Conf-751106--5

A CESIUM VAPOR CYCLE FOR AN ADVANCED LMFBR

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Paper submitted for presentation at the ASME
Winter Annual Meeting
Houston, Texas
November 30-December 5, 1975

*Research sponsored by the Energy Research Development Administration under contract with the Union Carbide Corporation.

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Introduction

A. Amobosi of the Argonne National Laboratory has proposed that a mercury vapor cycle be employed in place of the intermediate sodium loop in an advanced, higher temperature LMFBR.[1] This would increase the thermal efficiency of the plant and would have several advantages from the reactor safety standpoint. However, past experience with mercury vapor cycles indicates that mercury gives serious difficulties with corrosion if used in iron-chrome-nickel alloys at temperatures above 900°F.[2] Further, the cost is high, the heat transfer coefficients are erratic because they depend on the composition of complex surface films, and the toxicity of mercury vapor is such that the mercury inventory would represent a hazard potential roughly equivalent to that of the volatile fission products in the reactor.[3]

In searching for other candidate working fluids for high temperature Rankine cycles, a variety of studies have favored the alkali metals, notably potassium and cesium,[3-12] and work on a potassium vapor cycle for use with fossil fuel is currently going forward under an NSF/RANN-ORNL program.[11] The boiling point of potassium is a bit high for use with an advanced LMFBR, but cesium is a logical candidate as the working fluid for a Rankine cycle employed as the intermediate heat transport circuit coupling the primary sodium circuit of the reactor to the steam cycle. The boiling point of cesium is approximately 160°F below that of potassium, hence the proportions of a cesium turbine are large but tolerable with a cesium condenser temperature as low as 800°F, whereas around 960°F appears to be as low a temperature as one would care to go for the bottom end of a potassium vapor cycle.

Cost and Availability of Cesium

There has been essentially no commercial market for cesium, and, as a consequence, the price of pure cesium metal is quite high, about $100/lb.

However, ample quantities could be made available if a market were to develop, and the price would be in the range of $2 to $10/lb, substantially less than for mercury when allowances are made for differences in the densities and hence weights of fluids required.[13, 14]

Cycle Conditions

The major components of the combined cycle considered here are shown in Fig. 1. The heat from the reactor sodium coolant would be used to boil cesium. The cesium vapor would expand through a cesium turbine and then would condense and give up its heat in a steam generator.

A cesium vapor topping cycle offers two important advantages from the thermal efficiency standpoint. The first is that the peak cycle temperature can be substantially higher than for steam, and this will yield an improvement in thermal efficiency. A second, but more subtle, major advantage is that the combined cesium vapor-steam cycle can be designed to give a more nearly rectangular temperature-entropy diagram and thus will approach more closely the ideal Carnot cycle. This effect can be deduced from Fig. 2 which shows temperature-entropy diagrams for a typical set of Rankine cycles. Note that in going from the saturated vapor cycle in the upper left corner of Fig. 2 to the same basic cycle but with superheating, the area under the temperature-entropy diagram is increased, but not by as much as would have been the case if the peak pressure had been increased to yield the same peak temperature for a saturated vapor cycle. The quantitative effects of the peak steam temperature on the efficiency of steam Rankine cycles are shown in Fig. 3 as a function of the peak temperature for both a series of cycles in which the turbine inlet temperature is that for the saturation pressure and for a series of cycles with various amounts of superheat. It is evident from this plot of cycle efficiency that superheating to a given temperature yields less of an increase in cycle efficiency than would be obtainable if the same increase in cycle operating temperature could be obtained with an increase in pressure so that the turbine would be supplied with saturated vapor.

Physical Properties

The physical properties of cesium that are important from the standpoint of power plant design are summarized in Table 1 along with the corresponding values for steam and potassium.[15-18] Data are given for typical turbine inlet and outlet conditions. Note the similarity between the cesium and potassium on the one hand and water on the other except for the thermal conductivity.

Corrosion and Mass Transfer

The compatibility of cesium with structural materials is essentially the same as for potassium. There is essentially no difference in the tendency toward solution corrosion and mass transfer. It is particularly important to note that in a stainless steel system of the type contemplated in Fig. 1 the recirculating cesium boiler acts to block mass transfer; the cesium in the hot zone (the recirculating boiler) becomes saturated with Fe, Cr, and Ni, and no further solution corrosion takes place because only the vapor leaves the boiler — the solute remains in the recirculating liquid. Extensive tests show that corrosion in potassium systems of this type is trivial up to a boiler temperature of ~1600°F, and limited tests with cesium indicate that it behaves in essentially the same fashion.[19]

A reference here would make a stronger case

Limitations Imposed by Turbine Design Considerations

Extensive experience with steam turbines has shown that it is not economically worthwhile to build turbines for discharge pressures below approximately 0.5 psia. At these low pressures the vapor density is so low that the turbine wheel diameter becomes excessive for a given power output, and the cost of additional stages (which increase in size rapidly) more than off-sets the savings in fuel cost from the improvement in thermal efficiency that would be obtainable. In compromising between turbine cost and cycle efficiency, the latter was favored and the turbine discharge temperature for the cesium vapor cycle was chosen to be 800°F, which corresponds to an absolute pressure of 0.66 psi.

Fundamental factors determining the size and cost of a turbine are the number of stages required and the diameter of the last stage. For a given power output the diameter of the last stage is directly proportional to the specific volume of the gas or vapor leaving the turbine and inversely proportional to the sonic velocity and the enthalpy drop across the stage. The power output per stage for a given Mach number in the blading is proportional to the adiabatic head associated with the sonic velocity. This in turn depends on the molecular weight. (Cesium and potassium vapors are monatomic.) Values of these parameters for steam, cesium and potassium at typical conditions are given in Table 2. These data were plotted in Figs. 4 and 5 to show the effects of turbine outlet temperature and pressure on the outlet stage diameter. Table 2 and Figs. 4 and 5 indicate that there is a major advantage from the turbine design standpoint to using cesium rather than potassium as the working fluid, an advantage given particular attention in the course of the space power plant program.[7,9,15–29] Note also that Table 2 assumes the same relative Mach number in the blading for all three fluids, that the rotor diameter for a given discharge pressure is lower for both cesium and potassium than for steam, and that, for a topping cycle designed to operate at a given condenser temperature, a cesium turbine will have a diameter about 60% of that for a potassium turbine.

The number of stages required is also an important factor affecting the cost of turbines. The prime limitations on the work per stage are limitations on the tip speed imposed by stresses in the rotor from centrifugal forces, compressibility losses associated with relative velocities greater than sonic at the inlets to rotors, and turbine bucket erosion by moisture in wet vapor. If only the first of these, the stresses from centrifugal forces is considered, the number of stages can be lower for vapors having low sonic velocities, e.g., cesium. Inasmuch as the work output per unit weight of working fluid per stage varies as the square of the rotor tip speed, the number of stages required varies directly as the square of the sonic velocity for a given tip speed. The last line of Table 2 indicates the implications of this consideration. In point of fact, compressibility losses become large at high Mach numbers so that design compromises must be made. In studies of turbines for space power plants, for example, it has appeared that a good compromise is given by using about 60% as many stages in a cesium turbine as in a potassium turbine.[9,20]

Turbine bucket erosion by moisture in the vapor can be kept to an acceptable level even at centrifugal stress-limited tip speeds by proper design.[20, 26] Thus the prime considerations in determining the size and number of stages of a turbine are the turbine discharge pressure and the molecular weight of the working fluid because rotor stress limitations are about the same for the operating ranges for the three fluids considered here. Clearly, cesium is a more attractive working fluid than potassium.

For an attractive set of turbine proportions, the turbine ought to have a temperature drop of at least 300°F. Thus, if the cesium turbine discharge temperature is chosen to be 800°F, one ought to go to a turbine inlet temperature in excess of 1100°F. Thus, for preliminary analysis purposes, the three points chosen were for cesium vapor temperatures and pressures at the turbine inlet of 1150°F/8.9 psia, 1250°F/15.8 psia, and 1350°F/25.9 psia, respectively.

Temperature Differences in the Heat Exchangers

When allowances are made for the temperature rise necessary in the primary sodium circuit to keep the pumping power requirement to a reasonable level, the three typical cycles chosen for analysis would require reactor sodium outlet temperatures of at least 1300, 1400, and 1500°F, respectively. The local temperature in each of the fluid streams passing through both the primary sodium-cesium heat exchanger and the cesium-steam generator are shown in Fig. 6 as functions of the fraction of the heat transferred to the cold fluid for a typical case. Because of the relatively poor heat transfer coefficient characteristic of superheated steam, the peak temperature in the steam cycle was taken as 100°F below the cesium condensing temperature. This assures that the amount of heat transfer surface area required will not be excessive. The "pinch point," or minimum temperature difference between the sodium and the boiling cesium was taken as only 20°F because the heat transfer coefficients are very high for both fluid streams.

A somewhat higher cycle efficiency can be obtained for a given reactor coolant outlet temperature by following the approach employed in the British gas-cooled reactors, i.e., using two boiler pressures, one somewhat higher than the other to give a system temperature distribution such as that in Fig. 7. This implies the use of two cesium turbines, one supplied by the higher pressure cesium boiler and the other by the lower. This would not represent a serious complication in a very large plant — which an LMFBR will have to be because of economic considerations — because a number of turbines will have to be used in parallel under any circumstances. Optimization studies would be required to establish the best temperature differences for good proportions.

Results of Calculations

The calculations for the performance of the cesium vapor cycle were carried out following the same procedure as used for the earlier potassium and cesium vapor cycle studies at ORNL. Use was made of some 14 reports prepared in the course of a study carried out for NASA by ORNL on the relative merits of cesium and potassium as working fluids for nuclear electric space power plants.[7,17-29]

Thermodynamic Calculations

Thermodynamic calculations for a cesium vapor cycle superimposed on a steam cycle are presented in Table 3 for the three sets of conditions considered here. A turbine efficiency of 85% was assumed for both the cesium and steam turbines.

It may be noted that the cesium expansion is carried to a vapor quality of around 85%, and this implies that there might be difficulty with moisture causing turbine bucket erosion in the cesium vapor turbine. However, both analyses and experiments indicate that moisture in the vapor in either cesium or potassium vapor turbines represents a much less serious problem than in steam turbines.[21,26] Further, the development of turbines for water reactor plants has progressed to the point where moisture removal techniques are effective in avoiding not only turbine bucket erosion but also most of the moisture churning losses that have formerly reduced the turbine efficiency.[26]

Overall Thermal Efficiency

The cycle efficiencies presented in Table 3 are for cycles operating with no regenerative feedwater heating. Past experience indicates that including feedwater heating in the steam cycle should increase the overall thermal efficiency by about 6 points for the range of temperature conditions under consideration here. This effect is shown in Fig. 8, and allowance for it has been included in Table 4, together with an allowance of 3.5 points for heat losses and pumping power requirements. Thus, it appears that the overall cycle efficiency for a cesium vapor cycle coupled to an LMFBR could be around 50%. This would serve to reduce the fuel costs about 15% and would in effect reduce the reactor capital cost by about the same amount relative to a conventional steam cycle system. Further, it will reduce the heat rejection to the environment by about 30%.

The effects of reheat on steam cycle efficiency are substantial, as can be seen in Fig. 9. They are not as great as those of regenerative feedwater heating, and the extra complication and cost of more than one stage of reheat is usually not economically justifiable in fossil fuel plants. As a consequence, only one stage of reheat was used for the steam cycle employed here, although a detailed study may favor two stages of reheat, particularly because the length and cost of the reheater piping between the steam and cesium turbines will be much less than between steam turbines and furnaces of conventional fossil fuel plants.

If a fossil fuel heat source were employed for the temperature range under consideration here, heat losses to the stack gas would reduce the thermal efficiency to about 45%. As indicated earlier, the reason for the marked improvement in thermal efficiency obtainable with the cesium vapor topping cycle with a relatively small increase in peak temperature stems from the more nearly rectangular temperature-entropy diagram that can be obtained by using a high pressure steam cycle with relatively little superheat. This effect can be deduced from Fig. 2.

Maintenance Characteristics

Maintenance operations on the intermediate fluid circuit of a conventional LMFBR present some difficult problems. These stem mainly from the fact that operating experience with sodium, NaK, and lithium systems has shown that a major set of operating problems is presented by imperfect drainage of systems — a substantial amount of liquid remains on the tube walls in the form of droplets or lies in little depressions in pump casings, valves, etc. As a consequence, if maintenance on the system is required, even after draining one cannot allow the system to be exposed to atmospheric air because residual liquid in the system would oxidize, excessive amounts of oxide would form, and these would be likely to cause corrosion during the shutdown. Further, these oxides would present a major cleanup problem when the time came to refill and restart the system. As a consequence, maintenance is handicapped by the requirement that the system first be thoroughly flushed with alcohol or steam followed by water to remove all traces of the alkali metal before maintenance work can begin. Then the system must be thoroughly dried out to remove all traces of water. Otherwise the system must be flooded with inert gas and the inert gas kept somewhat above atmospheric pressure throughout the course of the maintenance work. These operations have proved to be very time consuming. Fortunately, experience with potassium vapor systems under the space power plant program showed that, by draining and then holding the system at a temperature of 600°F or more while venting it to a 200°F dump tank, the potassium can be distilled off so that no liquid droplets will remain in the system to present problems in the course of maintenance. The lower boiling point of cesium should make it still easier to distill off residual droplets. Thus, access to components in the intermediate fluid circuit for maintenance should be greatly eased relative to the usual sodium or NaK circuit of a conventional LMFBR, and the cost and time required for maintenance should be much reduced, particularly for the steam generator.

Reactor Safety Considerations

The use of a cesium vapor cycle as an intermediate loop between the primary sodium system and the steam system in an LMFBR has a marked advantage from the safety standpoint. This stems from the fact that the vapor volume in the cesium condenser will be very large and, as a consequence, if a steam leak into the cesium condenser were to develop, the pressure disturbances in the intermediate system would be much less severe than if the system were filled with liquid. In the first place, the reaction between steam or water and alkali metal vapor is much less severe than the corresponding reaction between steam and water and liquid alkali metal. Further, the large volume of the vapor region acts as a buffer that can absorb a large volume of hydrogen released from an alkali metal-water reaction with relatively little increase in pressure.[30, 31]

Cesium Condenser-Steam Generator

The steam generator is one of the most vital elements in an LMFBR system from the safety standpoint. In a typical system of the sort envisioned here the cesium condenser-steam generator unit would be mounted directly beneath the cesium vapor turbine in much the same fashion as steam condensers are mounted under steam turbines.[3] The steam generator tubes would be of the re-entry type indicated in Fig. 10 with the feedwater admitted to the bottom of a central tube about 1/4-in. in diameter in which it would boil as it rose to the top of the tube.[32] The steam emerging from the top of the inner tube would then flow vertically downward in the annulus between the inner and outer tubes and would emerge at the bottom superheated. Cesium vapor would condense on the outside of the outer tube, and a film of liquid cesium would flow down over the outside of the outer tube to the pump at the base of the condenser. This arrangement was designed to avoid large thermal stresses that would be induced by the high heat transfer coefficients inherent in the boiling of water and condensing of cesium if these processes took place on opposite sides of a single tube wall when the temperature difference between the boiling water and the condensing cesium became large under transient conditions.

Leak Detection

If a steam leak into the cesium vapor region were to develop, it will probably be very small initially and will develop slowly because the short-time tensile stress is many times the design stress for high temperature structures, and hence a burst type of rupture is exceedingly unlikely. As a consequence the prime leak detection problem is to detect trace leaks at an early stage. This is quite easy to do in a condenser. By following conventional steam condenser design practice the vapor passages can be made so that non-condensibles are swept to a pocket at one end. In steam condensers an air ejector is used to remove them, but in a cesium condenser a vacuum gage inside a palladium capsule can be used to give a very sensitive indication of the presence of hydrogen. The high vapor velocity (∿300 ft/sec) and the high efficiency for collection and concentration of noncondensibles in a cesium condenser with a system of this type gives a much more effective method of detecting small steam leaks than is possible in a liquid sodium or NaK-heated steam generator. The response is rapid, and there is relatively far less difficulty with hydrogen bypassing the leak detector as small bubbles entrained in the liquid are diluted by dissolution in a large inventory of liquid metal. A very sensitive hydrogen leak detector is available; it makes use of the high diffusion rate of hydrogen through a palladium diaphragm into a vacuum gauge.[31]

Complete Tube Rupture

Although it is highly unlikely that complete rupture of a steam generator tube will occur abruptly, it is still important to consider the effects of such a contingency. Taking as a point of departure a typical system similar to the potassium vapor cycle system of Ref. 26, it was estimated that the pressure in the cesium condenser would increase only about 40 psi in 40 sec in the event of a double-ended tube failure in the steam generator, assuming that no action whatsoever was taken. Certainly this would be a more than adequate time interval for either operator or automatic control action, and thus the effects of a major leak in the steam generator appear to be suprisingly mild and substantially less serious than in a sodium or NaK-heated steam generator.

Capital Costs

The principal cost items in the proposed cesium system are the cesium turbine, boiler, and condenser. The heat transfer coefficients and log mean temperature differences in these two heat exchangers will be about the same as in a conventional LMFBR, hence the surface areas and costs should be about the same. The cesium turbine will be a new item, but its cost will be more than offset by the electric power it will produce. In fact, the increase in the overall thermal efficiency of the plant will lead to an increase in the net output from a given reactor, and this will reduce the overall capital cost of the plant per kilowatt of power produced.

Summary

A review of the above indicates that a cesium vapor topping cycle appears attractive for use in the intermediate fluid circuit of an advanced LMFBR designed for a reactor outlet temperature of 1250°F or more and would have the following advantages:

1) It would increase the thermal efficiency by about 5 to 10 points (from ~40% to ~45 to 50%) thus reducing the amount of waste heat rejected to the environment by 15 to 30%.

2) The higher thermal efficiency should reduce the overall capital cost of the reactor plant in dollars per kilowatt.

3) The cesium can be distilled out of the intermediate fluid circuit to leave it bone-dry, thus greatly reducing the time and cost of maintenance work (particularly for the steam generator).

4) The large volume and low pressure of the cesium vapor region in the cesium condenser-steam generator greatly reduces the magnitude of pressure fluctuations that might occur in the event of a leak in a steam generator tube, and the characteristics inherent in a condenser make it easy to design for rapid concentration of any noncondensibles that may form as a consequence of a steam leak into the cesium region so that a steam leak can be detected easily in the very early stages of its development.

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Table 1. Comparison of Physical Properties of Potassium, Cesium and Water for Condensing Conditions

	Potassium		Cesium		Water	
	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor
Temperature, °F	1040.0		800		115.6	
Pressure, psia	1.50		0.66		1.50	
Specific volume, ft^3/lb	0.02269	267.15	0.0091	610	0.01619	228.65
Enthalpy, Btu/lb	283.0	1170.4	69	267	83.56	1111.8
Heat of vaporization, Btu/lb	887.4		232		1028.14	
Specific heat, Btu/lb·°F	0.1823	0.1266	0.056	0.06	0.998	0.43
Viscosity, lb/ft·hr	0.37	0.0189	0.50	0.054	1.42	0.029
Thermal conductivity, Btu/hr·ft·°F	21.0	0.00363	11.2	0.0055	0.371	0.012
Prandtl No., $c_p\mu/k$	0.00321	0.659	0.0025	0.589	3.82	1.04
Surface tension, lb/ft	0.0041		0.0038		0.00469	

Table 2. Comparison of Principal Parameters Affecting the Design of the Last Stage for Steam, Cesium, and Potassium Turbines

Working Fluid	Steam		Cesium			Potassium	
Molecular weight of vapor	18		132.9			39.1	
Temp. at turbine outlet, °F	102	89	846	800	945	990	945
Press. at turbine outlet, psia	1.0	0.66	1.0	0.66	2.1	1.0	0.66
Specific vol. at turbine outlet, ft^3/lb	330	475	94	125	57	416	606
Sonic velocity at turbine outlet, ft/sec	1275	1260	878	863	904	1740	1690
Isentropic enthalpy drop, Btu/lb	32.4	31.7	15.4	14.8	16.3	60.4	57.0
$\bar{v}/\Delta h_s v_s$	0.00799	0.01189	0.00695	0.00979	0.00387	0.00396	0.00629
Relative diameter for a given power output per stage	1.0	1.49	0.87	1.23	0.48	0.495	0.787
Relative no. of stages for a given stress-limited tip speed	1.0	0.98	0.47	0.46	0.50	1.86	1.76

Table 3. Thermodynamic Cycle Calculations

Fluid	Temperature (°F)	Pressure (psia)	Enthalpy (Btu/lb)	Entropy	Change in Enthalpy (Btu/lb)	Superheat (°F) or Vapor Quality (%)	Specific Volume (ft^3/lb)	Cycle Thermal Efficiency (%)
Cs	1350	25.92	307.0	0.3467	238.0	100%		
	800	0.66	246.2	0.3467	60.8	76.6%		
	800	0.66	259.0	0.3576	51.7	82.5		
	800	0.66	69.0	0.202	190.0	0%		21.7
Cs	1250	15.83	305.7	0.3536	236.7	100%	8	
	800	0.66	253.2	0.3536	52.5	79.6%		
	800	0.66	260.6	0.3590	45.1	84.5%	604	
	800	0.66	69.0	0.202	191.6	0%	0.0100	
	1250	25	94.5	0.2195	25.5	0%	0.0109	19.05
Cs	1150	8.9	304.4	0.3592	235.4	100%	12.5	
	800	0.66	261.0	0.3592	43.4	83%		
	800	0.66	267.5	0.3648	36.9	85.7%	610	
	800	0.66	69.0	0.202	198.5	0%	0.0100	
	1150	20				0%	0.0109	15.7
H_2O	700	2000	1242	1.380	1172	65°F		
		500	1127	1.380	105	89.7%		
		500	1153	1.408	89	93.2%		
	700	500	1358	1.612	205	230°F		
	102	1.0	890	1.612	468	79.5%		
	102	1.0	961	1.720	397	83.7%	280	35.3
	102	1.0	70					
	635	2000	671					

Table 4. Effects on the Combined Cycle Efficiency of Regenerative Feed Heating, Heat Losses, and Power to Auxiliaries for a Cesium Vapor Topping Cycle Coupled to a High Temperature LMFBR

	Cs Turbine Inlet Temperature		
	1150°F	1250°F	1350°F
Ideal combined cycle efficiency, no regenerative feed heating.	45.4%	47.6%	49.3%
Ideal combined cycle efficiency, with regenerative feed heating.	51.4%	53.6%	55.3%
Overall thermal efficiency with pumping and heat losses.	48.0%	50.1%	51.7%

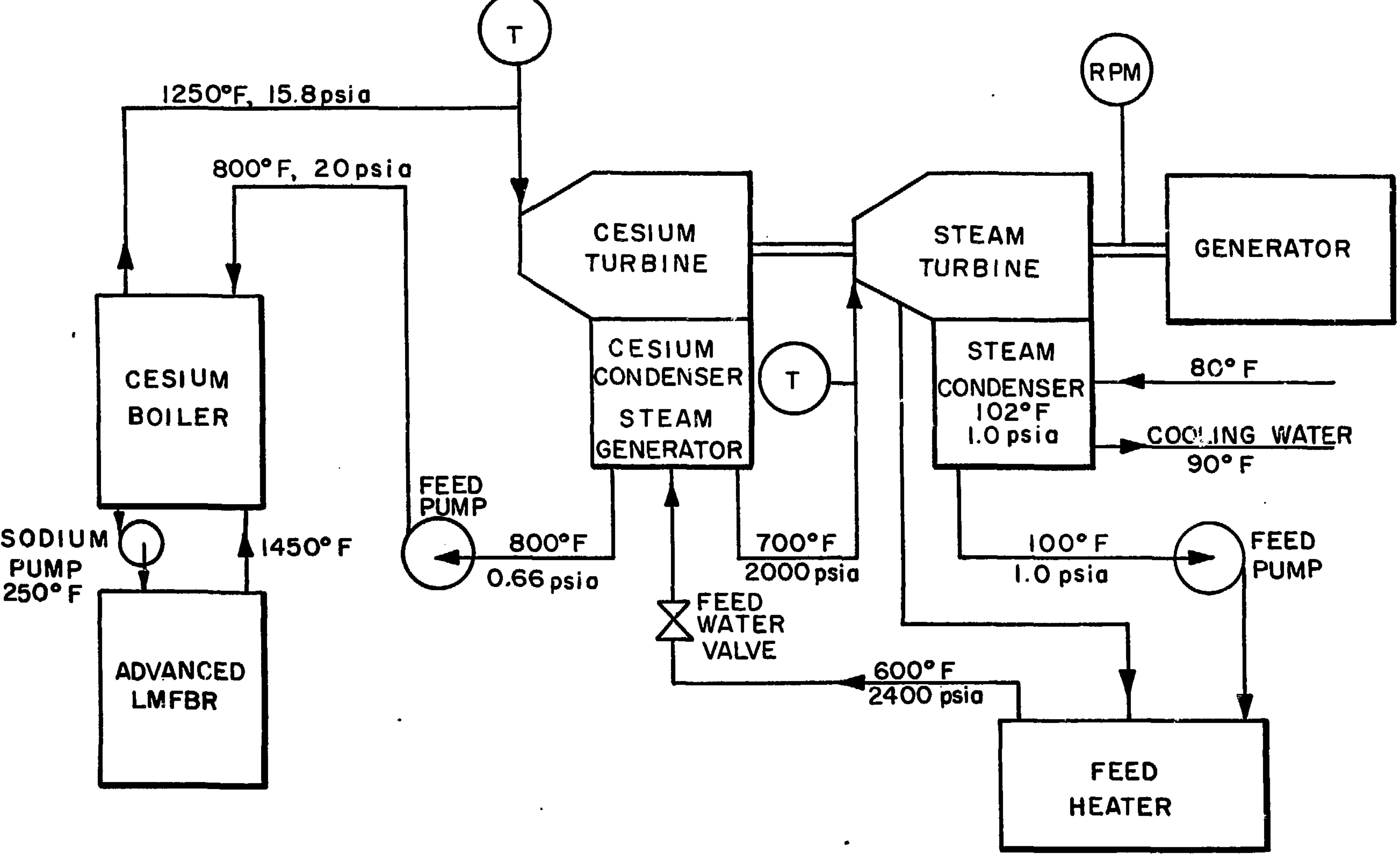

Fig. 1. Flow sheet for the proposed cesium vapor topping cycle system showing the relationships of the major components together with the local temperatures and pressures for a typical case.

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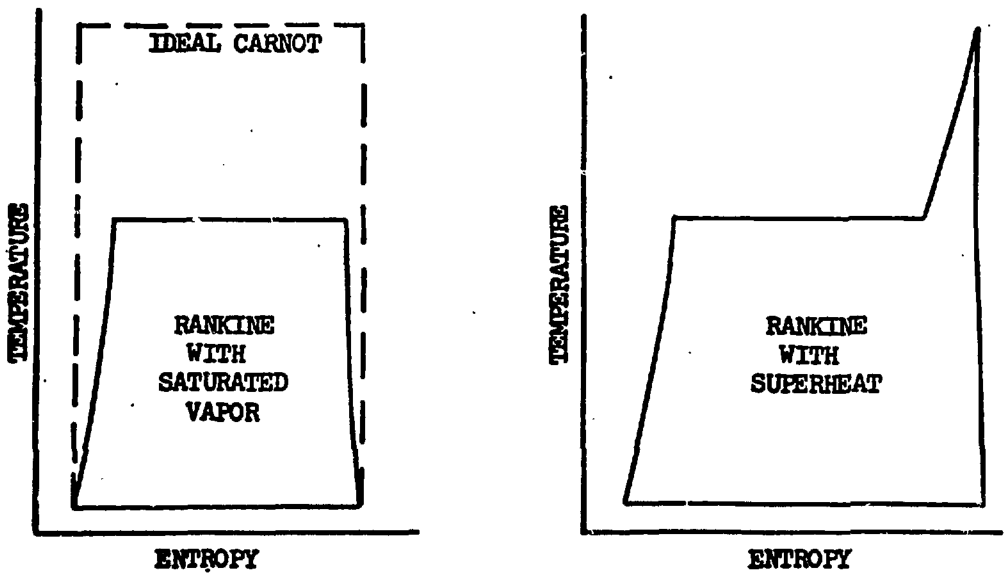

Fig. 2 Temperature-entropy diagrams for saturated and superheated Rankine cycles.

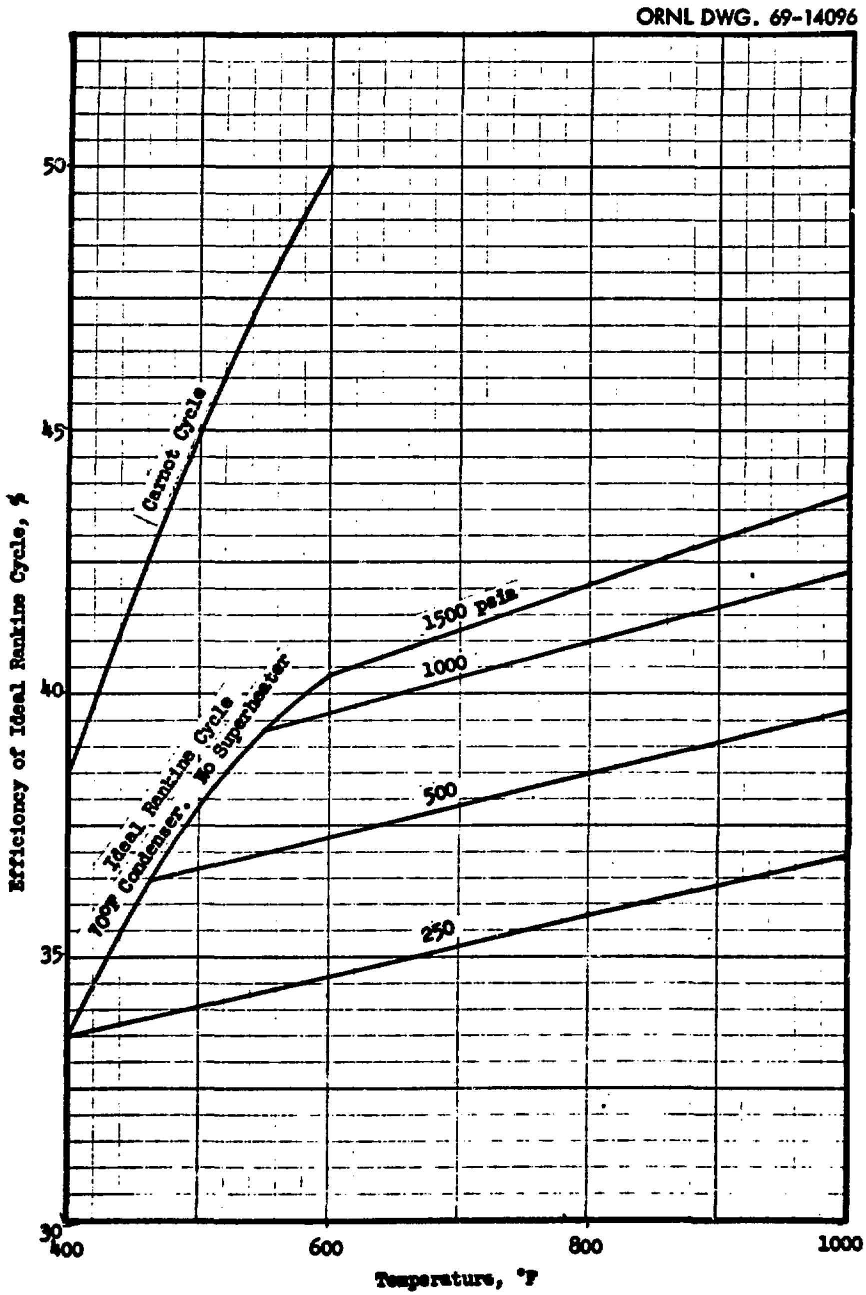

Fig. 3. Effects of turbine inlet temperature on the efficiency of ideal Rankine cycles for both saturated vapor and four cases of superheat at a constant pressure.

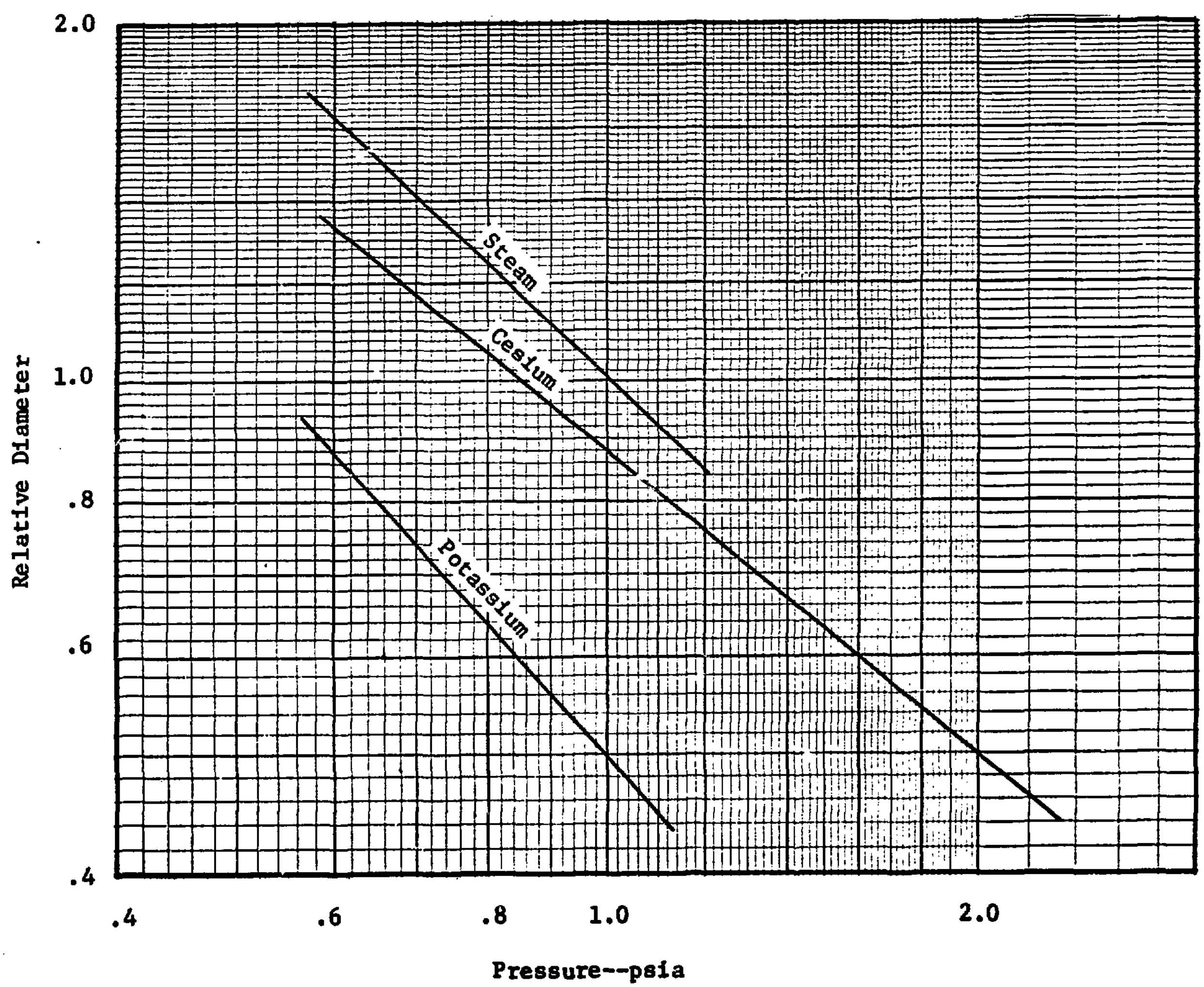

Fig. 4. Effects of turbine outlet pressure on the rotor outlet diameter for turbines having the same Mach numbers in the blading.

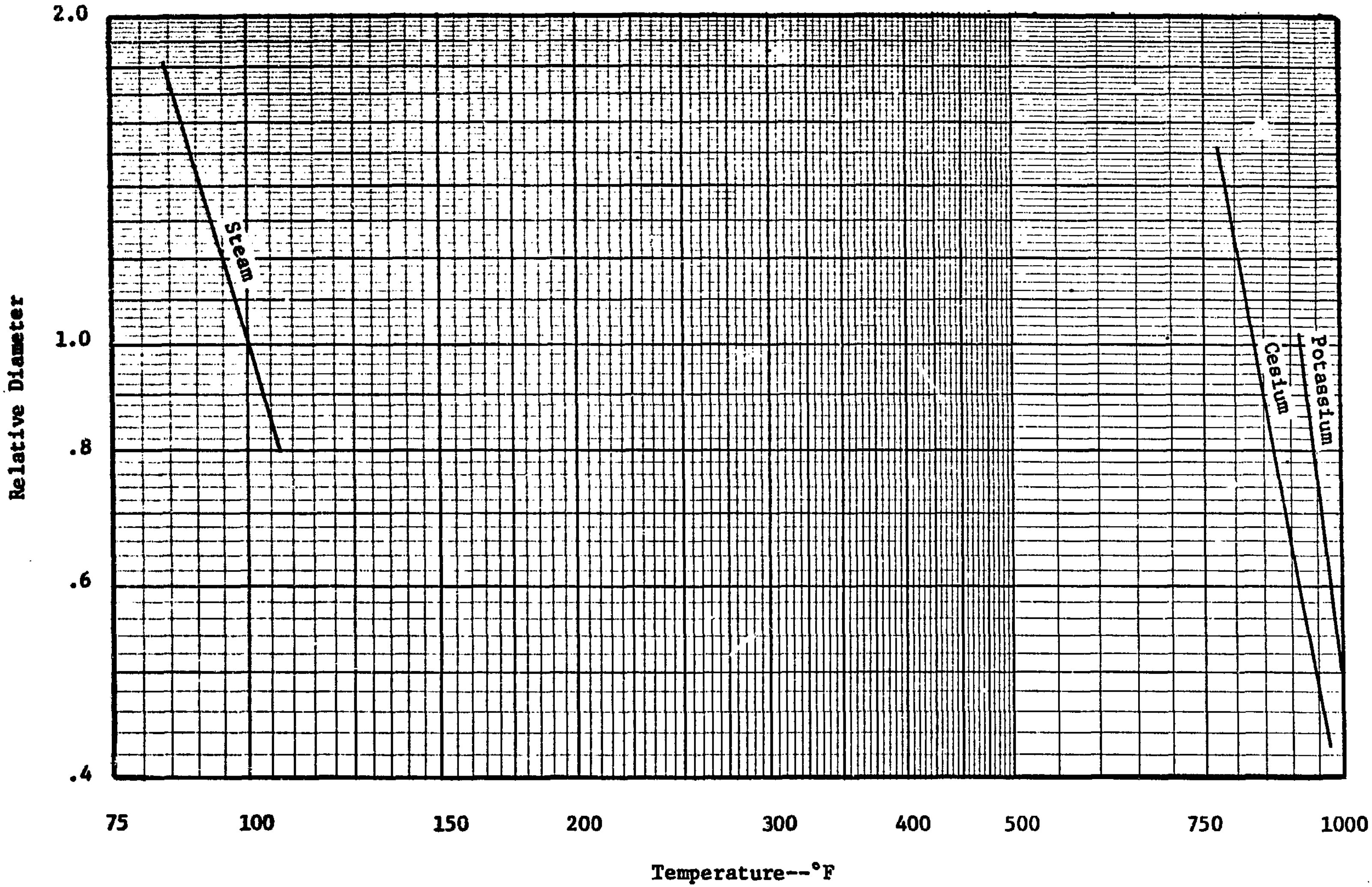

Fig. 5. Effects of turbine outlet temperature on the rotor outlet diameter for turbines having the same Mach number in the blading.

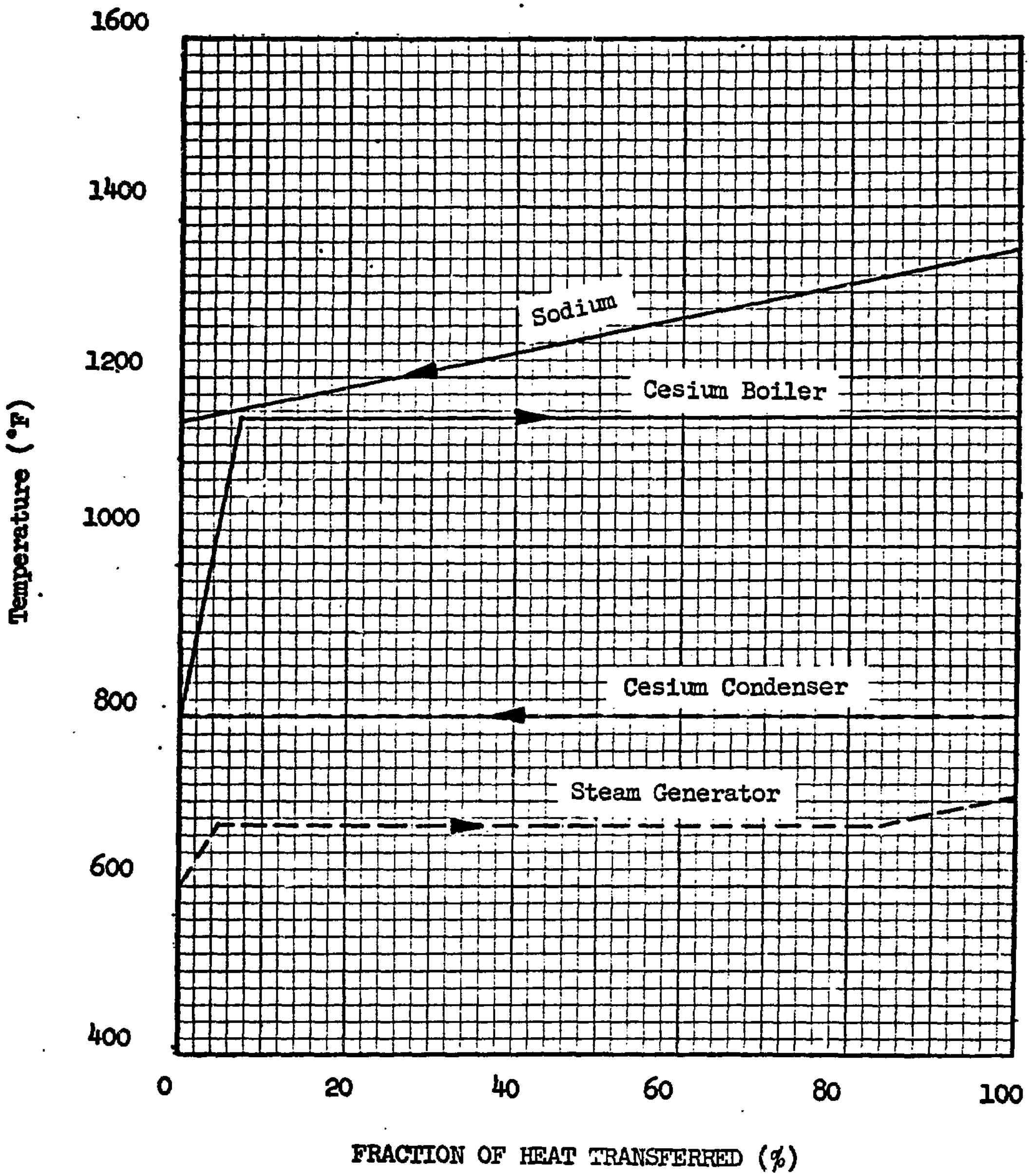

Fig. 6 Temperature Distribution in the Fluid Streams in the Cesium Boiler and Condenser Heat Transfer Matrices.

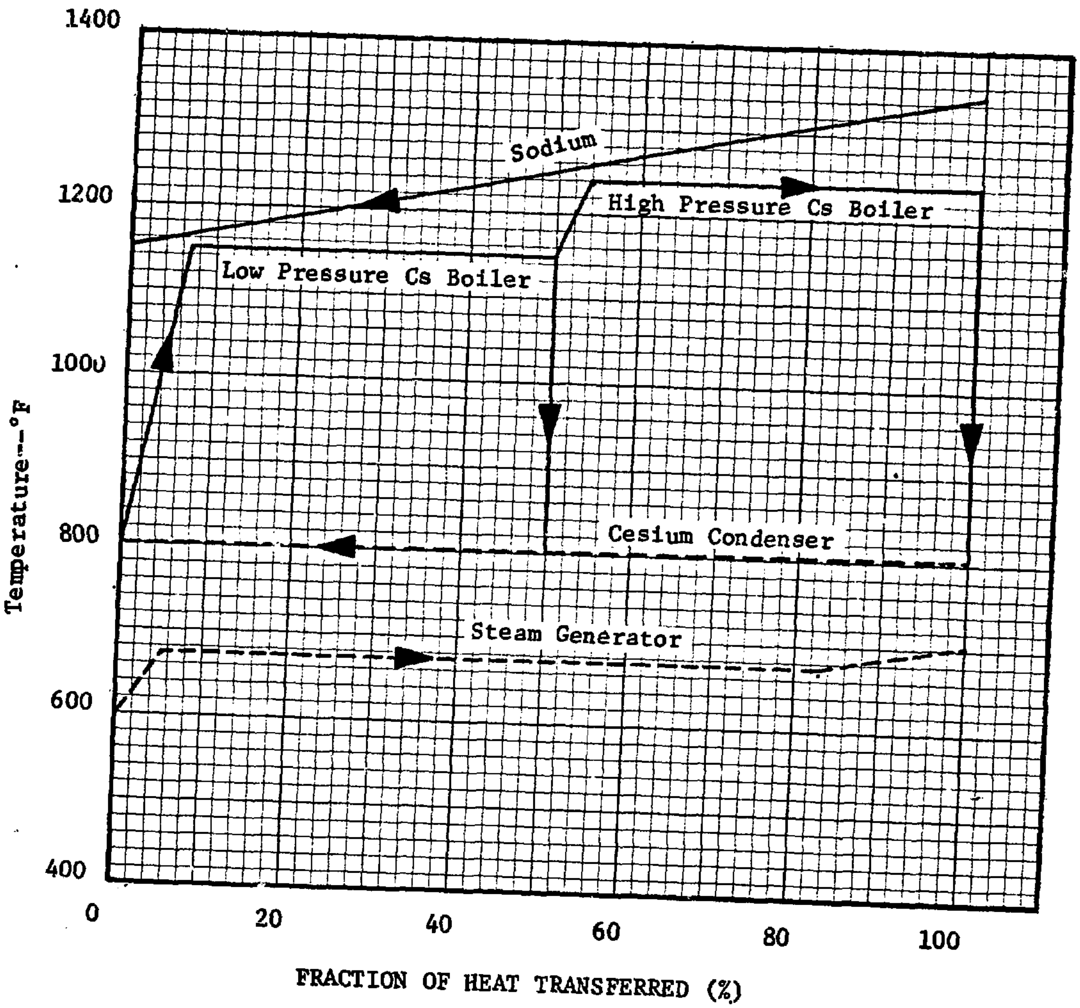

Fig. 7. Typical temperature distributions in the fluid streams in cesium boiler and condenser heat transfer matrices for a dual-pressure cesium boiler system.

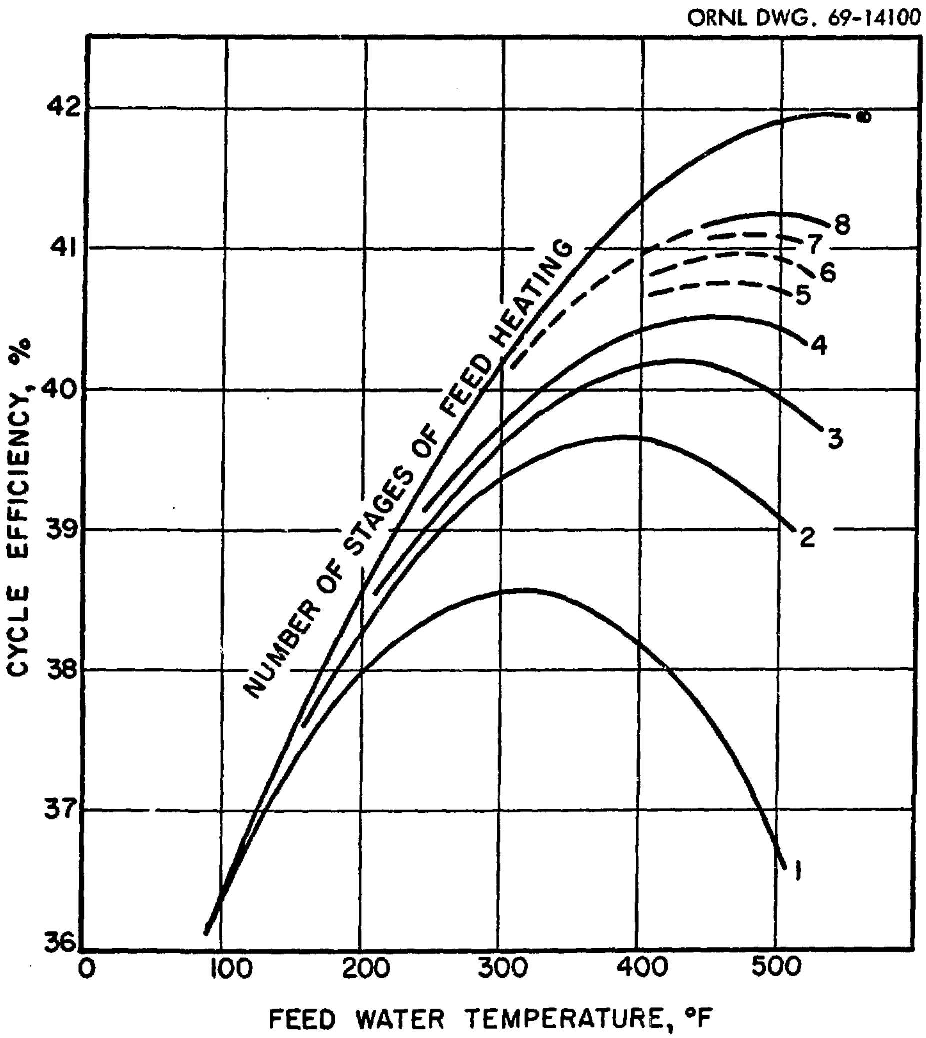

Fig. 8 • Effects of Number of Stages of Feed Water Heating and Boiler Inlet Temperature on the Thermal Efficiency of a Typical Rankine Cycle

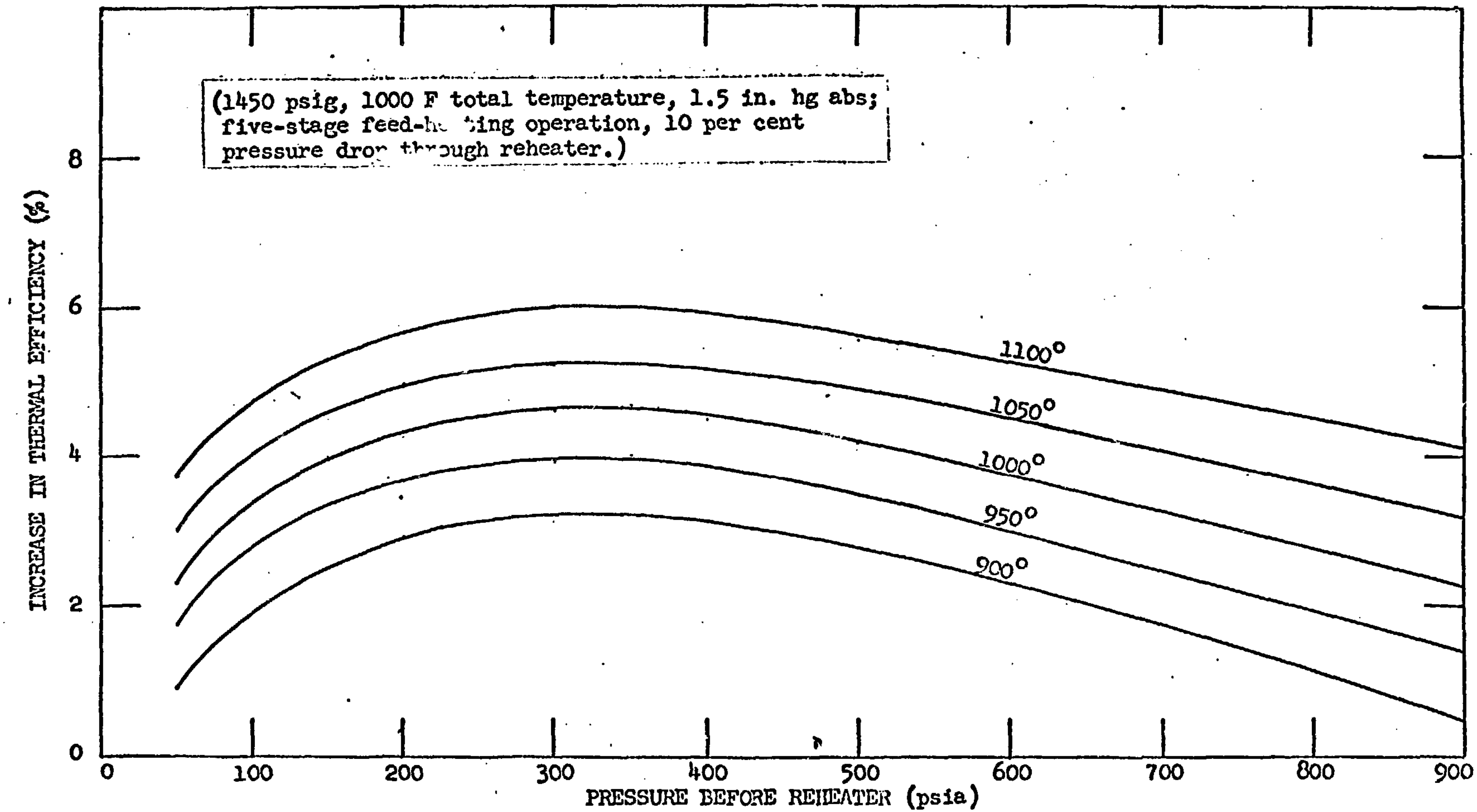

Fig. 9 . Increase in Thermal Efficiency of Steam Cycles Obtainable Through Reheat (Ref.)

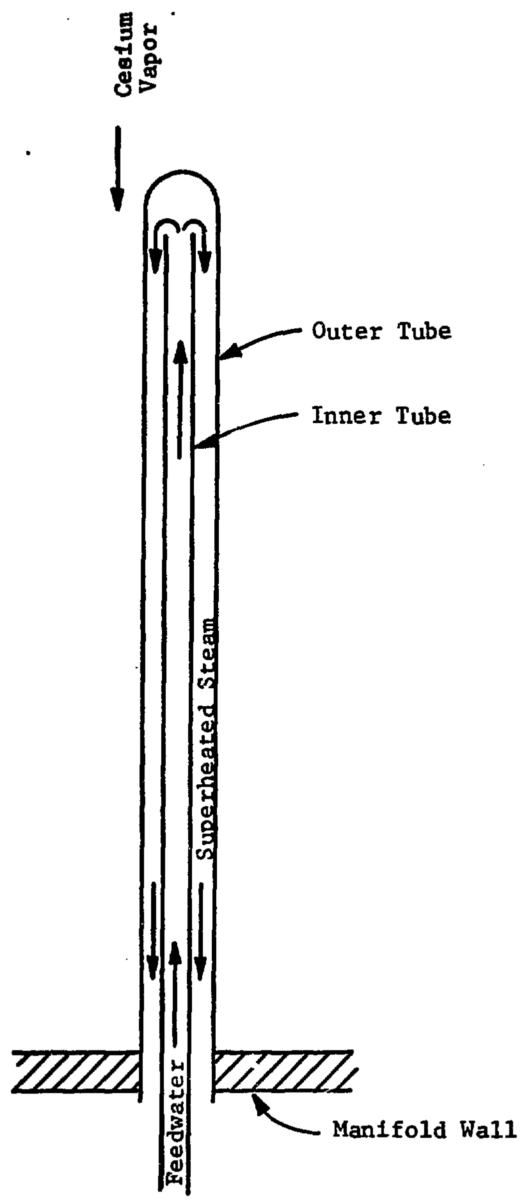

Fig. 10 Section through a reentry tube for the cesium condenser-steam boiler.