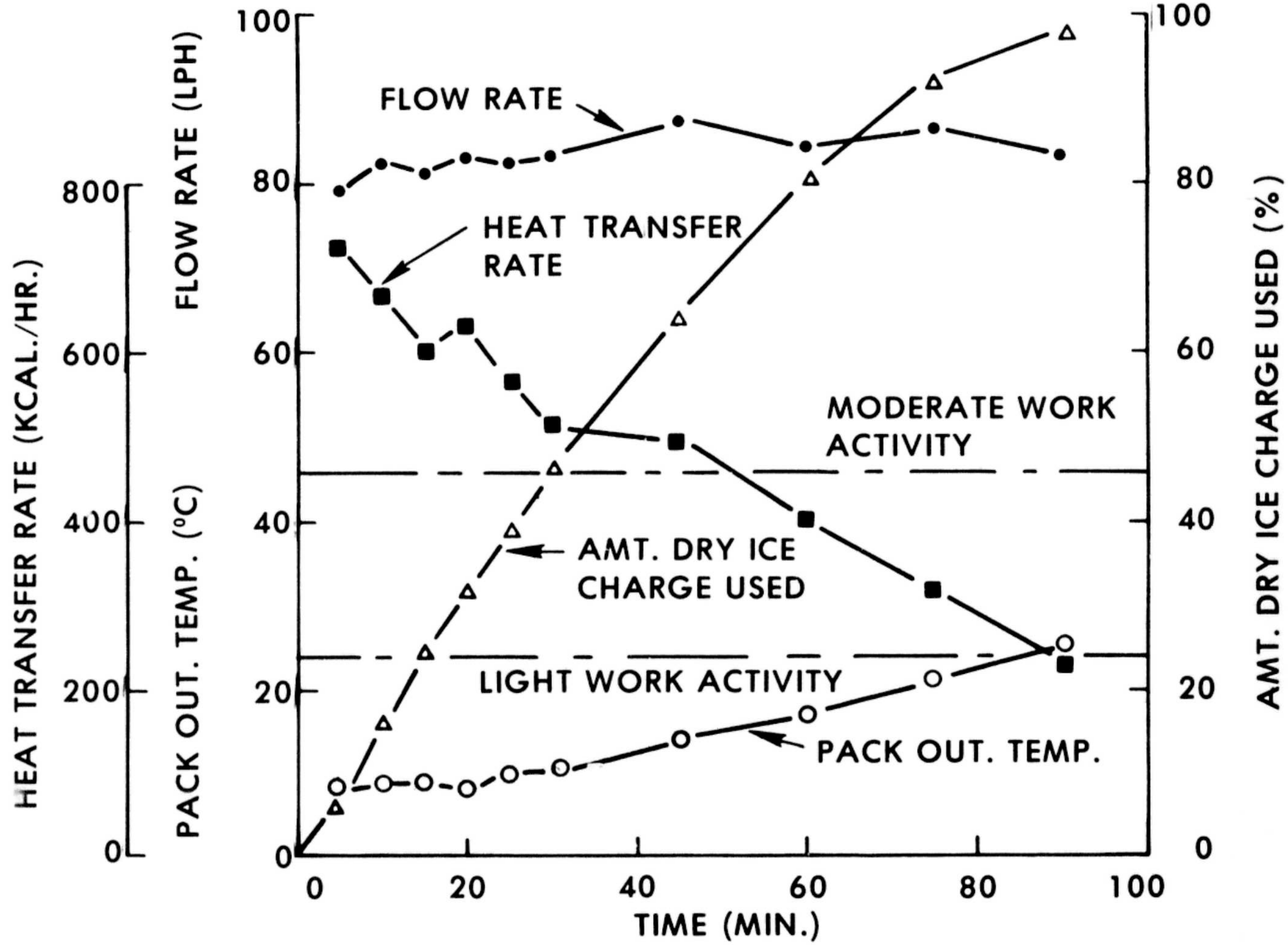


FIG. 11 THERMAL PERFORMANCE OF SYSTEM AT MAXIMUM HEAT TRANSFER RATE

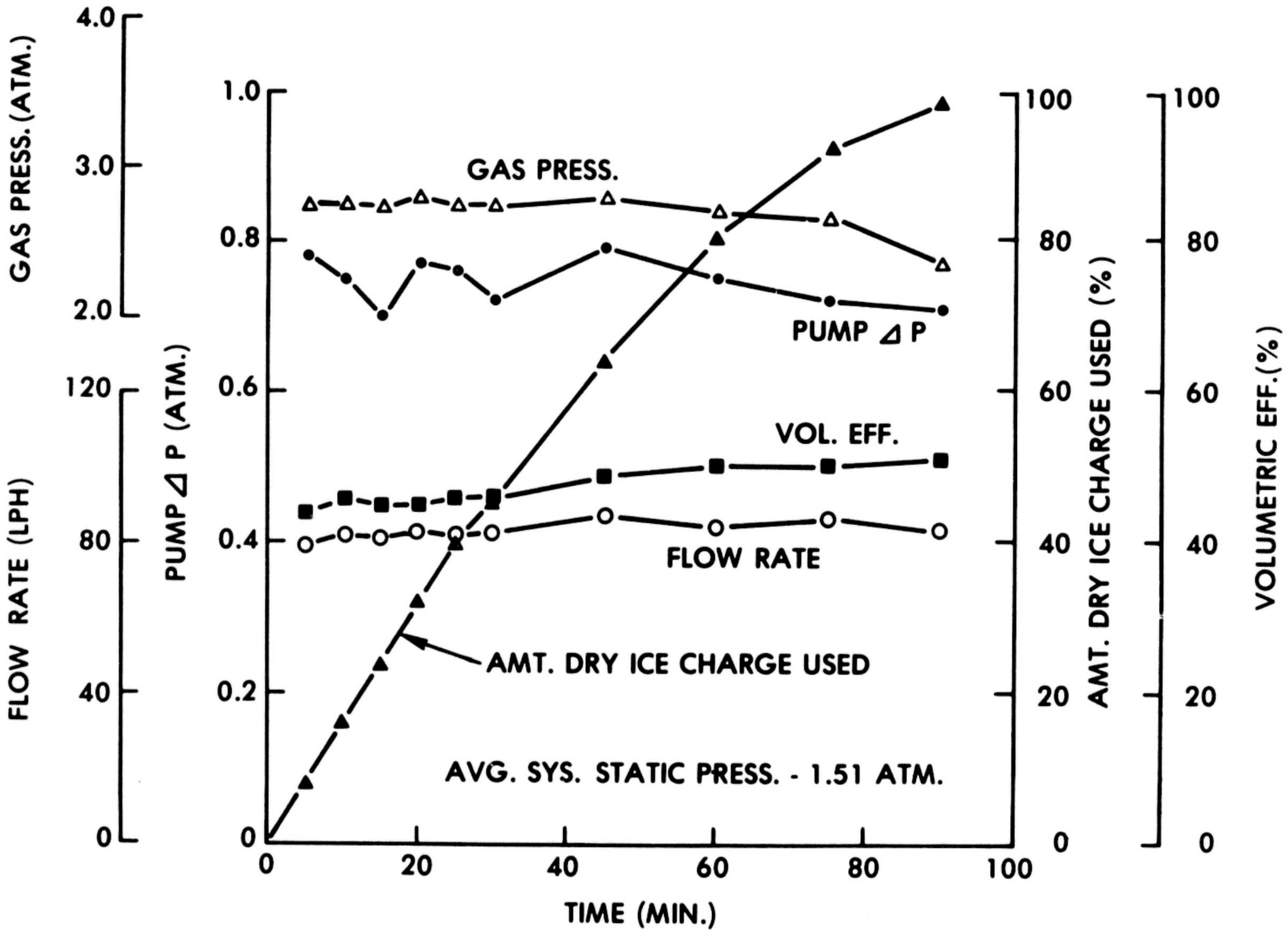


FIG. 12 PUMP PERFORMANCE WITH SYSTEM