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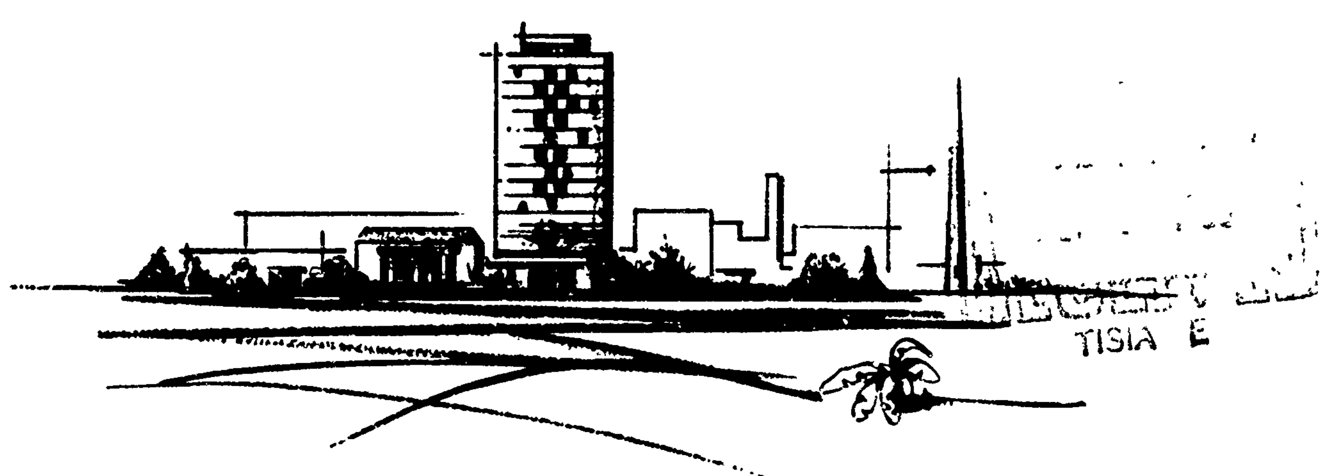

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SUMMARY REPORT OF RESEARCH
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on

DESIGN AND COST STUDY OF HEAT EXCHANGERS FOR AIR-CYCLE SHELTER COOLING SYSTEMS

Sub Contract No. B 64207-US
Subtask 1422A

to

STANFORD RESEARCH INSTITUTE

March 10, 1965

by

John D. Hummell

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September 2, 1965

Civil Defense Technical Office
Stanford Research Institute
Menlo Park, California

Attention Mr. W. L. White, Director

Dear Mr. White:

Attached is our Summary Report, "Design and Cost Study of Heat Exchangers for Air-Cycle Shelter Cooling Systems", prepared under Subcontract No. B-64207-US. Copies of this report are being mailed in accordance with the supplied distribution list.

Under this contract the design and performance factors of heat exchangers were studied in relation to the cost effectiveness of complete air-cycle systems to supplement previous work under Contract No. OCD-OS-62-191, Subtask 1422A, "Methods for Disposing of Excess Shelter Heat".

This study showed that air-cycle systems would be economically feasible for cooling shelters if low-cost compressor expanders could be developed. The heat-exchanger costs would be a small part of the total cost of the system. Thus no further work on heat-exchanger designs is needed at this time.

Future work on the air-cycle system should include the design and development of the expander-compressor and a study of power sources to drive it at high speeds.

The results of this project have shown that air-cycle systems are attractive for cooling shelters. We recommend that further research be undertaken, especially toward the design and development of the expander-compressor package.

Sincerely,

James A. Eibling

James A. Eibling
Chief, Thermal Systems Division

JAE/mln
Enc.

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DESIGN AND COST STUDY OF HEAT EXCHANGERS FOR AIR-CYCLE SHELTER COOLING SYSTEMS

by

John D. Hummell

A study of both novel and conventional cooling systems was described in a Battelle report to the Office of Civil Defense, "Methods for Disposing of Excess Shelter Heat", by J. D. Hummell, D. E. Bearint, and L. J. Flanigan, under Contract No. OCD-OS-62-191, Subtask 1422A. That report included a section on an air-cycle system describing its advantages and pointing out the need for additional study on the cost effectiveness of the heat exchanger and the piping. This report covers the additional work done for the Stanford Research Institute under Subcontract No. B-64207-US to evaluate the cost and design of heat exchangers and piping in relation to the performance and cost of the complete cooling system.

REVIEW OF RESULTS OF PREVIOUS STUDY

In the previous study it was determined that using air as the working fluid, the air-cycle system provides the following advantages for cooling shelters:

(1) Pressurized standby storage of working fluid is not necessary

(2) Small leaks can be tolerated without serious loss of performance

(3) Shelter occupants will not be endangered by leakage

(4) Centrifugal compressors and expanders are suitable for the machine

(5) The shelter can be ventilated without additional blowers

(6) The load for the power source can be provided should the power source require periodic exercise at full load.

The significance of the first three of the above items is that maintenance requirements are low during standby and operation is safe and reliable. The fact that small leaks would not impair the operation makes application attractive in blast-type shelters where equipment may be subjected to considerable physical shock. While a major fracture of a component would render the machine useless, shelter occupants would not be endangered, as they might be if some conventional refrigeration machines and working fluids were used.

Centrifugal compressors and expanders [Item (4)] are best suited for a shelter cooling system because of their comparatively low projected cost, ability to deliver oil-free air, and high-frequency sound characteristic which can easily be muffled. Single-stage machines with the compressor and expander wheels mounted back-to-back on a common shaft seem most appropriate for shelter application.

Items (5) and (6) are capabilities of the air-cycle machine which affect the design and maintenance of the entire cooling, ventilating, and power-source facilities. With proper ductwork and dampers at the compressor intake, the ventilating air could be supplied and mixed with the recycled air from the shelter; thus, a separate ventilation blower would not be needed. Also to be considered is the maintenance of the power source. Presently it is common practice to run emergency standby engines periodically at full power for several hours. Some means of loading the engine is required. Loading could be handled by running the air-cycle machine while maintaining the shelter temperature by controlling the heat rejected to the sink.

The disadvantages of the air-cycle system are the large power requirement, the need for adequate cooling water as the heat sink, and the lack, at the present time, of suitable compressors and expanders and the uncertainty of their costs. The power and sink requirements are discussed later in this report. Air-cycle systems of the type required for cooling shelters have never been built, and therefore the performance and cost of such systems must be estimated based on judgment and engineering experience in related fields. Equipment availability is discussed briefly in the Appendix.

Figure 1 shows schematically the air-cycle system considered most appropriate for cooling shelters. The superior merits of this arrangement compared with other possible arrangements were discussed in the report "Methods for Disposing of Excess Shelter Heat". As shown in Figure 1, warm and humid air from the shelter passes through the compressor where its pressure and temperature are raised. In the heat exchanger, water flowing counter current to the air absorbs heat. Under most circumstances the air will be cooled below the dewpoint at the higher pressure, and water will condense to be drained from the heat exchanger. On passing through the expander, which is coupled to the compressor, the air does work to aid in driving the compressor, while the air temperature and pressure decrease. In the expander the temperature of the air will drop below the dew point and possibly below the freezing point, and the expander and its downstream piping will have to be designed to handle the water or frost formed.

The remainder of this report summarizes the work done to evaluate the heat exchanger and piping design and cost as related to the performance and cost of the total cooling system.

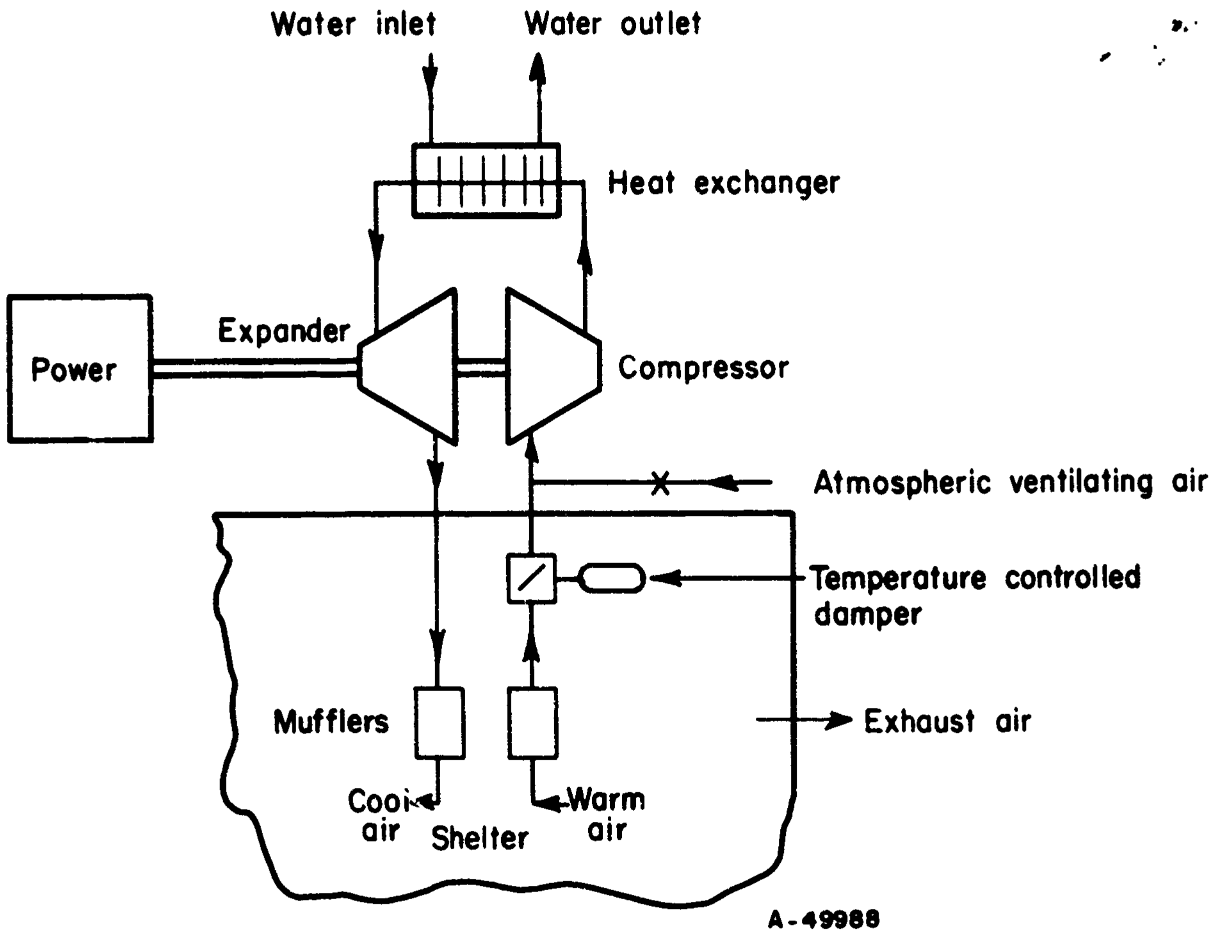

FIGURE 1. SCHEMATIC DIAGRAM OF AIR-CYCLE SHELTER COOLING SYSTEM

SUMMARY AND CONCLUSIONS

In this study calculations were made to show how the heat transfer and the pressure loss of the heat exchanger and piping would affect the pertinent air-cycle parameters. Appropriate values for heat exchanger characteristics, efficiencies of compressors and expanders, pressure ratios, and air temperature at the expander inlet were used to determine the power, equipment, and heat-sink requirements for an air-cycle system in a shelter. Then, for various possible operating conditions, cost estimates were made for each of the major components of the system. Cost estimates were made assuming that the compressor-expander costs per cfm would be independent of the pressure ratio and the flow rate.

The major conclusions of this work are

(1) Air-cycle cooling systems would be comparable in cost to conventional air-conditioning machines for compressor expanders costing $3 per cfm if driven by gasoline engines, or $1 per cfm if driven by diesel engines

(2) For the lowest cost cooling system, the temperature of the air at the expander inlet should be not more than 10 F above the temperature of the water at the heat-exchanger inlet

(3) Systems for pressure ratios of 3 or 4 are significantly lower in cost than for a pressure ratio of 2.

(4) A system designed for a pressure ratio of 3 and a thermal efficiency of 0.75 would cost less than a system for a pressure ratio of 4 and a thermal efficiency of 0.70.

(5) A finned-tube heat exchanger in a custom-built housing and piping arrangement would represent no more than 10 to 30 per cent of the cost of the total system

(6) Water leaving the heat exchanger could be at least 160 F without significantly affecting the performance or cost of a system having a pressure ratio of 3 or 4

(7) Future work should include a study of the applicable power sources and the development of compressor expanders.

In regard to future work, the power requirement of the air cycle is sufficiently large that a program is warranted to determine if low-cost engines would be reliable for two-week operation. Even so, the economic and practical feasibility of using an air-cycle system to cool survival shelters rests primarily on the performance characteristics and costs of the compressor expander. It is not known whether compressor-expander costs would be a function of the volume flow only and independent of the pressure ratio and thermal efficiency. However, the air-cycle system appears so attractive for shelter cooling that further work should be done to determine:

(1) The relation between compressor-expander cost, pressure ratio, thermal efficiency, and capacity

(2) The conditions which might prevail in the expander when the air temperature falls below the dew point or the freezing point

(3) The best method for muffling the high-frequency sound produced by high-speed compressor expanders.

DISCUSSION OF DESIGN/COST RELATIONSHIPS

In an air-cycle system, the significant design parameters which can be varied, within limits, are

Design Parameters

Pressure ratio of compressor and expander
Mechanical efficiency
Pressure loss of system
Air temperature at expander inlet
Thermal efficiency.

These, in turn, establish the cycle parameters:

Cycle Parameters

Air temperature at the compressor outlet
Air temperature at the expander outlet
Power requirement
Heat to the sink
Cooling capacity per lb of air flow.

The design and cycle parameters which govern the heat exchanger and piping are:

Heat Exchanger and Piping Parameters

Air temperature at the compressor outlet
Pressure ratio of the compressor
Water temperature at inlet of heat exchanger
Water temperature at outlet of heat exchanger
Air temperature at the expander inlet
Heat to the sink
Air velocity and pressure loss through heat exchanger.

For discussion of the above factors, the following definitions and symbols are used:

$$\text{COP (Coefficient of Performance)} = \frac{\text{net cooling}}{\text{power input}}$$

η_m = mechanical efficiency

η_{th} = thermal efficiency.

Design and Cycle Parameters

The following steps were used to calculate the values of the cycle parameters for practical values of the design parameters:

(1) Estimated wet and dry bulb temperatures for an effective temperature of 85 F at the compressor inlet

(2) Calculated compressor work

(3) Calculated temperature at compressor outlet

(4) Calculated heat rejected by heat exchanger

(5) Calculated water removed in heat exchanger

(6) For the water vapor, calculated the expansion work and temperature at the expander outlet

(7) For the air, calculated the expansion work and temperature at the expander outlet

(8) Calculated power requirement

(9) Calculated air-water mixture temperature at expander outlet if water vapor, liquid water, and air were mixed adiabatically

(10) Calculated the wet and dry bulb temperature of air at the compressor inlet after heat addition

(11) Compared assumed temperatures of Step 1 with those of Step 10 and repeated all calculations necessary.

Steps 1 through 8 were done using the customary equations for polytropic compression and expansion processes, heat balances, and energy balances.

Step 9 was required because the calculated temperature of the air at the expander outlet was considerably less than the temperature of the water vapor. To obtain temperature equilibrium, it was assumed that the air and water vapor would mix adiabatically, resulting in the formation of liquid water or ice, depending upon the final temperature of the mixture.

In Step 10, the wet and dry bulb temperature for an 85 F effective temperature at the compressor inlet was calculated assuming addition of heat at a sensible ratio of 0.47 to the saturated air from the expander.

Mechanical Efficiency

The cycle parameters were calculated for a mechanical efficiency of 1.0 and also for compressor and expander efficiencies of 0.95 each. The result was approximately a 10 per cent larger power requirement for the latter case. The 0.95 value would be applicable to the smaller machines, but might be too low for the larger ones. The power input to overcome mechanical losses would be converted to heat in bearings and gears and this heat would be 10 to 15 per cent of the total heat to the sink.

In this report, mechanical efficiencies of 0.95 are used. It is assumed that the resultant heat would be dissipated by bearing cooling water and would not impose an additional heat load on the air-cycle machine.

Pressure Loss

The pressure loss between the compressor outlet and the expander inlet would depend upon the piping and heat exchanger design. Pressure loss results in a lower COP because less power is developed by the expander, and less cooling is obtained per lb of air flow. A larger machine is required for a given cooling capacity.

Figure 2 shows, for various pressure ratios, the COP and air flow requirements. The dotted line represents no pressure loss between the compressor and expander, which, of course, is not a real situation. The solid curves were calculated using a pressure loss of 2 psi.

Table 1 shows in more detail the effect of pressure loss between the expander and compressor at pressure ratios of 2, 3, and 4.

TABLE 1. EFFECT OF PRESSURE LOSS ON COP AND HEAT ABSORBED PER LB OF AIR FLOW

Compressor Pressure Ratio	Pressure Loss, psi	COP	Heat Absorbed, Btu/lb air flow
2	0	0.53	12.6
2	2	0.36	9.6
3	0	0.51	23.0
3	2	0.48	21.9
4	0	0.49	29.5
4	2	0.46	28.3
4	4	0.45	27.8

Note: η_{th} = 0.75; temp expander inlet, 120 F.

The decrease of the COP and the heat absorbed per lb of air is much more severe at a pressure ratio of 2 than at pressure ratios of 3 and 4. At pressure ratios of 3 and 4, a pressure loss of 2 psi lowers the COP and the heat absorbed per lb of air flow about 4 to 6 per cent. This means that the power requirement and the machine size would have to be increased 4 to 6 per cent. As will be shown later, the cost of keeping pressure loss below 2 psi is less than the cost of the larger machine and greater power supply. To be somewhat conservative and to allow for unpredicted losses, the cycle parameters, including the temperature of air at the expander outlet, were calculated for a 2-psi loss.

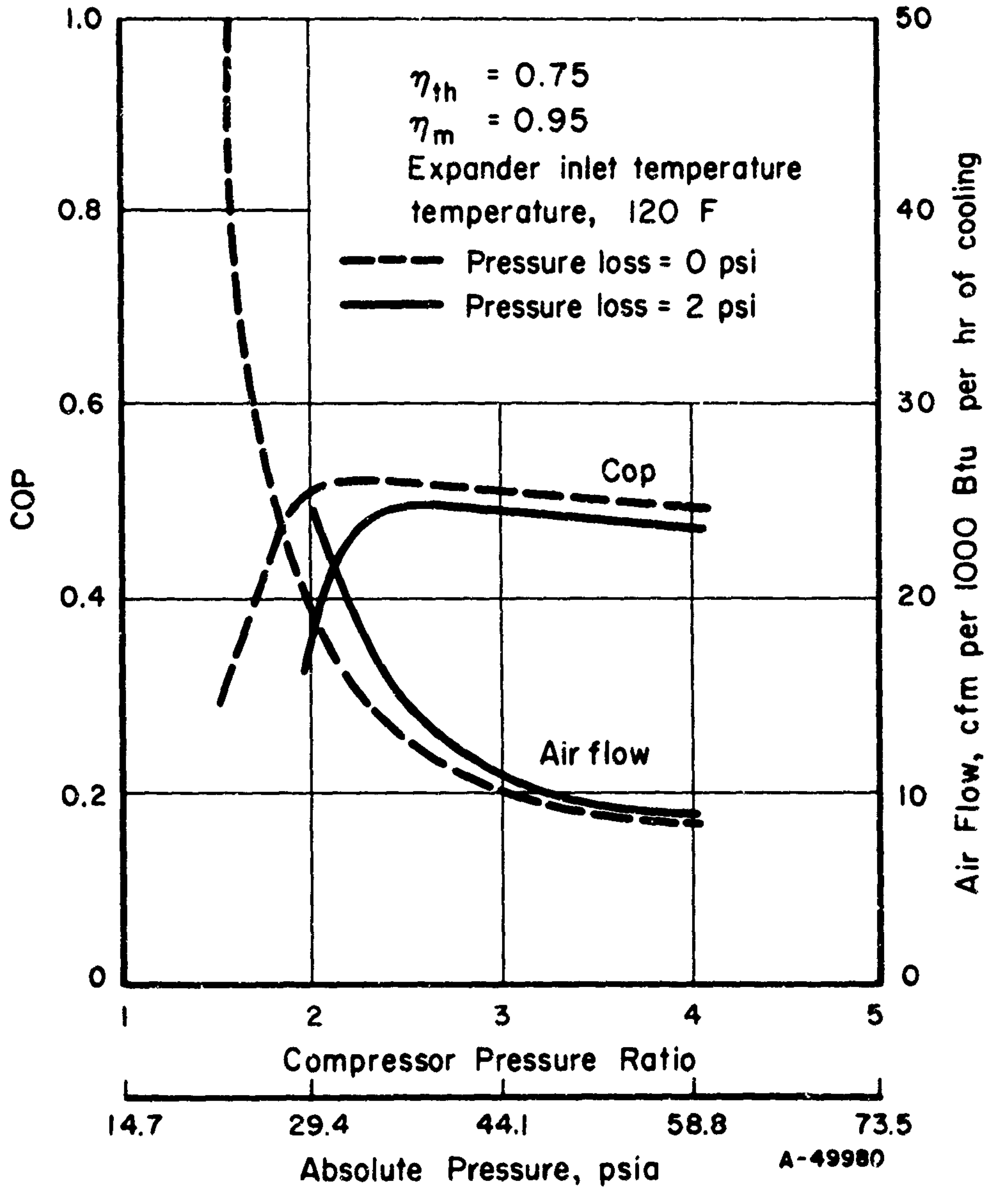

FIGURE 2. COEFFICIENT OF PERFORMANCE (COP) AND AIR FLOW REQUIREMENTS FOR PRESSURE LOSSES OF 0 AND 2 PSI BETWEEN THE COMPRESSOR AND THE EXPANDER

Expander Inlet Temperature

Figure 3 shows the effect of air temperature at the expander inlet on the COP and the heat absorbed per lb of air flow for compression ratios of 3 and 4. The COP's for these two compression ratios are, for practical purposes, the same while the heat absorbed per lb of air flow at a pressure ratio of 4 is about 1.3 times that at a pressure ratio of 3.

The steep slope of the curves emphasizes the need to cool the air as much as possible in the heat exchanger. A change of 10 F makes a 20-per cent difference in the COP and power requirement and a 15-per cent difference in the heat absorbed per lb of air flow. The implications of this and the heat exchanger design are discussed later.

Thermal Efficiency

Figure 4 shows the effect of the thermal efficiency of the compressor and expander on the COP and heat absorbed for three air temperatures at the expander inlet. A decrease in thermal efficiency of 5 percentage points has about the same effect on the COP as increasing the air temperature by 10 F. The heat absorbed per lb of air flow changes only about 5 per cent for a change in thermal efficiency of 5 percentage points.

Thermal efficiencies of about 0.75 are obtainable for single-stage high-speed centrifugal compressors and expanders mounted back-to-back on a common shaft. Experience has shown that for best efficiency the specific speed of a compressor should be about 80 and that for the expander 100. To obtain this would require the compressor and expander to run at different speeds on different shafts connected by appropriate gears. If a compromise were made from the ideal speeds to obtain the simplicity of a single-shaft machine, the compressor and expander specific speeds for the back-to-back configuration would be:

	Specific Speed	
	Compressor	Expander
Pressure ratio = 3	69	120
Pressure ratio = 4	65	128

It is estimated that this deviation would reduce the thermal efficiency only about 1 percentage point from the maximum.

Table 2 shows the speeds and diameters of the compressors and expanders for pressure ratios of 3 and 4, a thermal efficiency of 0.75, and various cooling capacities and air temperatures at the expander inlet. As wheel sizes decrease below about 4 inches which would be used for shelters housing fewer than 600 occupants, the maximum thermal efficiency obtainable may decrease below 0.75. A study of compressors and expanders would be required to determine the minimum sizes applicable for cooling shelters.

Temperatures at Compressor and Expander Outlets

Table 3 shows the calculated air temperatures at the compressor and expander outlet for compressor and expander thermal efficiencies equal to 0.75 and a pressure drop of 2 psi between the compressor and expander.

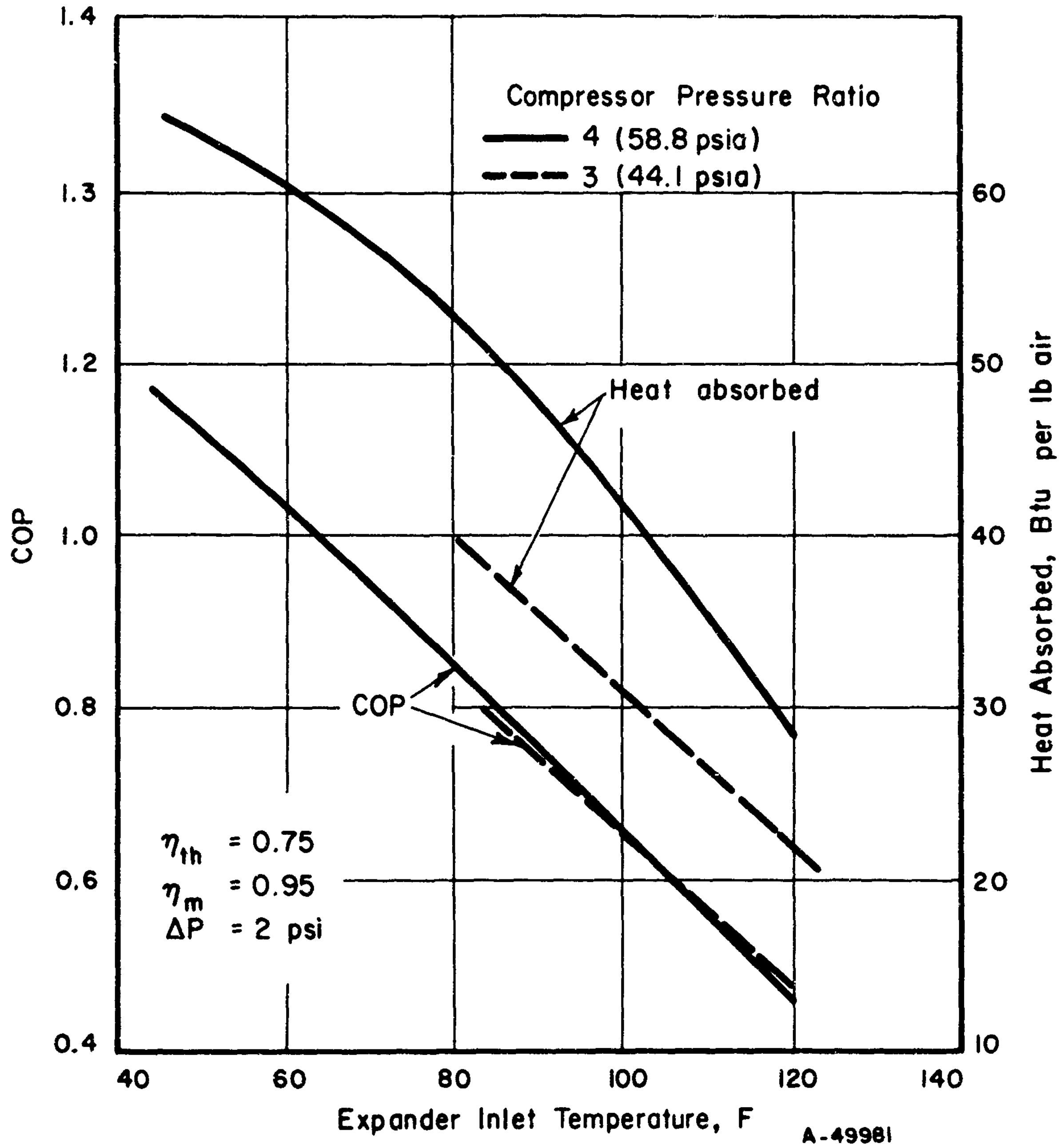

FIGURE 3. EFFECT OF EXPANDER INLET TEMPERATURE ON COP AND HEAT ABSORBED PER LB OF AIR FLOW

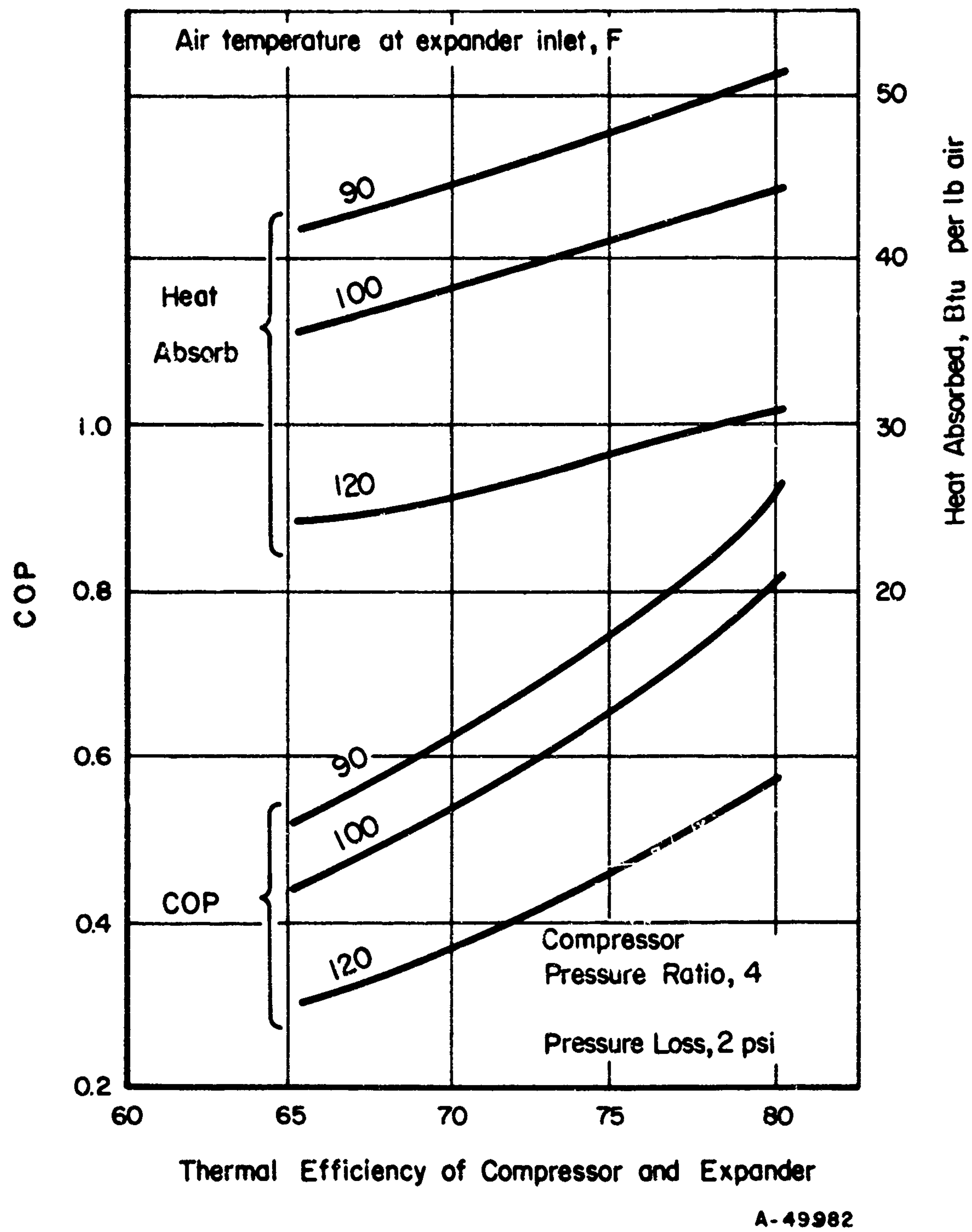

FIGURE 4. EFFECT OF THERMAL EFFICIENCY AND AIR TEMPERATURE AT THE EXPANDER INLET ON THE COP AND HEAT ABSORBED PER LB OF AIR FLOW

TABLE 2. APPROXIMATE COMPRESSOR AND EXPANDER DIAMETERS AND SPEEDS

Cooling Load, 1000 Btu per hr	Air Temp at Expander Inlet, F	Compressor Pressure Ratio, 3			Compressor Pressure Ratio, 4		
		Speed, 1000 rpm	Diameter, inches		Speed, 1000 rpm	Diameter, inches	
			Compressor	Expander		Compressor	Expander
50	90	115	2.6	1.8	209	2.0	1.4
50	100	105	2.8	2.0	193	2.1	1.5
50	120	89	3.3	2.4	160	2.6	1.8
100	90	81	3.6	2.6	148	2.8	1.9
100	100	75	4.0	2.9	137	3.0	2.1
100	120	63	4.7	3.4	113	3.7	2.6
300	90	47	6.3	4.5	36	4.8	3.3
300	100	43	7.0	4.9	79	5.2	3.7
300	120	36	8.1	5.9	66	6.4	4.5
1000	90	26	11.4	8.2	47	8.8	6.1
1000	100	24	12.4	9.0	43	9.5	6.7
1000	120	20	14.8	10.7	36	10.7	8.1
5000	90	12	25.5	18.5	21	19.6	13.6
5000	100	11	28.0	20.0	19	21.4	14.8
5000	120	9	33.1	23.9	16	26.2	18.2

TABLE 3. AIR TEMPERATURES AT COMPRESSOR AND EXPANDER OUTLET

Compressor Pressure Ratio	Expander Inlet Temp, F	Compressor Outlet Temp, F	Expander Outlet Temp, F
3	90	364	23
3	100	365	35
3	120	366	52
4	50	435	-63
4	90	440	-6
4	100	446	11
4	120	450	40

It would be necessary to duct the cooled air to various portions of a shelter where it could mix with the bulk of the air for cooling the occupants. The sensitivity of the cooled air temperature to the air temperature at the expander inlet suggests that a simple control system could be devised which would regulate the quantity of cooling water to the heat exchanger.

Heat Exchanger and Piping Design

Figure 5 shows schematically the piping and heat-exchanger layout for the air-cycle system. The air would leave the compressor at about 200 fps and then pass through a diffuser where the velocity would decrease with a gain in static pressure. Heat would be rejected in a counter-flow heat exchanger and some water would condense to be drained from the casing. The flow would then have to be turned approximately 180 degrees to return the air to the expander inlet where a nozzle would be used to increase the velocity to about 200 fps.

A detailed piping and heat-exchanger design and cost study were made for a finned-tube heat exchanger, and one example was worked out for a concentric-tube heat exchanger.

Finned-Tube Heat Exchanger

Finned-tube heat exchangers are particularly suited to an air-cycle system in which air is to be cooled by water. They are designed for specific applications and constructed of standard finned-tube sections installed in an appropriate housing. For this study heat-transfer, pressure-drop, and air-flow data obtained with a typical finned-tube design were used to estimate the heat-exchanger and piping requirements.

Finned-tube performance data are available for atmospheric pressure air. However, the pressure of the air flowing over a finned-tube bank affects the flow velocity at which water would be blown off the surfaces, the heat-transfer coefficients, and the pressure loss. Air-viscosity differences are small enough to be neglected. Therefore, the atmospheric-pressure data were modified for higher pressure air-cycle applications.

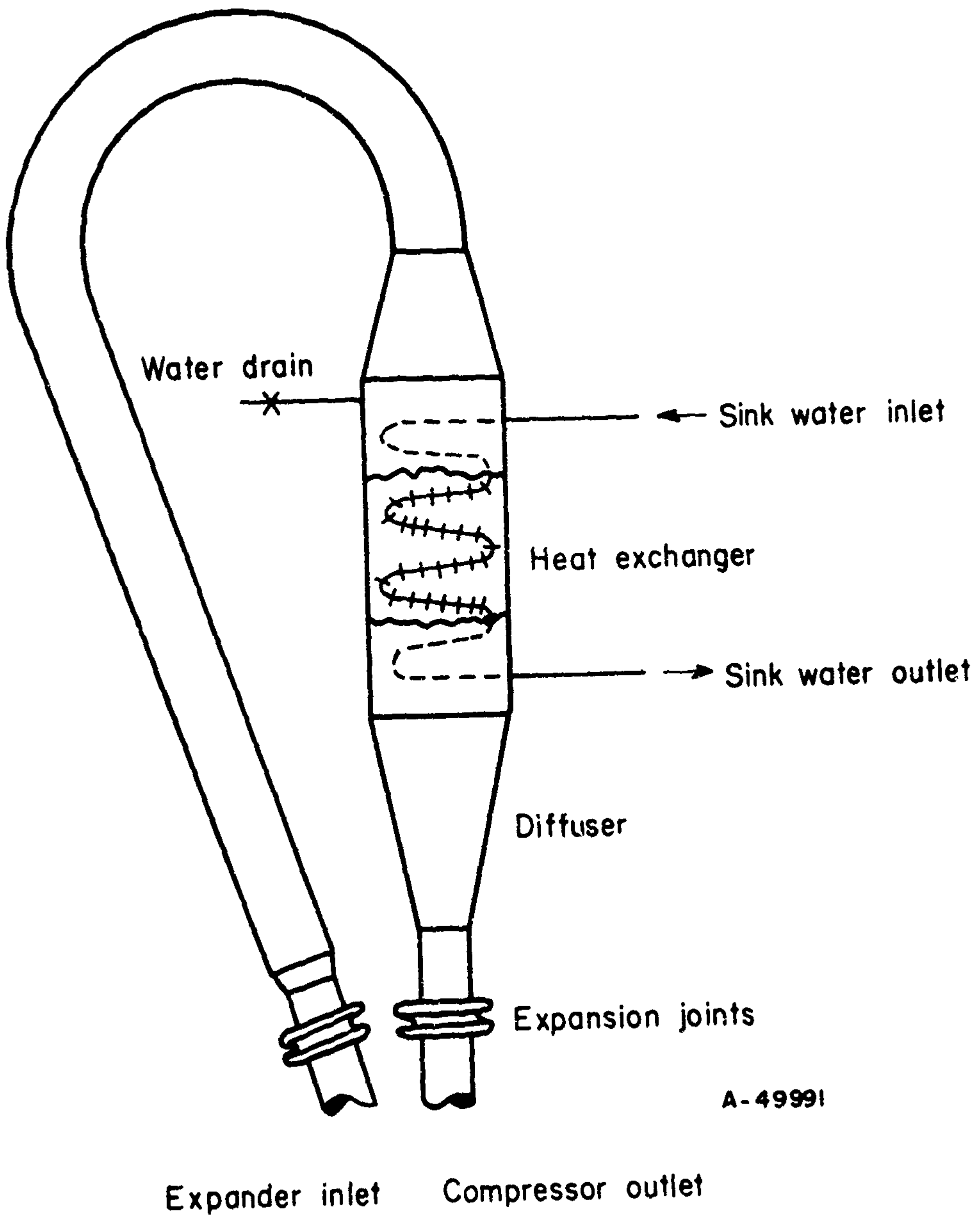

FIGURE 5. PIPING AND HEAT EXCHANGER LAYOUT FOR AN AIR-CYCLE SYSTEM

The water, which would be condensed out of the air on the cooler heat-exchanger surfaces, could run down the fins and tubes into a collecting plenum and a drain. Air velocities would have to be low enough to avoid blowing the water off the surfaces and carrying the drops into the expander. The drag force acting on the water-covered surfaces would be proportional to the density times the velocity squared. At atmospheric pressure, the maximum face velocity which could be used without water carryover is about 700 fpm. This establishes the relation between the density and velocity at any pressure as:

$$\rho V^2 = \text{constant} = 37{,}000 \quad ,$$

where

ρ = air density, lb per cu ft

V = air velocity, cfm.

The heat-transfer coefficient of finned-tube banks varies approximately as the Reynolds number to the 0.6 power. The effect of higher pressure, using ρV^2 = constant, is the heat-transfer coefficient is proportional to the density to the 0.3 power.

The pressure drop through a tube bank is proportional to ρV^2. Therefore, because ρV^2 = constant, the pressure loss data at atmospheric pressure are directly applicable to this higher-pressure application.

Figure 6 shows three piping layouts studied using the above relations for the heat exchanger and data on pressure loss through piping systems for the air-cycle parameters of Table 4.

TABLE 4. CYCLE PARAMETERS FOR PIPING STUDIES

Pressure Ratio	4
Thermal Efficiency	0.75
Cooling Load	300,000 Btu/hr
Heat to Sink	635,000 Btu/hr
Air Flow	6,230 lb/hr
Air Velocities	
Compressor Outlet	200 fps
Expander Outlet	200 fps
Temperatures	
Compressor Outlet	445 F
Expander Inlet	90 F
Water Inlet	80 F
Water Outlet	130 F
Log Mean Temperature Difference	89 F

Table 5 lists the details of the heat exchanger, the piping, and the pressure losses for the configurations of Figure 6. The heat exchanger and downstream piping are the same for all three systems because these depend only on the air pressure, air flow, and the log mean temperature difference. The finned-tube heat-exchanger configuration selected had 5/8-inch tubes, tube spacings of 1-1/2 and 1-3/4 inches, 93 fins per foot

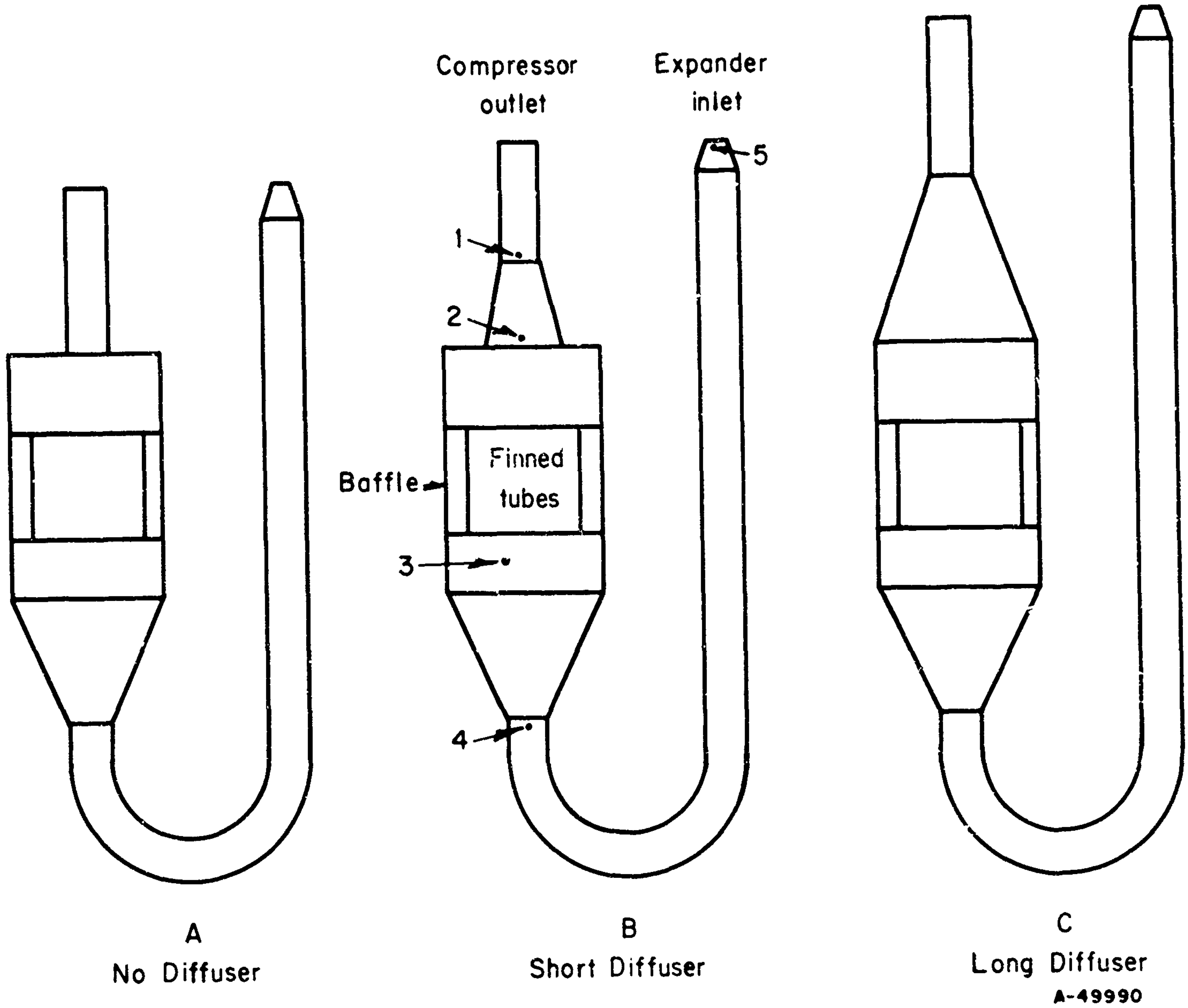

FIGURE 6. PIPING AND HOUSING DESIGNS FOR FINNED-TUBE HEAT EXCHANGERS

(Details of Finned Tube Section in Appendix.)

of tube, and about 23 square feet of heat-transfer area per square foot of face area per row of tubes. This configuration would have an over-all heat-transfer coefficient of 290 Btu per (hr, sq ft face area, log mean temperature difference, row). With a face area of only 1 square foot and a depth of 45 inches, heat would be dissipated to the sink at the rate of 635,000 Btu/hr. The heat exchanger and its headers could be housed in a 20-inch-round pipe.

TABLE 5. DETAILS OF PIPING AND HEAT EXCHANGER FOR CONDITIONS OF TABLE 4 AND CONFIGURATION OF FIGURE 6

	System		
	A	B	C
Diameter at 1, in.	3	3	3
Air Velocity at 1, fps	200	200	200
Diameter at 2, in.	--	4	20
Length from 1 to 2, in.	--	8	120
Diameter at 3, in.	20	20	20
Velocity at 3, fps	2.75	2.75	2.75
Face Area Heat Exchanger, sq ft	1	1	1
Depth of Heat Exchanger, in.	45	45	45
Number of Rows of Tubes	26	26	26
Length 2 to 4, ft	12	12	12
Diameter at 4, in.	4	4	4
Velocity at 4, fps	65	65	65
Diameter at 5, in.	2.4	2.4	2.4
Velocity at 5, fps	200	200	200
Length Pipe from 4 to 5, ft	15	16	25
Pressure Losses, psi			
1 to 2	0.71	0.25	0.15
2 to 3	0.13	0.13	0.13
3 to 5	0.18	0.18	0.20
Total Pressure Loss, psi	1.02	0.56	0.48

The difference in the three systems is in the diffuser at the compressor outlet. The pressure loss of the 8-inch-long diffuser of System B would be about 0.5 psi less than the pressure loss of the abrupt enlargement of System A. The use of the short diffuser of System B would significantly reduce the power requirements, but little advantage could be gained by using the long diffuser of System C.

Total pressure loss in the system between the compressor and expander inlet would be 1 psi or less using velocities of 65 fps in the piping between the heat exchanger and the expander inlet.

Figure 7 shows the estimated installed cost for finned-tube heat-exchanger coils. Using these and a figure of $0.50 per lb for a custom-built piping system, the cost of the heat exchanger and piping layout for System B would be:

Coils	$800
Piping	450
Total	$1250

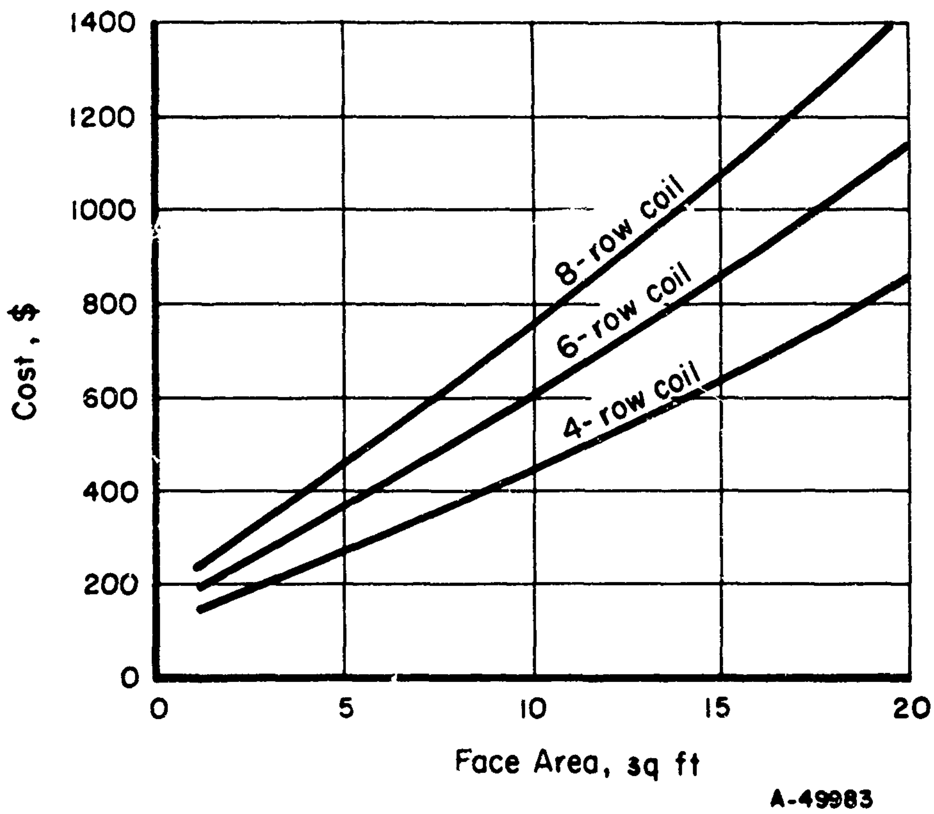

FIGURE 7. INSTALLED COST OF FINNED-TUBE HEAT-EXCHANGER COILS

Concentric-Tube Heat Exchanger

Figure 8 shows two arrangements for concentric-tube heat exchangers, the difference being the location of the diffuser which is downstream from the heat exchanger in System D and upstream in System E and System F. Assuming the same air cycle as used for the finned-tube design, the pipe sizes, air velocities, and pressure losses of these designs are shown in Table 6.

TABLE 6. DETAILS OF PIPING AND CONCENTRIC-TUBE HEAT EXCHANGERS SHOWN IN FIGURE 8

	System D	System E	System F
Pipe Size, in.			
At 1	3	3	3
At 2	--	4	5
At 3	4	4	5
At 4	4	4	5
Air Velocities, fps			
At 1	200	200	200
At 2	--	110	70
At 3	68	68	44
At 4	68	68	44
Pressure Loss, psi			
1 to 3	2.9	1.4	0.7
3 to 4	1.1	1.1	0.5
Total, 1 to 5	4.0	2.5	1.2
Overall Heat-Transfer Coefficient, Btu per (hr, sq ft, F)	60	33	25
Length of Heat Exchanger, ft	145	195	205

The simplicity and ruggedness of such heat exchangers are attractive for shelter cooling systems. The water condensed from the air could be removed with some type of a water separator. A cyclone-type separator was considered which would remove all drops above about 10 microns in Systems D and E. For System F a larger separator was considered to permit lower pressure loss even though the larger separator would remove less of the water from the air stream.

The pressure losses of Systems D, E, and F are greater than for the finned-tube design. To obtain lower pressure losses would require larger piping and water separators. This would, in turn, increase the costs, which for the concentric tube systems shown in Table 6, are

	System D	System E	System F
Heat Exchanger	$1100	$1750	$2260
Water Separator	200	200	240
	$1300	$1950	$2500

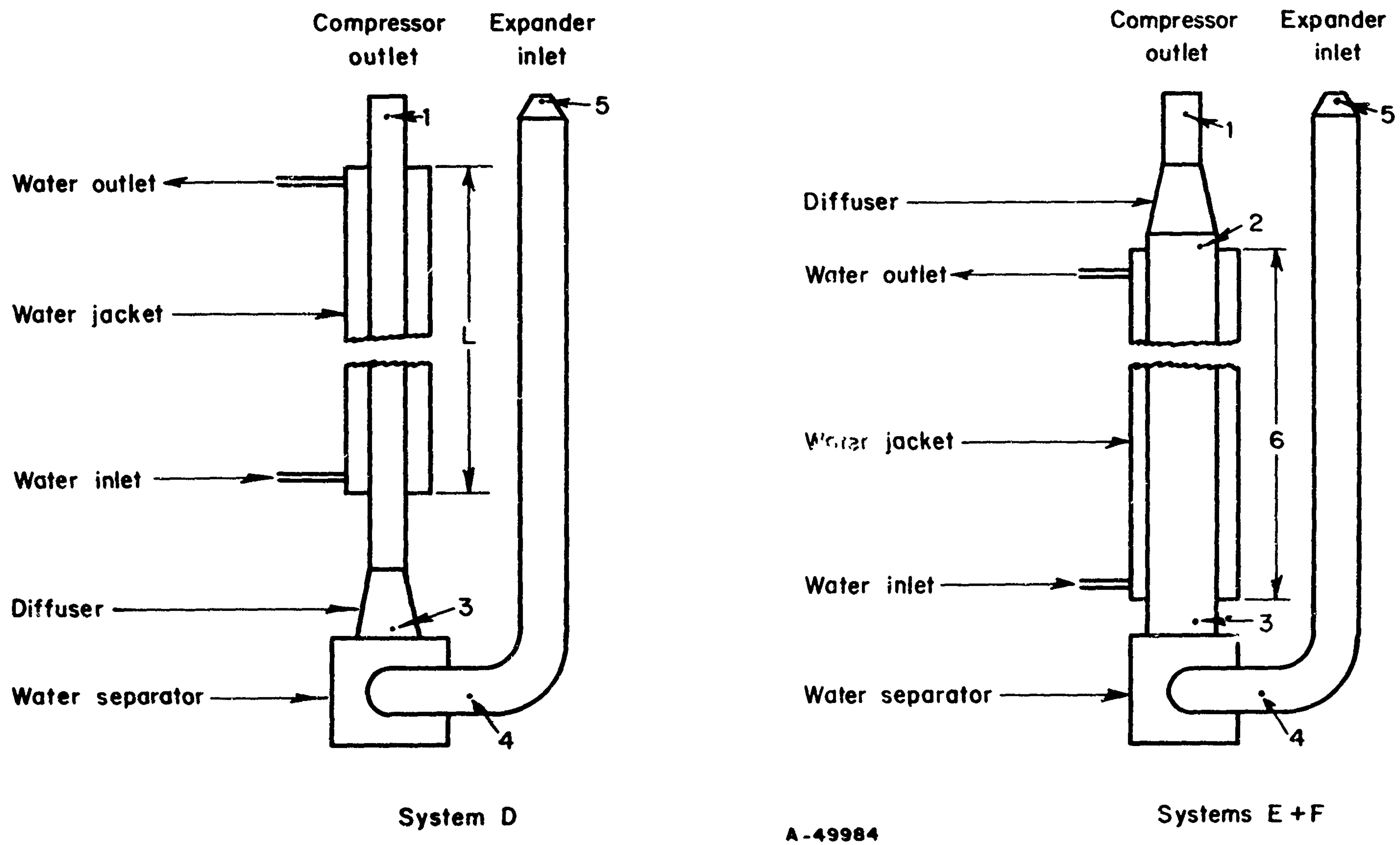

FIGURE 8. CONCENTRIC PIPE HEAT EXCHANGERS

ESTIMATED COST OF THE COMPLETE AIR-CYCLE SYSTEM

The heat exchanger affects the total cost of an air-cycle system primarily because it governs the inlet temperature to the expander. Previously discussed were the relations between the expander inlet temperature, COP, heat-rejection factor, air-flow requirements, and power. The costs of complete cooling systems were estimated using these relations, the estimated cost for the finned-tube heat exchanger, and the considerations of Table 7.

TABLE 7. CONSIDERATIONS FOR COST ESTIMATING

(1)	``-t-exchanger coil costs proportional to the heat-rejection rate, inversely proportional to the log-mean-temperature difference, and inversely proportional to the density of the 0.3 power
(2)	Piping and coil-housing cost proportional to the square root of the flow rate and inversely proportional to the density to the one-fourth power
(3)	Power to drive the compressor, $15 per horsepower for industrial gasoline or LP-gas engines, or $40 per horsepower for heavy-duty diesel engines
(4)	Compressor and expander, $1 per cfm at compressor inlet
(5)	Ample supply of water at no cost
(6)	Expansion joints and water pumps as noted in following tabulations

It was outside the scope of this project to specify the type or service duty of engines which would be satisfactory for driving an air-cycle system for two weeks in a survival shelter. Yet the cost of the system depends largely upon the cost of power. Therefore, two values were used which bracket power costs between $15 per horsepower for industrial gasoline or LP-gas engines and $40 per horsepower for heavy-duty diesel engines.

Table 8 shows in detail the parameters of several air-cycle cooling systems and the estimated installed cost of each major item. The costs as dollars per 1000 Btu per hour of cooling are summarized in Figure 9.

The significance of the results are

(1) The cost of power is the major one

(2) The cost of the air-cycle system varies significantly with expander inlet temperature

(3) The cost of the piping and heat exchanger is 10 to 20 per cent of the total cost of the diesel-engine-powered system and 20 to 30 per cent of the cost of the gasoline-engine-powered system

TABLE 8. AIR-CYCLE COOLING SYSTEM PARAMETERS AND ESTIMATED COSTS FOR A COOLING CAPACITY OF 300,000 BUT PER HR

Pressure Ratio	2	2	3	3	4	4	4	4	4
Thermal Efficiency	0.75	0.75	0.75	0.75	0.75	0.75	0.75	0.70	0.70
Temperatures, F									
Air at Expander Inlet	90	120	90	120	50	90	120	90	120
Water Inlet	80	80	80	80	40	80	80	80	80
Water Outlet	130	130	130	130	130	130	130	130	130
Log Mean ΔF	45	75	74	113	86	89	133	95	142
Heat to Sink, 1000 Btu/hr	780	900	658	843	530	635	878	743	1,030
Water, lb/hr	15,600	18,000	13,160	16,850	5,900	12,700	17,600	14,900	20,600
COP	0.73	0.45	0.76	0.48	1.3	0.75	0.47	0.63	0.37
Air Flow, lb/hr	14,300	27,000	8,300	13,700	4,650	6,230	10,600	6,620	11,500
Air Flow, cfm	3,320	6,300	1,930	3,180	1,080	1,450	2,470	1,540	2,680
Horsepower	160	262	155	245	90	160	250	187	318
Costs, dollars									
Piping, Housing, Expansion Joints	870	1,290	640	830	520	570	780	620	840
Heat Exchanger Coils	2,200	1,500	1,010	850	600	800	900	880	990
Compressor and Expander	3,320	6,300	1,930	3,180	1,080	1,450	2,470	1,540	2,680
Pump	600	650	500	600	300	500	650	600	700
Subtotal	6,990	9,740	4,080	5,460	2,500	3,320	4,800	3,640	5,210
Diesel Power	6,400	10,480	6,200	9,600	3,600	6,400	10,000	7,480	12,720
Gasoline or LP-Gas Power	2,400	3,930	2,320	3,680	1,350	2,400	3,760	2,810	4,790
Total									
Diesel Power	13,390	20,220	10,280	15,060	6,100	9,720	14,800	11,120	17,930
Gasoline or LP-Gas Power	9,390	13,670	6,400	9,140	3,850	5,720	8,560	6,450	10,000

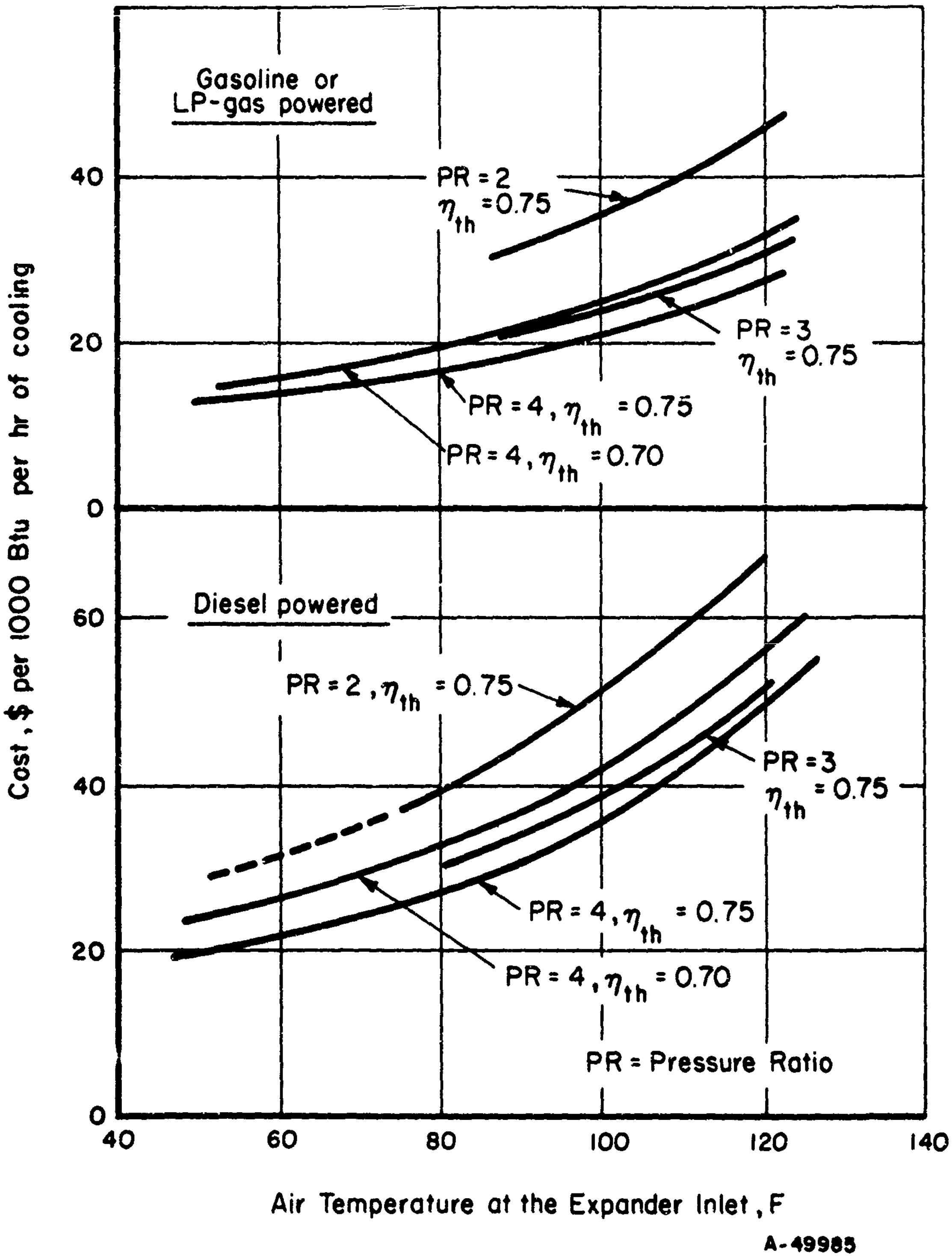

FIGURE 9. SUMMATION OF ESTIMATED COSTS FOR AIR-CYCLE COOLING SYSTEMS

(4) The costs of the heat exchanger and piping are not significantly different for air temperatures at the expander inlet 10 or 40 F above inlet water temperature

(5) The water temperature at the outlet can be increased appreciably with little additional cost and with no loss in performance.

It should be emphasized that the cost estimates were made assuming the compressor expander would cost $1 per cfm. No compressor expanders are available for air-cycle cooling system service, and while the $1 per cfm is strictly speculative at this time, this value is considered by automotive gas-turbine specialists as realistic for future machines. The significance of the expander-compressor cost for the cost of the complete system is shown by Figure 10.

In the previous Battelle report, "Methods for Disposing of Excess Shelter Heat", a cost of $43 per 1000 Btu per hr of cooling was shown for a water-cooled mechanical-vapor-compression system driven by an electric motor and diesel-engine generator set. If the compressor were driven directly by a gasoline engine, the cost would be about $34 per 1000 Btu. Comparison of this value with those of Figure 10 reveals that air-cycle cooling systems of comparable costs could be built using compressor-expanders costing as much as $4 per cfm.

In making comparison between the air-cycle and other cooling systems, the sink requirements should not be overlooked. The total heat to the sink of this air-cycle system and the heat to be dissipated from its driving engine are greater than those of conventional mechanical air-conditioning machines. This would be of small consequence if adequate natural water were available, but it would be important if cooling towers and limited natural water, or cooling towers and stored water were used to dissipate the heat. The relations between the ratio of heat to the sink to the net cooling load, the COP, and engine thermal efficiency are shown by Figure 11.

Because the cost of the heat exchanger and piping would be only 10 to 30 per cent of the total cost, refinements of these components would have little effect on the total cost of the system.

Not expected was the fact that the cost of the heat exchanger and piping would be nearly independent of the air temperatures at the inlet to the expander. The reason for this is that as the expander inlet air temperature decreases, the air flow and the quantity of heat to be rejected to the sink decrease because the COP increases. Consequently, for the lower air temperatures at the inlet, the increased depth of the heat exchanger in the direction of air flow is offset by the smaller cross-sectional flow area.

Because of the relatively high temperature of the air at the compressor outlet, the temperature of the water to the sink can be increased appreciably for a small increase in heat-exchanger size and cost. Assuming a given temperature of water at the heat exchanger inlet and of air at the expander inlet, the increase in temperature of the water leaving the heat exchanger lowers the log mean temperature difference for heat transfer. Consequently, the heat-exchanger size would increase inversely proportional to the log mean temperature differences. Table 9 includes a few log mean temperatures for various temperature combinations to show that significant increases in water temperature at the outlet could be obtained with a minor decrease in the log mean temperature.

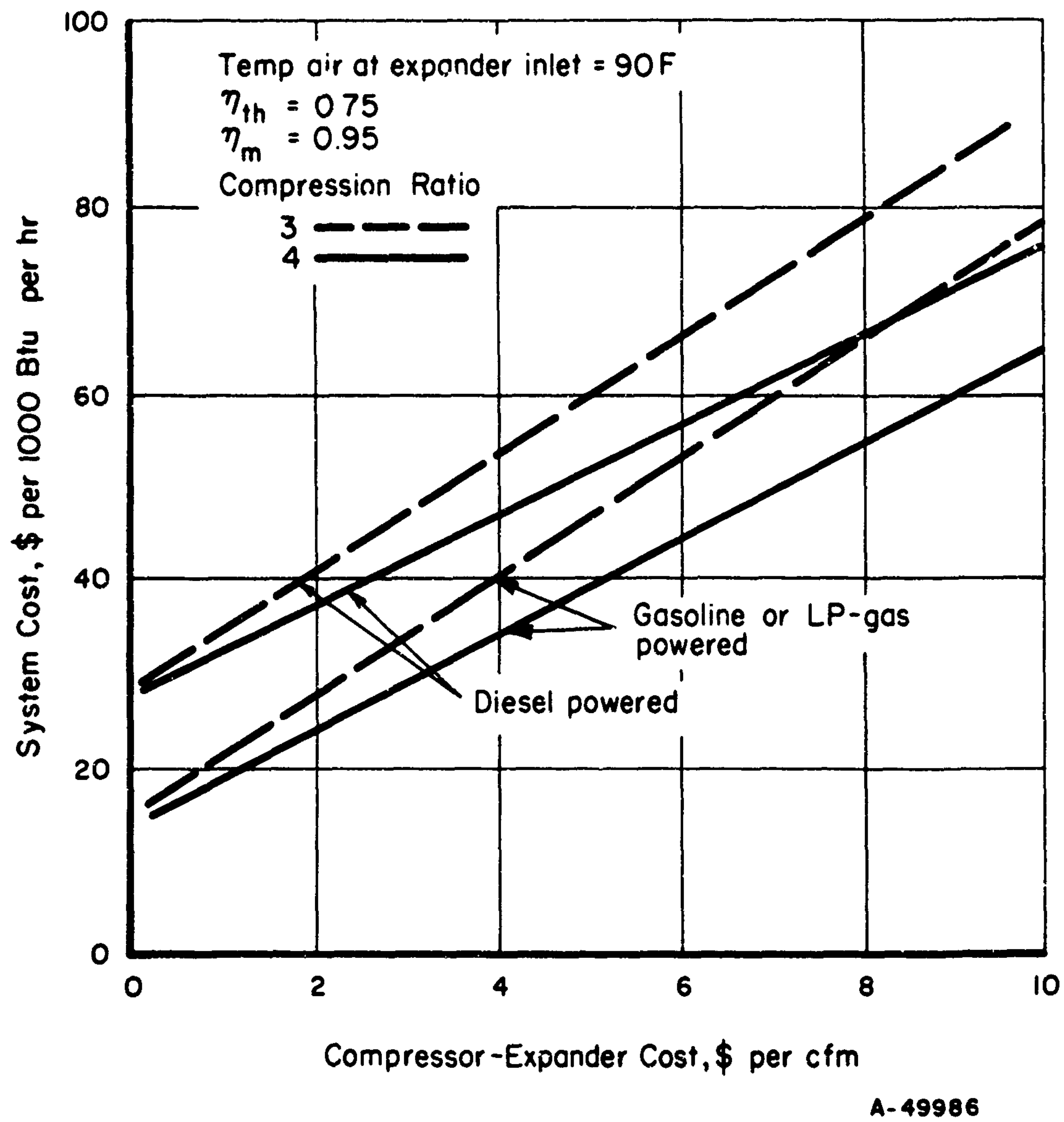

FIGURE 10. RELATION BETWEEN AIR-CYCLE COOLING SYSTEM COST AND THE COST OF THE COMPRESSOR-EXPANDER

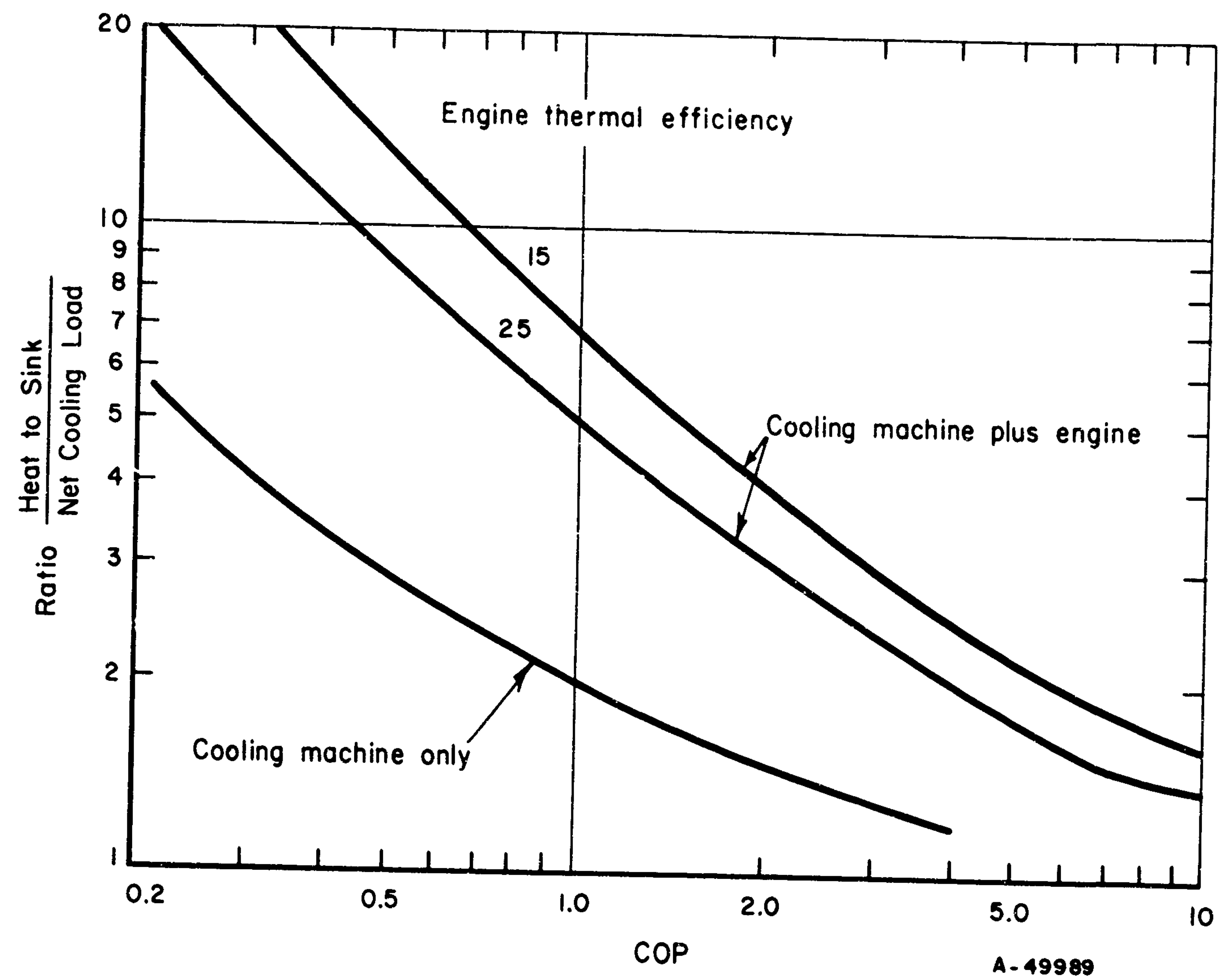

FIGURE 11. RATIO OF HEAT TO A SINK TO NET COOLING LOAD FOR VARIOUS COP AND ENGINE THERMAL EFFICIENCIES

TABLE 9. LOG MEAN TEMPERATURE DIFFERENCES

Water Inlet Temp, F	Air Expander Inlet Temp, F	Pressure Ratio = 3; Water Outlet Temp, F			Pressure Ratio = 4; Water Outlet Temp, F		
		100	130	160	100	130	160
40	50	78	71	65	93	86	79
40	90	142	131	121	153	144	135
40	120	158	147	135	182	172	162
60	90	110	102	94	129	121	113
60	120	141	131	121	163	154	144
80	90	80	74	67	95	89	82
80	120	122	113	104	141	133	125

Whether or not higher temperatures of the water leaving the heat exchanger could be used to advantage depends upon the capacity of the sink. If the supply of water available is adequate, even for a low increase in the water temperature, no benefit will result. On the other hand, if the quantity of water is limited, higher cooling capacities can be obtained by operating with higher water temperatures at the outlet. Should the water supply be so small that a cooling tower is required, higher water temperatures would permit use of a smaller tower.

The estimated cost, on a cooling unit, basis does not vary appreciably with cooling capacity because the costs for power and the compressor expander are dependent only on the characteristics of the cycle and not on the total cooling capacity of a system. The costs for the heat exchanger, piping, and water pump would change, but these account only for the minor portion of the total cost. Estimated costs for various capacities are shown in Figure 12.

FUTURE WORK

Further investigation of the air-cycle cooling system should include the development of the compressor-expander combination and a study of the power sources best suited to this application.

Technical problems which must be studied in connection with the compressor-expander are related to wheel designs, manufacturing techniques, lubrication requirements, drive mechanisms and connection to the power source, compressor cooling by water injection, sound suppression, and difficulties which might occur in the expander when the air temperatures go below the dew point and possibly the freezing point of the water in the air. Some of these are strictly design problems which can be analyzed on the basis of present-day design technology. Others will require the construction and testing of a full-scale system.

Gas turbines are attractive for the power supply because the compressor-expander must run at high speed. At the present time, conventional gas turbine costs are too high for an economic application, but the costs might be reduced for a two-week operation by water or steam injection. Power output by aircraft gas turbines is increased on take-off by water injection. Also, a gas-turbine manufacturer is studying, for cost-reduction

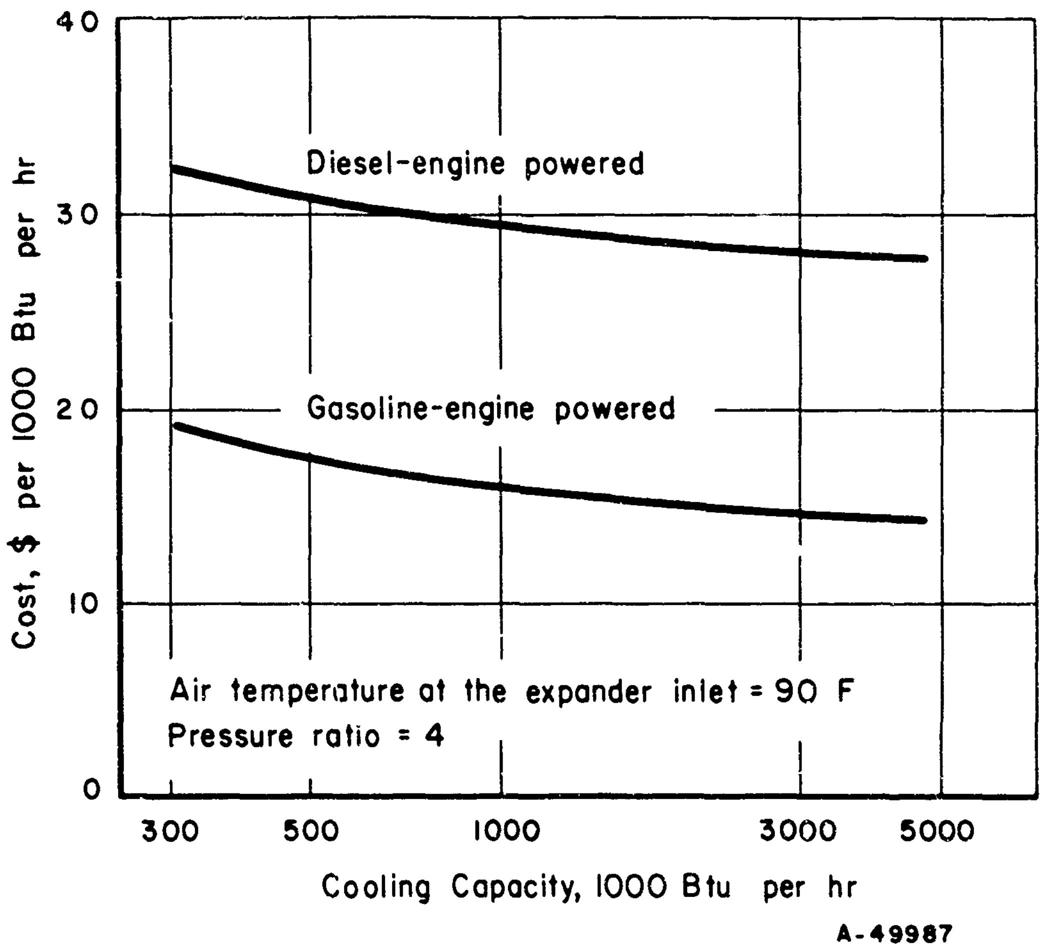

FIGURE 12. ESTIMATED COSTS FOR VARIOUS CAPACITIES

purposes, the injection of steam into a gas-turbine combustion chamber to increase the mass flow rate, to control the turbine-inlet temperature, and to increase the power output for peaking power generation. Similar considerations should be given for gas-turbine power plants for survival shelters.

The cost of power is such a major factor in air-cycle cooling that low-cost engines should be evaluated to determine their reliability for the short service required in a survival shelter.

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APPENDIX A

EQUIPMENT AVAILABILITY

APPENDIX A

EQUIPMENT AVAILABILITY

Table A-1 summarizes the availability of equipment required to build up an air-cycle system for cooling survival shelters.

TABLE A-1. AVAILABILITY OF AIR-CYCLE EQUIPMENT

Research and Development Required	Custom-Built, But Little Research or Development Required	Commercially Available
Expander-compressor	Mufflers	Power source
	Heat-exchanger housing	Heat-exchanger core
	Piping transitions	Expansion joints
		Temperature controls
		Water pump
		Cooling tower

The expander-compressor is the only component of the air-cycle system which would require substantial research and development. Based on experience with gas turbines and high-speed gearing, a single-stage centrifugal machine having the expander and compressor mounted back-to-back on a common shaft seems most appropriate for shelter cooling. The maximum temperature of the expander-compressor would not exceed 450 F, and therefore all the components could be made of low-cost, easily machined materials. The compressor of the air-cycle machine would be nearly identical to the compressor of a gas turbine. The expander of the air-cycle machine would have to be designed for an air inlet temperature between 90 and 120 F and an expansion ratio between 3 and 4. Development and testing would be required to design an expander which could tolerate temperatures of air and water vapor mixtures below the dew point or frost point at the expander outlet.

The expander-compressor could be driven through a speed-changing gear box by a reciprocating engine or by an electric motor. Direct drive by a gas turbine would be equally satisfactory but more costly at the present prices of gas turbines.

The mufflers, heat-exchanger housing, and piping transitions would have to be made specifically for an air-cycle system. These components could be made easily by metal fabricators to the specifications of the designer.

The heat-exchanger core could be selected from the standard stock of the many manufacturers of finned-tube heat exchangers. In some cases the core would consist of more than one standard component in parallel or in series to obtain the desired air and cooling water flow paths. A typical example is shown in Figure A-1.

The temperature controls, water pump, cooling tower, and hardware items such as expansion joints are commercial items which complete the requirements for an air-cycle system.

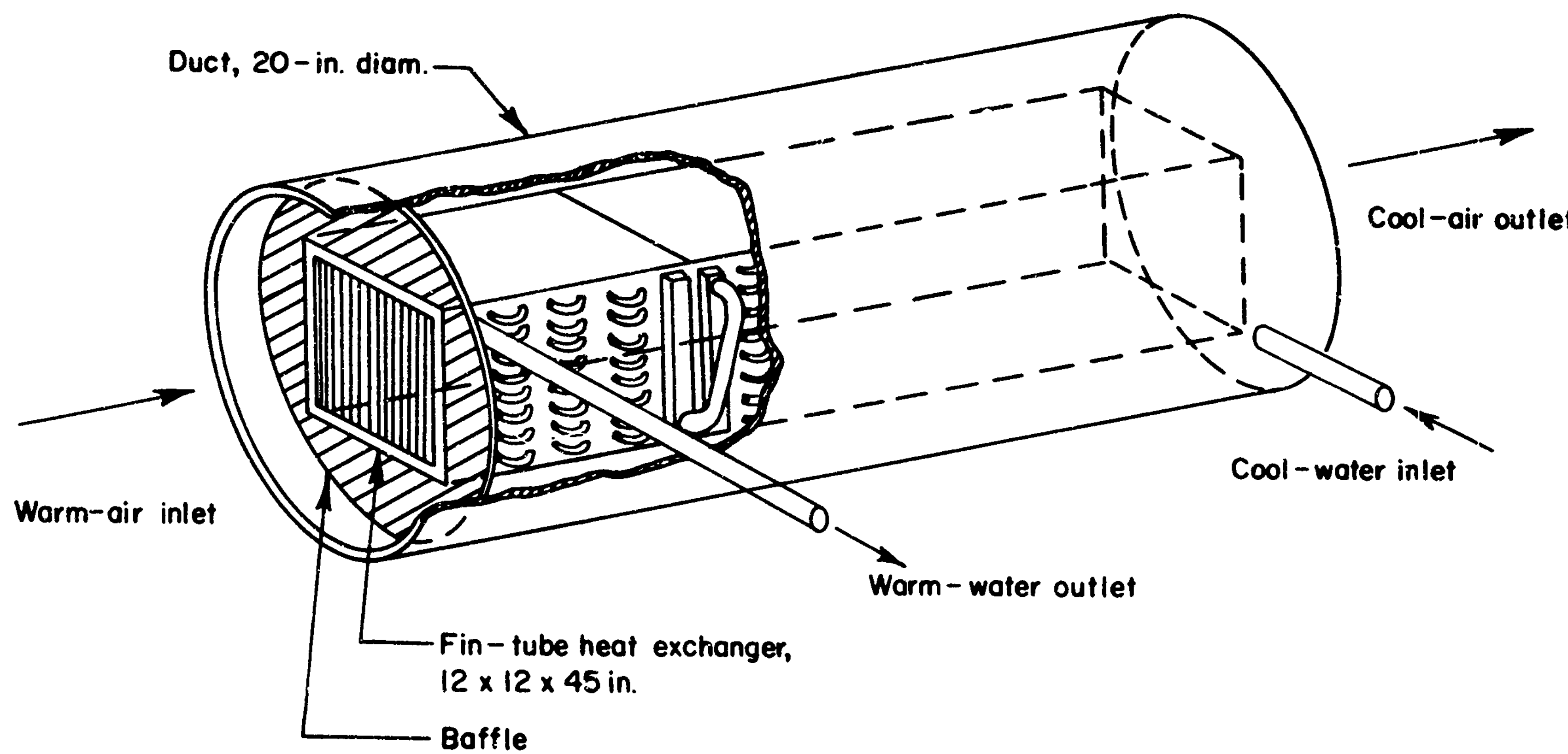

FIGURE A-1. SECTIONAL VIEW OF HEAT EXCHANGER