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SOME CONSIDERATIONS AND PRELIMINARY EXPERIMENTS ON AN AIR-CYCLE SYSTEM FOR REFRIGERATION AND PRODUCTION OF DROPS IN CONNECTION WITH AN ICING WIND TUNNEL

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ABSTRACT

A study was made of the possibility and feasibility of an air cycle for refrigerating an icing wind tunnel. This study was influenced by the assumption that Government surplus equipment could be utilized. The relative merits of various cycles and arrangements are evaluated and discussed. The selection of a particular cycle for any given design must be intimately geared to the flexibility required and, of course, the expense. The low-pressure open air cycle analyzed would afford considerable financial saving, although some flexibility would be sacrificed.

A preliminary experimental study of an expansion turbine indicates that the droplets formed are substantially smaller than those normally encountered in natural clouds and artifically produced sprays. Ice accretions formed on tunnel models do not resemble the type usually experienced.

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INTRODUCTION

The problems associated with an icing wind tunnel are many and varied.[1,2] Refrigeration equipment is essential if the tunnel is to be operated in icing conditions for the entire year. In a closed cycle system, where a conventional heat exchanger is used as the refrigeration medium, some difficulty is experienced in that ice particles from the cooling coils can conceivably "seed" the supercooled cloud. In this case the cloud will transform to ice particles at a fairly rapid rate. Another disadvantage of the heat exchanger is its high initial cost. The above considerations indicate the judiciousness of analyzing various air cycles, where most of the components could be drawn from government surplus. For instance, turbo-superchargers, turbo-jet engines, and outdated reciprocating engines could be utilized.

The University of Michigan icing wind tunnel has no provision for refrigeration and is quite limited on the icing runs possible. With the above thoughts in mind an area of research was initiated to study experimentally the use of a simple air-cycle system in connection with the tunnel. The components drawn from surplus are limited in capacity and thus, operating conditions are confined to modest tunnel velocities. However, valuable insight can be gained into the refrigerating capabilities of expansion machinery, as well as the mechanism of drop formation in a rapid expansion machine.

In view of the above, this report is separated into two parts. In the first part the results of the air-cycle analysis are presented and discussed. The second part is concerned with the experimental study of the expansion turbine and its refrigeration and drop-making characteristics.

PART I. REFRIGERATING CYCLES FOR USE IN AN ICING WIND TUNNEL

General Considerations

The problem to be considered is that of cooling a mass of air to a subfreezing temperature with an appropriate liquid water content. The magnitude of this task is dependent on the size of the icing tunnel, as well as the operating conditions. On a temperature entropy diagram, Fig. 1, the problem is to go from State A to State B, where the initial temperature, T_A, is reduced to a subfreezing temperature, T_B. The most direct path is, of course, along the constant pressure path, P_1. This process corresponds to the conventional vapor-compression refrigeration system. Point B may also be attained by the process, A-C-D-B; where path A-C represents an isentropic compression to P_2, C-D a constant-pressure heat exchange, and D-B an isentropic expansion back to the original pressure. The heat-exchanger effectiveness is now higher because of the greater temperature level, but the inefficiencies of compression and expansion are not indicated. The effect of these inefficiencies is to impose a greater load on the heat exchanger if the same final temperature is to be achieved.

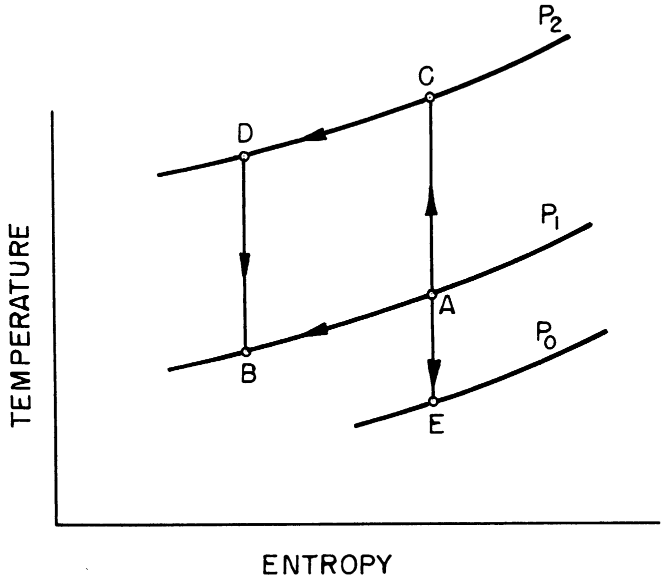


Figure 1. Temperature - Entropy Diagram.

Another possibility is to expand initially to a subatmospheric pressure P_O (Point E) and then test before compressing back to the discharge atmospheric pressure. This process eliminates the need for a heat exchanger. The ambient conditions are of importance in an open cycle system; that is, where the tunnel discharge air is not ducted back into the inlet. Accordingly, Fig. 2 is included to show the annual variation of temperature and humidity. This is based on a 7 year mean with the humidity taken at 1:30 p.m. EST.[3] From the curve it is evident that only about 2-1/2 months out of the year have average temperatures below freezing. Furthermore, the absolute humidity is too high to allow for refrigeration without water removal. Superimposed on the refrigeration requirements are the problems pertinent to an icing wind tunnel, in addition to those of any subsonic wind tunnel. These problems are:[1,2]

(a) A supercooled cloud must be formed which adequately represents a natural cloud as to drop size, distribution, and liquid water content. This implies that freeze-out does not occur.

(b) The cloud should be distributed uniformly throughout the test section.

(c) The humidity of the air at the test section should correspond to saturation.

The particular design criteria considered in this cycle analysis are as follows:

(a) A test section area of 4 ft^2.

(b) A maximum velocity of 500 ft/sec.

(c) The total temperature should be down to 0°F.

(d) The liquid-water contents should be up to 2 gm/m^3.

(e) Government surplus equipment should be utilized where possible.

(f) The tunnel should be flexible and relatively simple and inexpensive to operate.

(g) The tunnel should be capable of operating continuously for at least 2 hours in icing conditions.

(h) Turbine efficiencies are assumed to be 80 percent while compressor efficiencies are taken as 75 percent.

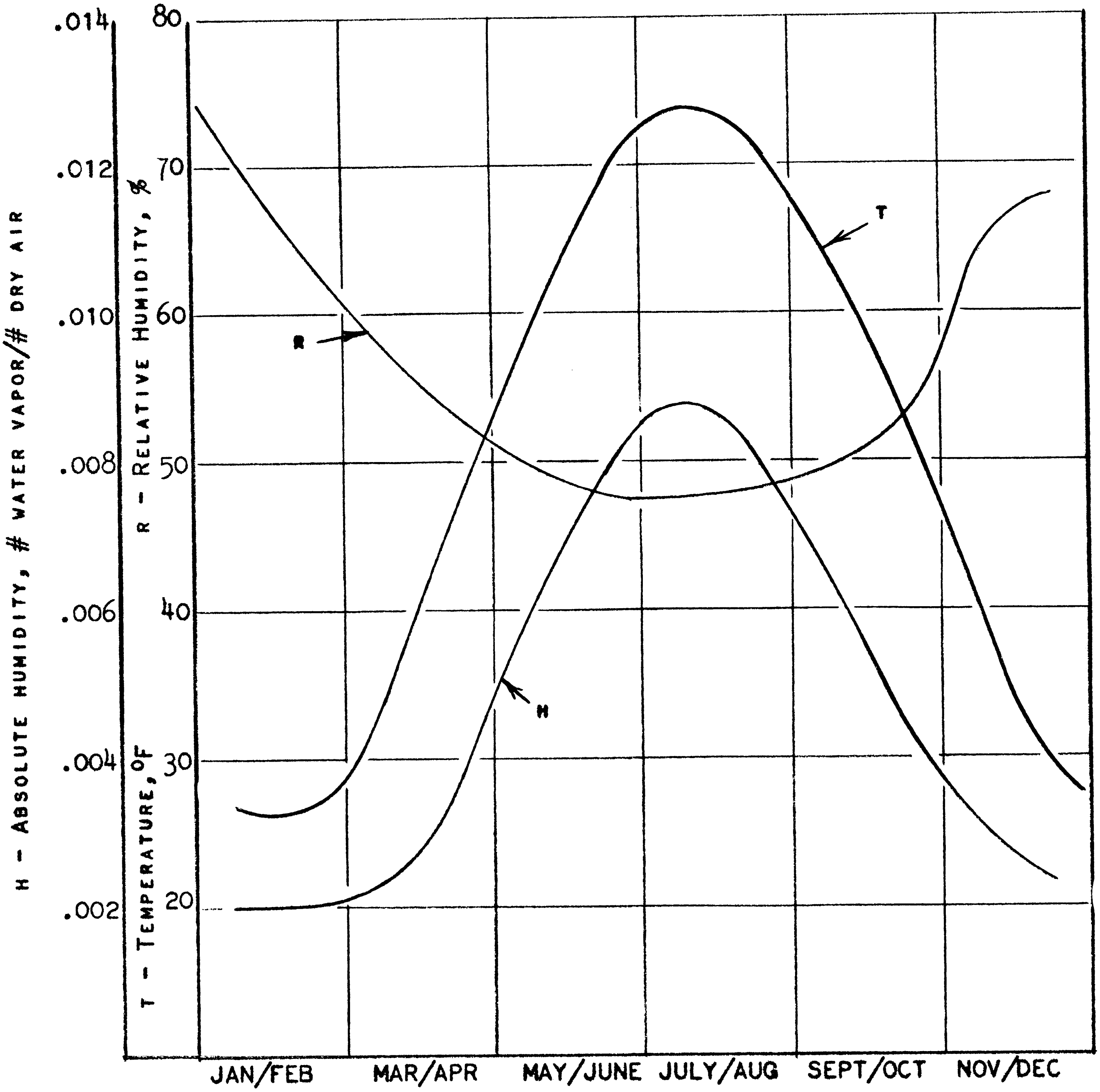


FIGURE 2 - LOCAL CLIMATOLOGICAL DATA (YPSILANTI, MICHIGAN; DATA BASED ON SEVEN-YEAR MEAN)

In the following sections the various ways of refrigerating the assumed tunnel are considered and evaluated.

Vapor Compression System

As mentioned earlier, the straightforward method of providing refrigerated air for icing wind tunnel purposes is to pass the air over the evaporator coils of a conventional vapor-compression refrigeration system. This amounts to a simple heat exchange from the air to the working medium of the vapor system.

Although these vapor systems have been applied to icing tunnels, they are not without disadvantages. Mass flows required for high-subsonic tunnel velocities dictate a costly vapor-refrigeration unit and more economical methods of refrigeration might be sought. This is especially true if government surplus equipment can be utilized.

There is also a possibility that in this system ice accretion on the evaporator coils may seed the supercooled icing cloud and freeze-out may occur if there is sufficient distance between the evaporator coils and the tunnel test section. This effect can be avoided by appropriate control of the dew-point of the stream, which, of course, implies high dryer capacity and the associated expense.

Air Cycle Refrigeration

The air cycle is seldom used commercially for refrigeration because of its low coefficient of performance. That is, the energy realized in refrigeration is small compared to the input energy. However, for icing tunnel work, where the use is intermittent and where the components may be procured cheaply, this system offers certain advantages.

The air cycle can be operated as a closed system or as an open system. In the former, the air is continuously recycled and except for second order effects the system is independent of ambient conditions. The open system consists of refrigerating the ambient air and then discharging to the atmosphere after passing through the tunnel. Both methods of operation are considered and evaluated. In making these analyses certain symbols which are defined in Fig. 3, will be used to represent the various components.

The first cycle considered is a low-pressure open air cycle (Fig. 4). In this system the excess water vapor is first removed from the air, then the air expands across a turbine which drops the temperature. The outlet temperature depends on the pressure ratio, the inlet temperature, and the assumed

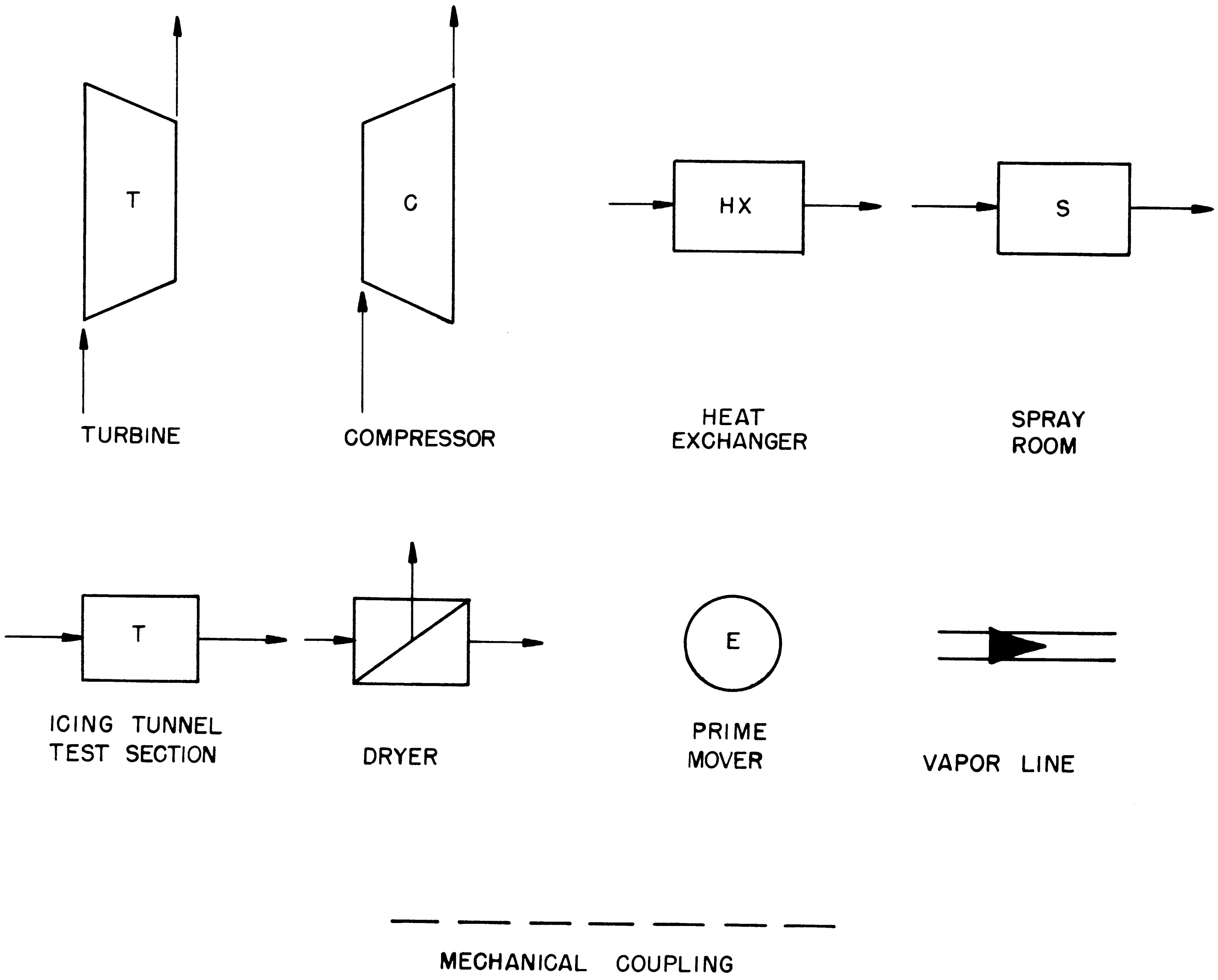
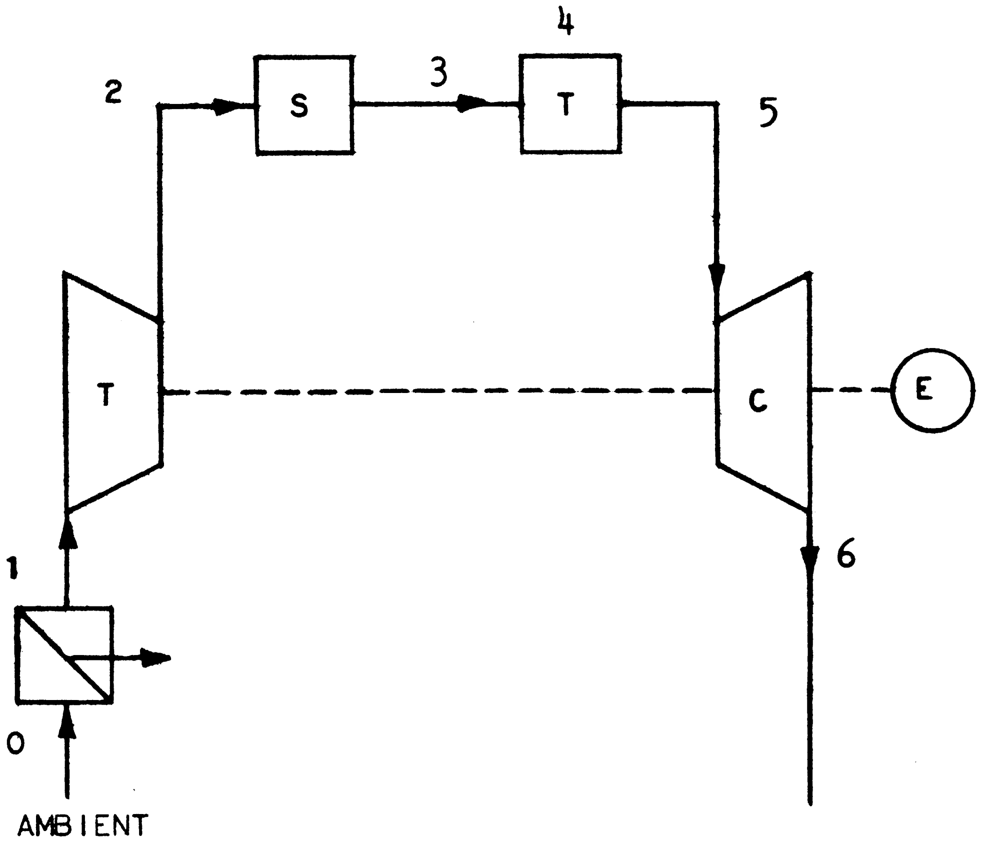


Figure 3. Air-Cycle Components



STATION	STATIC PRESSURE PSIA	STATIC TEMPERATURE °R	WATER-VAPOR CONTENT # VAPOR / # AIR	LIQUID-WATER CONTENT # WATER / # AIR
0	14.3	510	.004	0
1	14.3	510	.0001	0
2	9.54	460	.0001	0
3	9.54	460		.0025
4	8.04	438		.0025
5	9.1	460		.0025
6	14.3	547		.0025

FIGURE 4 - CYCLE #1, LOW PRESSURE, OPEN

turbine efficiency. The latter is taken to be 80 percent and is defined as the ratio of the actual-enthalpy change to the isentropic-enthalpy change across the turbine. The spray room serves as a settling chamber for the tunnel and allows for the icing cloud formation. This cloud may be formed directly by condensation of the water vapor through the turbine or by spray nozzles. From the air-cycle standpoint the more stringent design condition is that where the air must be sufficiently dried so that the liquid water content of the cloud is governed solely by the spray nozzles. Accordingly, this design condition will be incorporated, and the spray room is assumed to be at 0°F.

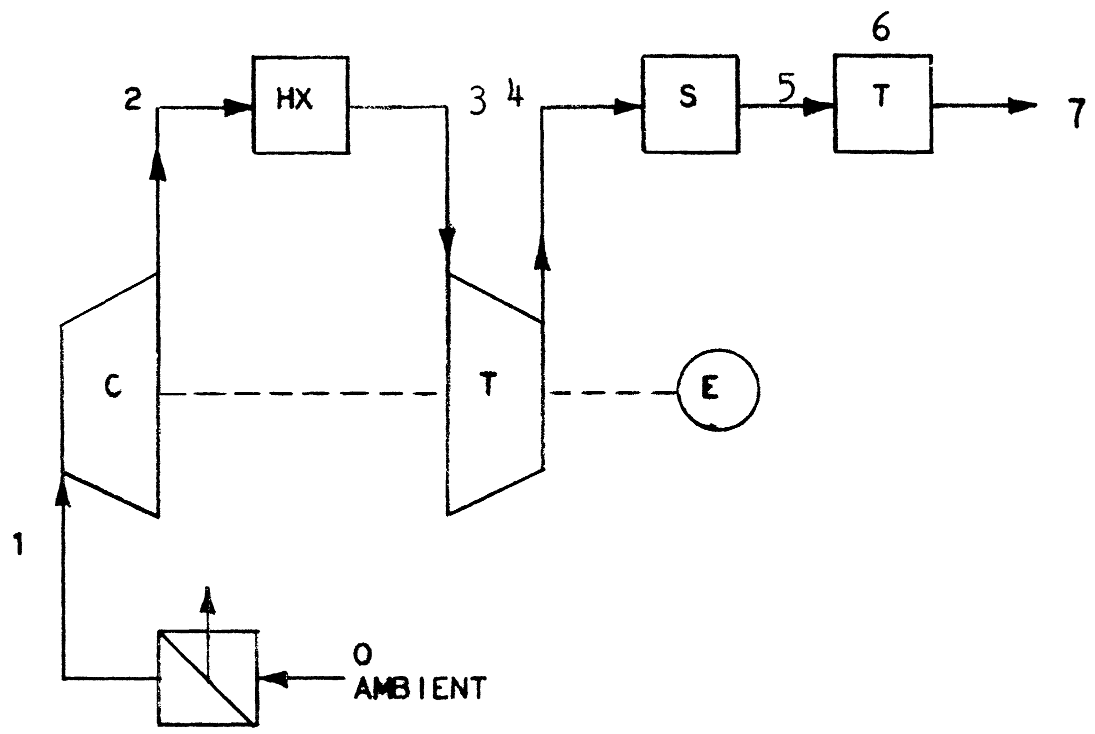
The tunnel test-section conditions have been previously described as an area of 4 sq ft with a velocity of 500 ft/sec. The static pressure of the test section will vary with the different cycles. In Cycle No. 1 this pressure, and hence the mass flow, will be quite low. There is a loss in total pressure across the tunnel which will be taken as one third of the test-section dynamic head throughout the calculations. The state conditions after the compressor are calculated in the same manner as with the turbine. The adiabatic efficiency is assumed to be 75 percent. The energy, E, can be provided by any prime mover, such as a reciprocating engine through an appropriate gear box, an electric motor, or a jet engine. The turbine and compressor indicated in the cycle can be part of a turbo-jet engine or a turbo-supercharger. Included in Fig. 4 is a table of the thermodynamic properties at different phases of the cycle. The numbered stations are indicated on the schematic drawing.

Cycle No. 2 (Fig. 5) provides for testing at a total pressure equal to atmospheric pressure. This cycle necessitates the use of a heat exchanger. However, at relatively high temperatures after compression this heat exchanger is quite efficient. The cycle calculations assumed no pressure drop across the heat exchanger. Also, in this case as well as with Cycle No. 1, the temperature rise due to the condensation in the dryer was neglected.

A closed air cycle system is included as Cycle No. 3, Fig. 6. Flexibility is now good because the tunnel may be operated at variable density. Both a heat exchanger and dryer are necessary, but the loads are greatly reduced from the equivalent open air cycle. The dryer need only remove the water introduced as the icing cloud, while the heat exchanger load is merely the summation of the losses in the system.

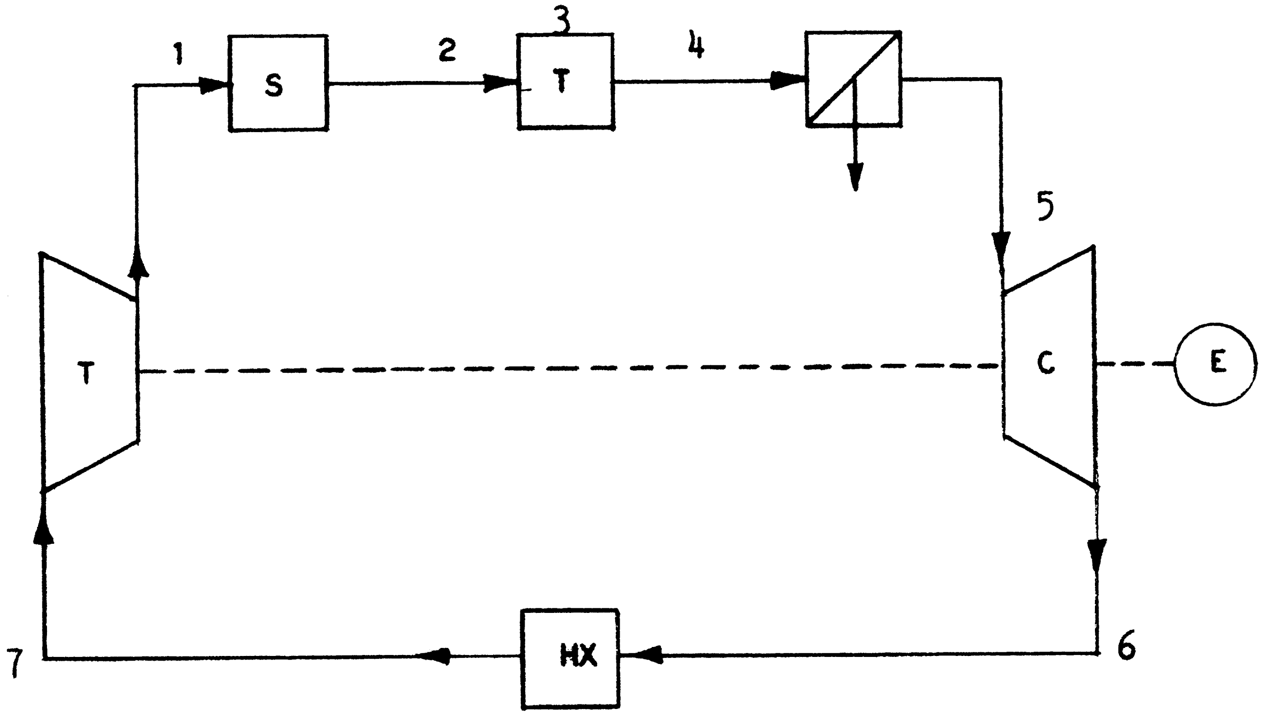
Cycle No. 4 (Fig. 7) differs from Cycle No. 1 only in that the power for driving the tunnel is now realized through ejector action. Any suitable source of compressed air could be used to drive this ejector.

Cycle No. 5 (Fig. 8) is an open system using a conventional vapor-type heat exchanger.



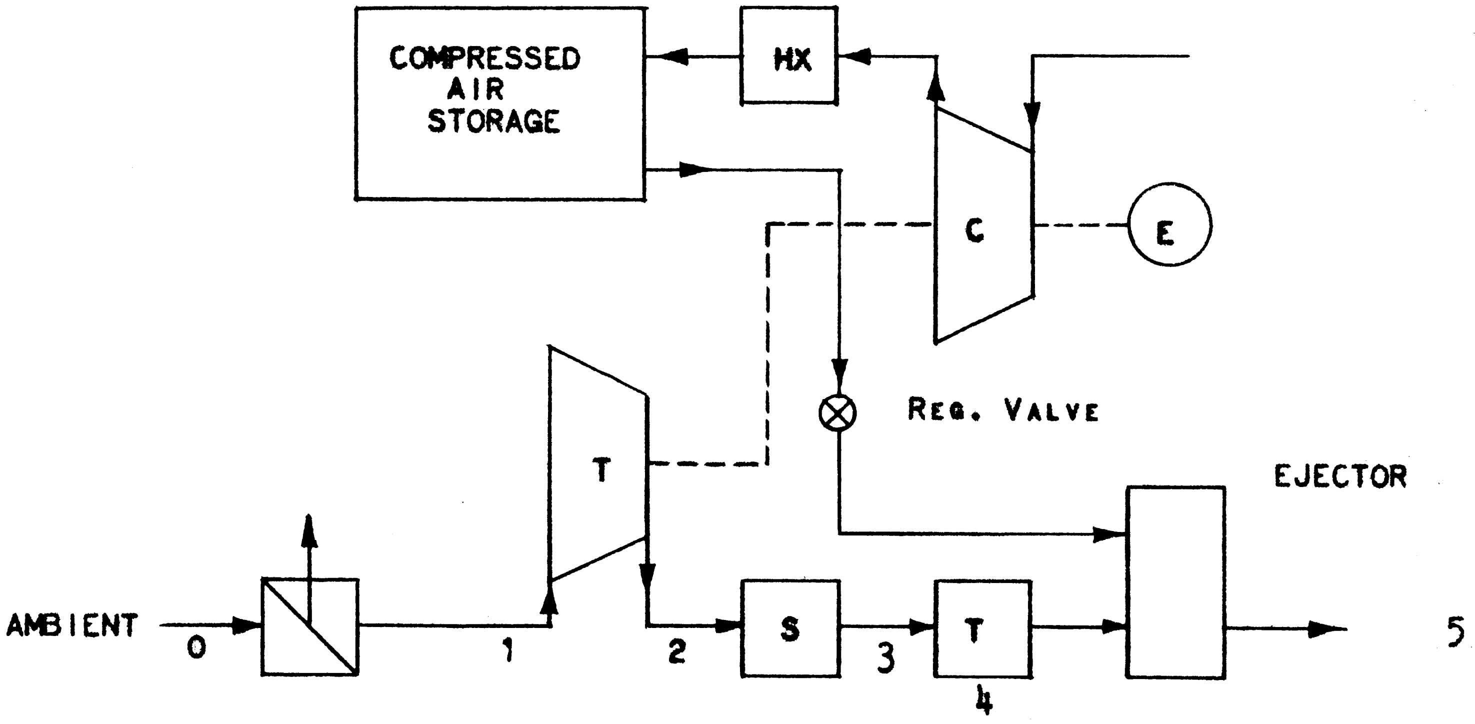
STATION	STATIC PRESSURE PSIA	STATIC TEMPERATURE °R	WATER-VAPOR CONTENT # VAPOR / # AIR	LIQUID-WATER CONTENT # WATER / # AIR
0	14.3	510	.004	0
1	14.3	510	.0001	0
2	27.5	623	.0001	0
3	27.5	530	.0001	0
4	14.97	460	.0001	0
5	14.97	460		.0025
6	12.6	438		.0025
7	14.3	460		.0025

FIGURE 5 - CYCLE #2, HIGH PRESSURE, OPEN



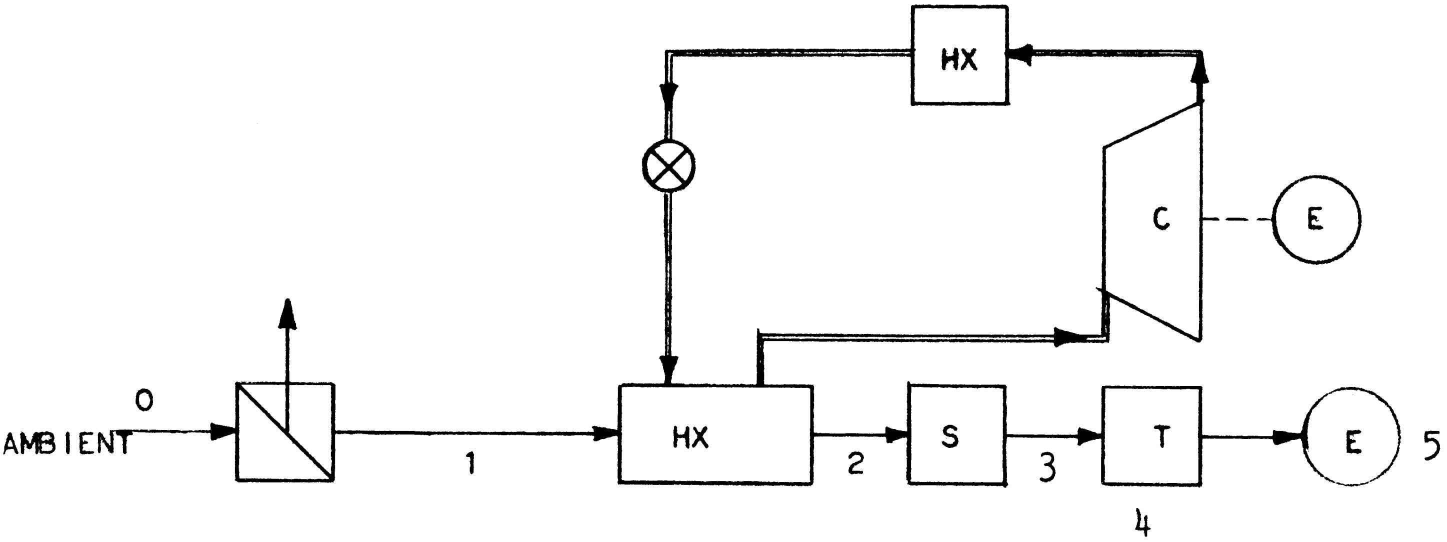
STATION	STATIC PRESSURE PSIA	STATIC TEMPERATURE °R	WATER-VAPOR CONTENT # VAPOR / # AIR	LIQUID-WATER CONTENT # WATER / # AIR
1	14.3	460	.0001	0
2	14.3	460		.0025
3	12.0	438		.0025
4	13.6	460		.0025
5	13.6	460	.0001	0
6	24.8	580	.0001	0
7	24.8	520	.0001	0

FIGURE 6 - CYCLE #3, CLOSED SYSTEM



STATION	STATIC PRESSURE PSIA	STATIC TEMPERATURE ^{0}R	WATER-VAPOR CONTENT # VAPOR / # AIR	LIQUID-WATER CONTENT # WATER / # AIR
0	14.3	510	.004	0
1	14.3	510	.0001	0
2	9.54	460	.0001	0
3	9.54	460		.0025
4	8.04	438		.0025
5	14.3			.0025

FIGURE 7 - CYCLE #4, EJECTOR DRIVEN, OPEN



STATION	STATIC PRESSURE PSIA	STATIC TEMPERATURE °R	WATER-VAPOR CONTENT # VAPOR / # AIR	LIQUID-WATER CONTENT # WATER / # AIR
0	14.3	510	.004	0
1	14.3	510	.0001	0
2	14.3	460	.0001	
3	14.3	460		.0025
4	12.0	438		.0025
5	14.3			.0025

FIGURE 8 - CYCLE #5, VAPOR-COMPRESSION SYSTEM

Choice of Components

The choice of components is somewhat restricted since conventional standard components may not necessarily fulfill the requirements of air-cycle refrigeration. This means that in some cases either specifically designed components or modified standard components must be used. In the following paragraphs some of the problems influencing the choice of components are presented.

Single large air turbines that will handle mass flows of about 150 pounds per second are not available as standard units. A number of standard smaller units (such as a turbo-supercharger) could be operated in parallel at the design pressure ratio for various mass flows, and hence, tunnel velocity. In this way the optimum turbine efficiency could be preserved.

It is possible to use compressor-turbine combinations from a conventional turbo-jet engine where the combustion chambers are inactive, but the common shaft would require an external drive. Either axial flow or centrifugal compressors would be adequate for the described use. Reciprocating compressors, however, may introduce undesirable pulsations. In those cases where the compressor is downstream of the icing tunnel the centrifugal compressor is advantageous in that it is not as susceptible to icing conditions. This is not true for the axial compressor, and hence, the supercooled water drops and ice particles would have to be removed prior to entering the compressor. As with the turbine, a number of smaller compressors might offer better control and efficiency.

The conventional chemical dryer having a sufficient capacity for icing tunnel purposes would be prohibitive in cost. Also in installation where it is necessary to remove the moisture as ice, chemical drying is impractical. The solution to the drying problem seems to lie in the use of a centrifugal-type separator. These centrifugal separators have been used for moisture removal in some of the low-temperature tunnels in England and might well be applied to the icing tunnel.[4] A combination of the chemical and separator-type dryer might be used to advantage.

Heat exchangers are another source of great expense and it becomes highly advantageous to minimize these requirements or, preferably, to do away with them altogether. Single-pass tubular heat exchangers are available and adequate. Also, surplus aftercoolers and intercoolers could be used by arranging them in parallel.

It is interesting to consider the possibility of using an hydraulic compressor as a means of driving the icing wind tunnel. This type of compressor is quite unique in that it utilizes a difference in natural water elevation to produce compressed air without the use of moving parts. As the water falls to an underground cavern the induced air is compressed isothermally and dried. Therefore, by the nature of its operation the hydraulic compressor incorporates a heat exchanger, a dryer, and a compressor, Obviously, it offers many advantages for operation with an icing wind tunnel.

The original cost of a hydraulic compressor would, no doubt, prohibit its use, but at this writing an abandoned unit exists in the upper peninsula of Michigan.[5] This particular unit is capable of producing 36,000 cu ft of free air per minute at a pressure of 117 psia or about 45 lb of air per second. The oxygen content of the air furnished by this compressor is somewhat less than that of standard air, due to the increased solubility of oxygen in water in relation to nitrogen at high pressures.

The greatest utilization could be derived by expanding the compressed air across a turbine into the spray room of the tunnel. The expansion process would furnish the cooling and water could then be added as desired. This particular mine could not run continuously at higher mass flows but an icing run of sufficient duration could be made. For continuous operation at higher mass flows, the available compressed air could serve as an ejector driver to induce the main flow through the tunnel.

Discussion

The results of the cycles considered in this report are given in Figs. 4 through 8, along with the flow diagrams. The important parameters are summarized in Table I while the advantages and disadvantages of each cycle are itemized in Table II. In general, the open cycles are conservative as to power requirements while the closed cycle offers more flexibility. Also, the open cycle systems are probably less expensive to construct.

Cycle No. 1 has some distinct advantages because no heat exchanger is required and the power requirements are low. This is the result of a low-testing pressure, which markedly reduces the mass flow of air through the system. This implies, however, a pressure-tight spray room as well as a well-sealed test section. Thus, accessibility to the test model and spray room is somewhat complicated. Flexibility is limited because the test section pressure is intimately tied to the velocity. The dryer capacity on warm humid days with an open cycle system becomes quite prohibitive. Substantial savings can be achieved by not designing for these more stringent conditions. This implies that no icing runs could be made on certain days in the summer. In the calculations made, an outside temperature of 50°F and a relative humidity of 50 percent were assumed.

TABLE I

THE IMPORTANT PARAMETERS OF EACH CYCLE

Cycle No.	Type	External Power Required	Assumed Inlet Conditions	Turbine Pressure Ratio	Mass-Flow Rate	Heat-Exchanger Capacity	Test-Section Pressure	Dryer Capacity
1	open	1230 hp	$P = 14.3$ psia $T = 50°F$ $W = .004 \frac{\text{lb } H_2O}{\text{lb air}}$	1.51	99 lb/sec	none	8.05 psia	$.396 \frac{\text{lb water}}{\text{sec}}$
2	open	2260 hp	$P = 14.3$ $T = 50°F$ $W = .004 \frac{\text{lb } H_2O}{\text{lb air}}$	1.84	155 lb/sec	$3460 \frac{\text{Btu}}{\text{sec}}$	12.6 psia	.62 lb/sec
3	closed	2950 hp		1.73	148 lb/sec	$2140 \frac{\text{Btu}}{\text{sec}}$	12.0 psia	.393 lb/sec
4	open	approximately 1600 hp (ejector is relatively inefficient)	$P = 14.3$ $T = 50°F$ $W = .004 \frac{\text{lb } H_2O}{\text{lb air}}$	1.51	99 lb/sec	none	8.05	.396 lb/sec
5	open vapor system	approximately 533 hp (for refrigeration) 1850 hp for tunnel	$P = 14.3$ $T = 50°F$ $W = .004 \frac{\text{lb } H_2O}{\text{lb air}}$		148 lb/sec	$1780 \frac{\text{Btu}}{\text{sec}}$	12.0 psia	.59 lb/sec

TABLE II

THE ADVANTAGES AND DISADVANTAGES OF EACH CYCLE

Cycle No.	Advantages	Disadvantages
1	(A) Low power requirements (B) No heat exchanger necessary (C) Dryer capacity is relatively low (D) Inexpensive system	(A) Limited Flexibility (B) Low pressures complicate design (C) Possible complications due to ice particles into compressor
2	(A) Medium power requirements (B) Atmospheric pressure in the spray room with slightly reduced pressure in the test section (C) No problems connected with ice particles into rotating machinery	(A) Limited Flexibility (B) High heat-exchanger requirements (C) High dryer capacity necessary (D) Expensive due to dryer and heat exchanger
3	(A) Flexible so as to allow variable density	(A) High power requirements (B) Both heat exchanger and dryer necessary although each of medium capacity (C) Expensive (D) Possible complication due to ice particles into compressor might necessitate a separator
4	Same as Cycle No. 1 although the ice particles are no longer a problem.	
5	(A) Medium power requirements (B) Good control over operating temperature (C) Atmospheric pressure in spray room	(A) Limited flexibility (B) Possible complications due to ice particles into rotating machinery (C) Very expensive heat exchanger equipment although less than cycles 2 and 3.

In Cycles 1, 3, and 5 the ice particles, or supercooled water, represent a problem in that they can impinge on the diffuser walls on the blades of rotating machinery. Seemingly, some sort of separator would be required. The ejector system does serve to keep the diffuser clear of ice.

Cycle No. 2 has some serious disadvantages. Not only is the flexibility limited but the heat exchanger and dryer requirements are extreme. The expense associated with these two units alone would be extremely high. The same might be said of Cycles 3 and 5 although the closed cycle at least affords some flexibility.

It would appear that if surplus turbojet engines or turbo-superchargers were to be used to an advantage in an air-cycle refrigeration system for an icing wind tunnel, the choice would be confined to Cycles 1 or 4.

This conclusion is premised on the assumption that heat exchangers of the capacity indicated are not available as surplus.

PART II. PRELIMINARY EXPERIMENTS USING EXPANSION TURBINES FOR AIR REFRIGERATION AND DROPLET PRODUCTION

General Remarks

This section of the report will deal with some preliminary experiments conducted at the icing wind tunnel at the University of Michigan to determine the feasibility of the utilization of an expansion turbine for the production of a supercooled cloud of droplets for icing simulation studies.

Up to the present time all the aircraft icing facilities have employed some type of spray to obtain a cloud of droplets that approximate the size and distribution of the droplets found in natural supercooled clouds. One of the problems associated with the production of an artificial cloud is that the droplets tend to be larger than the droplets found in natural clouds. Furthermore, if the spray is introduced into air that is unsaturated, the smaller droplets having a higher vapor pressure evaporate faster than the larger droplets, until the air is saturated. After this time the smaller droplets still evaporate, while the larger droplets grow. Both of these effects tend to force the mean-effective droplet size to ever larger values. In many cases, even with well designed sprays, a mean-droplet size of 10-15 microns, e.g., the size found in most natural supercooled clouds, is difficult to obtain.

It was felt that if the adiabatic cooling processes which form a natural cloud could be experimentally duplicated, the possibility of obtaining a cloud more closely simulating a natural cloud would logically be very good. In aircraft icing studies air temperatures well below freezing are desired; therefore, it was decided to investigate the type of cloud formed by expanding air below its dewpoint in an air-cycle turbine. At the same time it was believed that the refrigerated air passing from the turbine could be utilized on days when the outside ambient temperature was slightly above freezing, and by mixing the refrigerated air with outside air in the spray room it would be possible to obtain icing conditions in the tunnel. This would be a great advantage inasmuch as there are many such days during the winter months in this locality.

Experimental Equipment

Two Packard-built Rolls-Royce engines were used as the air supply.[1] They were naturally aspirated and drove their own superchargers to furnish the compressed air for the turbines (see Fig. 9). The air from each compressor first passes through two aftercoolers and from there to the turbine. The engine-driven air supplies are arranged so that they may be coupled in series, if high-pressure ratios across the turbine are desired; or in parallel, if large mass flows are desired. When the compressors are coupled in series the air into the first stage compressor is obtained from outside the building, thus, making it possible to get air of constant and accurate specific humidity.

Either of two B-22 type turbo-supercharger units are used to expand the air (Fig. 10). One is used in the conventional manner, i.e., the air is expanded through the turbine buckets with the compressor acting as the load. Since the efficiency of this turbine is quite low, it was decided to reverse the normal direction of the air flow and expand the air through the centrifugal compressor of another unit so that it operated as a centripetal turbine.* In this case, the power was absorbed by passing a portion of the air through the turbine of the unit and using it as an irreversible air brake.

The air is expanded through either of the two turbines directly into the spray room. In the past the spray room had been used as a settling chamber at the top of which air was introduced through louvres with the spray nozzles located near the top of the chamber. To accommodate the turbine installation, the room was weather-stripped and doors were placed over the louvres. The refrigerated air with the entrained water droplets is then passed out through the tunnel test section. A perspective drawing of the whole facility with one turbine unit appearing is shown in Fig. 11. A schematic drawing of the installation showing both turbines appears in Fig. 12.

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*Tests performed in this manner on a larger centrifugal compressor at NACA[6] indicated that efficiencies over 70 percent were obtainable.

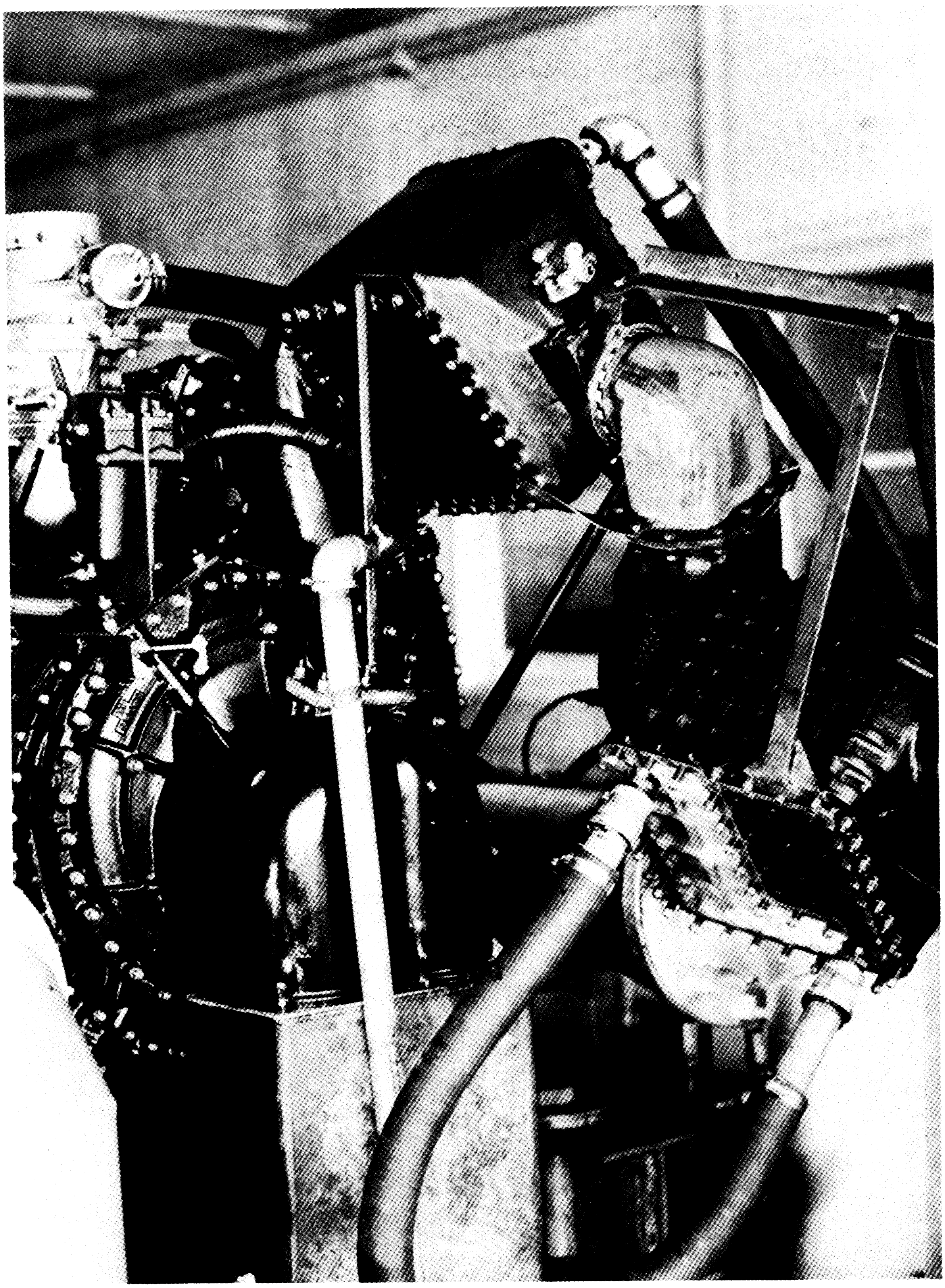
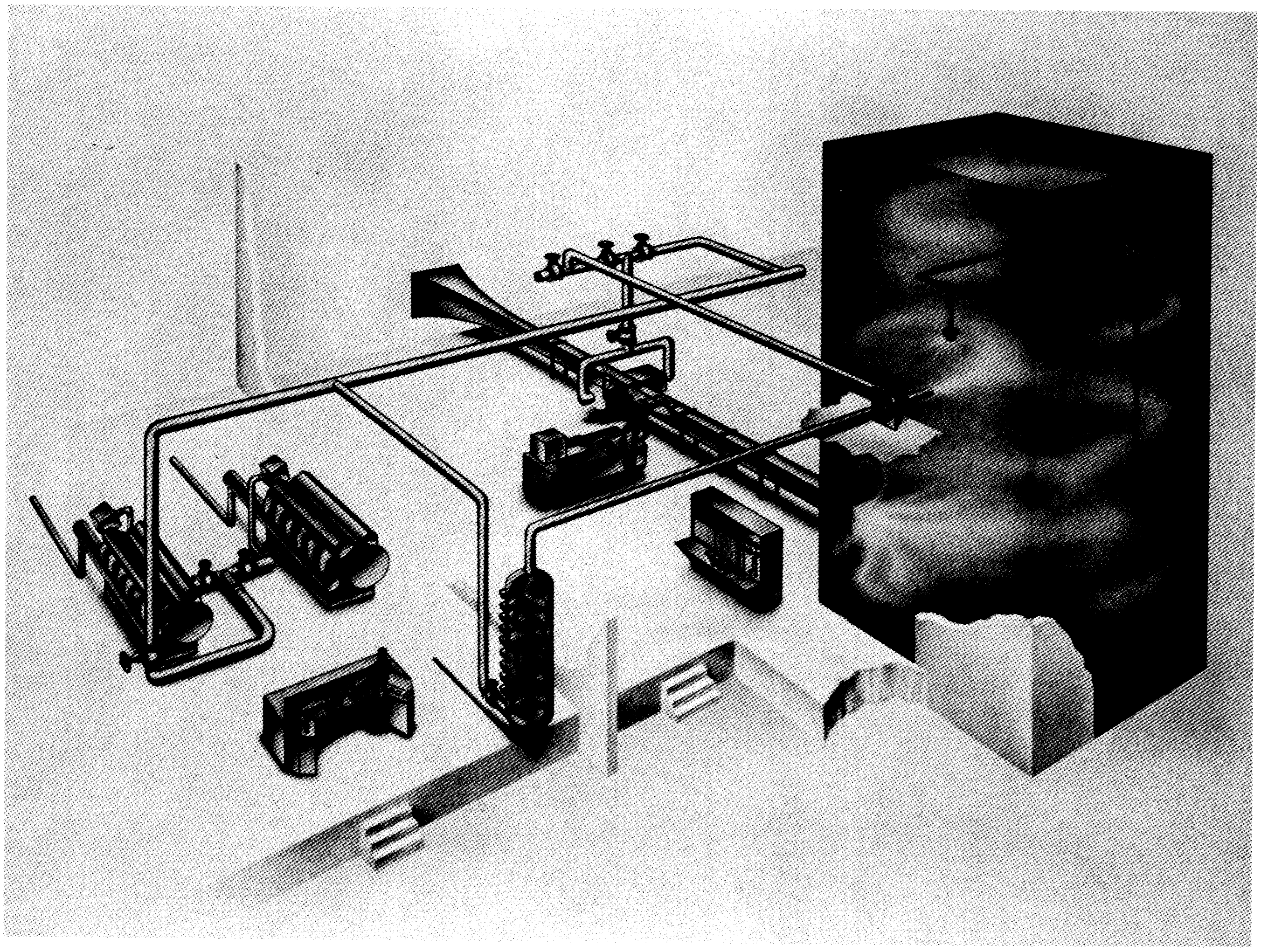


Fig. 9 - Packard Rolls-Royce Air Supply

Fig. 10 - Photograph of a B-22 Type Turbo-Supercharger Used as an Expansion Turbine



ICING WIND TUNNEL

THE ICING TUNNEL PROVIDES A MEANS OF STUDYING AIRCRAFT ICING AT VELOCITIES UP TO 400 FT. PER SEC. IN A 1 FT. BY 1 FT. TEST SECTION. NORMALLY, THE TUNNEL RELIES ON NATURAL REFRIGERATION WHERE THE CLOUD IS FORMED BY PNEUMATIC SPRAYS. AN AIR CYCLE EXPANSION TURBINE ARRANGEMENT IS PROVIDED FOR MODERATE REFRIGERATION WHEN AMBIENT TEMPERATURE IS ABOVE FREEZING.

THE TUNNEL IS POWERED BY EJECTORS WHICH ARE DRIVEN BY THE OUTPUT OF A ROLLS ROYCE SUPERCHARGER.

Fig. 11

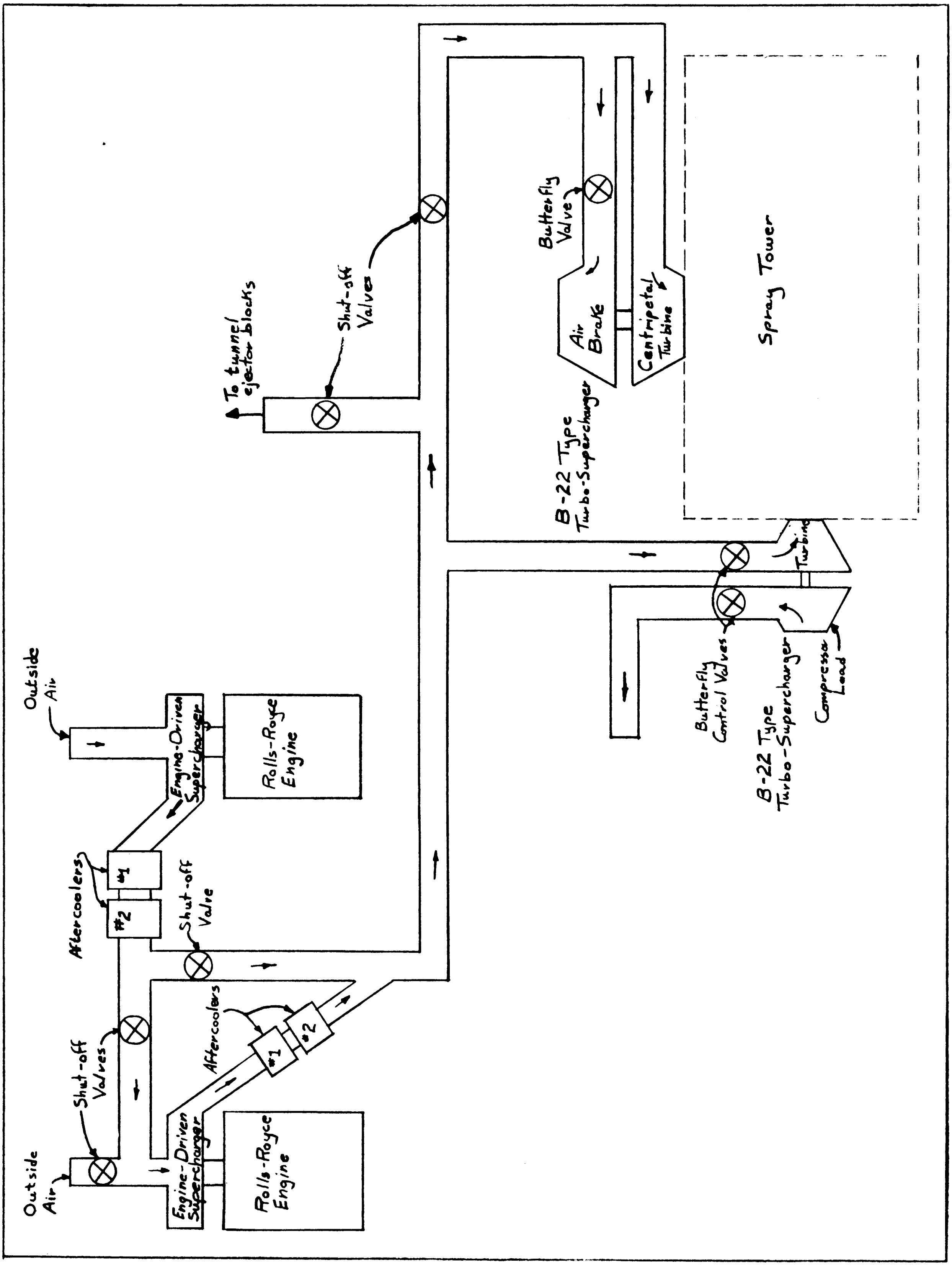


Fig. 12 - Schematic Drawing of Air-Cycle Refrigeration System

Experiments and Results

Prior to October, 1953, only one engine air supply was available, and the pressure ratios across the turbine were naturally limited. With an area opening of 5 sq in. at the nozzles of the number one turbine a pressure ratio of about 2.3 was all that was possible. From October, 1953, through mid-February, 1954, several runs were made using the two air supplies compounded in series. Pressure ratios of 3 and slightly greater were obtained. During these runs temperatures as low as 0-5°F were obtained at the exit of the turbine with eqiulibrium temperatures as low as 23°F in the spray room.

No attempts were made to measure the liquid water content of the air in the spray room because it was believed that if the specific humidity of the air entering the system from the outside were known this information could be obtained. It was found that the aftercoolers in the system had developed perceptible leaks so that any calculated liquid water content would be low. However, droplet samples were taken in the spray room for the runs of 11 November and 22 December 1953.

The technique used consisted of obtaining samples on glass slides coated with Shell Spherex, a technique developed by the Low Temperature Laboratory, N.A.E. in Ottawa, Canada. Since the velocity at the test section in the tunnel was extremely low, about 20-40 ft/sec, it was not possible to impinge the droplets on the slide at the test section; thus, the samples were obtained in the spray room by holding the slide at arm's length and rapidly moving it through the cloud.

Photomicrographs of the droplet samples appear in Figs. 13 and 14, with existing spray room conditions for these runs 36 and 23°F, respectively. The mean-effective droplet diameter computed from samples made during these two runs were 4.8 microns for the condition of 36°F in the spray room and 4.7 microns for the condition of 23°F in the spray room. Considerable doubt exists as to whether this is a valid means of obtaining a true droplet sample, inasmuch as the catch efficiency for exceedingly small drops on a flat plate is not too high. However, these samples give mean drop sizes which are, if anything, larger than existed in the actual cloud.

For comparison a droplet sample was obtained at the test section of the tunnel on 4 March 1953, when an icing experiment was being conducted. The tunnel velocity was 200 ft/sec and the temperature at the test section was 14°F. This sample was taken with the sprays operating in the normal manner with air being induced by the tunnel ejector from the outside. During this run the turbine was not operating. Figure 15 shows a droplet sample taken during this run. A mean drop diameter of 14 microns was obtained using this sample along with others. This is a representative magnitude of one of the smallest mean droplet

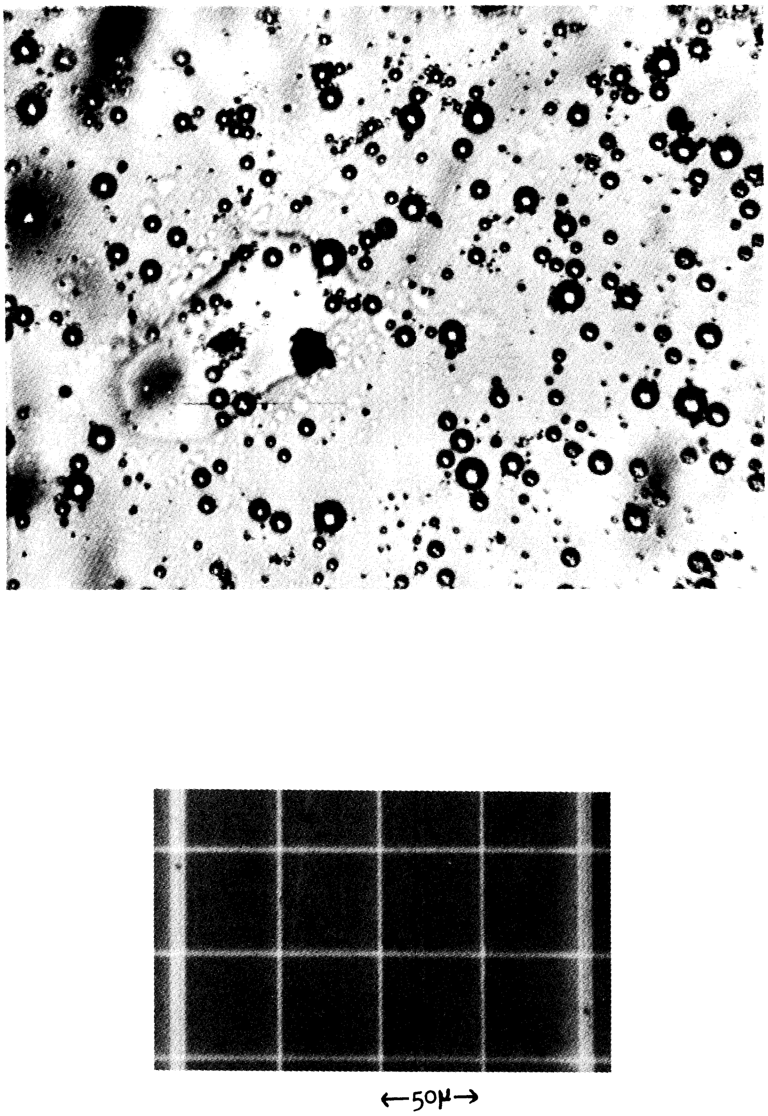


Fig. 13 - Photomicrograph of Droplet Sample Produced by Turbine, Temperature 36°F

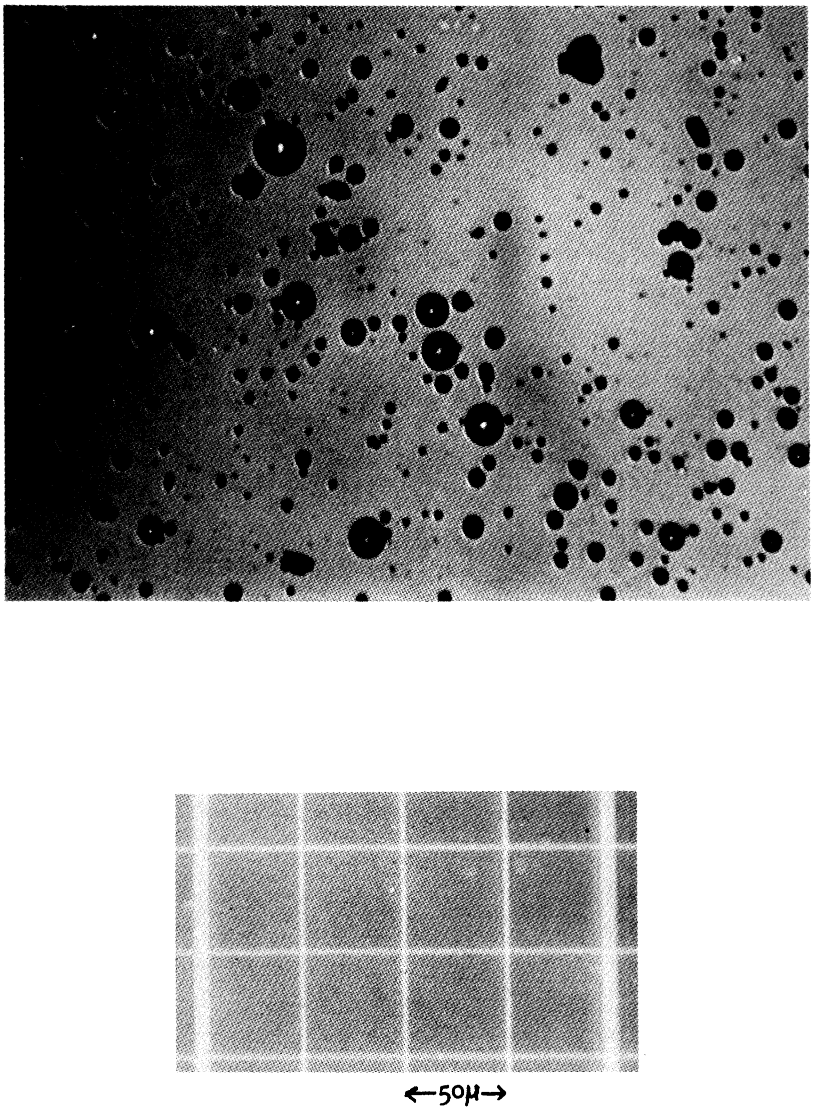


Fig. 14 - Photomicrograph of Droplet Sample Produced by Turbine, Temperature 23°F

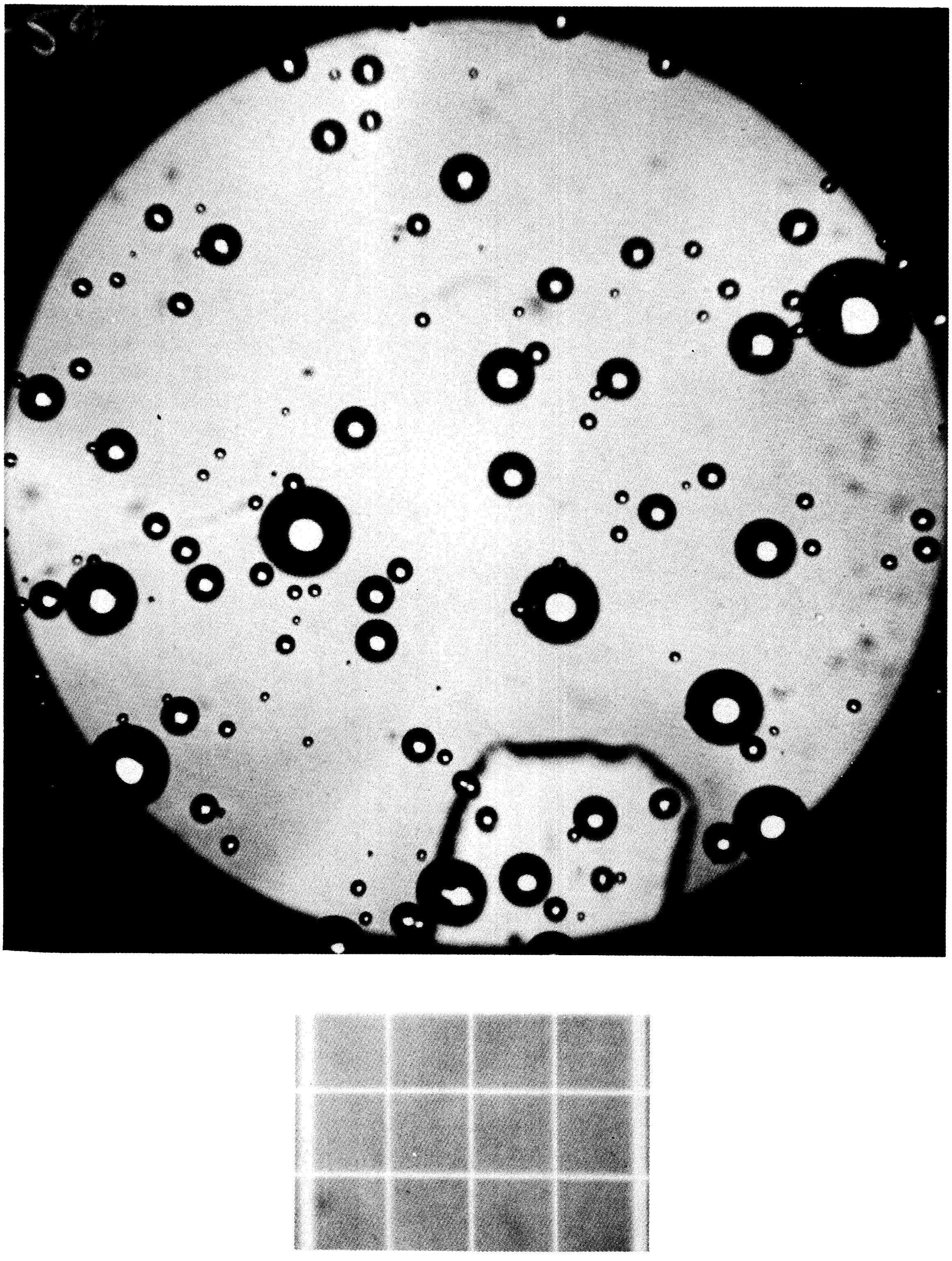


Fig. 15 - Photomicrograph of Droplet Sample Produced by Air-Aspirated Spray Nozzles, Temperature 14°F

diameters obtained in the icing tunnel to date, using an air-aspirated water spray system patterned after the one in use at NACA. Figure 16 shows a comparison of the droplet distributions obtained with the two turbine-produced clouds and the cloud produced with the air-aspirated spray.

Until the beginning of February, 1954, the turbo-supercharger unit (the centrifugal compressor which was to be used as a centripetal turbine) was being adapted and instrumented. About the time this unit was ready a failure of one of the Rolls-Royce engines discontinued any tests on that unit using high-pressure ratios across the turbine. Several runs were made with only one air supply operating. With the area blocked down to 3 sq in. at the turbine nozzles, a pressure ratio of 2.7 was obtained. However, with the leaks in the aftercoolers stopped and a limited pressure ratio available the temperature in the spray chamber could not be lowered sufficiently below the dewpoint to obtain a stable cloud.

A qualitative observation of the type of ice forming during several runs was made. A cylinder was placed in the low-velocity air stream at the test section of the tunnel. At temperatures as low as 23°F a porous type of ice was deposited that was not nearly as dense as the ice formed when the sprays were operating at the same temperatures. This could possibly be due to the small droplet size, the low velocity of the air, or a combination of both.

Also, it was observed that no ice had been deposited on the turbine buckets at temperatures as low as 0-5°F at the exit to the turbine. However, ice was deposited on the walls of the pipe connecting the outlet of the turbine with the spray room. Furthermore, even at the lowest temperatures obtained in the spray room, it appeared that no freeze-out had occurred when observing bright lights shining through the cloud.

Conclusions

It is to be emphasized that any conclusions drawn are strictly qualitative due to the preliminary nature of the experiments performed to date.

First of all, it appears that the mean droplet size for the turbine produced cloud is substantially smaller than that produced by the air-aspirated type of spray in use at the University of Michigan icing wind tunnel and is smaller than that observed in most natural supercooled clouds. Perhaps with longer residence times in the spray room the droplets would grow considerably. Further studies should be made, perhaps on a smaller scale, of the growth rate of droplets as a function of distance from the outlet of the turbine. It is apparent from the droplet photographs that the spectrum of droplet sizes is narrower for the turbine-produced cloud than it is for the spray-produced cloud.

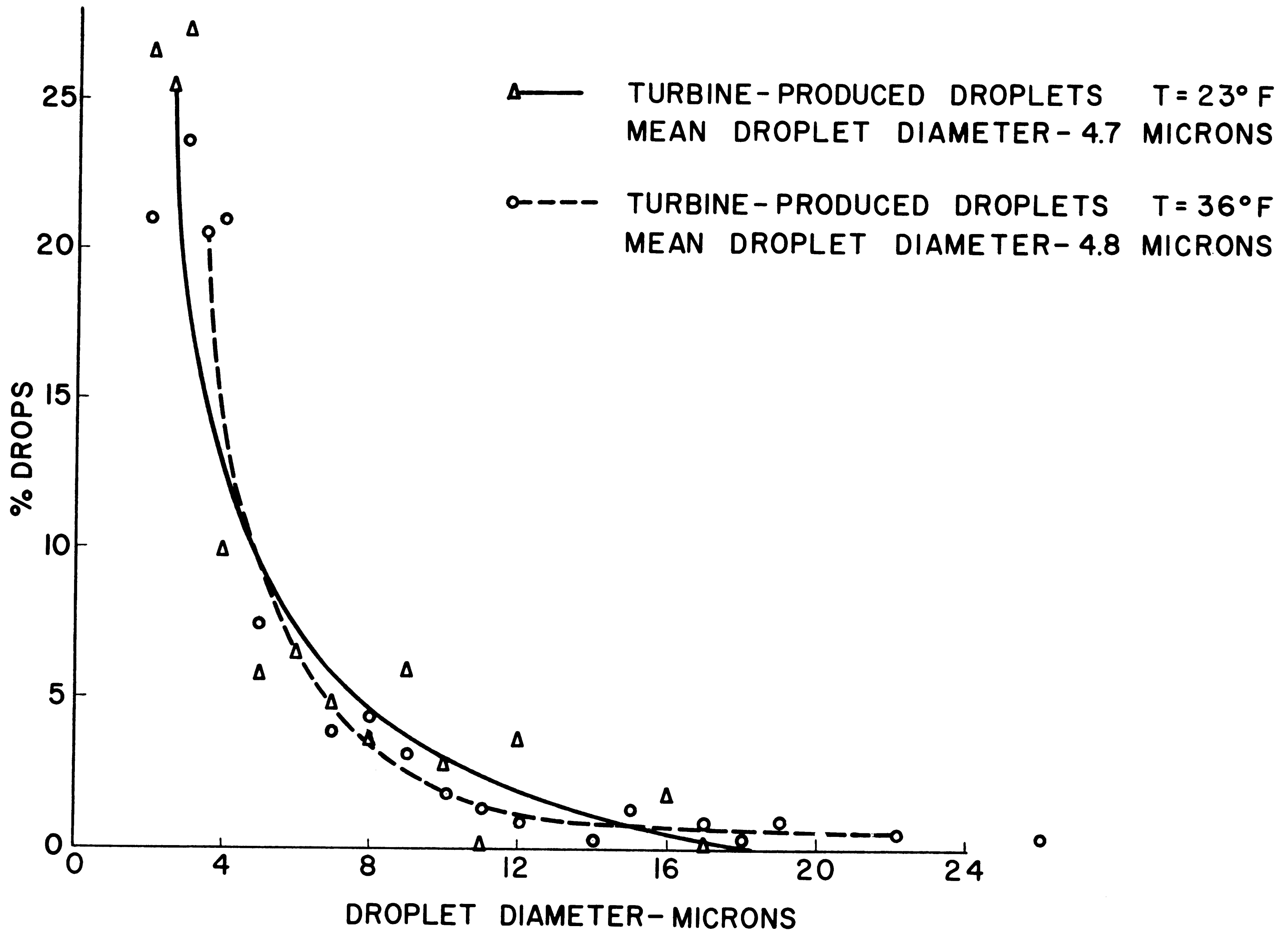


FIG 16A. COMPARISON OF DROPLET DISTRIBUTION FOR TURBINE-PRODUCED CLOUDS AND CLOUDS PRODUCED BY AIR-ASPIRATED SPRAY NOZZLES

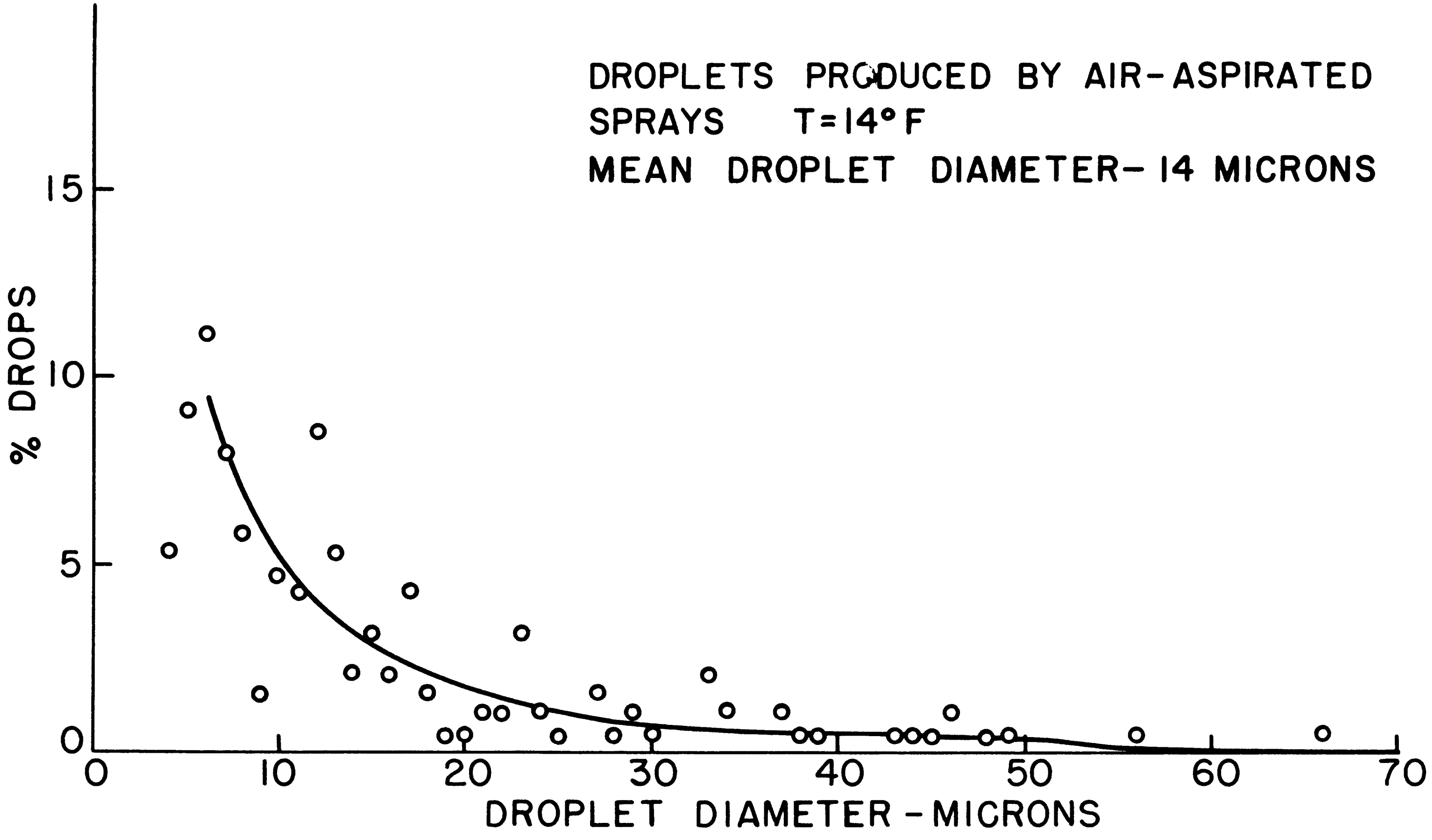


FIG. 16B. COMPARISON OF DROPLET DISTRIBUTION FOR TURBINE-PRODUCED CLOUDS AND CLOUDS PRODUCED BY AIR-ASPIRATED SPRAY NOZZLES

Again, it is possible that different droplet distributions could be obtained by different designs of the calming chamber ahead of the entry into the tunnel. If the outlet from the turbine were carefully designed it might be possible to obtain droplets having essentially the same size and growing somewhat uniformly as the cloud moved into the tunnel. It would be interesting to observe various icing phenomena with droplets essentially of a given size. Of course, it is evident that this might pose serious design problems due to the swirl of the air coming from the turbine outlet. It would undoubtedly be an easier problem to design a settling chamber in which turbulence is purposely produced to simulate more closely the droplet size and distribution found in nature.

It should be noted that two phenomena, one of freeze-out in the cloud and one of icing of the turbine buckets, did not occur. Possibly, at temperatures below 0°F at the turbine exit, either or both of these phenomena could occur and pose serious problems connected with the study of highly-supercooled clouds formed by the turbine. One problem appears evident; however, this is the structure of the ice formed on the rod placed in the test section as supercooled turbine-formed droplets were passed by. The ice formed resembled neither glaze nor rime ice, in that it was much more porous than rime ice and had very low adhesion properties. A further investigation should be made of the type of ice formed under various droplet sizes and distribution.

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