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FINAL REPORT
LIGHTWEIGHT MOVING RADIATORS FOR HEAT REJECTION IN SPACE

by
Karl Knapp

ARC-TN-1107

10 November 1981

ASTRO
RESEARCH
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CARPINTERIA, CALIFORNIA

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SECTION 1
INTRODUCTION

Large heat-rejection radiators of advanced design will be required for future systems where large amounts of power are generated in space. High-temperature radiators will be needed for solar- and nuclear-powered heat engines where the amount of heat rejected is typically large compared with the power generated. Lower-temperature radiators will be needed to cool concentrating photovoltaic systems and to reject heat from systems where power is used in space, including manned stations and industrial processing operations. These lower-temperature radiators will require large radiating areas in order to compensate for the reduced radiation intensity which, at 316 K, is 100 times lower than 1000 K. In order to establish these large radiators in space, they must be either deployed or assembled on-orbit after being transported from Earth in a compact packaged condition by a vehicle such as the Shuttle. The economic practicality of these large systems depends on the development of very lightweight radiators which can be established readily in space.

In order to reject heat to space, the radiator must provide a surface area with a view of space and a means of efficiently distributing the heat over this surface area. The trend in the development of advanced radiator designs has been to consider heat-pipe and fin designs with fluid-loop headers. The radiators are initially oversized so that the loss of some heat pipes due to micrometeoroid damage leaves sufficient operating area at the end of a specified mission life. Fluid-header pipes are heavily shielded to avoid this type of damage. These radiators use a combination of pumped fluid, heat pipes, and conduction to distribute the heat. The tubes and fins provide the radiating surface area. Boeing's Space Power Satellite (SPS) design for a thermal heat engine includes a large radiator with a specific mass of slightly less than 5 kg/m^2 of radiating surface (ref. 1). Somewhat lighter radiators have been developed for small heat loads, but the proposed SPS design probably has the lightest mass achievable for a very large system using this type of technology.

There is another class of heat-rejection devices referred to as moving radiators. These devices distribute the heat over the radiating surface by repeatedly moving the surface past a contact with the heat source. They operate in a manner similar to a disc brake where the rotor is heated by friction at the point of contact with the brake pads and subsequently cooled by convection and radiation. Weatherston and Smith are credited with the first suggestion to use a moving belt radiator in space (ref. 2). The belt surface provides the radiating area, and the heat is supplied to the belt by one or more heated rollers. Their preliminary calculations indicated that the radiator system specific mass equivalent to 2.5 kg/m^2 could be achieved with this concept. This is a significant 65-percent reduction from the Boeing design referenced above. The belt radiator also appears to be a design that could be stored efficiently and subsequently deployed for use in space.

Recently, John Hedgepeth suggested the concept of another type of moving radiator which uses a stream of small particles to radiate the heat (ref. 3). The particles would be captured, reheated, and subsequently rejected at two or more stations working in cooperation with each other. Preliminary calculations conducted at Astro Research Corporation (Astro) indicate that a heat-rejection system specific mass below 1.5 kg/m^2 is potentially achievable with this technique (ref. 4). Subsequently, Mattrick and Hertzberg have pointed out that small liquid droplets can be formed and propelled by pumping a fluid out in narrow streams (ref. 5). The instability of the narrow streams causes them to break up into droplets, as observed in the operation of a shower head. They have also identified a number of liquid metals with very low vapor pressures which can be used with an acceptably small penalty in evaporation losses.

This report summarizes a study by Astro of several moving radiator configurations. Droplet-stream radiators suitable for operation at peak temperatures near 300 and 1000 K have been studied using both freezing and nonfreezing droplets. Moving-belt radiators have also been investigated for operation in both temperature ranges. The potential mass and performance characteristics of both concepts have been estimated on the basis of parametric variations of analytical point designs. These analyses have included all consideration of the equipment required to operate the moving radiator system and have taken into account the mass of fluid lost by evaporation during mission lifetimes.

Preliminary results of this study, presented at a midterm review held at NASA MSFC on 11 December 1980, indicated that the low-temperature droplet-stream radiator appeared to offer the greatest potential for improvement over conventional flat-plate radiators. Since it appeared that this type of radiator would also require the lowest development cost, the decision was made to concentrate on this system in finishing the study and this is reflected in this report.

SECTION 2
DROPLET-STREAM RADIATORS

Mattrick and Hertzberg have shown that the liquid mass required in a radiating droplet stream is very small compared with conventional flat-plate radiator system masses, even when evaporation losses are taken into account. The use of liquid droplets in a particle-stream radiator provides feasible methods of forming the droplets in a stream, collecting them with a centrifuge, and reheating them in a heat exchanger. In order to make a fair comparison with existing systems, the masses of the equipment required to spray and collect the droplets need to be established and added to the fluid mass. As part of this study, we developed a preliminary analytical model of a droplet radiator system with the intention of minimizing the total mass of the system by an appropriate selection of geometry, stream size, droplet size, droplet velocity, etc. Both freezing and nonfreezing droplets have been considered.

2.1 ANALYSIS

A cylindrical geometry was selected for the droplet stream for compatibility with a spinning collector and to minimize the size of the droplet generator, which would appear to be the most dense piece of the equipment. One possible configuration, illustrated in Figure 1, shows two stations working in cooperation with each other. The analysis, which is reported in reference 6, included several important assumptions which are discussed in the following paragraphs.

An average view factor for the droplets has been conservatively estimated by calculating the view factor of space for a droplet in the center of the stream surrounded by a uniform density of equal-diameter droplets. In reality, the droplets near the edge of the stream would cool slightly faster than those on the inside, unless the diameters were deliberately adjusted to compensate for the differences in view factor. External radiation loads on the droplets have not been taken into consideration. Since it is unlikely that the droplets would have a low ratio of solar absorptivity to thermal emissivity, a low temperature stream must be shielded from solar radiation.

The mass of the reheating station, including droplet generation and collection equipment, is assumed to be proportional to the circular cross-section area of the stream. The area density of this equipment is represented by the parameter m_c. The additional pump mass required to form the liquid jets is assumed to be proportional to the two-thirds power of the liquid volume flow rate. A parameter m_p represents this effect. The mass of fluid within the reheating station is assumed to be a fixed fraction β of the mass of the fluid stream itself.

In addition to the above assumptions, the kinetic energy of the droplets is restricted to a small fraction γ of the heat released in each traverse. This is done to limit the pump power required to maintain the droplet stream. In the case of the nonfreezing droplets, the temperature drop had to be limited to some value above the freezing temperature of the material. The resulting expression for the total specific mass of the system, including fluid to make up evaporation losses, for the nonfreezing case is

$$\frac{M_T}{\dot{Q}} = \frac{3\dot{Q}^{1/3}}{2(1-T_2/T_1)^{4/3}} \left\{ \frac{(1+\beta)\ [(T_2/T_1)^{-3} - 1]}{BcVT_1\sigma T_1^4} \right\}^{2/3} \left(\frac{m_c\pi}{18}\right)^{1/3}$$

$$+ \frac{m_p}{[\rho cT_1(1-T_2/T_1)]^{2/3}\ \dot{Q}^{1/3}} + \frac{\eta P(T_1)(T_1/T_v)\tau}{F\sigma T_1^4\ \sqrt{2\pi RT_1}(1-T_2/T_1)}$$

which is obtained by differentiating an earlier expression with respect to the length-to-diameter ratio in order to establish the value of L/D which minimizes the mass of the total system. The parameter B in the above equation depends on the emissivity and view factor as evaluated in reference 6, and τ is the length of the mission. This expression can be used to determine numerically the value of the final temperature T_2 which minimizes the system mass for fixed values of the other parameters and properties of the selected fluid. However, it was observed that as the total heat-rejection rate is increased in numerical examples, the value of L/D, which results in a minimum system mass, becomes very large. If L/D is restricted, a slightly different equation results.

$$\frac{M_T}{\dot{Q}} = \frac{(1+\beta)\sqrt{L/D}\,[(T_2/T_1)^{-3}-1]^{1/2}\,\dot{Q}^{1/2}}{cVT_1\sqrt{3B\sigma T_1^4}\,(1-T_2/T_1)^{3/2}}$$

$$+\frac{\pi m_c[(T_2/T_1)^{-3}-1]}{12B\sigma T_1^4(L/D)(1-T_2/T_1)}+\frac{m_p}{(\rho cT_1)^{2/3}(1-T_2/T_1)^{2/3}\,\dot{Q}^{1/3}}$$

$$+\frac{\eta P(T_1)(T_1/T_v)\tau}{F\sigma T_1^4\sqrt{2\pi RT_1}\,(1-T_2/T_1)}$$

In both of the above equations

$$v = \sqrt{2\gamma c(T_1-T_2)}$$

and the vapor pressure of the fluid had been represented as

$$P(T) = ke^{-T_v/T}$$

The results for a freezing droplet stream are much simpler because of the constant temperature process. For the fixed L/D case, the specific mass can be expressed as

$$\frac{M_T}{\dot{Q}} = \frac{(1+\beta)\sqrt{L/D}\ \dot{Q}^{1/2}}{V\Delta h\sqrt{B\sigma T_1^4}} + \frac{\pi m_c}{4B\sigma T_1^4 (L/D)} + \frac{m_p}{(\rho\Delta h)^{2/3}} + \frac{\eta Q P(T_1)\tau}{\sqrt{2\pi R T_1}\ \sigma T_1^4}$$

where

$$v = \sqrt{2\gamma\Delta h}$$

2.2 NUMERICAL EXAMPLES

Both nonfreezing and freezing droplets at temperatures near 300 and 1000 K, respectively, were considered in our studies. The fluids were selected on the basis of low vapor pressures and appropriate freezing points. Gallium and aluminum with freezing temperatures of 303 and 933 K, respectively, were selected as examples of freezing droplets. Dow 705 silicone oil and tin were selected for use as nonfreezing droplets at low and high temperatures. The properties of these fluids are well known, except for their emissivities, when formed into very small droplets. As a starting point, the emissivities of the metal droplets were assumed to be 0.2, and the emissivity of the silicone was assumed to be 0.7 after pigmentation. It has been suggested in reference 5 that carbon or some other material might be used to darken the liquid metal surfaces up to a value of 0.8, but considering the number of other undemonstrated features of the droplet-stream radiator, this optimistic value was not used in the present examples.

In order to evaluate the numerical examples, it was necessary to select values for several of the parameters. For each fluid, two cases were considered: one with parameters which are believed to be optimistic, and a second case with parameters which are believed to be conservative. Future point design studies will provide a more rational basis for establishing these parameters. Three of the parameters were varied in these two cases with the following values selected for each:

PARAMETER	CASE I	CASE II
β Nonfreezing	0.10	0.20
β Freezing	0.20	0.30
m_c	40.0 kg/m^2	100.0 kg/m^2
m_p	1000.0 $kg/s^{2/3}/m^2$	2500.0 $kg/s^{2/3}/m^2$

The ratio of the kinetic energy to the heat rejected for flight was selected as 0.005 in each case. The L/D ratio was limited to a maximum of 250.

The results of the numerical analysis are shown in Figures 2 through 11. The total heat-rejection rate per stream was varied between 10 kW and 10 GW for each fluid. The minimum specific mass occurred near 100 kW for each example, except for the freezing aluminum droplet system which had a minimum in the region of 1 GW. The characteristics of each of these examples are tabulated in Tables I and II for both the cases described above. The high-temperature examples are strongly dependent on the length of the mission as shown in Figures 2 and 3 because of the large evaporation rates. The results show a distinct advantage for freezing droplet streams when the mission is 10 years long or less. The Case I results indicate that the nonfreezing droplets would require a lighter system if the mission was to extend beyond 15 years. A comparison between both types is shown in Figure 4 for a lifetime of 20 years.

The low temperature results of the examples are only slightly affected by evaporation losses, and, as shown in Figure 5, the nonfreezing droplets of silicone appear to offer a slight advantage over the freezing gallium droplets. If the parameters chosen here prove to be realistic, these results indicate that either system would have a mass that is an order of magnitude smaller than existing systems. This is clearly illustrated in Figure 6 where the specific masses of various droplet stream systems are compared.

2.3 PARAMETRIC VARIATIONS

One of the reasons for an analytical study was to establish the effect of various parameters in a droplet-stream radiator system. The influence of the various parameters on the specific mass of a silicone droplet-stream system is shown in Figures 6 through 11. In Figure 6, the influence of variations of the two mass parameters m_c and m_p are shown. Since the fluid mass is a small part of the system total, variations in β have a much smaller influence on the total mass. The influence of the thermal emissivity ε is shown in Figure 8, where it is evident that the system specific mass is still very much smaller than that of flat-plate radiators even with low values of the droplet thermal emissivity. The effects of limiting L/D are illustrated in Figure 9, and the influence of the limitation on stream kinetic energy is shown in Figure 10. The influence of the temperature level of this type of system is illustrated in Figure 11. The evaporation of the silicone oil droplets becomes significant if the peak temperature is increased beyond 300 K.

The comparison between freezing and nonfreezing droplet-stream radiators shows relatively little difference in performance, particularly in the 300 K examples. Since they both indicate the possibility of substantial improvements over conventional systems, it is our recommendation to pursue further development of the non-freezing system since it appears to be easier to collect liquid droplets rather than solid particles. The relative scarcity of gallium probably makes it an impractical choice of fluid in any case.

2.4 COMPONENT DESIGN CONSIDERATIONS

There are two components of the droplet-stream radiator system which require new technology: the droplet generator and the droplet collector. In contrast, the pumping and internal heat exchanger components can be the same as those which might be used with conventional flat-plate radiators. The following discussion is limited to consideration of systems which would operate at low and moderate temperatures, and at 300 K in particular. High-temperature systems must deal with the corrosive characteristics of liquid metals at those temperatures, a problem which is not discussed here.

2.4.1 Droplet Generator

The basic concept of the droplet generator is identical, in principle, to the shower head that utilizes a large number of small holes to generate a droplet stream. Birch and McCormack (ref. 7) and others have shown that droplets of uniform size and spacing can be generated in this type of device by imposing small mechanical vibrations at an appropriate frequency. Lack of this capability appears to eliminate most other nozzle designs since they do not have the ability to produce a well-defined stream. Even the proposed approach, without the applied vibration, will produce a variety of droplet sizes and velocities. A well-defined stream is essential to this system since small differences in direction, associated with the formation of a variety of droplet sizes, greatly increase the required collector area or result in an unacceptable fluid loss rate.

The low mass of the droplet stream is a direct result of very small droplet diameters which, in the case of the Dow Silicone fluid, have been estimated at 40 μm in Table I. This droplet size requires orifice sizes near 20 μm which is in the range currently used in ink jet printers. Reference 8 discusses the use of sharp edge square orifices, etched in single crystal silicone wafers, with sizes between 15 and 40 μm. Eight of these orifices, on 0.3-mm centers, produced streams of water solution inks that were parallel within ±1 mrad. It is conceivable that many of these thin wafer strips containing 10 to 100 orifices each might be mounted in an array to form a large stream. In order to generate the number of droplets implied in Table I, the array would need to contain approximately 1.4×10^5 orifices.

The combination of small orifices, moderate velocities, and high viscosity associated with the silicone fluids will result in very small Reynolds numbers at the orifices. It can be shown that for Reynolds numbers below 10 the ratio of static head to kinetic energy of the stream at the orifice is approximately given by the equation

$$\frac{2\Delta P}{\rho V^2} = \frac{66}{R_e}$$

where the Reynolds number is based on the orifice diameter. The Reynolds number for the case described in Table I is about 3.5, which means that the pumping power required to generate the stream would be 19 times the rate of kinetic energy generation. If this is multiplied by the value of γ = 0.5 percent (assumed in the analysis), the pumping power would be nearly 10 percent of the rate of heat rejection. This is not acceptable in space systems where typical energy conversion efficiencies are about the same value. Two additional problems are associated with the extremely small orifices: the power required to overcome surface tension effects, and the necessity of filtering the fluid to avoid blockage of the orifices.

In order to investigate an analytic point design that might eliminate the above problems, the parametric equations were used to evaluate a new example which is similar to the previous Case II except that γ was reduced to 0.0005 which is one-tenth the original assumed value. The results of this numerical example are shown in Table III for values of L/D equal to 100 and 250, and total heat rejection rates of 100 kW and 250 kW per stream. The resulting droplet sizes range from 90 to 370 μm in diameter with significant reductions in the required pumping power and moderate increases in the total specific mass. In addition, the number of orifices required is reduced by an order of magnitude.

2.4.2 Droplet Collector

The droplet collector must perform three functions:

- Collect the droplets without splashing
- Coalesce the droplets into a fluid stream
- Develop sufficient hydraulic head to feed a pump

A number of different concepts potentially capable of meeting these requirements have been identified. Mathematical Sciences Northwest, working under an Air Force subcontract with Professor A. Hertzberg at the University of Washington, has identified eight candidate approaches which are shown in Figure 12. Our preference is for a nearly flat spinning conical disc as shown in Figure 13. The disc surface would be coated with a material intended to suppress splashing, and a stationary collector at the rim would combine with the spinning disc to create a centrifugal pump. This pump would be used to feed another, probably a positive displacement device driven by the motor used to spin the disc.

Considerable design and experimental work will be required to develop the droplet collector system. T. Mattick, an associate of A. Hertzberg at the University of Washington, has begun both analysis and experimental work on the problems associated with splashing.

2.5 DEPLOYMENT TECHNIQUES

It appears that the droplet-stream radiators are most suitable for large, relatively quiet spacecraft. If solar power is being used, the large Sun-oriented collection areas probably ensure the appropriate conditions and provide a shield from direct solar illumination of the droplet stream. If, for example, solar cells are being used to generate 200 kW which must ultimately be rejected, the total blanket area will be approximately 2000 m^2. A configuration for the solar array and radiator system might look like the one illustrated in Figure 14 where the array is protected from heating by thin aluminized plastic film reflectors.

It is assumed that the structure supporting the droplet-stream radiator will have other primary purposes. In the above case, it would be used to deploy and support the solar array. The main requirements of the droplet stream system will be for very accurate alignment which might be provided by an active control system. A laser beam and detectors could be used to operate a two-axis positioning system. Thin droplet generator and collector systems of 1-m diameter are easily packaged in the Shuttle cargo bay.

Once deployed, the droplet generator and its matching droplet collector must be coaxial with an error in the angular position of the droplet generator of less than 1 mrad. Increases in this error would require a larger droplet collector with an associated mass penalty. Angular rotation, or translation, rates of the spacecraft structure will also place requirements on the collector diameter. For example, a droplet stream with a length of 210 m and a velocity of 7 m/s will take 30 seconds for the droplets to travel from the generator to the collector. An angular rotation rate of 1×10^{-5} rad/s would cause an error in stream position at the collector of 0.63 m unless the direction of the droplet generator had been adjusted in anticipation of the rotation.

One of A. Hertzberg's students has devised a unique version of the droplet radiator stream which results in a flat spiral as shown in Figure 15 (appeared originally in ref. 9). This approach minimizes the overall dimensions of the system and provides a method of operating a single flat stream. It appears to be less sensitive to angular motions of the spacecraft, although it is equally sensitive to translational movements.

2.6 START-UP AND SERVICING TECHNIQUES

The start-up conditions in a droplet radiator require careful consideration, especially if the fluid is frozen prior to application of the heat load. Whatever the start-up scheme, a frozen or gelled material might initially limit the amount of heat that can be applied. The fluid could be initially contained in an accumulator which surrounds the heat exchanger or its supply piping. The preheated fluid could then be fed to the main pump while isolating the collector with a one-way flow valve. However, depending on the fluid, consideration must be given to the initial low temperatures encountered in the droplet generator and collector. The fluid could be recovered during a shut-down procedure by diverting the pumped fluid from the droplet generator to an accumulator.

A similar approach could be used to provide make-up fluid to the system. Prepressurized containers of replacement fluid could be periodically attached to the system, using appropriate quick-connect fittings, at the same location in the system as the initial fluid resevoir.

2.7 VARIABLE HEAT LOADS

There are several techniques that might be used to handle variations in heat load. If the fluid is not inclined to freeze, the system might be left to change its mean temperature without changing the fluid flow rate. Otherwise, temperature-controlled valves could be used to bypass portions of the droplet-generating system and reduce the number of droplets in the stream as the heat load is reduced.

2.8 NEW TECHNOLOGY REQUIREMENTS

Several advances in present technology are required to reduce the concept of a droplet-stream radiator to practice. They include:

- Development of droplet generator systems capable of producing a large number of directionally accurate streams of droplets with diameters in the range of 50 to 500 μm
- Development of droplet collector systems suitable for operation in the zero-gravity, vacuum environment of space
- Identification of fluids suitable for efficient operation at moderate heat rejection temperatues and establishment of their thermal emissivity properties as small droplets

It appears that these key experiments can be accomplished in vacuum chambers on the ground. If, for example, the droplet collector can be operated against the gravitation field, a great deal of confidence would be developed about its chance of successful operation in space. In order to create the appropriate droplet velocities at the collector, the droplet generator might be placed very close below the collector and pressurized to provide an initial velocity excess.

2.9 OTHER RESEARCH

A. Hertzberg and his associates at the University of Washington are conducting parallel studies on droplet radiator systems with financial support from NASA and the Air Force. They are particularly interested in the use of liquid metals in conjunction with heat engine power systems. A recent report on their activities is contained in ref. 10.

SECTION 3
MOVING-BELT RADIATORS

In order to compare the potential performance of moving-belt radiators with droplet-stream radiators, Astro has considered two types of moving belt designs as previously reported in ref. 11. A design suitable for low-temperature operation, with the belt heated by contact with a drum, is illustrated in Figure 16, and a design suitable for high-temperature use, with the belt heated directly by the fluid, is illustrated in Figure 17. A parametric analysis was completed for both approaches with the mass of the belt, drum, and heat exchanger taken into account. The results of the analysis are briefly summarized below.

3.1 HEATED-DRUM BELT RADIATORS

The concept of heating the belt by contact with the drum is considered suitable for low-temperature operation where a low vapor pressure grease, such as silicone, could be used to enhance the heat transfer rate. Several belt materials were considered, and results are shown in Table IV for belts of aluminum, Kapton, and beryllium each rejecting 100 kW per belt segment with an initial fluid temperature of 300 K. The results indicate that the specific mass can be arbitrarily reduced by using thinner belts as illustrated for Kapton in Figure 18. Moving-belt radiators become more mass efficient, under the assumptions of our analysis, as they become smaller which is shown in Figure 19. Moving-belt radiators might be suitable for Shuttle-related heat rejection systems where the anticipated operational lifetimes are relatively short and the need for supplemental heat rejection is usually in a range between 10 and 100 kW.

3.2 CONVECTION-HEATED BELT RADIATORS

The high-temperature belt design considered the use of direct convection heating, as originally proposed by Weatherston and Smith in ref. 12. This design adds the complication of sealing against the moving belt with the benefit of very high heating rates at the interface between the belt and the fluid. A single design was considered for operation near 1000 K using beryllium for both the belt and the heat exchanger. The results for one example are shown in Table V. The variation in the total specific mass with the peak operating temperature is shown in Figure 20.

3.3 OPERATIONAL CONSIDERATIONS

The moving-belt radiators discussed have very competitive specific masses as compared with droplet-stream radiators. They also are relatively easy to deploy and start up since the belts can be temporarily rolled up for transport in the Shuttle. Initial low temperatures are unlikely to degrade the belt material so long as the storage radius is not too small relative to the material thickness. Moving belts should accommodate variations in heat load without difficulty. The main concerns with these designs are listed in the following section.

3.4 NEW TECHNOLOGY REQUIREMENTS AND KEY EXPERIMENTS

Several technology issues require resolution before moving-belt radiators can be considered for long-term operation in space. The key issues include:

- Heat transfer rates between heated drums and thin belts (Will a grease enhance these rates over the long term?)
- Lifetime of belts (loads on thin belts must be limited to avoid plastic deformation)
- Lifetime of seals on heated drum systems

- Lifetime of seals and fluid loss in direct convection systems

Experimental evidence is needed to resolve these issues and verify potential solutions to the problems.

3.5 WET BELT DESIGN

The material contained in this report was presented at NASA MSFC as part of a final review of the contract. At that presentation, Gene Commer of NASA pointed out that if a silicone grease coating is acceptable for the low-temperature, heated-drum belt design, then direct convection heating of the belt with a silicone fluid should be considered. The seals would not have to operate perfectly since the evaporation losses should be acceptably low, based on the droplet-stream calculations. This suggestion should lead to lower masses than those predicted by the current analysis since the high-temperature example has shown the significance of eliminating the drum mass.

SECTION 4
CONCLUSIONS

Moving radiators have the potential of significantly reducing the mass of heat rejection systems in space. Although significant technical challenges remain to be satisfied by the development of technology, the development of practical hardware is feasible.

Low-temperature droplet-stream radiators, using dyed non-metallic fluids, can be used to radiate large amounts of waste heat from large space facilities. Moving-belt radiators are suitable for use on a smaller scale, radiating as few as 10 kW from Shuttle-related operations. If we assume that appropriate seal technology can be developed, moving-belt radiators may prove to be important for high-temperature systems as well.

Both types of moving radiators require substantial experimental verification and new detailed design of components. Most of the experimental work can be completed in Earth-based vacuum and thermal vacuum chambers. The most challenging requirement is the development of a low-mass droplet collector, including appropriate experimental demonstrations.

The development costs associated with the technology requirements should be moderate. Although it is difficult to predict the rate of technical progress that will be achieved, we estimate that work leading to a demonstration of an engineering model in space should cost less than one million dollars for either system.

Section 2 of this report contains information previously published in ref. 13.

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TABLE I. NONFREEZING LIQUID DROPLET STREAM RADIATORS

PARAMETER	CASE I	CASE II
Dow 705 Silicone Oil - T_1 = 300 K τ = 20 years, $\dot{Q}$ = 100 kW		
Fluid specific mass, $M_F/\dot{Q}$	0.071	0.092 kg/kW
Station specific mass, $M_C/\dot{Q}$	.276	.638
Pump specific mass, $M_p/\dot{Q}$	.093	.258
Evaporation specific mass, $M_E/\dot{Q}$	.027	.031
Total specific mass, $M_T/\dot{Q}$	.467	1.019 kg/kW
Stream length, L	230	224 m
Stream diameter, D	0.92	0.90 m
Length-to-diameter ratio, L/D	250	250
Droplet radius, a	20	26 μm
Temperature at end of stream, T_2	233	243 K
Stream velocity, V	33.2	30.9 m/s
Tin - T_1 = 1000 K, τ = 20 years, $\dot{Q}$ = 100 kW		
Fluid specific mass, $M_F/\dot{Q}$	0.0123	0.0137 kg/kW
Station specific mass, $M_C/\dot{Q}$	.0131	.0298
Pump specific mass, $M_p/\dot{Q}$	.0237	.0609
Evaporation specific mass, $M_E/\dot{Q}$	.0146	.0153
Total specific mass, $M_T/\dot{Q}$	.0637	.1197 kg/kW
Stream length, L	51.0	48.7 m
Stream diameter, D	0.204	0.195 m
Length-to-diameter ratio, L/D	250	250
Droplet radius, a	8.6	9.6 μm
Temperature at end of stream, T_2	510	530 K
Stream velocity, V	35.7	35.0 m/s

TABLE II. FREEZING LIQUID DROPLET STREAM RADIATORS

PARAMETER	CASE I	CASE II
Gallium - T_1 = 303 K, τ = 20 years, $\dot{Q}$ = 10 kW		
Fluid specific mass, $M_F/\dot{Q}$	0.044	0.048 kg/kW
Station specific mass, $M_C/\dot{Q}$	.348	.871
Pump specific mass, $M_p/\dot{Q}$	.076	.189
Evaporation specific mass, $M_E/\dot{Q}$	.0	.0
Total specific mass, $M_T/\dot{Q}$	.468	1.108 kg/kW
Stream length, L	83.3	83.3 m
Stream diameter, D	0.333	0.333 m
Length-to-diameter ratio, L/D	250	250
Droplet radius, a	1.2	1.2 μm
Stream velocity, V	28.3	28.3 m/s
Aluminum - T_1 = 933 K, τ = 20 years, $\dot{Q}$ = 1000 kW		
Fluid specific mass, $M_F/\dot{Q}$	0.0040	0.0043 kg/kW
Station specific mass, $M_C/\dot{Q}$	.0039	.0097
Pump specific mass, $M_p/\dot{Q}$	.0108	.0269
Evaporation specific mass, $M_E/\dot{Q}$	.0618	.0618
Total specific mass, $M_T/\dot{Q}$	.0805	.1027 kg/kW
Stream length, L	76.4	76.4 m
Stream diameter, D	0.306	0.306 m
Length-to-diameter ratio, L/D	250	250
Droplet radius, a	32	32 μm
Stream velocity, V	62.4	62.4 m/s

TABLE III. DOW 705 SILICONE OIL, T_1 = 300 K, τ = 10 YEARS, γ = 0.0005.

PARAMETER	L/D = 100		L/D = 250	
	$\dot{Q}$ = 100 kW	$\dot{Q}$ = 500 kW	$\dot{Q}$ = 100 kW	$\dot{Q}$ = 500 kW
$M_F/\dot{Q}$, kg/kW	0.32	0.58	0.34	0.60
$M_c/\dot{Q}$, kg/kW	1.46	1.58	0.68	0.79
$M_p/\dot{Q}$, kg/kW	0.29	0.15	0.09	0.05
$M_E/\dot{Q}$, kg/kW	0.02	0.02	0.01	0.01
$M_T/\dot{Q}$, kg/kW	2.09	2.32	1.12	1.45
L, m	137	318	234	531
D, m	1.37	3.18	0.94	2.12
T_2, K	252	243	234	231
V, m/s	6.3	6.9	7.4	7.6
a, μm	48	166	91	187
Re_d	1.8	4.8	4.1	8.6
$\frac{2\gamma\Delta\rho}{\rho V^2}$, %	1.8	0.7	0.8	0.4

TABLE IV. PERFORMANCE DATA FOR SEVERAL HEATED-DRUM BELT RADIATOR SYSTEMS.

SYSTEM	I		II		III	
Belt material	Aluminum		Kapton		Beryllium	
Belt density, ρ (kg/m^3)	2830		1420		1854	
Heat capacity, c (W/kg-K)	896		1092		1850	
Belt thickness, t_b (mm)	0.10		0.10		0.10	
Area density, m_b (kg/m^2)	0.283		0.142		0.185	
Drum material	Aluminum		Aluminum		Beryllium	
Mass parameter, B (kg/m^3)	2.83		2.83		3.36	
Heat rejection, $\dot{Q}$ (kW)	100		100		100	
Fluid temperature, T_f (K)	300		300		300	
Thermal emissivity, ε	0.80		0.80		0.80	
	a	b	a	b	a	b
Specific mass, $M_T/\dot{Q}$ (kg/kW)	2.444	2.746	1.605	1.708	2.005	2.157
Belt specific mass, $M_b/\dot{Q}$ (kg/kW)	1.492	1.935	0.940	1.138	1.193	1.437
Drum specific mass, $M_d/\dot{Q}$ (kg/kW)	0.952	0.811	0.665	0.570	0.812	0.720
Mean temperature, $\overline{T}$ (K)	214	238	202	229	203	233
Belt width, D (m)	5.1	11.7	5.8	12.7	5.7	12.3
Belt length, L (m)	102.7	58.5	115.1	63.6	113.6	61.4
Drum diameter, d (m)	1.45	0.89	1.14	0.71	1.16	0.74
Belt velocity, V (m/s)	5.8	4.7	4.7	3.6	4.1	3.4

a) Radiation from two sides, L/D = 20

b) Radiation from one side, L/D = 5

TABLE V. PERFORMANCE DATA FOR A TYPICAL CONVECTION-HEATED BELT RADIATOR.

Belt material	Beryllium
Chamber material	Beryllium
Belt thickness, t_b	0.1 mm
Chamber wall thickness, t_c	2.0 mm
Thermal emissivity	0.8 (one side)
Heating fluid	Sodium
Fluid temperature, T_f	1000 K
Mean belt temperature, $\bar{T}$	963 K
Belt specific mass, $M_b/\dot{Q}$	4.75×10^{-3} kg/kW
Chamber specific mass, $M_c/\dot{Q}$	0.58×10^{-3} kg/kW
Fluid specific mass, $M_f/\dot{Q}$	0.02×10^{-3} kg/kW
Total specific mass, $M_T/\dot{Q}$	5.35×10^{-3} kg/kW
Belt velocity, V	10.7 m/s
Total heat rejection rate, $\dot{Q}$	1000 kW for dimensions below
Belt width, D	3.91 mm
Belt length, L	25.6 m
Chamber height, H	0.13 mm

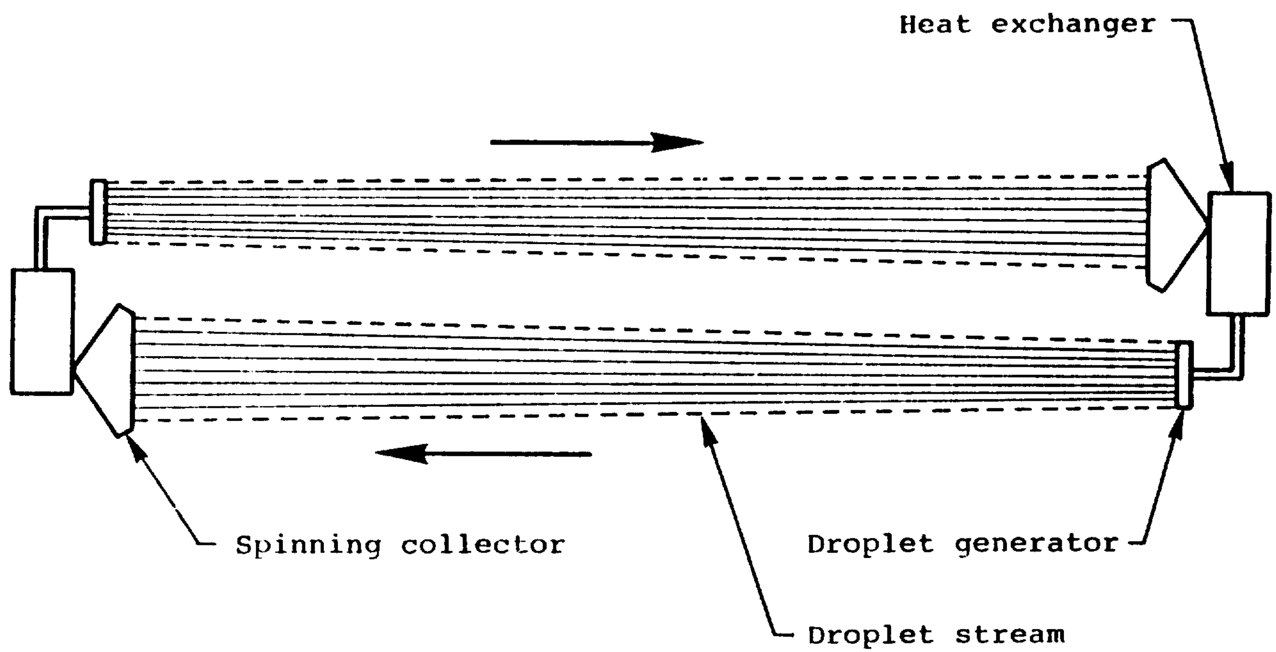


Figure 1. Schematic of a two-station droplet-stream radiator system.

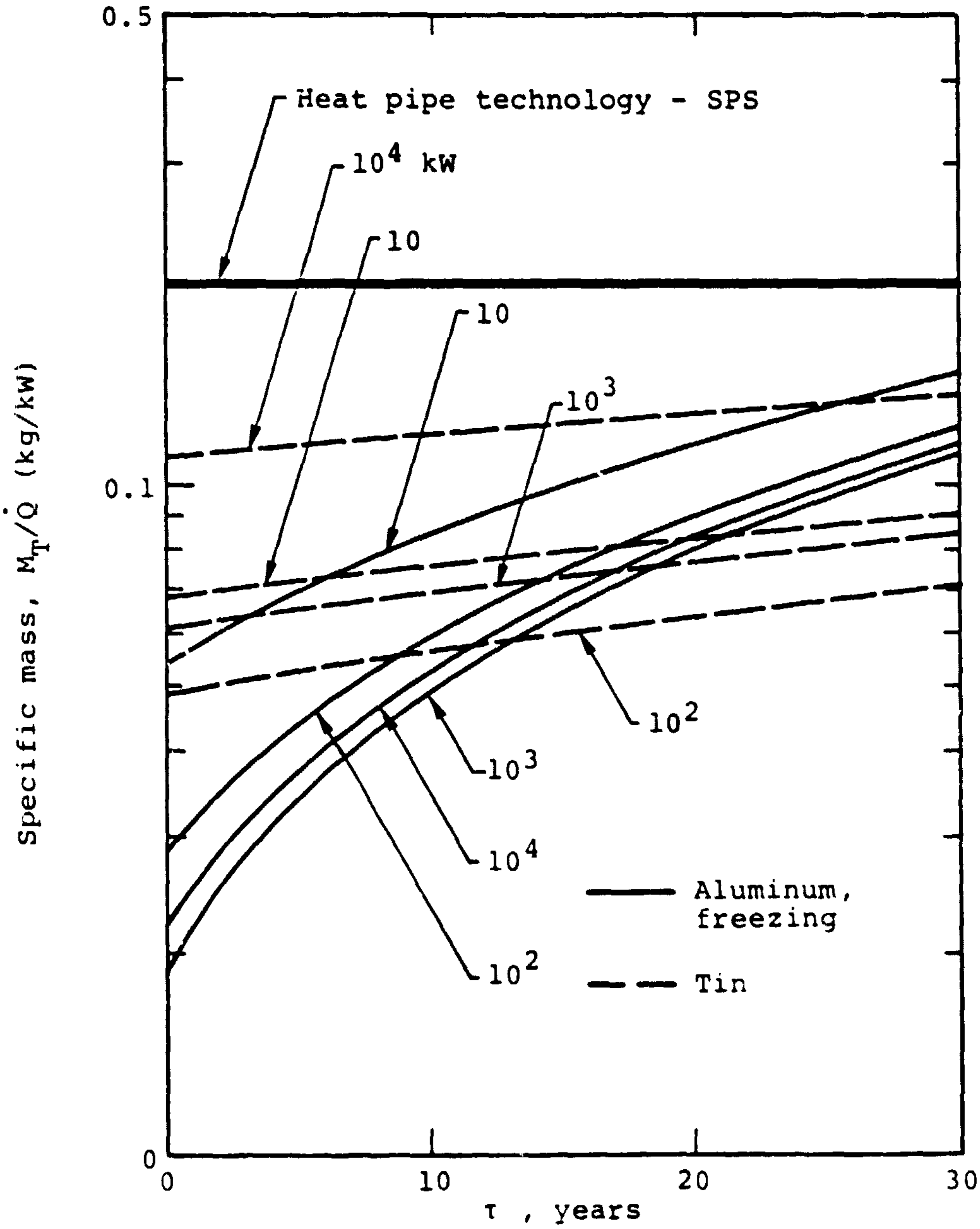


Figure 2. Specific mass of freezing and nonfreezing radiating streams at various heat rejection levels and system lifetimes, T = 1000 K, Case I.

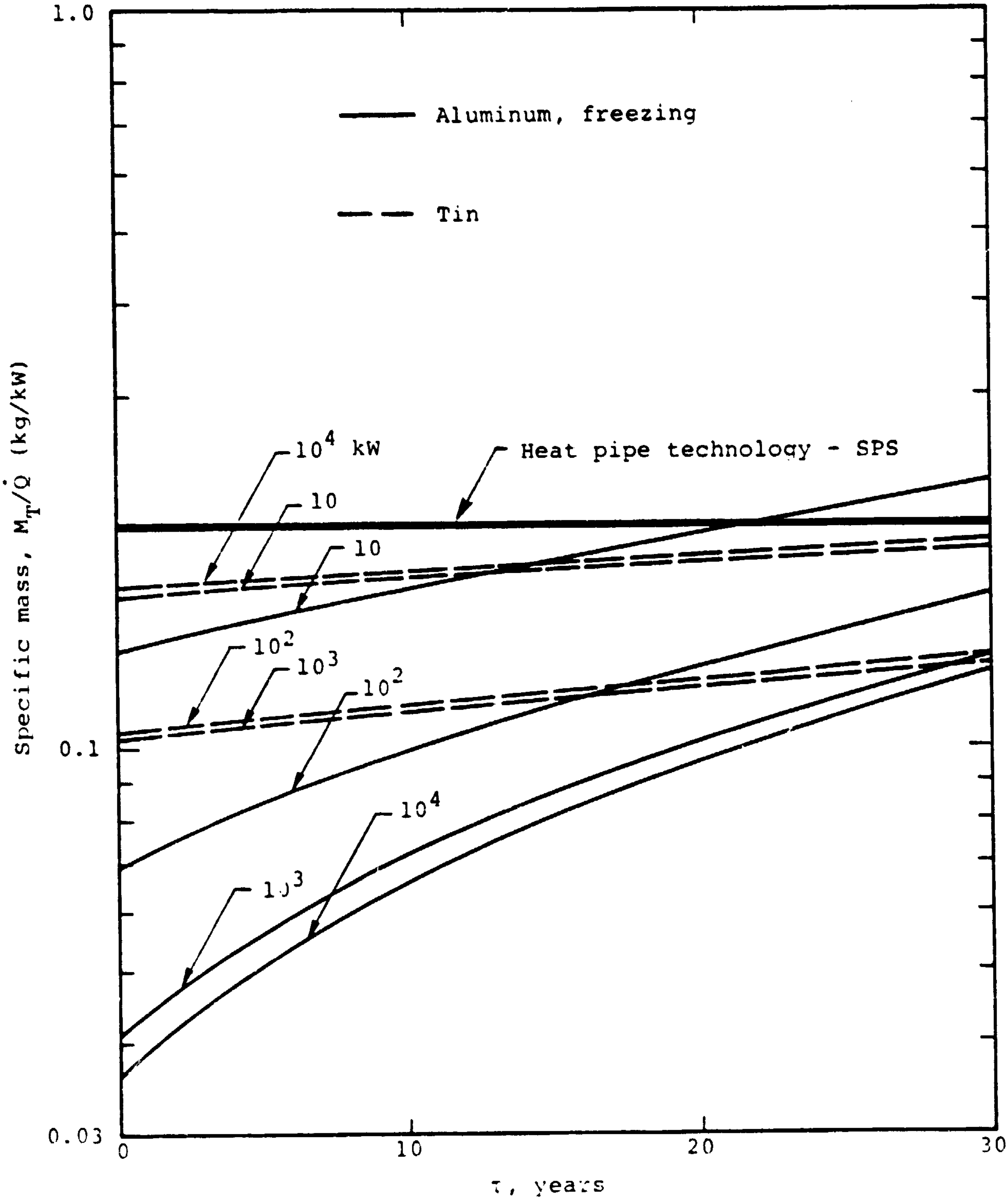


Figure 3. Specific mass of freezing and nonfreezing radiating streams at various heat rejection levels and system lifetimes, T = 1000 K, Case II.

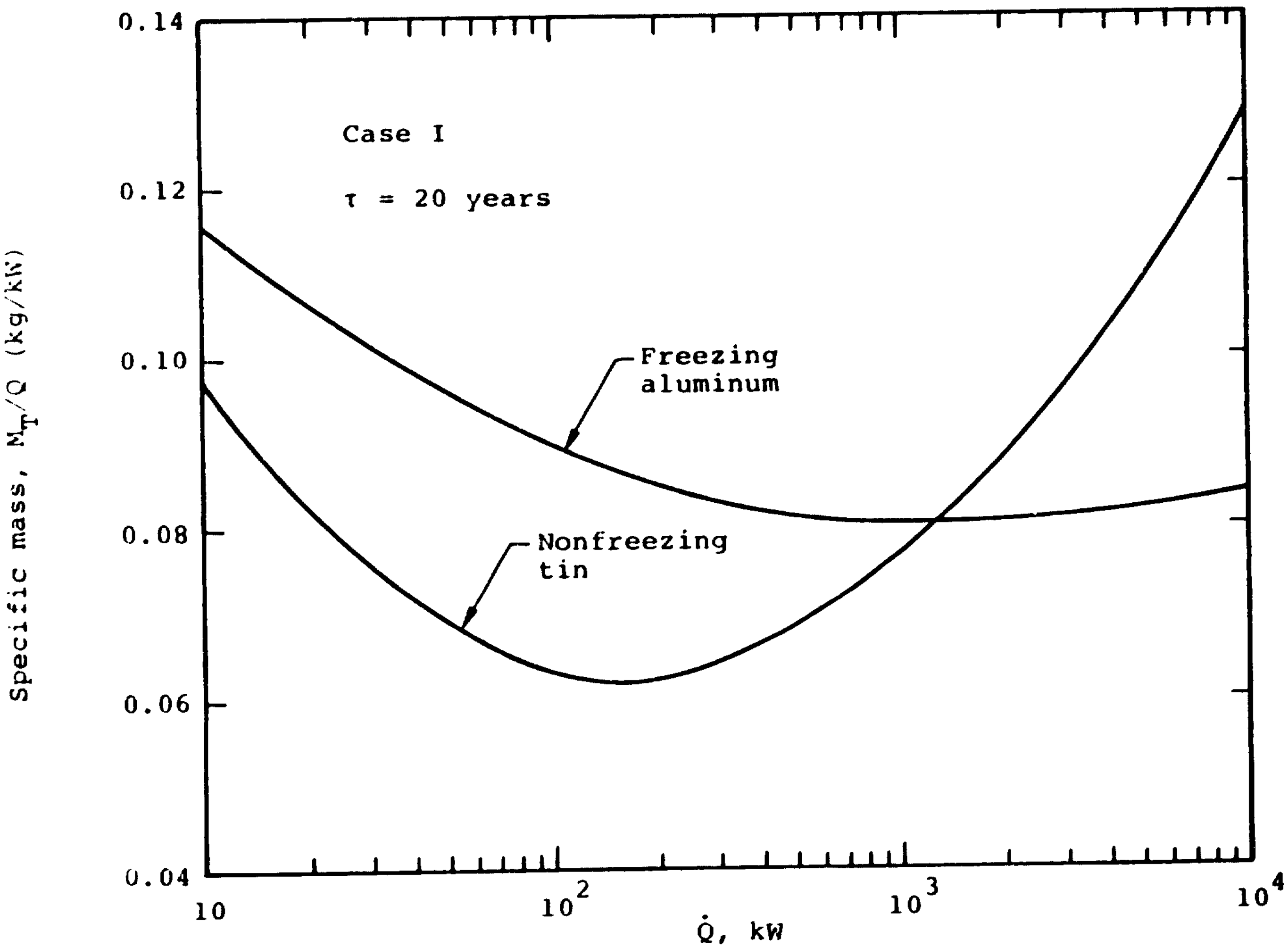


Figure 4. Variation of specific mass of freezing and nonfreezing radiating streams with total heat rejection rate, T = 1000 K.

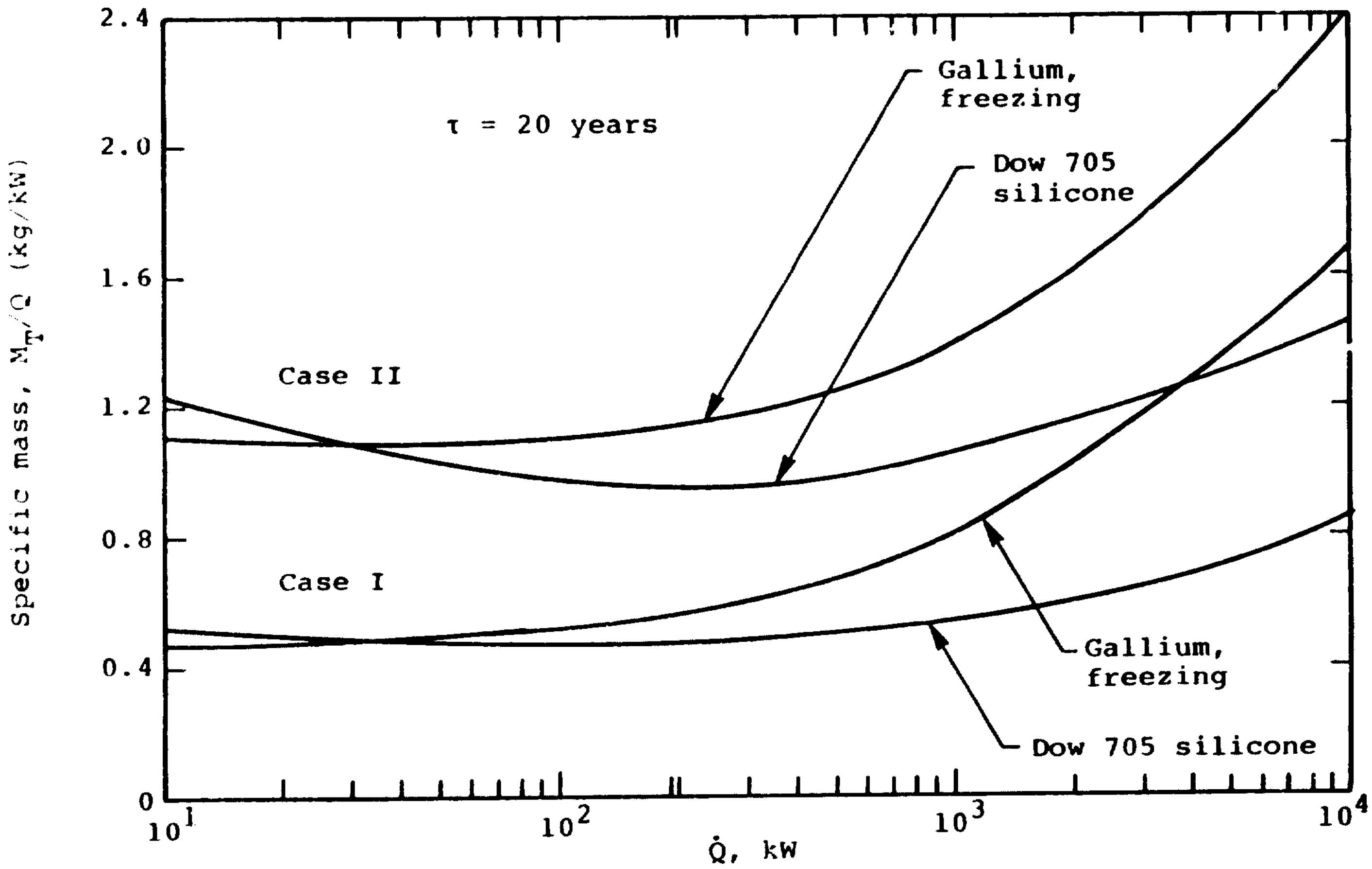


Figure 5. Specific mass of freezing and nonfreezing radiating streams at various heat rejection levels, $T = 300$ K, $\tau = 20$ years.

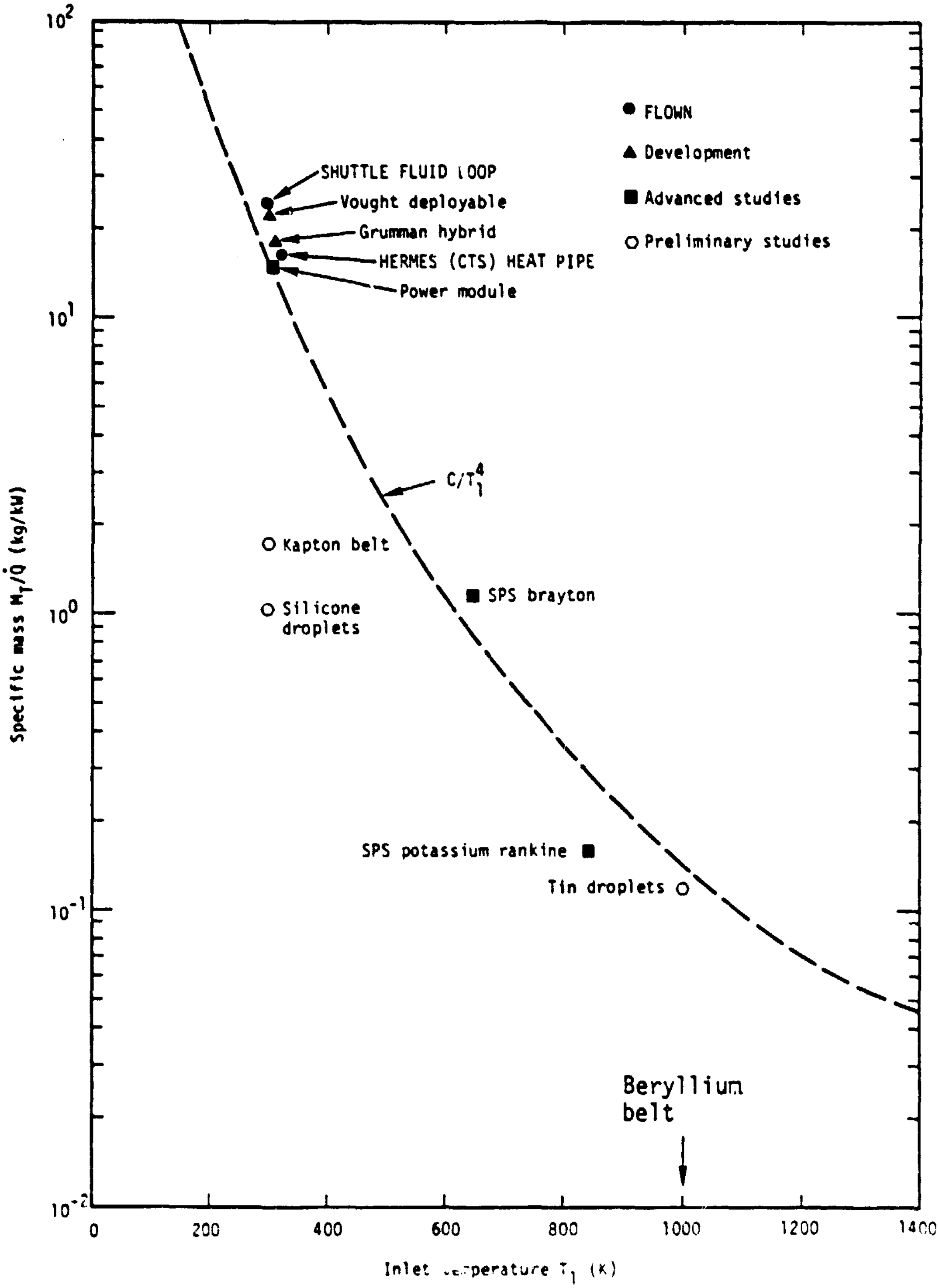


Figure 6. Variation of radiator specific mass with inlet temperature.

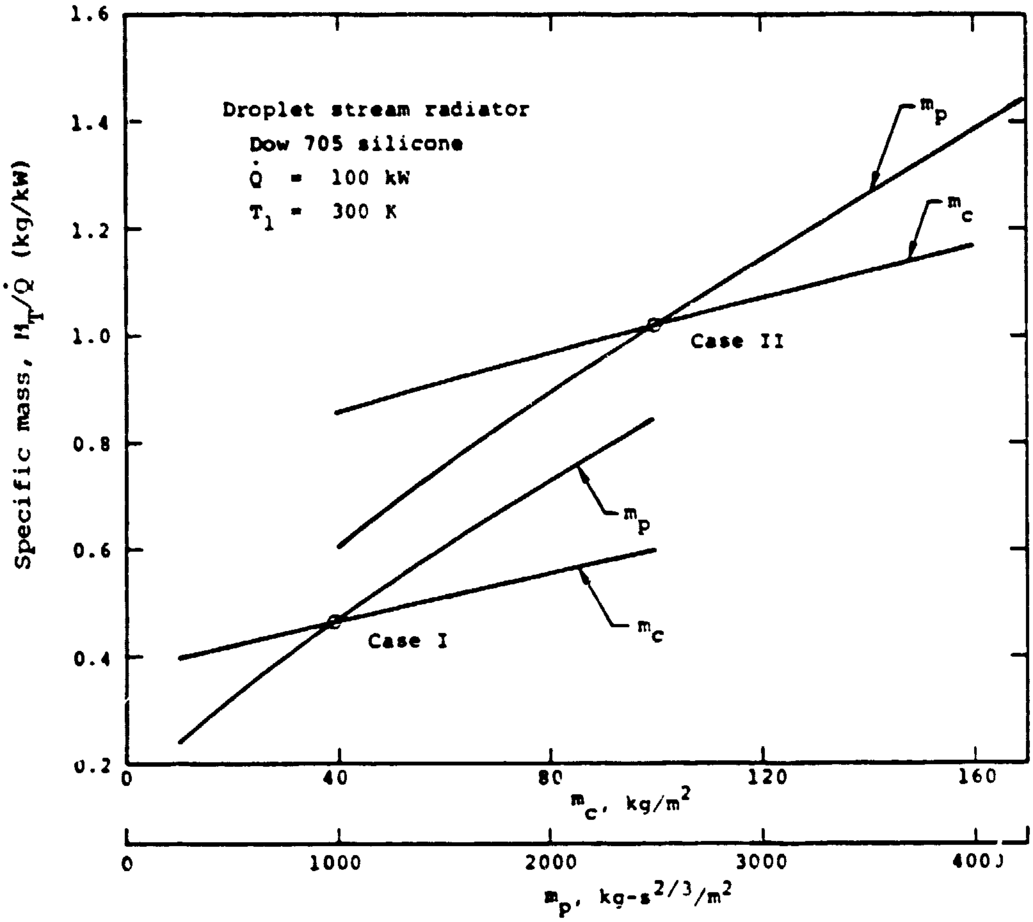


Figure 7. Influence of equipment mass on the specific mass.

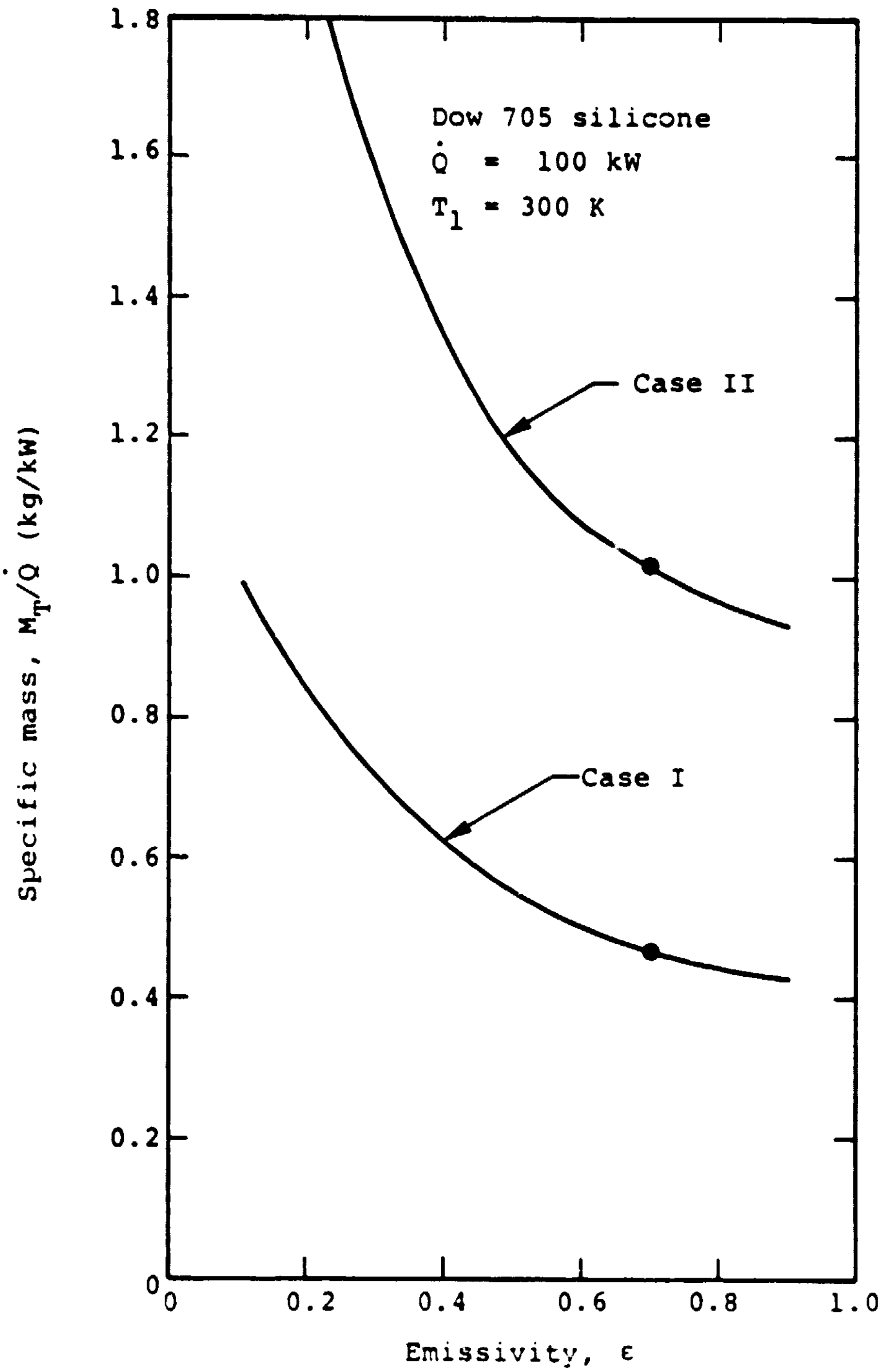


Figure 8. Influence of emissivity on the specific mass of a radiating droplet stream, T = 300 K.

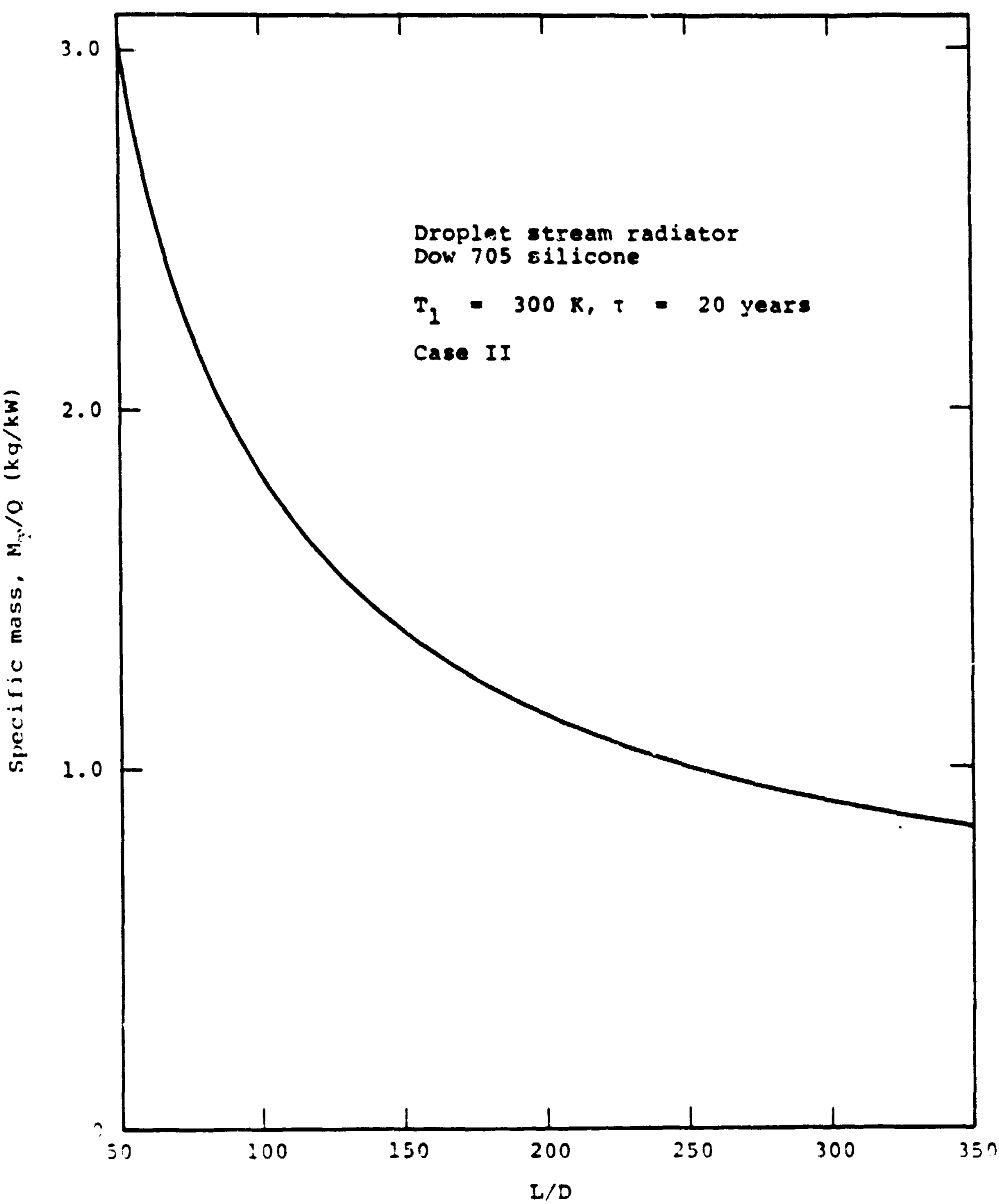


Figure 9. Variation of specific mass with L/D.

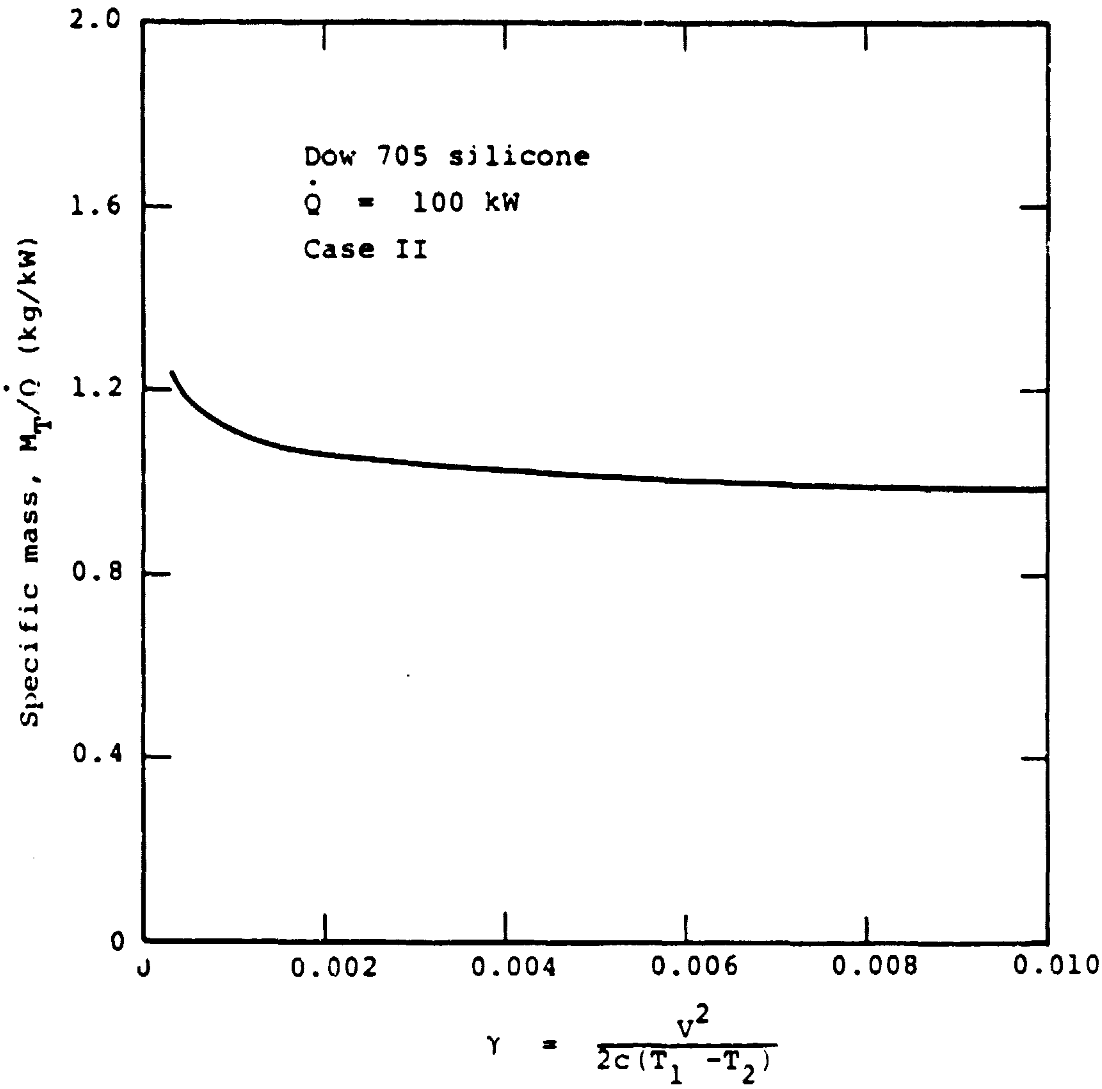


Figure 10. Variation of specific mass with the limitation on stream kinetic energy.

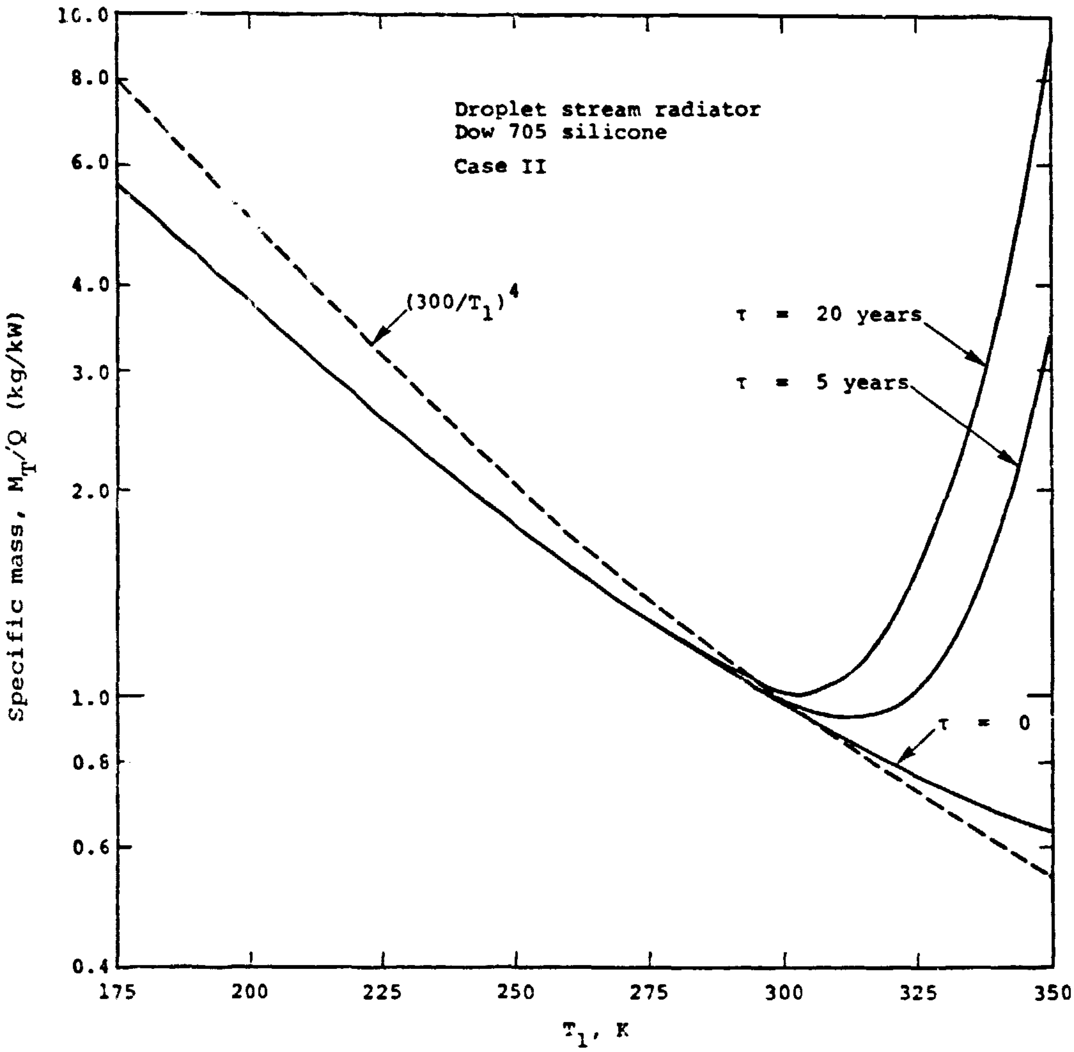
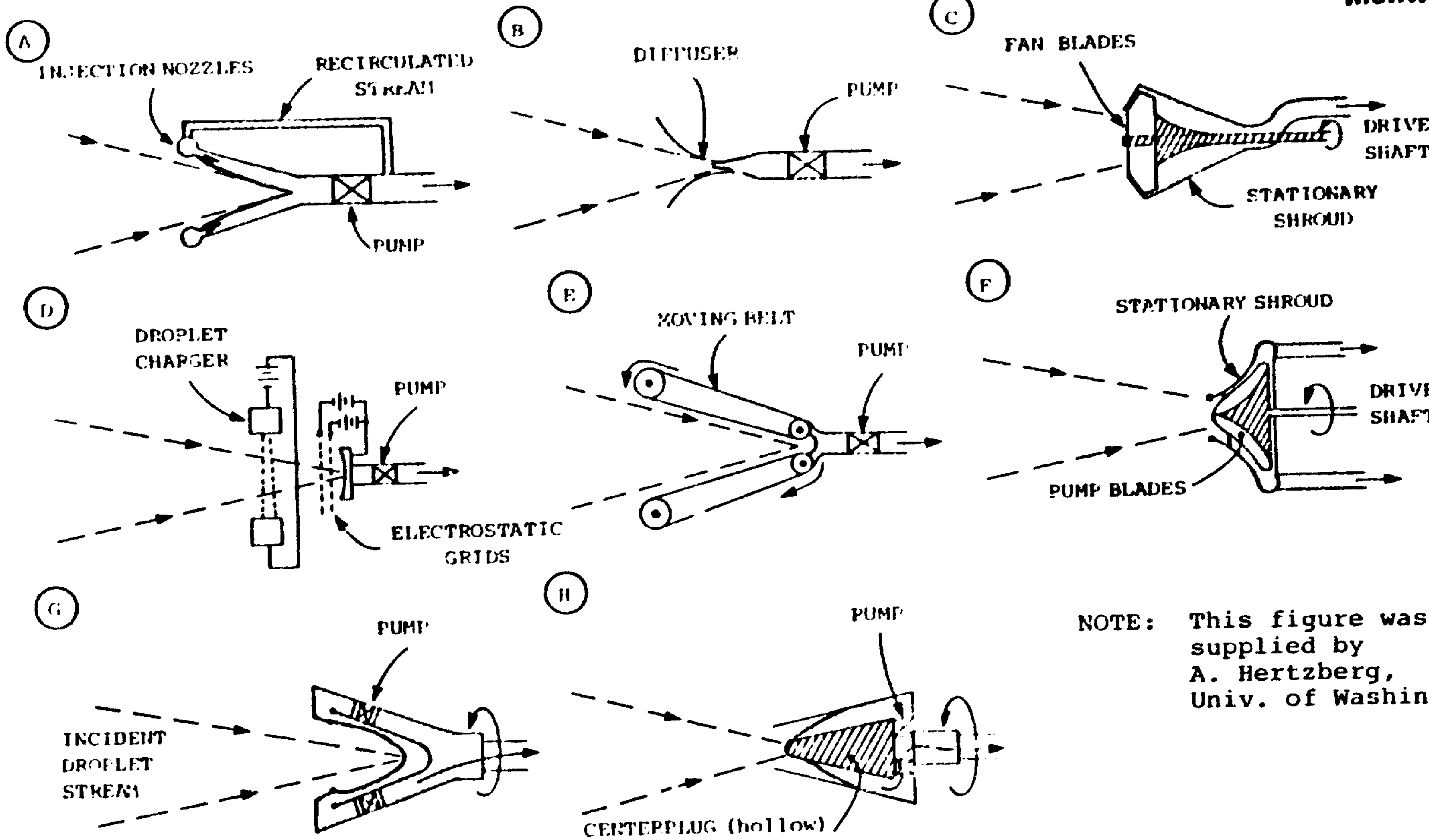


Figure 11. Variation of specific mass with peak temperature.



NOTE: This figure was supplied by A. Hertzberg, Univ. of Washington.

Figure 12. Candidate droplet collectors.

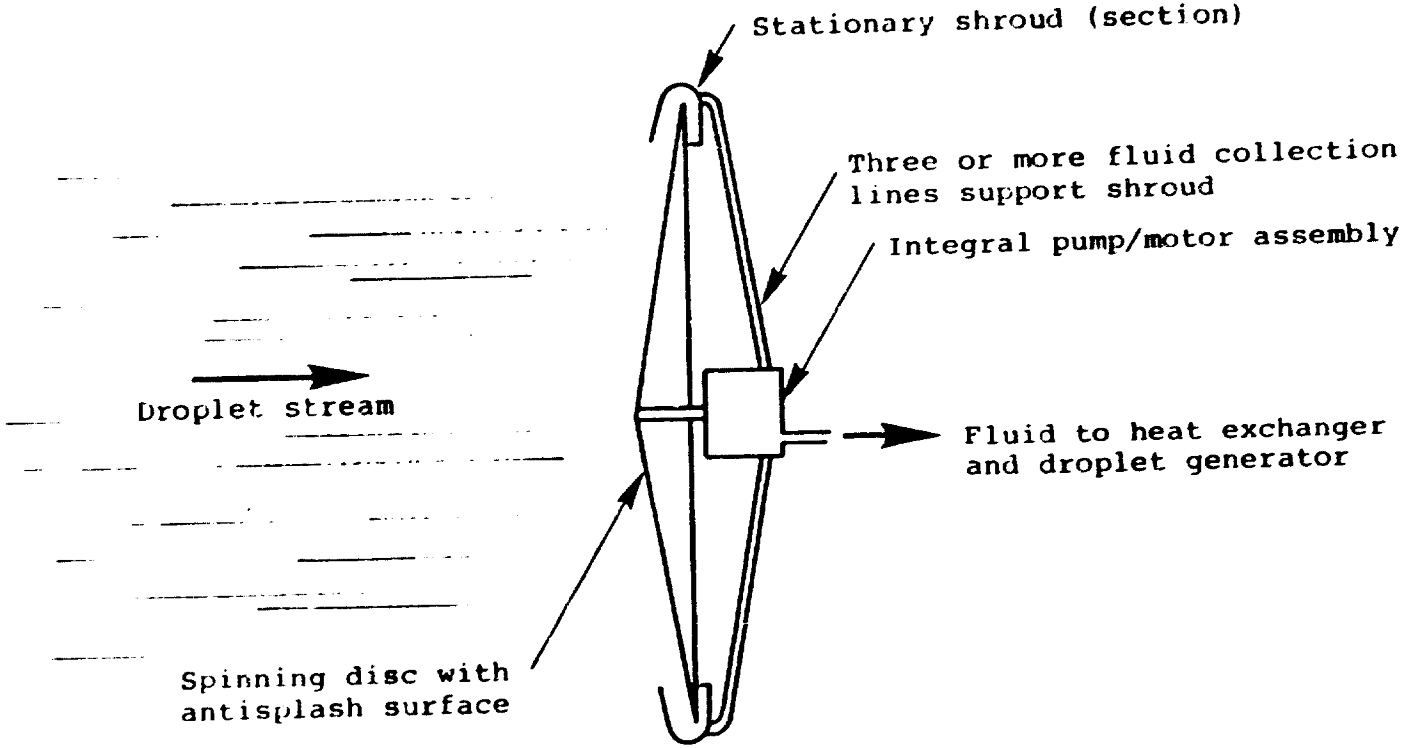


Figure 13. Side view schematic of a low-mass droplet collector.

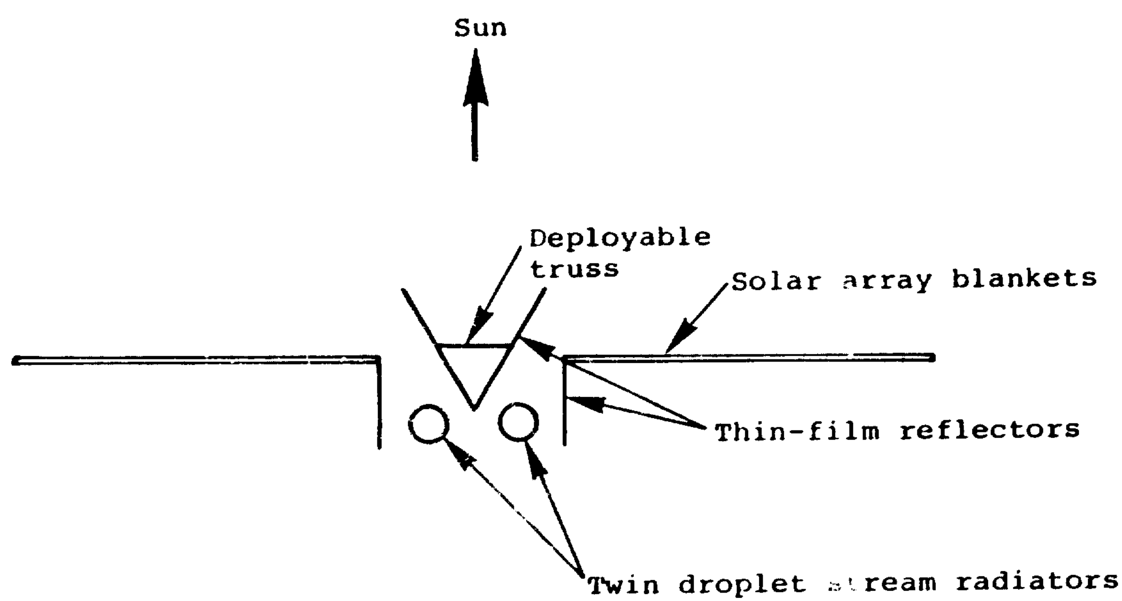
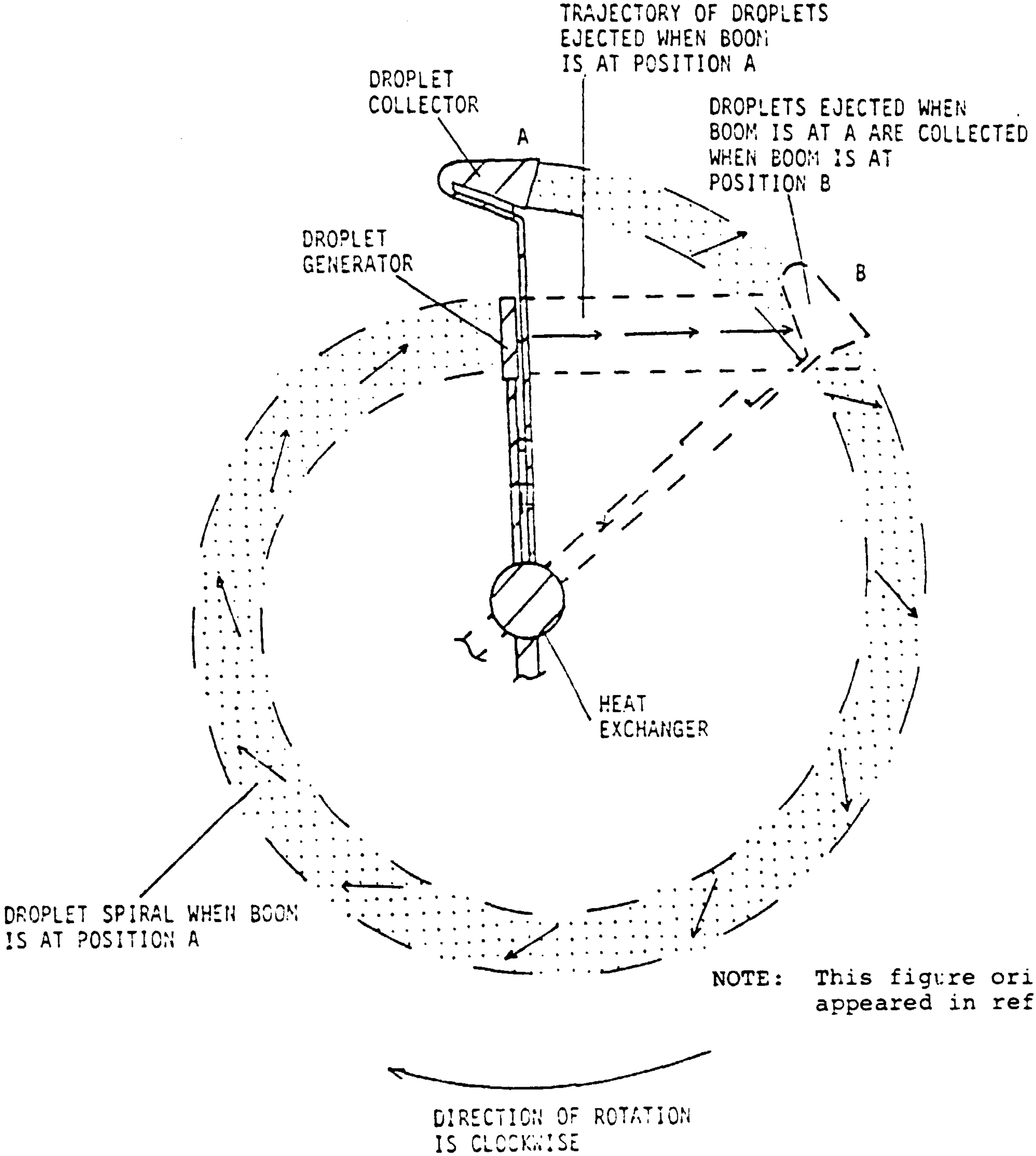


Figure 14. Schematic end view of solar arrays combined with droplet-stream radiators.



NOTE: This figure originally appeared in ref. 9.

Figure 15. Rotating boom droplet radiator.

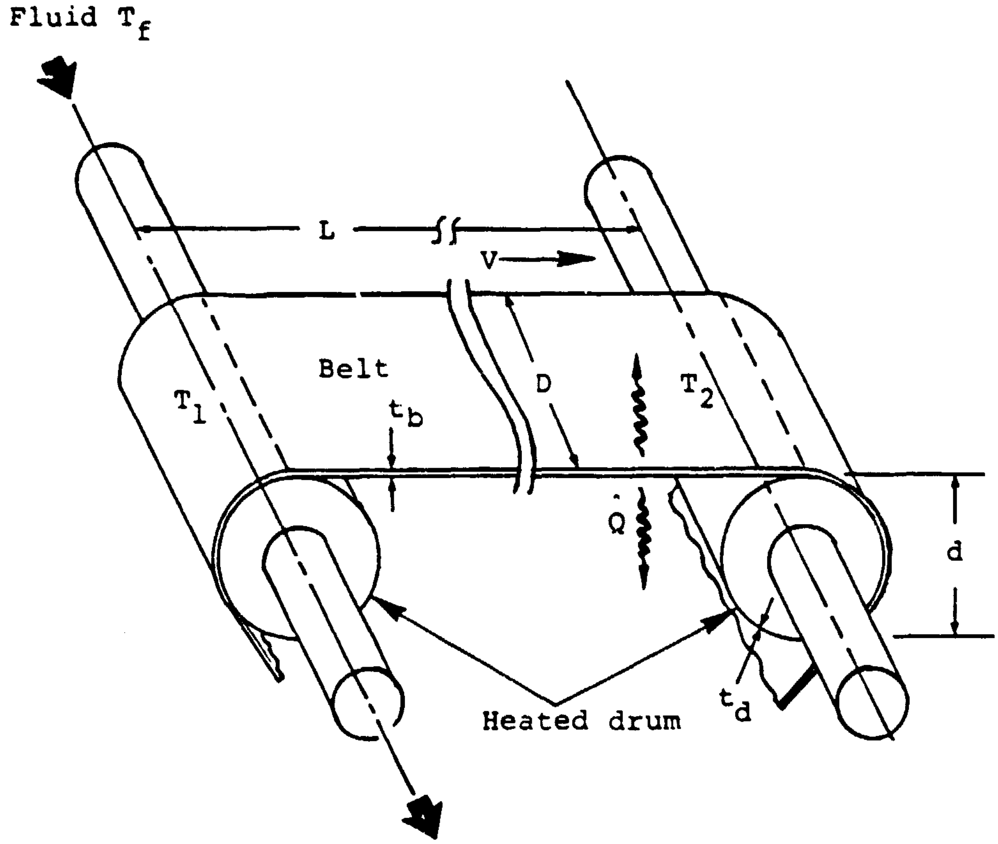


Figure 16. Schematic of a heated-drum belt radiator.

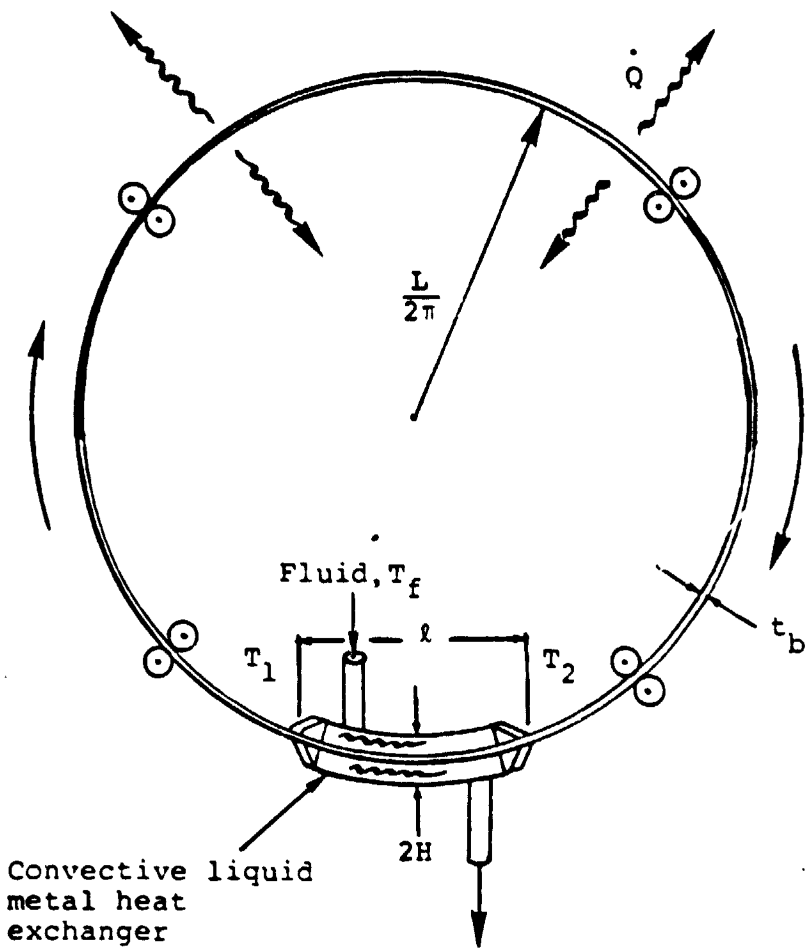


Figure 17. Schematic of a circular convection-heated belt radiator.

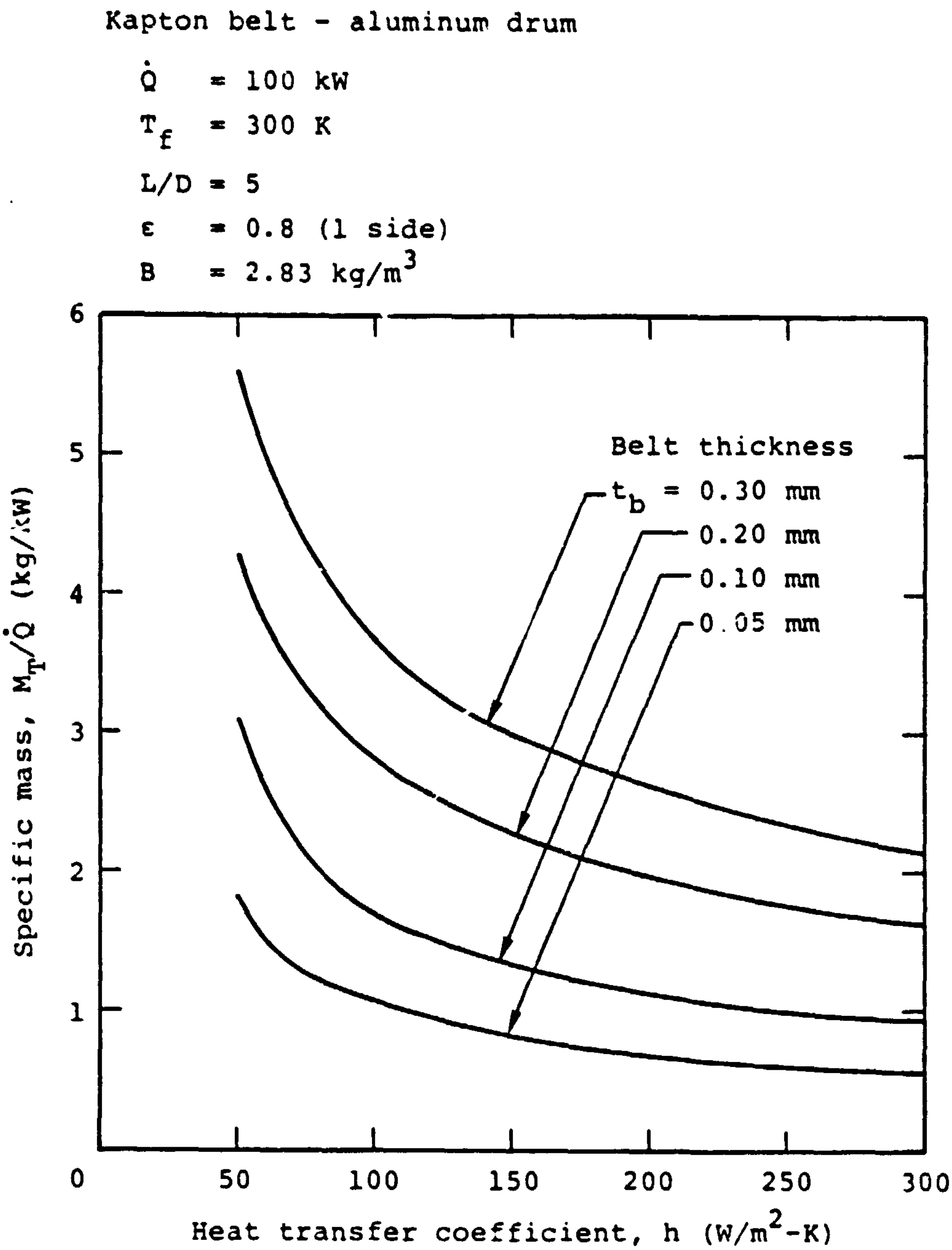


Figure 18. Variation of heated-drum belt radiator specific mass with heat transfer coefficient.

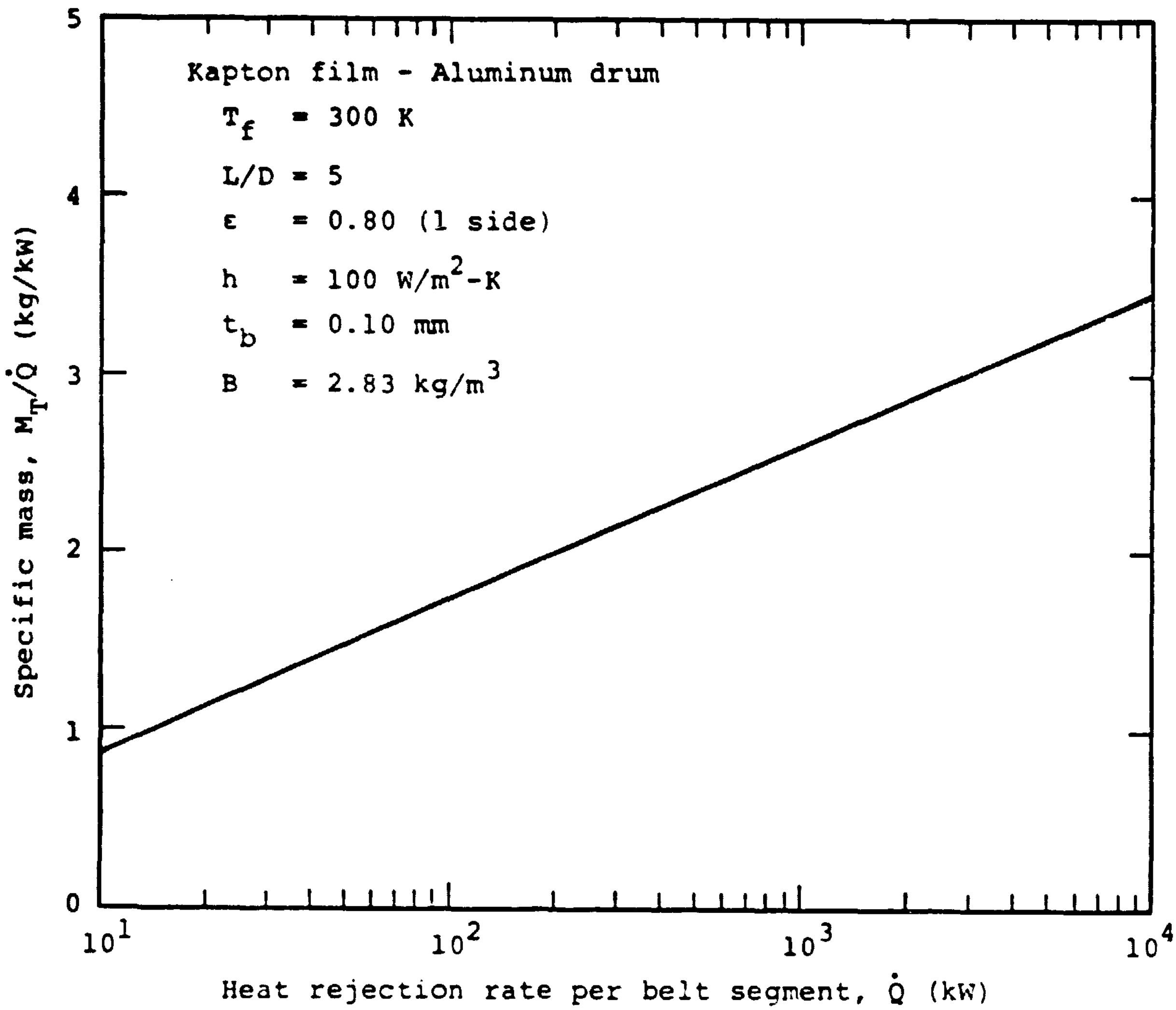


Figure 19. Variation of heated-drum belt radiator specific mass with heat rejection rate.

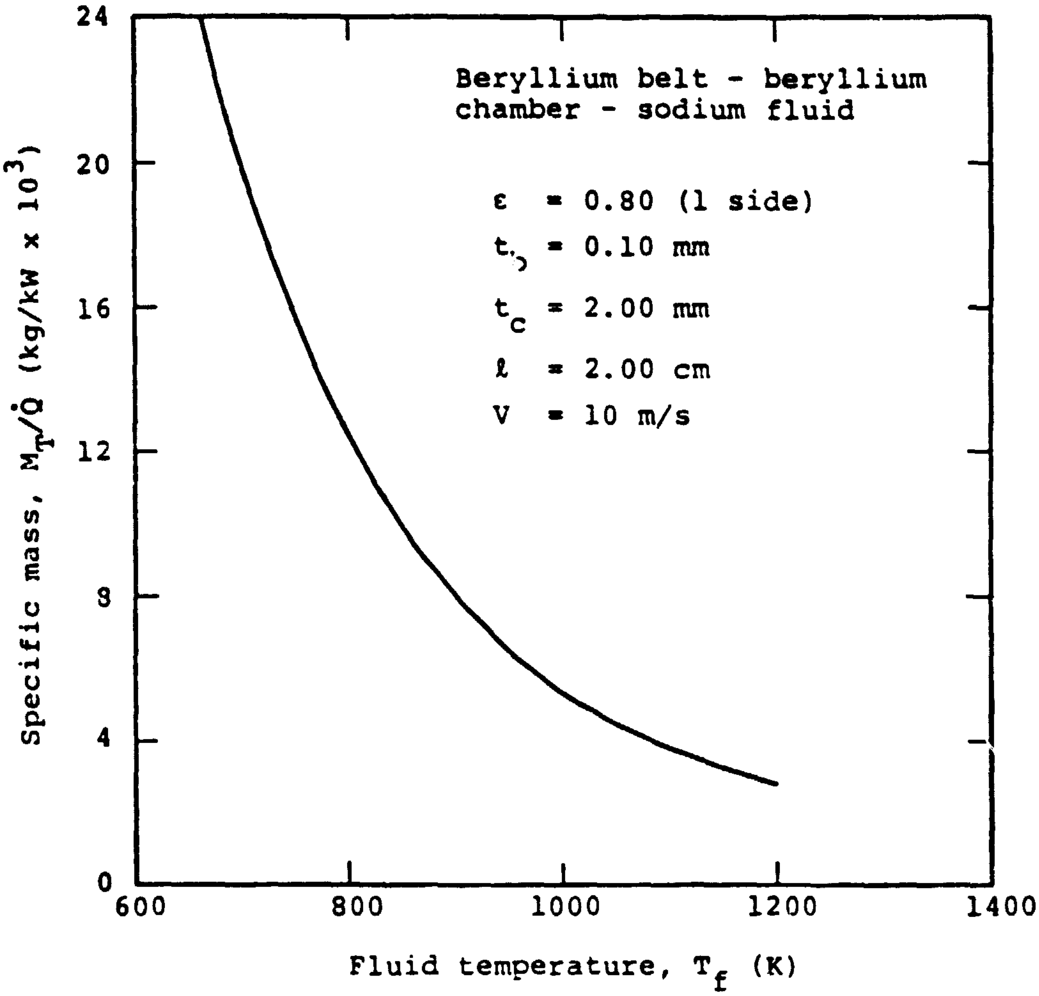


Figure 20. Variation of convection-heated belt radiator specific mass with fluid temperature.