This is an informal report intended primarily for internal or limited external distribution. (The opinions and conclusions stated are those of the author and may or may not be those of the laboratory.) ~~This report is not to be given additional external distribution or cited in external documents without the consent of the author or LLL Technical Information Department.~~

UCID- 16437

LAWRENCE LIVERMORE LABORATORY

University of California/Livermore, California

THE SHALLOW SOLAR POND SCHEME

Performance Assessment of a Model System

L. F. Wouters

October 29, 1973

MASTER

Prepared for U. S. Atomic Energy Commission under contract no. W-7405-Eng-48

DISCLAIMER

ACKNOWLEDGMENT

The several members of the fledgling LLL Solar Energy Team are participating in the conceptual, analytic and practical development of the Shallow Solar Pond scheme: Arnold Clark, Jim Day, Bill Dickinson, and this writer. Charles Backus and Frank Edlin are providing consultative advice. My thanks to John Wells, Senior Engineer, PG&E, for furnishing the typical load curves.

Contents (Part I)

(Tables are with text)

Synopsis

This paper discusses the division of energy among the various competing processes, in the sequence from the collection of the Solar flux, to the production of electrical power at a "busbar output", for the Shallow Solar Pond scheme. I also review the basic criteria and the logic for the particular configuration choices made here; this representative "fiducial" system is depicted in Figs. 4, 5 and 6. The Solar collector would utilize shallow flowing water to transfer thermal energy to a hot water reservoir (at $\sim$ 95°C). Several layers of plastic sheet would cover the collection area, to suppress heat losses. A Rankine Cycle thermodynamic system would convert part of the heat energy to shaft work and thence to electricity. It would utilize a Freon gas turbine, along with evaporator, condenser and pressurizing pump; the rejected heat would be removed by an evaporation pond (at $\sim$ 25°C).

The fiducial system used for this analysis, is assumed to have an area of 1 km^2. It would figuratively deliver an output of 8½ Megawatts; its mean efficiency for the reference input is 2.8%. The reference operating point corresponds to equinoctial noon, 33°N lat. (No attempt was made in this initial analysis to include a Summer-Winter optimization). The various losses and power expenses, are summarized in the "power account" of Table I. For some items, the available residue represents a difference between large phenomenological competitors. Other items represent unavoidably necessary "support functions" in the system dynamics. Despite that low efficiency, the salient point of this scheme, is the prospect of unit costs so low, as to more than compensate therefor.

This Part I addresses the complete numerical analysis. The Heat Exchanger design problem became sufficiently complex as to warrant its discussion as a separate Part II.

Note that as a low grade thermal source, this equivalence holds:
<u>1 km^2 Solar Pond thermal yield $\equiv$ 1000 bbls of oil daily.</u>

Table I

System Power Ledger

1 km² pond field

Solar input (Equinoctial noon, 33° N lat.)		900 Megawatts
Cosine elevation factor		750
Thermal Sub-system		
Blanket transmission factors	(-250)	
Fresnel reflection	-180	
IR absorption	- 45	
Absorber back-scatter	- 25	
Net Pond Field input		500
Blanket loss factors	(-170)	
Conduction, Convection	- 55	
Radiation	- 95	
Base conduction	- 20	
Net Pond Field thermal output		330 Megawatts
Thermodynamic Sub-system		
Average continuous thermal power (@ secular factor)		110 Megawatts
Ideal Rankine output	15	
Real Turbine output		10.5
Rejected heat	∿100	
System operation loads	(- 2.1)	
Collector circulation pumps	- .25	
Exchanger circulation pumps	- .6	
TD pressurization and Freon circulation	- .35	
Electrical generation losses	- .65	
Control and utility requirements	- .25	
Net Busbar Output		8.4 Megawatts

Introduction

My purpose here is to assess the Solar Shallow Pond System concept[1], to outline some new ideas and determinative constraints, and to discuss the rationale for a particular set of component choices, among candidate alternatives. The approach is generally similar to that used in the Greenhouse Analysis, but I omit much of the introductory basics presented in that study[2]. The design-economics relationships concerning suitable Heat Exchangers became so involved, that it was advisable to compose this Analysis in two parts. The Heat Exchange issue is not a trivial one.

The applicational, terrestrial, conversion of Sunlight on a major scale on Earth, is tempered by this summary observation: There have been a number of technically competent attempts at "Solar Power" during the past century. None operate today; none led to presently operating systems. Retrospectively, every system seems to have one or two "obvious" shortcomings; every proponent vigorously champions his "answers". The record speaks for itself.

The basic pragmatic issue thus remains: Can one contrive a Solar power configuration, using common industrial resources, which could produce economically marketable power today ? This analysis addresses a representative Shallow Pond configuration assuming that a cost-effectiveness cross-over is forthcoming: By a significant rise in costs of conventional systems (and of their fuels), and by a developmental reduction in costs for this Solar system.

A new influence has also emerged since the original Greenhouse study, at least locally: A conviction that Solar power will "get off the ground" most effectively by the earliest possible technical system demonstration - any reasonable demonstration - even one which might seem, at first, economically unfavorable. To such an end, one then adds the criterion of examining those configurations which generate the shortest and least demanding list of R&D questions.

---

1. Invention Case No. IL-5805; A Shallow Pond . . . System.

2. The Greenhouse Solar Power Scheme, SDK-73-10.

Technically, this scheme belongs to the class of Solar schemes known as "flat-plate non-concentrating systems"; such systems exhibit some performance penalties as a result of the seasonal variation in the ecliptic plane. I make no attempt at a detailed diurnal analysis here; I evaluate system performance at a judiciously chosen location and time. (Arizona, equinoctial mid-day.) Such a critical study would warrant a more painstaking delineation of all system components, as well.

<u>Concept Evolution</u>. The Solar Greenhouse explored the use of <u>air</u> as the thermal and thermodynamic fluids. Here I examine a particular set of geometries which use <u>water</u> to the same ends. (Note also the Voyageur's Guide III.)[3] The Shallow Pond retains a thermal container analogous to the Thermal Blanket, but with less severe technical and practical criteria.

It was evident from the Greenhouse study, that its performance goals generated a list of R&D tasks which was far from "immediately" satisfiable from an early demonstration standpoint. One way to ameliorate this situation, was to reduce the working temperature; hence the "Solar Poorhouse". At 150° C for example, the thermal blanket becomes a much more manageable entity, and one can also take resort in unsophisticated low-pressure steam cycles. A major feature of the Poorhouse which is carried over into the Shallow Pond scheme, is the separation of the collection and storage functions. Whenever the Solar flux falls below a useful level (at night or in inclement weather), the ponds can be drained into an insulated storage reservoir. The residual heat capacity of the dry ponds is low enough that they may cool down without great penalty on start-up.

Information concerning an isothermal Solar Pond project (GE-Dow, 1962)[4] provided the stimulus and justification for taking a final evolutionary step down to $\sim$ 100°C, and converting the collection scheme to what is, in effect, a "water-cooled" flat-plate interceptor.

Concurrent examinations of the stratified "deep" pond scheme (á la Tabor)[5] suggested that it also seemed to generate a long list of R&D questions. A shallow flow-pond, with a separate high-capacity storage element, appears to resolve many of the problems as seen from this viewpoint.[6]

---

3. Solar Energy - A Voyageur's Guide, Part III - "Starting Over" (October, 1972)
4. Private communication, F. Edlin
5. Solar Pond Project, Tabor and Matz; Solar Energy, Vol. 9,4,p.177 (1965)
6. Solar Ponds Extended, A. Clark, UCID-16317

Of course, such a conceptual trend has brought with it a significant thermodynamic penalty. The achievable Carnot temperatures are at once "locked in" to certain particular limits - by the boiling point of water on the one hand (T_1 = 100°C), and by the ambient atmospheric temperature on the other (T_2 = 20°C). The maximum possible Carnot efficiency is then about 20%; real thermodynamic systems will do much less than that (∿ 10%, with luck). Taking into account the limitations in collector performance, and also, a host of small peripheral power penalties, the overall efficiency prediction is about 3%. The salient point of the Shallow Pond scheme, is a perceptible prospect of unit costs so low as to more than compensate for this low efficiency.

Overall System Considerations

In the course of system development, it is often the case that the system definition at the end of the exercise is noticeably different than at the beginning. I tried to avoid this in the Greenhouse analysis by adopting a "fiducial module" as a case example - not necessarily an optimum one. Material and configurational variations and alternatives could then be evaluated relative to that fidu case. The slippery situation which I encountered thereby, lay in the initial assignment of a particular output number (like 10 Megawatts). Our analysis schemes are still too causal* and elementary to readily accommodate such a defined end-point. There are too many undetermined parameters operating in series. For this exercise, I will instead define a fiducial input: The collection area - One square kilometer.

$$A = 1\ \text{km}^2$$

This turns out to be a fortuitous choice from the standpoint of the design of several of the system elements. This dimension is also the nearest metric unit to a 10 Mw configuration assuming:

	ϕ = .07 watts/cm^2	(peak solar flux)
	σ = 1/3	(secular factor)
	ε = 3%	(overall efficiency)
Find:	A = 1.4 km^2	(for 10 Mw)

In the early stages of this exercise, it became clear that there were much closer relationships between the choices and designs of the individual system components, the economic trade-offs, and the performance of the complete system (than was the case in the Greenhouse). This is in part, a consequence of those "locked-in" Carnot temperatures; that considerably simplifies an analytic exercise (Temperature is no longer parametric), but it seems to bring in an embroidery of applicational possibilities and of alternative system configurations. It is also a consequence of the relative conservatism of the entire concept, and of its components. These are nearly all "off-the-shelf" - or constructible from off-the-shelf materials.

* not "casual" !

Without as yet specifying a fidu system, I reference Fig. 1, which is a simple block diagram of the major sub-systems. The insertion of the hot water thermal storage has major system design and performance implications: It effectively decouples the thermodynamic machinery from the Solar flux collector, at least on a daily secular basis.

Next, I outline some of the interesting relationships between:

1. Solar Pond thermal gradient
2. Buffer/storage location and technique -
3. TD sub-system configuration and fluid choice
4. Market address:

   Demand profiles, price premiums, location factors

This is perhaps our first encounter with the relatively delicate influence of the customer (for Solar power, at least). I suspect this is partly because the fixed-temperature framework removes the coarseness of gross parametric technical variations within the system, so as to make more apparent the impact of exterior, less-technical factors, such as $. Slightly different technical alternatives are more easily seen to better match different power demand models, or singularly restrictive resource situations.

An implicit feature of both the thermal and thermodynamic segments of the fidu system described hereunder is an accommodation for a relatively broad range of operating conditions. The daily integrated "fair weather" solar input may vary by a factor of 5 or 6 during a year, and the market demand certainly fluctuates by a factor of 2 - and not necessarily "in phase". Hence the power "thru-put" - and possibly the Carnot temperatures - will depart noticeably from this "single-point" analysis. The essential point of accommodation in the system, is the Freon turbine. The experience and design indications for such machines show that they can have an exceptionally broad range of efficient performance and of input and load tolerance, around their specification point. This initial analysis thus does not attempt a more delicate economically-oriented optimization. That is sufficiently complex to warrant a fairly sophisticated computational attack.

<u>The Thermal Gradient</u>. There is a fundamental thermal number for Solar Pond systems: The maximum rate of heating of a unit volume of water, assuming total flux absorption in it. (Either by close contact with a black interceptor surface, or by virtue of an opaque dye. It can be a moving element, as in the Shallow Flow scheme, and the number may be representative of an average, as in the case of mildly turbulent flow:

$$\frac{\delta T}{\delta t} = \frac{\phi}{\rho C_p} = .018°\text{C per sec.} \quad \text{(per cm depth)}$$

$$\underline{\text{or} \sim 1°\text{C per minute}} \quad \text{(max.)}$$

(Since ρ and C_p are unity in metric systems, the numerical value is the same as the Solar flux number.)

This is an "unappealable" number; it is certainly less, but it cannot be greater. Evidently if we start with 20°C water one cm deep, after about 4500 seconds (1-1/4 hours), the water is up to 100°C. (Provided that it is well insulated.) Beyond that, the Solar radiation starts being converted into latent heat of evaporation, with no further benefit to the thermal operation, and very likely some mechanical detriments. We can leave it out there longer, if we make it deeper; thus a quiescent pond ~ 8 cm deep can just about absorb one day's solar input starting from "scratch", and suppressing all losses. This is well validated by the experience in Solar still performance.[7]

Shortly, I will discuss the large penalty which the scheme suffers from every <u>avoidable</u> thermal and thermodynamic loss. Nevertheless, there is an <u>unavoidable</u> penalty due to such heat exchangers as may be required. Even with high fluid circulation rates, there must be some temperature differences across heat transfer surfaces, and in extracting thermal energy from the pond water. Assuming a low-loss thermal reservoir, the water returned to the pond cannot be much colder than the water discharged by the evaporator. It is sometimes argued that the thermal transfer fluid itself should somehow be cooled all the way down to near ambient, before being returned to the collector. In a thermodynamic analysis, it is easily shown that, for "simple" (one-loop) systems, this is essentially "impossible".

---

7. Solar Distillation, etc., Carl Hodges et al, Univ. of Arizona Solar Energy Lab, January, 1966

It is crucial to recognize that the Solar radiant absorption efficiency is imperceptibly affected by the mean water temperature. The effective source temperature is $\sim$ 6000°K; it is immaterial if the interceptor is at 293°K or 373°K.

The substance of these gradient arguments, is that, in a simple fidu system, the shallow ponds must be virtually isothermal. Assuming a 3° temperature excursion, the hot water recovers that heat in about 10 minutes of solar exposure. It must be a fast-flow system.

The economic circumstances pertinent to the competition between photon capture efficiency in the collector and thermal leakage from it, are summarily discussed in UCID-16386, Pt. 1[8]. For a given state-of-art, there appears to be a broad optimum in collector configuration and design temperature. That competition "cuts off the Sun" in designs attempting to achieve higher-than-optimum collector temperatures, such that collector efficiency falls drastically. It is fortuitous that today's materials technology leads to an optimum temperature domain not much above the boiling point of water. A considerable advance in mass-produced optical materials is required to predictably improve the situation. And then of course, water may no longer be an appropriate thermal medium.

---

8. On the Economics of Thermodynamic Solar Power Systems, September, 1973, Part 1 - The Efficiency Function.

Storage. Already today, a significant economic relationship can be argued as between the power market, and efficient, convertible storage. The impact of "Storage" per se thus goes further than Solar Power. For example, consider the persistent daily peak load requirement in strongly industrialized regions. The availability of a high-capacity, reliable, reasonably compact storage means would make possible a local "in-house" time-shift of energy bought at cheaper "midnight prices", to this daytime peak. As another aspect, a suitable storage device (e.g., a "super-flywheel") could reduce the severe transmission losses associated with peak demands, by serving as a local reservoir for peak shaving.

In such applications, we speak of very substantial energy storage quantities, of course. For example, consider a 20% saw-tooth peak on a 1000 megawatt network, with a base time of 6 hours. That corresponds to a storage requirement of 2×10^{12} joules (or about 1/2 kiloton of energy). If a singular storage accident releases this energy in 10^{-3} sec, one has the equivalent of a small nuclear explosion. A near-urban storage location would evidently necessitate special installation precautions.

There is one technical system feature which bears especially strongly on storage as applied to Solar power systems: The interaction between the system conversion efficiency and the storage point. If the collection efficiency is 5% as seen at the busbar output, then storage at the collector requires 20 times the capacity needed at the busbar, and it must be 1/20 as costly, per joule stored. As a corresponding opposite limit model, consider now a 1000 megawatt Solar plant, with three days full-output storage in the thermal loop. This calls for a total storage capacity of 5×10^{15} joules ($\sim$ one megaton !). Aside from the somewhat appalling physical implications with respect to meaningful 24-hour "Solar power", these numbers are quite striking in once again exhibiting the awesome dimensions of the existing U.S. energy complex.

A first examination thus suggested that perhaps we might examine two system alternatives here: (1) A true 24-hour, strongly-buffered system, and (2) A lightly-buffered, "peak-shaving" system. But the technical and market interactions in this latter situation appear sufficiently intricate as to warrant its separate consideration.

A conservative storage choice for a 24-hour water-based system, is a hot water reservoir. While its storage density is very low, it is probably the most straightforward and cheapest contemporary scheme, and it connects naturally into a shallow Solar pond concept. In essence, the storage feature of the deep Solar pond (e.g., per Edlin) is separately and more efficiently provided, and nighttime collector losses can be essentially eliminated. It seems reasonable to extrapolate known reservoir technology to the dimensions and temperatures involved here, although we may be in for mild surprises.

From a thermodynamic analysis standpoint, such a configuration has the side benefit of furnishing relatively fixed Carnot temperatures. The daily "ripples" due to the Solar input peak, and the power withdrawal peak, are, percentagewise, small. Conversely, a real-time, lightly-buffered system undergoes large performance fluctuations, the design-analysis problems are much more complex - and so too, are the performance problems.

Some Thermodynamic Issues. UCID-16386, Pt. 2[9] discusses the efficiency-cost tradeoff with respect to turbines suitable for thermodynamic Solar power systems. It is shown that this tradeoff is of order 30:1 - Each % gain in turbine efficiency is worth up to 30% in price (!). It is an argument which may be generalized to any downstream aspect with similar tradeoff impact - such as the Carnot temperatures themselves and the thermal differences in the heat exchangers. It would be best if the collector thermal efficiency could be improved - but the component price trade-off is nowhere near as attractive: A 1% efficiency gain is worth only 1% in permissible collector price increase, roughly speaking. That does not mean that such improvement isn't worth it - only that for the thermodynamic sub-system, one can in effect, throw away the price catalog.

---

9. On the Economics of Thermodynamic Solar Power Systems, September, 1973 Part 2 - The Turbine Trade-off.

Aside from the low Carnot efficiency, the narrow temperature domain (20°C to 100°C) presents certain special problems in choosing and thermodynamically manipulating a suitable fluid. Most TD power machines basically use the Rankine cycle or some derivative thereof, so that one wants as well a fluid which can be condensed just above $T_2 \sim 20°C$. Hence the vapor pressure ratio is likely to be low, which requires that the gas volumes and the machine elements (rotors, stators, nozzles, ducts, etc.) be very large. There is some economic compensation in the lower rotor speeds and in the avoidance of exotic high-temperature alloys.

Past experience with water as a TD fluid in this domain, has been unattractive (at best). The enthalpic (Joule) efficiency is very low ($\sim$ 4%) as a guideline limit; real TD machines (e.g. turbine bottoming stages) achieve perhaps 60% of that number, or $\sim$ 2-1/2% net TD efficiency. The alternative is to use an exotic fluid - in particular, one of a limited group of Freons, per Table II. As indicated there, the enthalpic TD efficiencies are noticeably higher ($\sim$ 14%), and one can hope for as high as 70% TD machine efficiency. Even though it may be a "custom job", the design of appropriate Freon trubines appears to be a reasonably well-practiced procedure ; note Appendix A.[10]

As configured here, the fidu scheme is to use heat exchangers at both T_1 and T_2, to accept heat from the thermal energy source for mechanical conversion, and to reject the residual heat to some exospheric cold sink, such as the atmosphere. The exchanger membranes permit complete physico-chemical isolation between the media, but they extract a penalty in cost, in pumping power, and in thermodynamic losses.

The design problem is compounded by the relatively large constant-temperature thermal excursion in the evaporation-condensation legs. The available temperature intervals for manipulating heat are small, so that large, quasi-isothermal volumes must be moved around, also threatening economically unacceptable pumping power demands. These various heat transfer problems are examined in some detail in Part II of this study.

10. Geothermal Power Plant on the Paratunka River, Moskvicheva and Popov, UN Symposium on Geothermal Resources, Pisa, 1970. Vol. 2, Part 2, p. 1567 (Geothermics/1970 - Special Issue 2) (This Soviet installation uses a 500 kilowatt Freon turbo-generator)

And then there is the rejected heat problem. Thermodynamic power plants in general, suffer today from the environmental problem of getting rid of the degraded, but excess, thermal power removed by the T_2 exchanger (or condenser) in Rankine cycles. If anything, a Solar power plant suffers from the inverse problem: In the general neighborhood, it would return 90% to 95% of the energy otherwise delivered by the Sun. The other 5% to 10% is exported to a different neighborhood. There are incidental quantitative complications due to changed albedos, changed convective air currents and so forth.

From a technical systems standpoint, it is difficult to escape the broad conclusion that the ultimate "dump" is indeed the atmosphere. (Unless the plant is on a sea coast.) Increasingly, large water-cooled inland power plants are being forced (ecologically) to insert thermal holding reservoirs or lakes. Their dominant long-term cooling mechanism is latent heat loss due to evaporation. Seen this way, the SMUD evaporative towers have an admirable, direct, functional simplicity.*

Market Address. There is a sort of "causality" in the address by scientists to energy systems considerations: Obviously one begins at the input and ends at the output. Accountants and investors have a very different view of "beginning" and "ending": Considering $ as the vehicle of interest (rather than energy), one begins at the output and works forward to the input ! Hence as soon as one entertains "Market Address", some attention must be paid to this different viewpoint.

From the inception of efforts towards applying geo-solar energy sources (Solar, wind, OTG, etc.) to modern power needs (actually starting almost 100 years ago), there was a strong awareness of the real-time mis-match between the variations of natural origin in the source power contributions, and the demand variations at the converted outputs. A variety of approaches have been considered for rectifying this, some technical, some economic, and even some dialectic. The first-order resolution (from the marketer's standpoint) is to simply specify that a Solar system should sell power in a manner completely competitive with conventional power producers (fossil, hydroelectric, nuclear). Besides costing no more, it should have essentially the same flexibility, and equivalent 24-hour capability.

---

* Sacramento Municipal Utility District, Nuclear Power Plant (S.E. Sacramento Co.)

That has invoked a wide variety of hopefully compatible energy storage schemes. (Also labeled "buffering" or "ballasting".) The available and reliable schemes today are very cumbersome and uncertainly expensive, such that one cannot face a "24-hour" requirement with much confidence or enthusiasm for very high power levels, at this writing. But in all this, there is an implicit (and unwarranted) assignment of adaptability as a feature of conventional power plants, which in fact, they don't possess either. In the past, power companies have gone to great lengths to try to operate their fossil and nuclear plants at constant loading, for reasons of technical necessity and operational economy, by:

Selling commercial/industrial power at cut rates at night,
Promoting electric heating (in winter) in summer A/C areas,
Peak shaving auxiliaries,
Load switching ("juggling"), so that the hydroelectric segment bears the brunt of load variation. (It has the largest and easiest load/cost latitudes.)

There has thus developed a psychology of "habit" or convention, which may be as much as obstacle in "selling" unbuffered Solar power as may be any particular technical issue. One can argue oppositely - that the ultimate energy demand and fossil fuel costs will become so critical, that power companies will buy any new busbar power for distribution at any time of day, and in particular, while the Sun shines. It is an interesting observation, that we have had a 24-hour Society, simply because we were able to easily exploit Nature's fossil energy reservoirs. Perhaps the most persuasive argument for producing and selling "real time" Solar power (no delayed-sales storage, per se), is that it might supplement oil-burning power plants in such a way as to significantly reduce fuel costs. Arguments have been made that, at $\sim$ \$7.-a barrel, it becomes economic to modify oil-burners so that their outputs can follow the real-time demand. The hydro plants then also begin to look more like "free fuel" sources, and they would be run "wide open" all the time instead.

A third view of market influence is to examine the typical metropolitan power demand curves, to see if there are some particular ways in which Solar plants could assist the established power grids in conjunction with some intermediate grades of time-delayed buffering. Figures 2 and 3 depict two typical load days, one in late summer, one in mid-winder. Both show noticeable, delayed, load peaks, relative to typical corresponding Solar inputs. (Dashed curves.) Such peaks are particularly costly to satisfy from remote conventional power plants, because the fuel requirement then approaches P^2, rather than P. The reason for this is that the increased transmission voltage drop requires a considerable increase in busbar voltage, on top of the increased output current.

A total energy supply system might utilize appropriately buffered Solar plants to satisfy these delayed peak demands as their principal function. This specification significantly reduces the requirements on the storage element, as compared to a 24-hour operation. Ultimately one might think in terms of a power grid in which existing oil-burners would be modified and operated primarily to furnish back-up power during singular events, such as extensive storms.

Concerning Basic Data. There is an extensive discussion of applied "source material" concerning pertinent thermal and thermodynamic processes and substances, in the Greenhouse Analysis (SDK 73-10, p.20). That remains generally applicable here; the emphasis on particular liquids as working media only changes the numerical and dimensional range of fluid physics Such supplementary data as are appropriate will appear as that is relevant to the development here, and in Part II.

Units. The duality of unit systems in applicational work persists. It is convenient to use both metric and English units, depending on topic. Some key conversions include:

Transfer Coeff: 1 cal/cm^2 sec °C = 7370 BTU/ft^2 hr °F

Thermal Content: 1 cal/gm = 1.8 BTU/lb

Materials Properties - WATER. Since water is a "big thing" here, it is useful to put down some of its basic thermal properties:

Thermal Conductivity	$k = 1.43 \times 10^{-3}$	cal-cm/cm^2-sec °C
Density	$\rho_o = 1.0$	gm/cm^3
Specific Heat	$C_p = 1.0$	cal/gm °C
Diffusivity	$\nabla = 1.43 \times 10^{-3}$	cm^2/sec
Viscosity	$\eta = .01$	$\frac{\text{dyne-sec}}{\text{cm}^2}$

System Description - The Fiducial Module

This Solar Shallow Pond scheme is examined in terms of three sub-systems:

A thermal input loop
A thermal storage element
A thermodynamic conversion loop

The heat transfer agent among these, is hot water. There is really a fourth sub-system - the thermal rejection element - but it is treated rather briefly below, as part of the TD sub-system.

There are some interactive design aspects in this scheme; as already mentioned, the strongly inertial thermal storage does decouple the two operative loops to a certain extent.

This fidu system is illustrated in Figs. 4 and 5. With respect to the several impact aspects noted earlier, it involves:

A low-gradient, fast-flow, shallow pond field
An "overnight" T_1 hot water storage reservoir
A relatively conventional, Freon, Rankine-cycle turbine
An evaporation-cooled T_2 lake
Market emphasis on general, 24-hour power supply

Thermal Sub-system - The Solar Collector. This is the major element of the thermal sub-system; the pond field consists of an array of flat, shallow, flowing-water troughs. A variety of fabrications are possible for a trough. One version involves a composite plastic sheeting layed on a compacted earth base with cemented curbs. The trough configuration indicated here (Fig. 6), consists of pre-fabricated plastic boxes, layed end-to-end on a graded, compacted base (no curbs). The butt ends are shaped to act as water leveling weirs or "riffles". Their purpose is to eliminate bare "hot-spots" due to small grade changes. The water head to be supported will be quite small; the water is to flow as a sheet about 1 to 3 cm deep.

Each trough is to be covered by a transparent "blanket", as in the Greenhouse scheme. Here it is shown as consisting of three semi-rigid sheets of thermally-stable, transparent plastic; these sheets would be assembled as a "sandwich" integral to the pre-fabricated box and separated and secured by means of an internal plastic "egg-crate" web. As before, the purpose of such covers is to inhibit convection and radiation losses.

In the alternative scheme, the blanket would consist of 2 or 3 plastic film sheets, supported by low air pressure, and fastened to the curbs along the trough edges. In a representative, 1 km^2 fidu scheme, that would require about 200 miles of curb-edge installation. The economic merit of the prefab sandwich may thus lie in eliminating these 200 miles of detailed field assembly, as well as in eliminating the associated air pressurization connections, edge sealing and air blowers.

The collector for a Solar pond system or "field" covering a square kilometer, would thus be subdivided into separate small ponds, for operational and maintenance reasons. (Flow regulation, cleaning, repairs, etc.) The analytic problem is then somewhat analogous to that of the Greenhouse, in that we are to determine pond length and width, water flow rates, number of ponds, and so forth.

The Pumps (and plumbing - flumes and weirs, actually) continuously circulate the hot water between collector ponds, storage reserviors, and heat exchangers, during the daily insolation period. As already noted, thermodynamic conditions impose very small permissible temperature differences, such that the pond field must operate in a quasi-isothermal, fast-flow mode. The pond water is replaced several times per hour. The pumping load is sufficiently critical that some design care is needed to keep it from becoming a major output penalty.

The Thermal Reservoir. This hot-water storage element is to supply a heat source for nighttime power generation and during inclement weather. Its thermal capacity therefore should be equivalent to several days of integrated insolation. Thermal energy would be withdrawn at about one-third the peak solar input, in order to maintain a balance between continuous power withdrawal and the secular solar input.

Again in the interests of minimizing pumping power, the system is tightly configured, and this is especially evident in the reservoir arrangement. We cannot afford to move the necessary quantities of water very far; here the reservoir also serves as part of the hot-water distribution scheme.

Thermodynamic Sub-system - The Freon Turbine. The fidu configuration utilizes a very conventional, closed Rankine cycle system. About the only special thing about it is the working fluid, which is one of the Freons. Several TD system concepts were examined, including one using a two-phase water machine (flash nozzles and bucket turbine). That one may not work at this low water temperature because most of the available heat goes into the latent heat of vaporization of the vapor fraction. The subject is not closed, but at this writing, the Freon loop appears to generate the shortest list of questions and problems. This sub-system includes:

an evaporative heat exchanger,
an impulse turbine (and electrical generator),
a condenser (with cooling lake)
a liquid-phase pressurization pump,
a TD fluid buffer tank.

The operation of this loop is a standard textbook exercise; it will not be further described here.

The subsequent analyses of these sub-systems are intended to be representative of the first-order procedures, and the numbers are typical rather than unconditional. The problem in more precise definition lies in the fidelity of detailed mathematical descriptions. The calculational task then becomes formidable enough to necessitate computer methods. The main thrust of these analyses is to develop justifiable entries, having plausible embodiments, for the Power Ledger of Table I.

Thermal Sub-system Analysis

Later, in the thermodynamic analysis, I will examine this thermal constraint: The various thermal transfer temperature increments need to be unconventionally small: 2 or 3°C. This implies comparatively large water volume flows, large thermal transfer areas, large thermal reservoirs, with relatively small temperature excursions. In conventional power plants, such features would be considered as unacceptably expensive technical luxuries, but here they are exceptionally cost-effective.

In a strict loop model, the thermal reservoir is part of both the thermal and the TD loops. But here, by inference, I may treat it as a thermal inertia so large, that it is a "constant-voltage battery". No matter how much or how little heat I take and return, in one day, the terminal temperatures are imperceptibly affected.

So I examine this input element as a thermal source tied to an infinite, constant-temperature sink.

The Shallow Pond Field. Evidently, when a Solar power plant gets big enough, "pond" is hardly appropriate as a descriptor; the collection area needs to be subdivided, for flexibility and convenience in operation and maintenance. (One might call it a "Solar lake", but that still infers an integral body.) The design/analysis problem here is to outline some logical technical criteria for subdivision, and to develop pertinent thermal specification numbers for a one km^2 fidu field. (Net thermal output, losses, flow requirements, etc.)

The fidu system is essentially a steady-state scheme, both thermally and thermodynamically. During the Solar day, the pond field must be operated so that the captured Solar energy is transferred out as fast as it enters, or else the water boils. The neatest way to do this, is to control the water flow rate through the pond field.

The Input. The mass flow must be sufficient to remove that part of the input Solar power which ultimately ends up in the water. The total, clear-day, vertical, peak input at 33°N latitude (Arizona), is:

$$\underline{\phi A = 900 \text{ megawatts}}$$

or $$\underline{2.2 \times 10^8 \text{ cals/sec}} \qquad (3.0 \times 10^9 \text{ BTU/hr})$$

A number of mechanisms operate to significantly reduce the water-borne power level; these may be analytically structured as two accountable thermal loss factors:

1. The captured fraction of input photons, accounting for geometry, for scattering and absorption in the blanket layers, and for absorption efficiency into the water. (η_p)
2. Thermal losses through the blanket due to radiation, conduction, and convection. (η_ℓ) Conduction loss into the ground can be made small.

These phenomenological features are depicted in their schematic relationships, in Fig. 7. Evidently the net collection efficiency will be:

$$\underline{\eta_c = \eta_p - \eta_\ell}$$

Photon Flux Capture. Besides the astronomical geometry factor (ecliptic plane, earth's rotation), several kinds of matter interactions can reduce the energy contribution of the Solar photon flux:

Fresnel reflection at (transparent) dielectric interfaces
Selective absorption in "transparent" sheets and films
Residual back-scattering from the absorber

Not all of such events end up by "losing" the heat. For instance, absorption in the plastic blanket layers may reduce the heat loss from the water layer underneath. Account is not made here for such compound effects; each gain or loss process is accounted for independently. The possible errors due to other design indeterminacies, are much larger, at this point.

Geometry. It is traditional to calculate Solar power problems at High Noon in mid-summer in Arizona. That is not an appropriate reference. Even at the Summer Solstice, there is a perceptible slant incidence angle there of about 10°; at the Equinoxes, it is the same as the latitude, of course $\sim$ 33°; and at the Winter Solstice we're down to $\sim$ 56°. So there is this multiplicative cosine factor:

Summer Solstice	$\sim$ 10° . . .	x .985
Equinoxes	$\sim$ 33° . . .	x .84
Winter Solstice	$\sim$ 56° . . .	x .56

The "Summer assumption" of x1 is technically justifiable, but here I will be somewhat conservative by invoking at least the Equinoctial value: $\underline{\Sigma = .84}$.

The daily variation is summarily accounted for, in the secular factor; a familiar "rectangular" model assumes the daily insolation as equivalent to 8 hours of mid-day sunshine: $\sigma = 1/3$

Note on Symbolism: As usual, we have more constants, variables and parameters than available symbols. Short of resorting to Coptic script, we must simply recognize by useage, which meaning applies where:

- σ stands for the Stefan-Boltzman constant; also, above for the secular factor
- n can be the index of refraction, a flow resistance, or the number of cover sheets
- η (with index) stands for an "efficiency" of some sort

Fresnel reflection depends on the refractive index. I use a value n = 1.5, which is representative of plastics (1.45 to 1.55); it is on the low side for glasses (1.5 to 1.8). Find:

$$\eta_f = .96 \text{ for each interface}$$

For water, n = 1.32 and $\eta_f = .98$
If there are n cover sheets, then:

$$\underline{\eta_f = .98 \times (.96)^{2n}}$$

Selective Absorption. Most plastics exhibit strong molecular absorptions in the IR, for $\lambda > 2$ microns. Very little of the Solar flux lies above this (in wavelength). Write $\underline{\eta_{ir} = .93}$

Absorber back-scattering. This is essentially a measure of "blackness". Relatively casual treatments, such as surface roughening, black paints, opaque dyes, do quite well: ∿ 95% absorption. $\eta_s = .95$

For all photon processes taken together, obtain:

$$\eta_p = \Sigma\ \eta_f\ \eta_{ir}\ \eta_s = .725\ (.921)^n$$

whence:

n	η_p	
1	.67	
2	.615	
3	.565	. . . . the fidu geometry
4	.525	
5	.485	

Number of sheets. n sheets define n blanket gas layers in this analysis. The bottom sheet as counted here, is not in contact with the water surface. The presence of water vapor is immaterial in this lowest space, because the H_2O molecules simply displace N_2 and O_2, and to first order, they behave in a thermally similar way in this temperature-pressure domain. There may be serious mechanical problems with a "contact" (or floating) configuration; if such a membrane exists, it should not be included as defining a thermal barrier.

Net Pond Field Input. 1 km^2 with n = 3 (3 sheets):

P_{th} = 500 megawatts with $\dot{q}_{th} = .05$ w/cm^2

Also $\eta_p = .56$

Thermal leakage. There is next, a competition for the heat deposited in the water: That removed by the pond flow and that lost by thermal leakage - mainly upward through the blanket (mattress, pillow, - - -). The problem is identical to that for the Greenhouse, but at a lower temperature. We will see that radiation loss and convection loss are of the same order. All three loss factors are less sensitive to geometric details, and it is worth verifying the ground loss.

We know, from the Greenhouse exercise, that a layered blanket will have sheet separations of a few centimeters. One expects several kinds of optimizations: Conduction vs. convection (to fix the separation h); Fresnel loss vs. thermal leakage loss (to fix the number of sheets n). The fidu choice of n = 3 with h = 5 cm, is based on the analysis and curves of the Greenhouse analysis.[2] The treatments of convection[11] and of radiation[12] are also elaborated there.

Conduction. This is straightforward:

$$\dot{q}_c = k_a \frac{\Delta T}{H} = 2.7 \times 10^{-4} \text{ cal/cm}^2 \text{ sec or}$$
$$\underline{\sim .0012 \text{ watt/cm}^2}$$

for the 3-layer blanket just defined. ($H = nh = 15$ cm, $k_a = 6.8 \times 10^{-5}$ cgs)

Convection. I modify the K&B formulation to extract the contribution attributable to convection:

$$k_d = (\varepsilon - 1)\, k_a$$

and

$$\dot{q}_d = (\varepsilon - 1)\, \dot{q}_c$$

2. SDK 73-10, ibid
11. Kutateladze and Borishanskii, A Concise Encyclopedia, etc. (Chap. 10)
12. Eastop and McConkey, Applied Thermodynamics, etc. (Chap. 17)

Following their prescription of dimensional analysis, I write the PrGr test as:

$$Z = 3\ \text{PrGr} = 3(\text{PG})(h^3\Delta T)$$

where (PG) is a universal parameter, as illustrated in Fig. 8. (Here I take refuge in K&B units, in which h is in meters.) For the defined blanket, the average temperature interval is $\Delta T = 20°C$, and the test function has the value $Z = 3 \times 10^5$. This is in the middle of their case (a) - as it was for the Greenhouse - so that the same convection coefficient function applies:

$$\varepsilon = .062\ (3\text{PrGr})^{1/3} = .062\ Z^{1/3}$$

Find at once: $\varepsilon - 1 = 3.6$

And: $\underline{\dot{q}_d = .0043 \text{ watt/cm}^2}$ $\qquad \dot{q}_u = \dot{q}_c + \dot{q}_d = .0055 \text{ watt/cm}^2$

Radiation: The applicable interchange factor formulations are taken from SDK 73-10.

The "worst case" calculation yields:

$$\dot{q}_r = \frac{\sigma T_o{}^4}{n+1} = .022 \text{ watt/cm}^2$$

The "indifferent case" expression is:

$$\dot{q}_r = (\sigma T_o{}^4)\ \frac{f}{n}\ [1 - (\frac{T_z}{T_o})^4]$$

where f is the interchange factor. For a "mediocre" black-body sheet: $f \simeq .4$; also here $n = 3$.

Find: $\underline{\dot{q}_r = .0085 \text{ watt/cm}^2}$ $\qquad \dot{q}_v = \dot{q}_r + \dot{q}_t = .0095 \text{ watt/cm}^2$ (N.B.)

Ground Loss. Some thermal care must be taken with the base material under the Ponds, or else it will act as a "thermal pump": It will cool down at night just enough to act as a heat sump for the day's heat. To see this, consider the case of a compacted, asphalt-binder earth base; for it, $k_g = .006$, and find:

$$\dot{q}_g \approx .012 \text{ cal/cm}^2 \text{ sec or } \underline{\sim .05 \text{ watt/cm}^2} \quad (!)$$

We can't even get started with this one.

Suppose instead that the base consists of a prepared material, such as pumice bonded with diatomaceous clay, about 6 in. thick. For this, $k_g = .00015$ and:

$$\dot{q}_g \approx 5 \times 10^{-4} \text{ cal/cm}^2 \text{ sec} \quad \text{or}$$

$$\underline{\sim .002 \text{ watt/cm}^2}$$

We can live with that.

Total Leakage. In sum:

$$\dot{q}_c = .0012$$

$$\dot{q}_d = .0043$$

$$\dot{q}_r = .0095$$

$$\dot{q}_g = .0025$$

$$\underline{\dot{q}_\ell = .017 \text{ watts/cm}^2} \qquad \text{or} \qquad \underline{\sim .0040 \text{ cal/cm}^2 \text{ sec}}$$

Correspondingly $\underline{\eta_\ell = .19}$

The radiation contribution is based on a "mediocre black body" model, and it is uncertain by perhaps a factor of 2 so that $\dot{q}_\ell$ spans a prediction range of $\sim$ .015 to $\sim$ .020. A related design uncertainty concerns the direct i.r. leakage through the plastic sheets or films. For an air-supported film "mattress", there's only $\sim$ 10-12 mils between hot water and the ambient atmosphere. For certain plastics such as Tedlar, this transparency may be as high as 20%. In the prefab box design of Fig. 6, thicker, semi-rigid Tefzel sheets would be used, amounting to $\sim$ 40 mils. The estimated i.r. transparency is $\sim$ 1%, and this is allowed for in $\dot{q}_t$ above.

These numbers also point to the difficult compromise represented by the choice n = 3, as discussed in Reference 7. For n = 2, the thermal leakage gets out of hand; for n = 4, the transmission losses over-compensate the reduced leakage.

Pond Field Thermal Output. We then have:

$$\underline{P_\ell \doteq 170 \text{ megawatts}}$$

and what's left to heat the water - and to be continuously extracted while the Sun shines - is:

$$\underline{\dot{Q} = 330 \text{ megawatts}} \qquad \text{and} \qquad \underline{\eta_c} = \eta_p - \underline{\eta_\ell = .37}$$

That's a disconcertingly small part of the Solar flux number itself: $\sim$ 36% thermal efficiency - and we still have to impose the thermodynamic penalties. It does suggest that there is room here for ingeniousness in making the Pond field area more effective. Obviously also, both the photon capture and leakage contributions deserve extensive parameter studies to better establish the optimum design values.

And, recalling the mid-winter cosine incidence factor (x.56), the operation becomes almost marginal, even at Noon, at the winter Solstice:

$$\dot{Q} \rightarrow 100 \text{ Mw}$$

Thermal Transfer. The other part of the Pond Field analysis is concerned with thermal power transfer into the thermal storage reservoir. In essence, this is constrained by the necessity of transferring the energy as fast as it is captured, in order to avoid boiling and "burn-out".

Pond Heating Rate. For purposes of general orientation, I earlier assumed a water heating rate based on complete absorption - that is - all .07 watt/cm^2 going into 1 cc. The residual thermal output just calculated, corresponds to a much lower thermal deposition rate $\sim$ .035 watt/cm^2. Correspondingly the specific temperature rate will be: .008°C per sec for a shallow layer 1 cm thick at $\sim$ 95°C.

Assuming a thermal exchange (in storage) which cools the water by 3°C in one passage, then the re-heating time is:

$\sim$ 360 sec or about 6 min.

The water enters the pond field from the reservoir at $\sim$ 92°C and returns at 95°C. Nothing is gained by holding it out there much longer; some hot spots would start boiling.

The initial heating rate (starting at ambient) would be somewhat higher, with a specific temperature rate of:

$\sim$ .01°C per sec (at $\sim$ 25°C)

Assuming a linear interpolation, the heating time for one filling from a cold start, is then:

$$\tau = \frac{T_1 - T_2}{\dot{T}_m} = 8000 \text{ sec}$$

(This does not allow for buffer storage or reservoir pre-heating.)
It does show that one should refill the pond field with hot water stored from the previous day, or else too much Sunlight is lost just getting started. (The heating function is discussed further in Appendix B.[13])

---

13. Memorandum: Shallow Pond Test Facility; LOC 73-9 revised.

Flow Rate. So we are next led to the question of how fast the water should move. For a "squarish" km^2, the maximum pond length would be $\sim$ 1000 meters, leading to a flow velocity $\sim$ 2 m/sec ($\sim$ 6 ft/sec), which is quite reasonable. The fidu configuration uses pond "strips" 200 m long (600 ft), for which the flow velocity would be much less:

$$v \sim .55 \text{ m/sec } (\sim 1.7 \text{ ft/sec})$$

These values are well clear of turbulence velocities ($\gtrsim$ 10 m/sec), above which the water sheet would tend to break up and tumble over itself.

Pond Slope. One way to get uniform flow in a pond may be to tilt the pond slightly. Marks gives a semi-empirical expression (English Units) for channel slope:

$$S \doteq \frac{n^2}{2.2}\,\frac{v^2}{d^{4/3}} \qquad \text{(ft per 1000 ft)}$$

where this n is a coefficient of roughness. Comparing with the tabulated values, I assign n = .001 as a probable situation. (E.g.: Smooth plastic sheet on graded and treated pumice sub-base.) I also continue with $d \simeq 1$ cm or 1/30 ft.

For the 1 km case, find S = .35 ft per 1000 ft. This pond field would need a drop of about 1 ft to attain the desired flow rate.

For the 200 meter case, we find: S = .020 ft per 1000 ft or a section drop of merely 5 millimeters !

That is a difficult gradient to achieve simply by earth grading methods. Perhaps the simplest scheme is to grade the ponds level, and to establish a flow gradient by means of adjustable metering weir edges, at each pond box joint (or riffle). The input flow is then controlled by an inlet weir, at the slightly elevated supply flume - e.g., Fig. 9.

<u>Volume/Mass Flow</u>. The total field water content is about 10^7 kgs or 10^4 metric tons (for 1 cm depth). This is to be replaced every 6 minutes leading to:

$$\dot{V} = \dot{M} = 2.8 \times 10^7 \text{ gms/sec or } 28 \text{ met}^3\text{/sec}$$

In "old" units, this is a flow of 10^3 cu ft/sec
$\sim$ 8000 gallons/sec
$\sim$ 30 tons/sec

<u>Pumping Power</u>. The required Pond flow head was seen to be trivial (a few millimeters). The central aspect of "pumping power" then relates to the power needed to overcome flow resistance in the water distribution network - pipes, flumes, etc. It is common practice to reference this in terms of equivalent pressure heads needed to sustain a desired flow volume. The fidu scheme configuration is shown as a gravity flow scheme as well: The six circulation pumps are really lift pumps which raise the water taken from the storage trough about two feet; the water then flows out the flumes, down the ponds and back to the reservoir entirely by gravity.

For the indicated water flow, and 85% pump efficiency, the power required per foot of head is:

<u>P/h = 115 kilowatts per foot</u>

Obviously we can only afford a very few feet of equivalent head: As I just indicated for the fidu system - only two feet and $\sim$ <u>.23 Megawatts</u> of pumping power.

Note also that even this head is no more than about 1 psi pressure loss in the "plumbing" - not much when one considers that conventional utility systems operate in the 60-80 psi domain. These numbers also speak for the severe penalties one would likely encounter if large water volumes were to be moved very far - horizontally or vertically.

There are many such small, accessory, and peripheral, "1/4 Megawatt" power consuming functions in a power plant. In conventional systems, they are easy to afford, but in a low-efficiency Solar plant, they add up to an uncomfortable subtraction. Each represents a potential dilemma in respect to trade-offs between different capital cost items.

Equivalent sectional area. Throughout the development so far, there has been a tacit implication of a "squarish" Pond field of area 1 km^2. (A geometry not more than 2:1 rectangular, say.) The fidu scheme uses a pond length of about 200 m; evidently, if such strips were laid side-by-side, they would occupy a total width w = 5000 m and we have assumed a water depth of $\sim$ .01 m. Consequently the equivalent flow section across these ponds, is:

$$\Sigma = 50 \text{ sq. met.}$$

This is a self-consistent number with the flow velocity calculated earlier.

Flumes and Weirs. These criteria for water-flow systems portray a situation very similar to those encountered in economically conveying large quantities of water for surface mining operations and for flat-land agricultural purposes. Both theory and practice show that conveyor geometries should have the least wetted perimeter and smoothest surfaces. To this end, smooth open flumes with a plastic lining should require the least head for a given flow.

Again I take recourse to Marks' semi-empirical flow equation, somewhat differently presented:

$$S = \frac{n^2}{2.2}\frac{v_f^2}{m^{4/3}} = \frac{h}{F}$$

where $m = \dfrac{\text{cross-section area}}{\text{wetted perimeter}}$ = ("hydraulic mean dimension")

F is the length of a flume

h is its gradient height

For "practical" flume geometries: $5d \lesssim m \lesssim .7d$ in which d is the actual water depth.

To avoid getting lost in details, I adopt the obvious average value: m = .6d and obtain:

$$h = \frac{n^2}{1.1}\frac{v_f^2}{d^{4/3}}\ F \qquad \text{(all English units)}$$

Here it is reasonable to take n = .009.

Note also that: $m = \dfrac{Wd}{W + 2d} = .6d$

whence find the approximate flume width:

$$W = 3d$$

The irrational power implies that we now have a closed non-degenerative description (as it did in the Greenhouse analysis). We get a unique solution in this way:

$$\dot{V} = .6N\ d^2\ v_f$$

in which N counts the number of "parallel paths".

Whence $$v_f = \frac{\dot{V}}{.6\ N\ d^2}$$

so that we get the trade-off expression:

$$\underline{h\ d^{16/3} = 2.5\ n^2\ \left(\frac{\dot{V}}{N}\right)^2\ F} \qquad \text{(still e.u.)}$$

There are all determinate terms. Note the strong dependence on the dimension variable; bear in mind that here this term carries both depth and width connotation - recall m = .6d. (For a pipe, the function is similar: $h\ \alpha\ d^{-5}$.) Evidently we're going to have to say something more detailed about the Pond field layout. Just to save time, let me indicate that a few "trial models" were calculated. These showed that a low-loss flume network would have a volume approaching that needed for "one-day" storage. The merger of storage and distribution was thus a logical design feature, further reinforced by the earlier constraints of proximity and "zero altitude".

Referring again to the fidu scheme, the feeder flumes add some numerical perspective to the problem. Consider the situation "half-way out":

$$\frac{\dot{V}}{N} = \frac{1}{2} \times \frac{1}{6} \times 10^3 = 83\ \text{ft}^3/\text{sec}$$

$$F = 800\ \text{ft}$$

$$h = \frac{1}{2}\ \text{ft}$$

Find: $d \sim 4.3$ ft, $W \sim 13$ ft

This is a flume about 5 ft high, 12 ft wide. (About 60 ft^2) As a cross-check the pumping power would be about:

$$2 \times \frac{4}{6} \times \frac{.115}{.7} = \underline{.22 \text{ Megawatt}}$$

The flume velocity would be:

$$\underline{v_f = 7.5 \text{ ft/sec}} \qquad (2.4 \text{ m/sec})$$

This example serves to show that there is a physically reasonable distribution scheme corresponding to a water pumping power assignment of the order of 1/4 Mw. But we have to carry this number forward as a debit to the final busbar figure, because the thermal power content of the water (330 Mw) remains to be converted into mechanical work and electricity. Note that there is an implication here, of a thermal manipulation power constraint of the order of .1% !

Module Segmenting. 1 km^2 is almost certainly too large an area to cover with a single unsegmented Pond, for many reasons:

Flow velocity and slope
Slope stability
Input water distribution
Local area maintenance

For modular design reasons, the convenient Pond length is 600 ft (or $\sim$ 180 m); that leaves the Pond width to be determined. An influence on this is the "end connection" problem. One senses that the individual weir width should be quite small as compared to pond length, in order to achieve uniform flow more easily. That suggests a Pond width in the range 10 to 20 meters. Another influence is the size limitation of practical plastic materials. At this stage of design, it seems advantageous to further partition the Ponds into 12 ft strips - being the width of commercial plastic sheet. The 1 km^2 fidu system is thus made up of about 1540 such strips, as suggested by Fig. 4. Using the prefab plastic box concept of Fig. 6, each strip consists of a series of 10 such boxes, with metering edges at each butt joint.

The Storage Element

As noted in the introductory discussion, the choice of a hot water reservoir as an energy storage buffer between the daily Solar input peak and the daily power demand cycle, was a natural one. The fiducial scheme is thus configured to "please everybody", by being able to deliver full output at any time during a 24-hour period. Each day, some of the hot water is pumped through the pond field; the reheated water is returned to the reservoir, where it mixes by convection. Each day also, some of the hot water is pumped through the thermodynamic evaporator (heat exchanger); the cooled water is likewise returned to the reservoir, for mixing by convection. In steady-state operation, the mean daily withdrawal power level is about 1/3 of the secular input power level.

How big should the reservoir be ? It acts in a role somewhat analogous to that of a large condenser in a reactive circuit: The degree of temperature "ripple" is comparable to the degree of voltage ripple, which is measured roughly by the ratio of charge removed to stored charge. The thermal equivalent is thus:

$$\frac{\delta T}{T_1 - T_2} \approx \frac{\delta Q}{Q} = \frac{\int \dot{Q} dt}{Q}$$

Another factor entering into storage dimensions, concerns just what form of "Sun-less" protection one desires. The smallest requirement is that related to bad weather - a cloudy period lasting 3 to 10 days. Much the largest would be that addressed to bridging the very low useful Solar input in mid-Winter, perhaps amounting to 3 months worth of thermal energy. A strong deterrent to such highly reactive schemes, lies in the extremely long time required to "charge up".

Ripple. There are various prejudices (as well as technical criteria) concerning the permissible thermal ripple associated with the charge-discharge cycles. For small ripples, note that the ripple expression also defines the change in TD efficiency, to first order:

$$\frac{\delta Q}{Q_r} = \frac{\delta T}{T_1 - T_2} = \frac{h}{\eta}\frac{\delta T}{T_1} = \frac{\delta\eta}{\eta}$$

(where h is the enthalpic factor)

One can show that the indirect economic constraints on thermodynamic efficiency[9] - and hence on temperature leniencies - are quite severe - so much so, that one can "throw the price book away" in achieving the highest possible TD efficiency. Correspondingly, the philosophy adopted here, is that the permissible thermal storage ripple ought to be no greater than the temperature loss allowed in sizing the TD heat exchangers: About 3°C. In a typical case, even that translates into an efficiency loss of ∿ 7% at the bottom of a ripple. This criterion also automatically causes most of the water in the reservoir to pass through both the collector and the evaporator no more (or less) than once each day.

Input vs. Output - The Secular Factor. A long-term steady-state operation requires that the long-term average input and output through the reservoir be the same. To first order this invokes (by self-evident derivation):

$$Q = \dot{Q}_{th} \times (8 \times 3600) = \dot{Q}_{td} \times (24 \times 3600)$$

and:

$$\underline{\dot{Q}_{td} = \tfrac{1}{3}\dot{Q}_{th} = 110 \text{ Megawatts}}$$

This is the 24-hour mean thermal power accessible to the fidu thermodynamic sub-system from the reservoir. Also note the daily thermal energy value:

$$Q = 6.6 \times 10^6 \text{ Megajoules} \quad \text{or} \quad \frac{2.3 \times 10^{12} \text{ Calories}}{9.0 \times 10^9 \text{ BTU}}$$

<u>1 km² of Sunshine is equivalent to 1000 barrels of oil per day.</u>

---

9. ibid.

Reservoir Size. The required minimum thermal storage mass follows at once:

$$\delta Q = \frac{2}{3} Q = MC_p \, \delta T$$

Find: $M = \frac{2}{3} \frac{Q}{C_p \, \delta T} = 5.2 \times 10^{11}$ grams or 5.2×10^{5} cu. met.

Also 18×10^{6} cu. ft or $\sim$ 430 acre-feet

The syphon constraint implies a reservoir depth of $\sim$ 30 ft; this means a reservoir area of $\sim$ 14 acres, or $\sim$ .05 km^2:

$$\frac{\text{Reservoir Area}}{\text{Collector Area}} \sim \frac{1}{20}$$

"Charging Time": Note the total thermal content:

$$Q_r = MC_p \, (T_1 - T_2) = 3.5 \times 10^{13} \text{ calories}$$

This is about 15 times the daily thermal "draw-down" - but is not "15 days worth". If we linearize the collector thermal yield, take a simple average value (420 Megawatts), and put it all into heating up this reservoir, that will take:

$$\tau_r \simeq 12 \text{ days}$$

(Of course the approach is asymptotic, but this is about when it should become worthwhile to operate the TD loop.)

Correspondingly a 3-month reservoir would require 3 years to "charge up" !

A Reservoir Concept. I have indicated the severe penalty offered by distribution schemes involving long flumes or conduits and correspondingly large pumping heads. It is really important to keep the entire configuration compact. Consequently we should seek geometries which avoid putting the thermal reservoir(s) far from the ponds. An interesting concept merges the distribution and reservoir requirements, by combining a good part of the flumes into the reservoir, as shown in the fidu depiction, Fig. 4. The area sacrifice ($\sim$ 4% - per area ratio defined earlier) is certainly acceptable. Besides, the T_1 flume now becomes a trough so large that we can exchange hot water for the TD power sub-system at either end with no perceptible flow penalty. This reservoir trough is 1000 m. long, 40 m. wide, 12 m. deep. Each of the 6 local supply flumes is supplied from the central reservoir-flume by a separate lift pump. Subsequently, each pond path operates entirely by gravity, back to the central reservoir thru similar return flumes. (Note also Fig. 5)

There are several obvious, more elegant variations on this geometric theme (including a "finer granulation"), but I'll stay with this initial primitive scheme for the fidu exercise. The figure also indicates a similar kind of disposition for the T_2 side of the thermodynamic system, and I'll get to that later.

Some Incidental Features. One of the principal reasons for separating the collection and storage functions was the significant reduction of "dark-time" thermal losses. The central reservoir will need at least a modest thermal-vapor cover, to suppress both radiative/convective loss and latent-heat evaportaion loss.

Another design detail will concern the proper "mixing" of input and output flows in the exchange with the ponds and the TD loop. Obviously one does not put these orifices side-by-side in the central reservoir. The fidu arrangement is designed to promote convective circulation locally at the pond take-offs, and to induce lengthwise flow by returning the evaporator flow to the far end.

Thermodynamic Sub-system Analysis

I have already indicated the fiducial choice of closed Rankine cycle thermodynamic machine, using a Freon as the TD working fluid. There appear to be no substantial "unknowns" in designing and fabricating such a TD system.[10] But the situation is not like it is for conventional steam machines, where a selection of proven "off-the-shelf" designs - and even of stock machinery - is available. Studies addressed to configurational optimization, materials specification, operating conditions, etc., must be carried through for a particular customer requirements. The Rankine choice appears to be the most conservative one in terms of demonstrated examples, immediate feasibility, and shortest list of likely problems.

Formally, a closed Rankine system analysis is a routine exercise, as is suggested by the "thermodynamic account" schematic, Fig. 10; but the temperature domain, thermal circumstances, and working media here provide some analytic entertainment.

The TD Fluid Choice. The thermodynamic properties of a number of candidate fluids ("refrigerants") were examined with primarily a machine efficiency criterion in mind. The Rankine cycle is bounded on the T-S and P-h diagrams by essentially constant, pre-determined, temperature lines. ($\sim$ 200°F top and $\sim$ 80°F bottom.) (Fig. 11) Evidently in making this choice, one wants to stay well clear (below) the critical point and "top-of-the-dome". More-or-less by trial-and error, one finds that the most advantageous vapor dome is one for which the saturated vapor line inflects near the thermal supply temperature (T_3). That furnishes the most leniency for turbine expansion, commensurate with the least input power. For the eligible fluids, the corresponding saturated liquid line (on the opposite side) is also fairly straight. - I.e.: The liquid phase is characterized by its specific heat.

As an interesting comment on the universal nature of chemical (atomic) bonding, those vapor domes can be laid practically on top of one another with small scatter, when linearly referenced to appropriate coordinate scales. There is no "great magic" to be found here. Heavier molecular weights appear to correlate with somewhat more favorable dome shapes, and, of course, with lower specific pumping (or pressurization) power.

10. Appendix A - ibid

To continue, the fidu fluid choice here is Freon 113. It is a fully halogen-saturated ethane: $CCl_2F{:}CClF_2$. Table II exhibits some of the better candidates (including 113), along with their summary properties. You really have to look at the vapor domes to make a choice among the three or four "obvious" ones. An earlier analysis choice was Freon 11. 113 looks better because the condensate "back pressure" is much lower (at 30°C): 8 psi vs. 18 psi; the working pressure (at 90°C) is also lower: 50 psi vs. 95 psi (note Fig. 11). From a turbine efficiency standpoint, the turbine friction loss is sensitive to the exhaust gas density. The pressure ratio is slightly favorable: 6.4 vs. 5.2. One can always build a bigger machine to compensate for the lower input pressure.

The TD dome for Freon 113 exhibits a singular entropic feature - probably related to its unusual molecular structure. There is an extended region of "reverse entropy" under the saturated vapor line, in our working domain. Superheating is less important, and supersaturation is less likely to occur, in the turbine expansion. The Rankine efficiency approaches the Carnot limit along these unique isentropic lines.

<u>The TD Cycle</u>. The fiducial closed Rankine cycle adopted here is also shown in Fig. 11, for Freon 113. The only "elbow room" available in this specification is in the loci on each side of the dome: Liquid phase pressurization, to the left of the sat. liq. line; gas phase expansion, to the right of the sat. vapor line. Let's look at this qualitatively: The cycle starts at "1" at essentially ambient conditions. A pressure pump takes the liquid Freon up to point 2, following an essentially vertical constant temp. line. ($\Delta T_{12} \sim 1°F$) The evaporative heat exchanger takes the fluid horizontally into and through the vapor dome, and slightly into superheat - point 3. This superheat is needed to ensure essentially isentropic expansion in the turbine towards "4". This side is shown as a band, because the Freon gas may follow somewhat different entropic histories in different regions of a turbine stage. Also the turbine efficiency is finite (< 100%). Consequently its exhaust gas is really going to end up somewhat hotter than "ideal" (point 3'), because it has to carry away that "friction loss" heat. (Otherwise the turbine melts.) The indicated points are located so as to correspond to a 70% mechanical efficiency in the turbine, as discussed further on. Finally the condenser takes the turbine exhaust back down to the starting condition, or putting it another way, the condenser provides the "vacuum" for turbine expansion.

Temperature Differences and Thermal Exchange. This cycle has some disguised impact features with respect to system design. Note that the evaporation segment for the Freon heating leg (h_2 to h_3) occurs at relatively constant TD fluid temperature, and it soaks up about 2/3 of the thermal power taken from the reservoir. Since our economics/efficiency arguments urges us to make T_3 (and h_3) as high as possible, the available temperature window on the water side is very narrow. We're "trapped" between 90°C and 100°C. For a practical fidu case, I adopt a hot water temperature of 94°C, a vaporization cooling increment of 4°C, and a final (counterflow) interface difference of 4°C also. Then the Freon gas temperature will be 90°C into the turbine. Another 2°C must be given up by the water for that remaining 1/3 of the thermal power needed to heat up the liquid Freon. So the hot water returns to the reservoir at 88°C.

Several ways for cheating on these numbers have been discussed, but my examination suggests that they end up by defining open TD cycles (perpetual motion machines of the n^{th} kind). After several degrading trips around the loop, the machinery stops.

There is a somewhat analogous situation on the condenser side. The condensation also takes place at essentially constant temperature, and here the cooling water has to warm up. However, it is easier to cheat here, because one could invoke evaporative cooling on the water side. But the primitive argument remains that we are again trapped by a narrow temperature window on account of economics. So I choose 30°C as the final condensate temperature, and 26°C as the cold water input. It returns to the evaporative pond at 32°C.

Such a specification of temperatures may seem rather arbitrary, but in effect they amount to criteria for specifying in turn, the heat exchanger configurations and thermal fluid flow rates, as detailed in Part II. There I will also show that these choices straddle a competition involving efficiency, exchanger cost, collector cost and operating cost.

P.S. Note that the indexing now does not correspond to the academic Carnot indices for hot and cold reservoirs. T_1 has become T_3, and T_2 has become T_3'.

Ideal TD Efficiencies. The working temperatures here, are "locked in" by the highly reactive thermal reservoirs; it takes a lot of heat to move far from:

$$T_2 = T_3 = 90°C \ (194°F) \qquad T_4 = T_1 = 30°C \ (86°F) \qquad \Delta T = 60°C$$

From the vapor dome, Fig. 11, we get the corresponding cycle enthalpies:

h_1 = 14 cal/gm (25 BTU/lb) $\qquad$ h_3 = 59 cal/gm (106 BTU/lb)

h_2 = 28 cal/gm (50 BTU/lb) $\qquad$ h_4 = 50 cal/gm (90 BTU/lb)

h_3' = 53 cal/gm (95 BTU/lb)

The corresponding bounding pressures are:

p_1 = 50 psi (3.4 atmos.) $\qquad$ p_2 = 7.8 psi (.53 atmos.)

with a ratio $p_2/p_1 = 6.4$.

The "ideal" efficiencies follow at once:

Carnot: $\eta_C = 16\frac{1}{2}\%$

Joule-Rankine: $\eta_R = 13\frac{1}{2}\%$

Ideal TD Power. Recalling $\dot{Q}_{td}$ = 110 megawatts (steady-state thermal demand), the ideal Rankine output can be no more than:

$$\underline{\dot{Q}_{td}\,\eta_R = 15.0 \text{ megawatts}}$$

It remains to take account of the finite efficiency of non-ideal real thermodynamic machinery - in particular, the turbine.

Turbine Efficiency. The last major physical element along the toute from Solar flux interception to busbar power, is the turbine machinery. A detailed turbine analysis and design optimization is a separate extensive exercise. Nevertheless one can adopt some machinery efficiency numbers based on experience and on design exhibits, which represent good probable expectations. An accepted design prescription expresses turbine efficiency in terms of a multiplicative series of semi-empirical efficiency multipliers, each of which addresses some identifiable power loss contribution.[14] These can be collected into several coefficients which converge asymptotically to unity (or to a number very close to unity - like .98); note Fig. 12.

The internal efficiency for a turbine is expressible as:

$$\eta_t = k_{ng}^2 \left(1 + k_b \frac{\cos \gamma}{\cos \beta}\right) \frac{\cos^2 \alpha}{2}$$

with: $$k_b = k_{co}\, k_p\, k_i \approx k_p^2\, k_i$$

Feasible designs include: $\delta \approx \beta$, $\cos^2 \alpha \sim .9$, whence:

$$\underline{\eta_t = .45\, k_{ng}^2 \left(1 + k_p^2\, k_i\right)}$$

In conventional turbine design, the design target values for the efficiency multipliers, comes out of a consideration of trade-offs concerning fuel operating costs and machinery capital costs. For conventional power systems, an economically optimum design may thus be one rather far from those asymptotes. For Solar systems, the turbine efficiency reflects economically so strongly via added collector costs, that it virtually pre-empts any other turbine design criterion. It is cost-effective to adopt a high-efficiency design philosophy almost irrespective of cost consequences on the turbine itself.

A primitive trade-off model illustrating this situation is discussed in UCID-16386, Pt. 2.[9] There I show for example, that a 1% increase in turbine efficiency is typically worth a 30% increase in turbine price, in this application.

---

9. ibid.

14. Lee, Theory and Design of Steam and Gas Turbines.

Returning to an inspection of Fig. 12, higher efficiencies means working with lower velocities and with mechanically less favorable blade angles, to reduce friction, turbulence, commutation effects, etc. That then requires larger and more stages, larger and more blades, slower rotors, etc. - all features that point to higher machine costs. But there are also technical limits as to how close one may approach the "ideal". For instance, the blade angle term optimizes at about .92; for yet shallower attack angles, the ideal turbine becomes power-less as well as friction-less. As a reference case, one sees (again by inspection) that the following criterial values are reasonable:

$$k_{ng} \sim .97 \qquad k_p \sim .97 \qquad k_i \sim .96$$

such that:

$$\underline{\eta_t \sim .80}$$

Nevertheless, in deference to unchallengeable conservatism, I invoke:

$$\underline{\eta_t = .70}$$

It follows that the turbine output is:

$$\underline{\dot{Q}_{td}\, \eta_R\, \eta_t = 10.5 \text{ Megawatts}}$$

Real TD Efficiency. The actual thermodynamic efficiency is:

$$\underline{\eta_R\, \eta_t\, \eta_x = 9.2\%}$$

where η_x expresses a small correction for the pressurization power discussed below ($\sim$ .98).

Correspondingly, 91% of the thermal power being withdrawn from storage has to be continuously disposed of, through the condenser. I write:

$$\dot{Q}_r = (1-\eta_R\eta_t\eta_x)\, \dot{Q}_{td} \quad \sim \underline{100 \text{ Megawatts}} \qquad (24 \times 10^6 \text{ cals/sec})$$

In the closed Freon loop, this is delivered to the condenser by the turbine exhaust gas at about 115°F (point 3').

At this point, the overall system efficiency is about 3%. We must still take account of the larger of a multitude of peripheral power "robbers" in this fidu system, such as heat exchangers and pumps. As indicated before, most of these deductions depend on just how far one cares to go in refinement and cost; to a certain extent, it also becomes a question of assigning an acceptable technical penalty and designing to it.

What there is left to say about the turbine proper, without getting into interior design details (such as blade shapes, staging, etc.), concerns bounding features and constraints which are determined by system requirements, by properties of matter, by the characteristics of other contiguous components. In particular we need the fluid flow numbers in order to calculate the (liquid) Freon pressurization work.

TD Fluid Mass and Volume Flows. Earlier we saw that it takes about 14 cal/gm (25 BTU/lb) to heat the Freon condensate, and about 31 cal/gm (56 BTU/lb) to vaporize it. Each gram of Freon through the evaporator removes 45 calories (188 joules) in the gas phase, at a density of .023 gm/cc (1.45 lb/ft^3). The required average Freon mass flow for satisfying the heat transfer criterion (110 Megawatts) is then:

$$\dot{M} = \frac{\sigma P_{th}}{\Delta h} = 590 \text{ kg/sec} \quad (\sim 1300 \text{ lb/sec})$$

The corresponding Freon volume flows are:

At turbine input: $\sim$ 50 psi (3.4 atmos.), 90°C and .023 gm/cc

$$\dot{V}_3 = \frac{\dot{M}}{\rho} = 27.0 \text{ met}^3\text{/sec} \quad (900 \text{ ft}^3\text{/sec})$$

At turbine output: $\sim$ 8.0 psi (.54 atmos.), 45°C and .0040 gm/cc

$$\dot{V}_4 = \frac{\dot{M}}{\rho} = 150 \text{ met}^3\text{/sec} \quad (5200 \text{ ft}^3\text{/sec})$$

In condensate phase: Liquid Freon, 30°C, 1.55 gm/cc

$$\dot{V}_{12} = \frac{\dot{M}}{\rho} = .37 \text{ met}^3\text{/sec} \quad (13 \text{ ft}^3\text{/sec, } 100 \text{ gals/sec})$$

Flow Velocities. In principle, there is just one mandatory flow velocity criterion for the thermodynamic loop: The severe non-linearities and high losses associated with the trans-sonic regime for gas flow in ducts and turbine working spaces. There is this traditional criterion in conventional turbine design: $v_g \lesssim .4\ v_s$ (also see SDK 73-10). Here for Freon gas: $v_s \sim 140$ m/sec. We obtain this rather low gas flow velocity limit:

$$\underline{v_3 \sim 56 \text{ m/sec}} \qquad (\sim 180 \text{ ft/sec})$$

This velocity number also gages the allowable tip (or edge) velocity of the largest turbine stage. It's going to be slow and big ($\sim$ 5 rps).

Inferentially, this also leads to a maximum flow velocity number for the liquid phase. I assume condenser-evaporator geometries which are not phase-sensitive. This could be a constant cross-section boiler - for instance, made of many parallel pipes - a conventional geometry. Then necessarily, by mass conservation:

$$\dot{M} = \rho_g\ v_g\ \Sigma = \rho_f\ v_f\ \Sigma$$

(where Σ denotes cross-sectional area)

Find: $\underline{v_{12} \sim 1.2 \text{ m/sec}}$ ($3\frac{1}{2}$ ft/sec) (liquid phase flow)

Of course, the permissible liquid flow velocity in other system locations has a much wider accommodation than is indicated by this criterion. Elsewhere, it would be limited only by the work required to overcome "plumbing friction". This is also briefly discussed in Part II; here I summarily assume that it is cost-effective to make the condensate plumbing large enough as to be essentially "loss-less": I.e., Less than .05 Megawatt residual head loss.

There is no strong criterion for sizing the exhaust side of the turbine. The efficiency here is so low that the gas kinetic energy criterion merely leads to a slightly lower exhaust gas velocity; this then obtains a conventional adiabatic thrust relationship:

$$\frac{\Sigma_2}{\Sigma_1} = \frac{\rho_1}{\rho_2} \approx 6.4$$

Admission-Exhaust Areas. There is an unavoidable flow bottleneck: The Turbine admission area. It would actually consist of the many first stage stator passage areas (or nozzles) in parallel. I assume 360° (full) admission, in the interests of maximum turbine utilization efficiency. The first and last stage passage areas are then sized, to first order, by the numbers indicated above, by virtue of:

$$\dot{M} = \rho\ v\ \Sigma$$

Find: $\Sigma_3 = .45\ m^2$ (5.0 sq ft) total admission area

$\Sigma_4 = 3.0\ m^2$ (31 sq ft) total exhaust area

It is not a very large machine. (Compare the Paratunka 500 kw unit - Appendix A.)

Auxiliaries. From a systems output standpoint, we must next account for these essential "support" loads and losses, in order to arrive at the net busbar output:

1. Thermodynamic fluid pressurization
2. Electrical conversion efficiency
3. Thermal system pumping power (that 2 ft lift, remember ?)
4. Control and operations requirements
5. Freon and water circulation work through the heat exchangers: evaporator and condenser

I subtract TD pressurization ahead of electrical generation because it is often customary to power this pump by means independent of the "main line", for reasons of systems stability and safety.

Thermodynamic Pumping Power. This is the last power output penalty directly attributable to the internal requirements of the thermodynamic loop. The condensed (liquid) Freon has to be pressurized before putting it back in the boiler. Virtually all of this work is vdp; the compression increment (pdv) is negligible. The required pumping power on this account is:

$$\frac{\dot{M}\Delta p}{\rho\eta_r} \simeq .14 \text{ Megawatt}$$

assuming a pumping efficiency $\eta_r = .85$ and $\Delta p = 2.9$ bars.

The same pump also pushes the Freon through both heat exchangers (evaporator and condenser). In Part II, I show that it is both necessary and reasonable to criterially assign a head loss figure to each thermal operation. This is nominally .1 Megawatt apiece, so that:

$$P_p = .35 \text{ Megawatt total}$$

The pressurization pump may be driven by a small independent turbine; customarily, an auxiliary starting engine can be clutched in, since this pump "tells the TD fluid which way to go" at the beginning.

The power ledger now reads:

$$P_{td} = 10.2 \text{ Megawatts}$$

Electrical Generation. Having reached the point of turning a shaft, we can invoke some "standard numbers" for the power generation and busbar manipulations:

Shaft conversion 95%
Transformer yard 98%

The gross electrical output is:

$$P' = 9.5 \text{ Megawatts}$$

Thermal Sub-system Power. In the thermal sub-system discussion, I assigned a 2 ft head to the circulation requirements of the Pond field itself, partly on the basis that one simply cannot afford much more. Recalling P/h = 115 Kilowatts, that assigns -.25 Megawatt to this function. Recall also that in the fidu configuration, this is split between six 40 kw pump-motor units.

Control and Operation Requirements. I rather arbitrarily assign -.25 Megawatt to this inescapable "plant load".

The Thermal Transfers. In the chosen fidu configuration, there are two places where all of the thermal energy must be transferred between Freon and Water. These correspond to the $T_1'-T_3$ leg (evaporation) and the $T_3'-T_1$ leg (condensation). Earlier I specified some relatively small permissible temperature increments for these operations: 2°C counter-flow differential at the Freon output, and 6°C change in the water passage. The dominating feature in binary heat exchanger design in this application, turns out to be the thin laminar film which bounds the moving fluids on both sides of the separating metal wall. It controls both the achievable heat transfer density and the pumping power penalty. This combination of constraints requires that very large volumes of fluid must be circulated along a thermal interface of exceptionally large area, as compared to conventional practice. Satisfying these technical requirements within acceptable capital cost and operating power limits, seems to be the next most challenging design problem in a Shallow Pond system. The analogy with the problems of OTG is inescapable.

In initially considering these design questions, one has the feeling that there simply must be a most advantageous exchanger design, within the large continuum of alternatives. Yet both theory and experience argue that this is not so; heat exchanger design is a relatively "flat" business. I conducted a fairly extensive analysis of the problem, and I was not disappointed in confirming this insensitivity. There are significant compromises between capital cost and operating costs, however; in this respect, the problem is rather analogous to the "Turbine Tradeoff" of Reference 9. Rather than engage in a lengthy discursive detour, the heat exchanger exercise and its numerical results will be deliberated in that separate Part II. Hereunder, I briefly summarize some of the outcomes most pertinent to our goal here.

An early, stark, design choice was this one: Do we immerse the heat exchangers in the reservoirs, or do we construct them as individual system components ? It turns out that this choice too, is economically immaterial. The film constraints operate so determinatively, that the fluid flow in a reservoir needs to be promoted and controlled as elaborately as if it actually were in a separate container. There is no "obvious" advantage. (A "natural convection" design turns out to require an exceptionally large interface: ∿ 20% collector area !) That is basically why I elected the more conservative choice in the fidu system, using conventional, parallel tube, enclosed assembles.

One then seeks tube configurations for evaporator and condenser which impose the least penalty in operating power and efficiency. Three basic analytic criteria can be written, involving:

Energy and mass conservation in steady-state transfer
Thermal transfer functions for laminar films
Pressure gradient in liquid flow due to laminar films

The derived first-order relationships between material properties, desired operating conditions, and geometry, turn out to be exceptionally degenerative for the geometric variables, yet strongly determinative for some functional ones. (Hence the "flatness" noted earlier.) Furthermore, this makes it difficult to construct a "closed" description, with a unique set of solutions.

Right at the start, one confirms that the laminar film limits the achievable thermal transfer rate to values well below that for just the metal interface. In the liquid phase, this film is of the order of .01 cm in thickness; it changes relatively slowly with fluid velocity (becoming thinner at higher v) - but the pumping power goes as v^3 ! In a realistic design, the achievable thermal transfer coefficient thru a Freon film is about 20% of that on the water side, so that the Freon side dominates the problem. A practical transfer rate is $\sim$ 2000 BTU/sq ft hr, and since we require a total thermal "thru-put" of the order of 2×10^8 BTU/hr, the exchanger interface area needs to be about 10^5 sq ft - not a trivial requirement, as it amounts to 1% of the collector area.

One also comes up with some other surprising geometrical features for an "optimum" design - like a Freon passage aspect ratio $\ell/d \sim 3000$! A tube 1" in diameter should be $\sim$ 250 ft long, with a surface area $\sim$ 60 sq ft, implying about 1600 tubes in all. It is a strange creature indeed - but made more plausible if you carefully inspect the Freon heat exchangers in an average household refrigerator - very tiny, very long tubes cleverly soldered to large air-coupled surfaces.

The significant numbers for the immediate purpose here, are the powers required to operate these exchangers. The parametric exercises to date, indicate that it is within technical and economic range to configure an evaporator and a condenser such that the required operating power is no more than .4 megawatt each. Of this, .1 megawatt is needed to circulate the Freon; this actually appears as an additional power requirement on the TD pressurization pump (and it was accounted for above). The other .3 megawatt is needed to circulate water; these circulation pumps would most likely be driven by electric motors. This requirement then represents another bus-bar penalty of: -.6 Megawatt.

The Water Flow Requirement. The evaporator must transfer 110 Mw, the condenser - 100 Mw, an almost negligible difference from a thermal design standpoint. Both sides are permitted a 6°C increment for this transfer. The mass flow rate is:

$$\dot{M} = \frac{P_{th}}{C_p \Delta T} = 4.5 \times 10^6 \text{ gms/sec}$$

In other language: 4.5×10^3 kg/sec   4.5 met^3/sec
160 cu ft/sec 10^4 lbs/sec
1200 gal/sec

The Pumping Heads. For the perspective purpose, one can here turn the problem around by asking: What are the implied loss heads in the thermal hardware, as indicated by the pumping powers, assuming incompressible fluids ?

$$P_p = \dot{V}\Delta p$$

Freon side: For $\dot{V}$ = .37 met^3/sec liquid phase, find:
$\Delta p \simeq 2.7$ bars or 56 ft head equivalent

Water side: For $\dot{V}$ = 4.4 met^3/sec, find
$\Delta p \simeq .7$ bar or 22 ft head equiv.

These numbers attest to the magnitude of the pumping problem. It is likely that one can do much better than this in the end, but the parameter studies so far, require this conservatism. Bear in mind that by these numbers, we would be moving around 150 Megawatts of heat at a cost of about 1/2 megawatt of power, or .3%. In conventional designs, a cost-effectiveness corresponding to $\sim$ 1% is considered to be "opulent".

"Lake Tee-Too". In the fidu configuration, the condenser is shown as transferring the rejected heat into a water body of sufficient area, as to be essentially in thermal equilibrium with the atmosphere, via wind cooling and evaporation. It is the latent heat of evaporation which dominates this final process; assuming that 24 x 10^6 cals/sec (3.5 x 10^8 BTU/hr) are ultimately dumped thereby into the air, we find a water utilization rate of:

$$\dot{M} = \frac{\dot{Q}}{L} = 4 \times 10^4 \text{ gms/sec}$$

$$\text{or} \sim 3600 \text{ m}^3\text{/day}$$

$$\underline{\sim 930 \text{ acre feet/year}}$$

For dry air, this involves an evaporation rate given by Marks as:

$$\dot{Q} = 100\ p_v A \qquad \text{(BTU/hr)}$$

where p_v is the water vapor pressure (inches Hg)

At 25°C, find:

$$A = 3.3 \times 10^6 \text{ sq ft} \quad \text{or} \sim 1/3 \text{ sq km}$$

$$\text{(75 acres)}$$

In effect, it takes about 33% of the collector field area to dispose of its degraded TD heat - a not unreasonable ecological prescription, as well.

This lake needs to be only 12 ft deep (average) in order to store one year's supply of water. (The evaporation rate is about one foot per month.) If I assume a 12" annual rainfall, then we need at least 800 acres (2 km^2) to replenish the lake (neglecting ground percolation). Considering the space required for water storage, power plant, access roads and service yards, 2 km^2 just corresponds to the site area needed for a 1 km^2 collector system. So it is not unreasonable to think in terms of a completely self-sufficient facility: Sunshine and rainfall in, electric power and humidity out.

Obviously "Lake Tee-Too" must also be "dirt cheap". In the fidu depictions, I have shown it as physically integral to the plant complex, again because we cannot afford to pump water very far - hot or cold. And it may need some kind of inexpensive sun shade, to prevent it from trying to behave like its Solar collector neighbor.

The Busbar Output.

Finally I collect these operating loads:

Pond field pumps	.3 Mw
Control and operations	.2
Heat Exchangers	.6
	1.1 Mw

We end up with:

$$\boxed{P = 8.4 \text{ Megawatts}}$$

System Efficiency. Based on a mean Solar input number of 300 Mw, the overall plant efficiency is then:

$\eta = 2.8\%$

A collection of salient numbers of this exercise is presented in a ledger, or "Power Account" of Table I (with Synopsis).

Table II

Properties of TD Fluids

Freon	Composition	Mol. Wt.	Density	Sp. Ht. (e.u.)	Thermal Cond. (e.u.)	Viscosity (poises)	Critical Pt. °F/psi	Pressure (psi) 70°F	Pressure (psi) 200°F	Pressure (psi) Ratio	Δ enthalpy BTU/lb
11	CCl_3F	137	1.47	.21	.05	4×10^{-3}	388/640	13	100	7.7	13
12	CCl_2F_2	121	1.29	.235	.05	2.5×10^{-3}	234/600	82	420	5.1	10
113	$CCl_2F\text{-}CClF_2$	187	1.56	.22	.037	6.4×10^{-3}	417/500	5.7	54	9.5	19
114	$CClF_2\text{-}CClF_2$	171	1.44	.245	.033	3.5×10^{-3}	294/480	27	175	6.5	17
216	$CClF_2\text{-}CF_2\text{-}CClF_2$	221	1.54	.17	n/a	5×10^{-3}	356/400	9	75	8.3	21

Note: "Δ enthalpy" is a purely comparative number taken on the saturated vapor line for 200°F and 70°F.

Commentary.

Analyses such as this tend to abrupt endings, since they are paper numerical exercises. Having defined the problem, there's not much of substance to puzzle over, until we get some "real numbers" Elsewhere the Solar team is developing an experimental plan for stepwise acquisition of data pertinent to the Shallow Pond scheme.

The estimated size of the competitive thermal and thermodynamic degradations in this scheme, make it seem relatively marginal. But Nature is marginal too in that sense. Yet it is a strongly successful and adaptable operation: The average Solar energy utilization efficiency of a forest is about 1%. One does not get a fair measure of the relative merits of Solar systems without an economic evaluation as well. As noted before, this scheme gives some promise of being so cheap as to compensate for the low efficiency, in terms of cost-effectiveness. A premature set of design cost estimates appears in Table III; it is by no means a "closed issue". Referencing Fig. 6, the essential economic question is: Can you make 16,000 such boxes for $5,000,000 ? If so, you're "in".

Table III

Cost Estimate

10 Mw. Shallow Solar Pond System

1. Average annual Arizona solar incidence - (24 hrs/day) ≈ 250 watts/m^2

2. Estimated overall conversion efficiency from incident solar energy to output electrical energy ≈ 3%

3. Solar Pond collector area required = 10^7 watts/[250 w/m^2 x .03]

   ≈ 1.33 Km^2

   (~ 1 mi^2 site area) (≈ 0.5 mi^2)

4. Estimated collector cost (includ. piping)

$$= \frac{\$0.50/ft^2 \times 10.76\ ft^2/m^2 \times 1.33 \times 10^6\ m^2}{10^4\ Kw} = \$715/Kw$$

5. Heat storage ⟶ 100/Kw

6. Turbogenerator, heat exchangers, etc. ⟶ 150/Kw

   $965/Kw

At 15% fixed charge rate and 85% load factor,

$$\frac{\$100}{Kw} = \frac{\frac{\$100}{Kw} \times \frac{0.15}{yr} \times \frac{10^3\ mills}{\$}}{8760\ \frac{hrs}{yr} \times 0.85} = \frac{2\ mills}{Kwh}$$

Solar Fixed charges	=	19.5 mills/Kwh
Operation & Maintenance	=	2.5 mills/Kwh
Fuel	=	0
Bus Bar ⟶		22 mills/Kwh

FINAL COST FIGURE = [22^{+10}_{-5}] mills/Kwh.

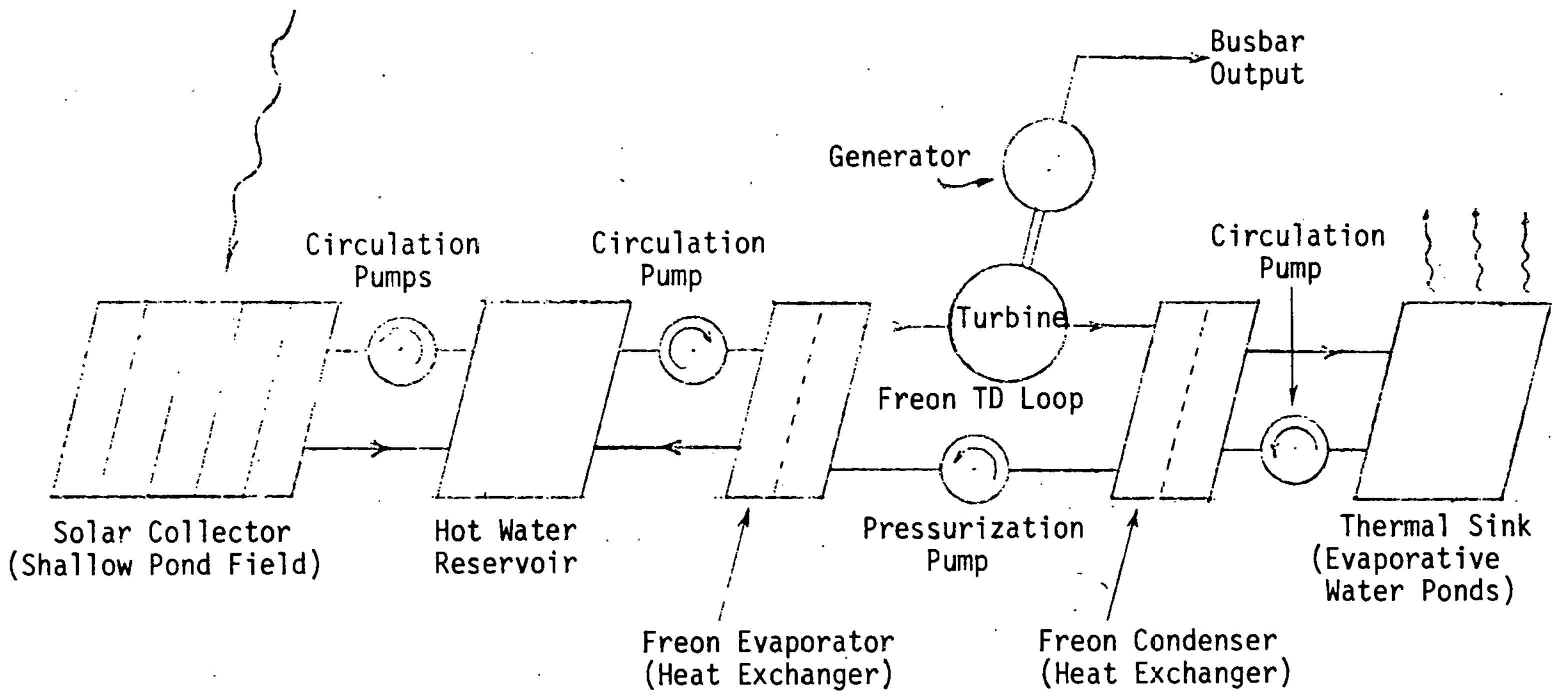


Fig. 1 Shallow Pond System Schematic

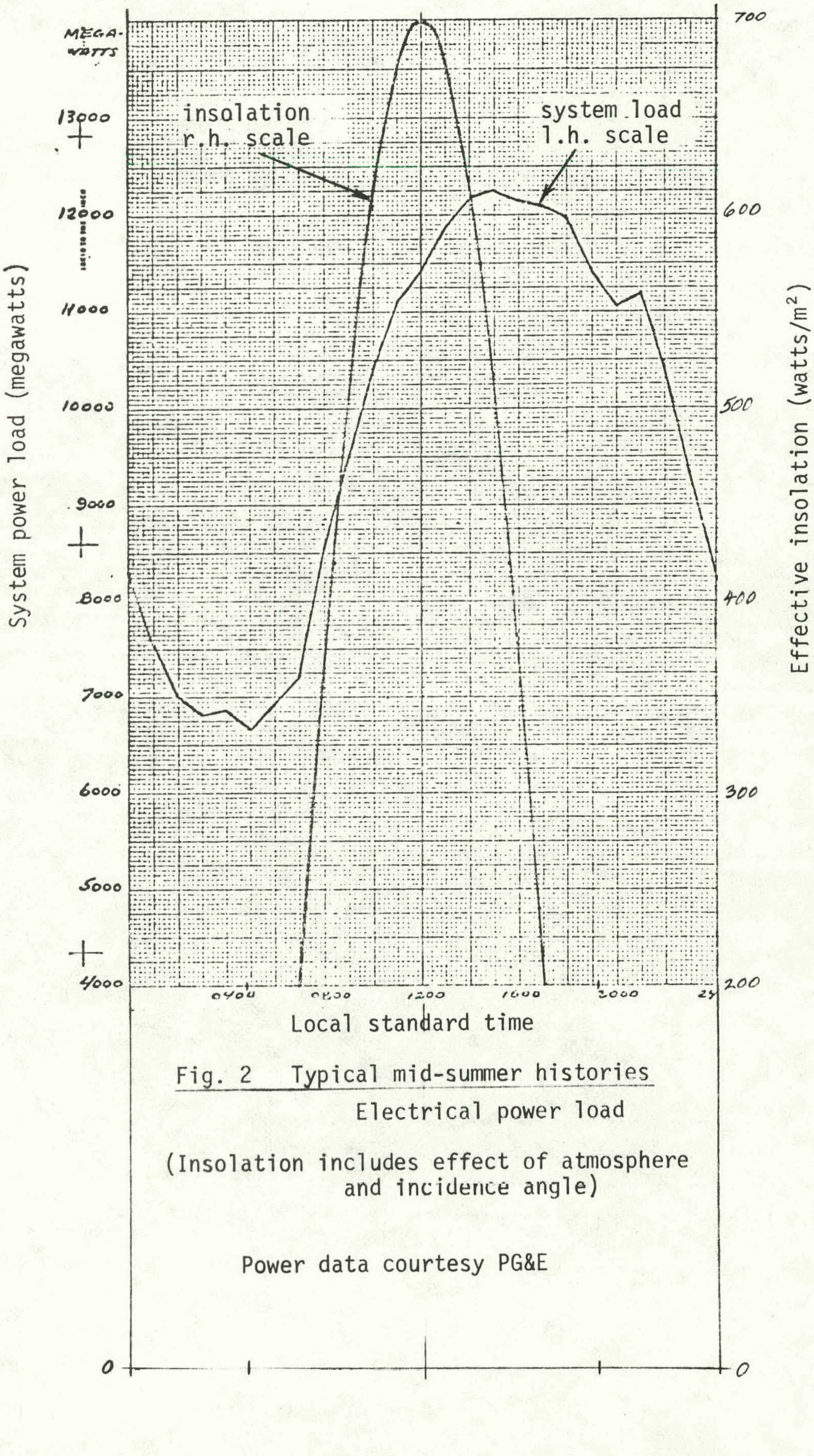
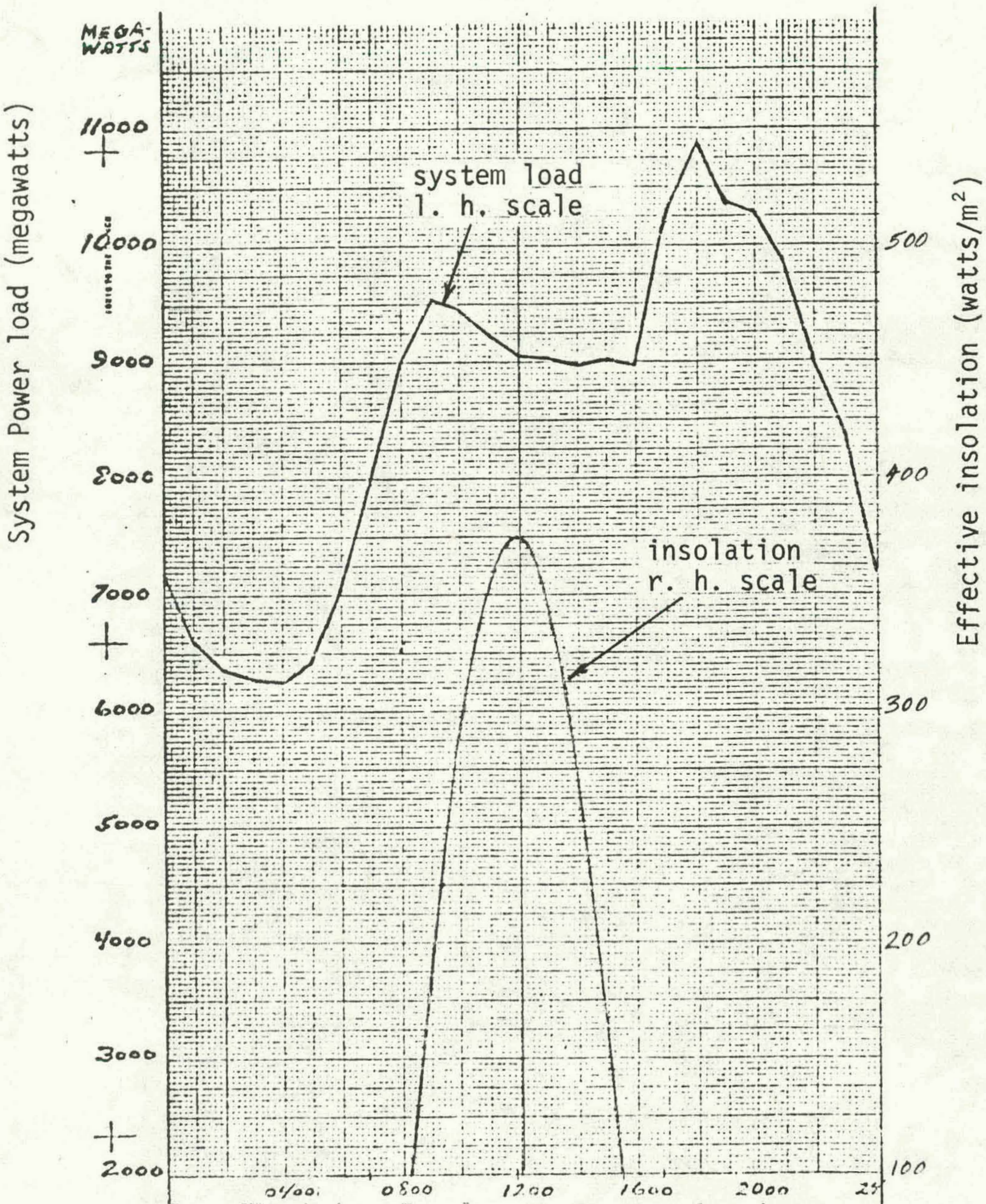


Fig. 2 Typical mid-summer histories
Electrical power load

(Insolation includes effect of atmosphere and incidence angle)

Power data courtesy PG&E



Local standard time

Fig. 3 Typical mid-winter histories (see Fig. 2)

Power data courtesy PG&E

THIS PAGE

WAS INTENTIONALLY

LEFT BLANK

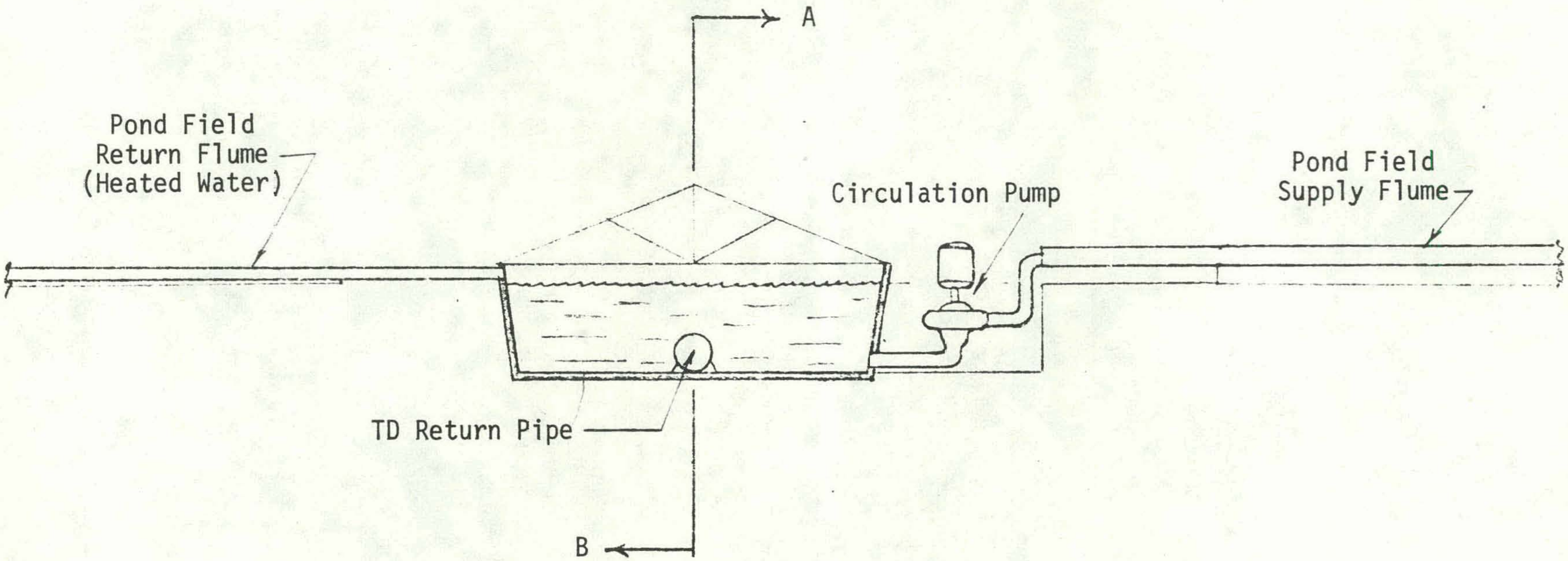


Fig. 5 Sections Thru Central Reservoir

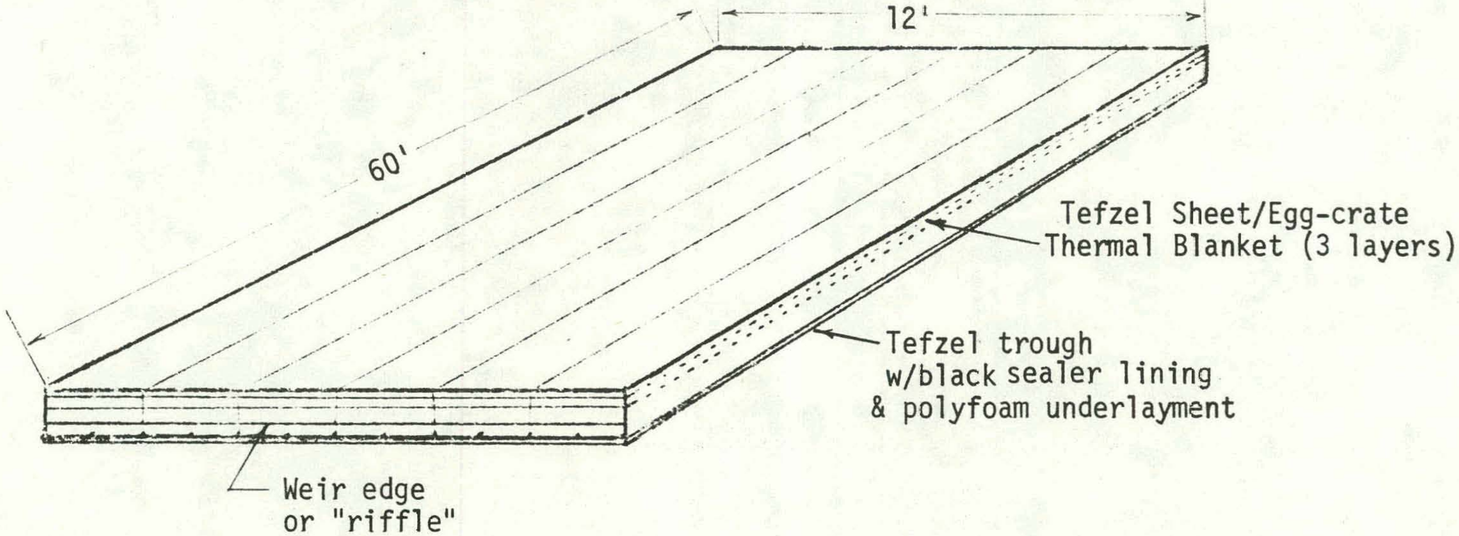


Fig. 6 Prefab Plastic Pond Box Concept

There would be 10 of these, laid end-to-end for each 600' pond strip.

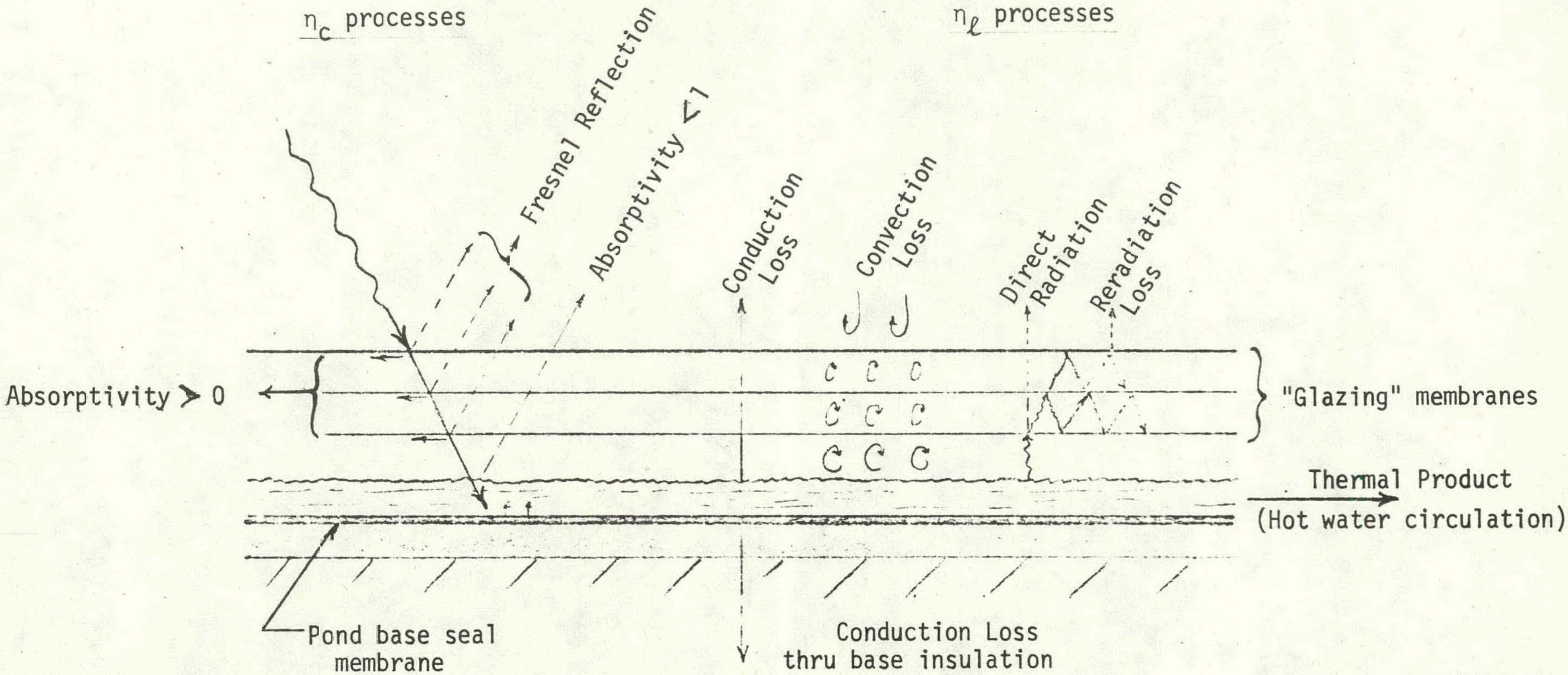


Fig. 7 Thermal Processes in Pond Geometry

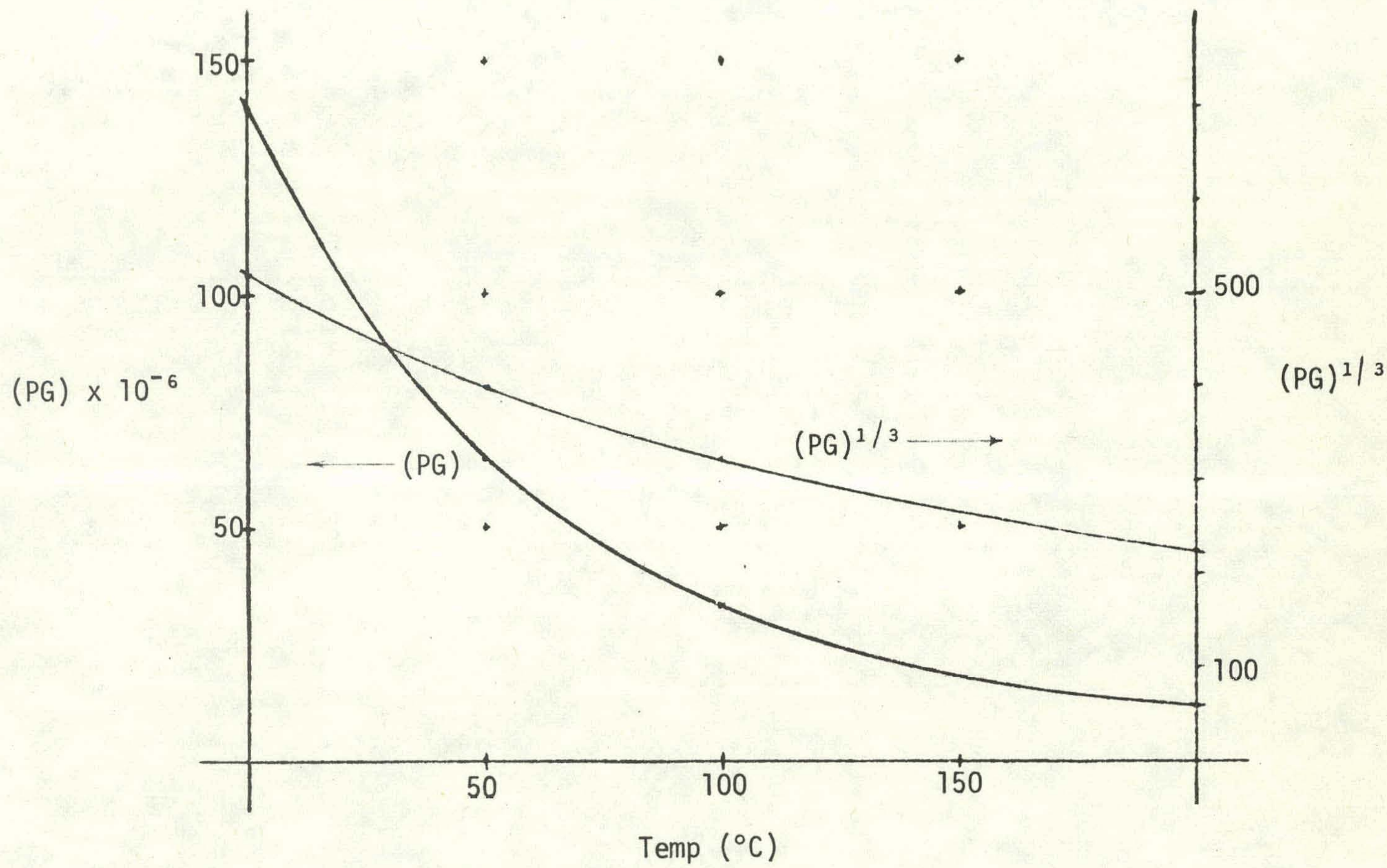


Fig. 8 The Convection Parameter (PG) for air

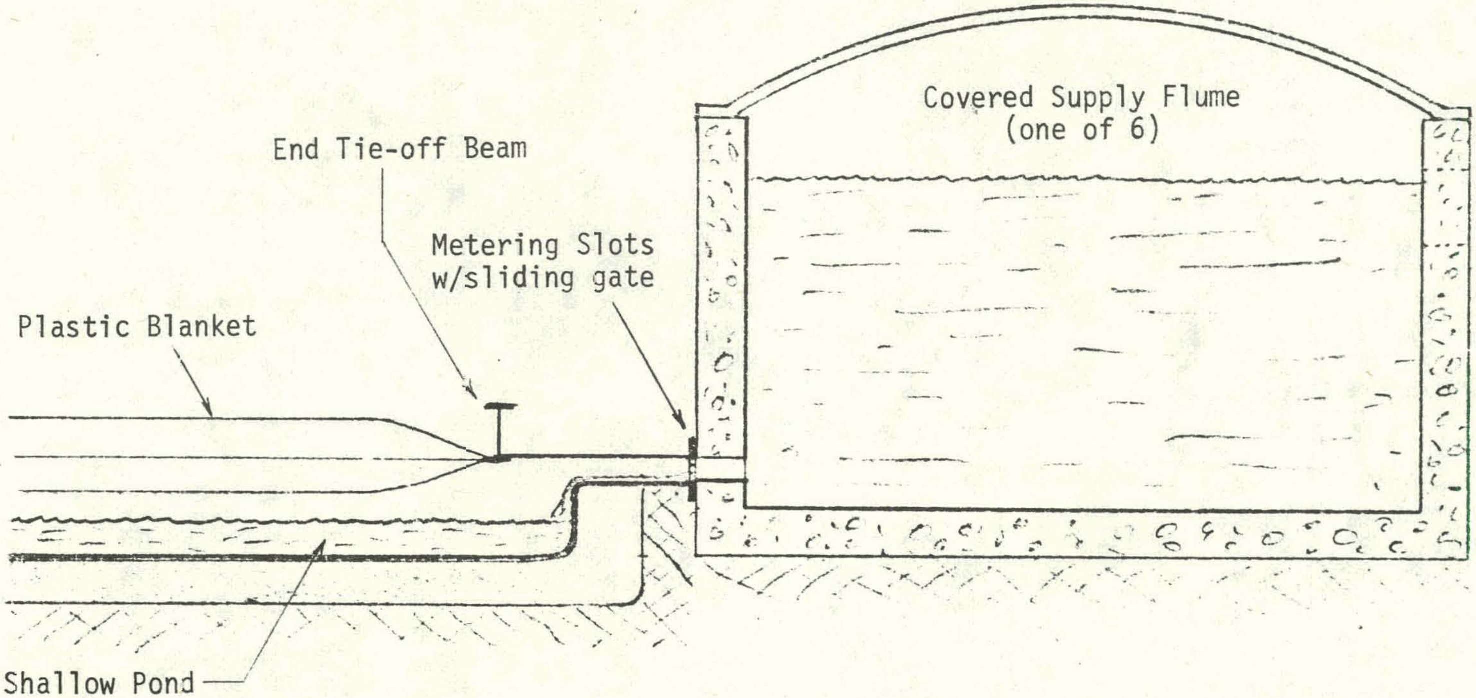


Fig. 9 Distribution Details

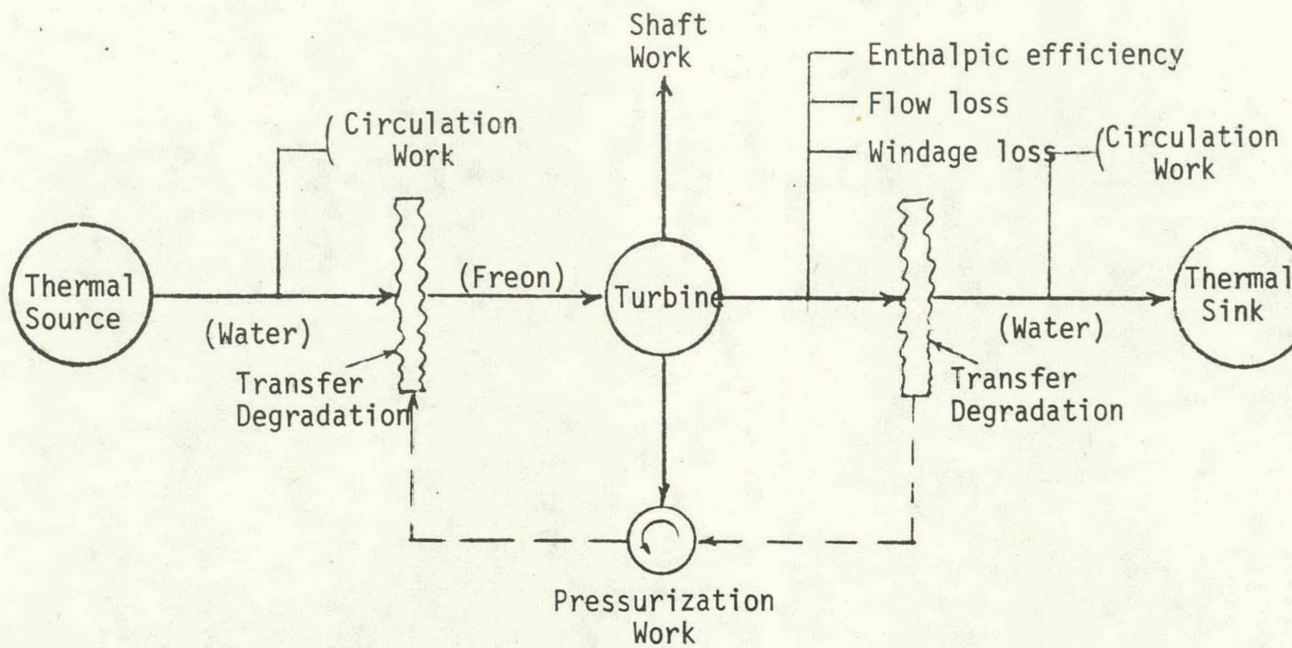
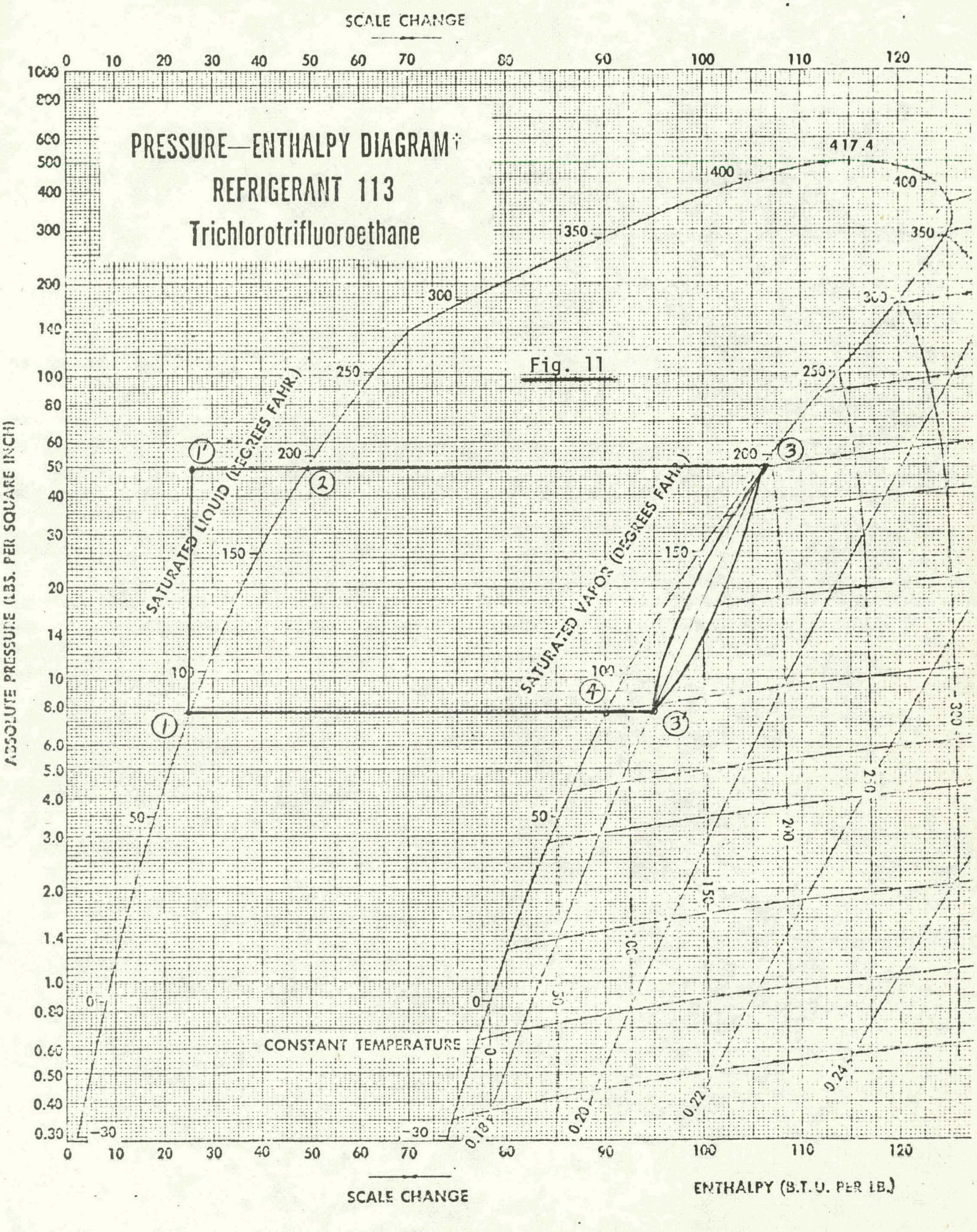
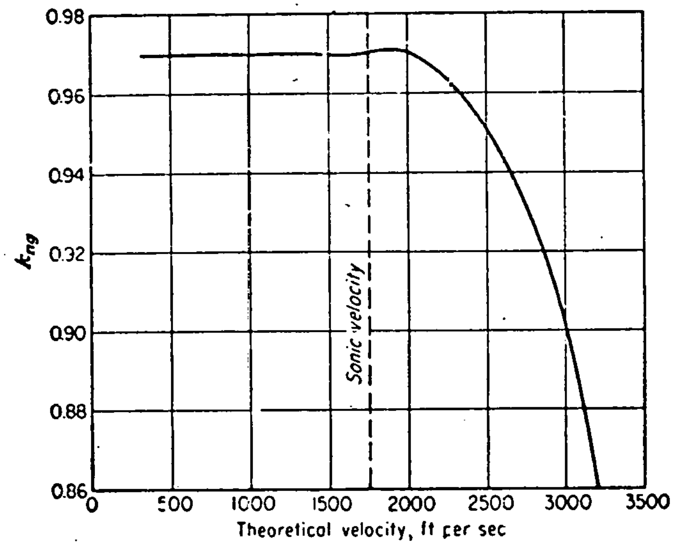
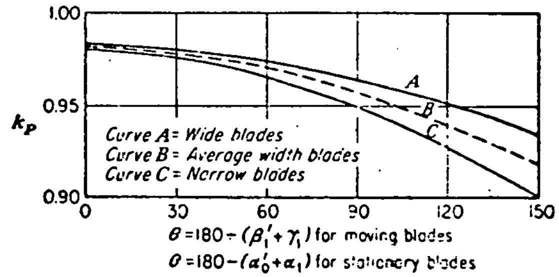


Fig. 10 Thermodynamic Account Schematic

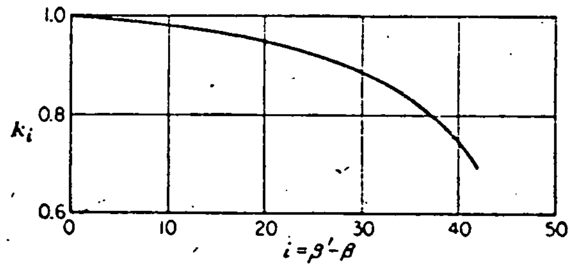

SCALE CHANGE
PRESSURE—ENTHALPY DIAGRAM†
REFRIGERANT 113
Trichlorotrifluoroethane
Fig. 11
SATURATED LIQUID (DEGREES FAHR.)
SATURATED VAPOR (DEGREES FAHR.)
CONSTANT TEMPERATURE
ABSOLUTE PRESSURE (LBS. PER SQUARE INCH)
ENTHALPY (B.T.U. PER LB.)
417.4
1'
2
3
3'
4
1



Typical values of velocity coefficient for superheated steam or gas.



Profile loss coefficient for blades and nozzle foils.



Incidence loss coefficient for standard impulse blades.

Fig. 12 Turbine efficiency parameters

$$\eta_t = k_{ng}^2 (1 + k_p^2 \, k_i) \frac{\cos^2 \alpha}{2}$$

Geothermics (1970) - SPECIAL ISSUE 2

U. N. Symposium on the Development and Utilization of Geothermal Resources, Pisa 1970, Vol. 2, Part 2

APPENDIX A

Geothermal Power Plant on the Paratunka River

V. N. MOSKVICHEVA * AND A. E. POPOV *

ABSTRACT

The problem of deep earth's heat utilization is a currently central one, both from an economic standpoint and for the purpose of power production in the areas distant from fuel bases.

The use of low boiling substances as working bodies in power plants opens up strong possibilities for the geothermal sources utilization.

This paper represents the first data on the Freon power plant operating in the USSR from 1967.

The interest in the problem of hot geothermal water utilization has recently grown in our country and also abroad. A number of reports dealing with this problem and short communications made by various authors at the VII International Conference were dedicated to this topic (Moscow 1968). Geopower projects are at present developed in Italy, Iceland, USA, Japan, Hungary, New Zealand and Mexico. A committee for geothermal studies was organized in our country headed by academician A. U. TIKHONOV at the Academy of Sciences of the USSR. Also a Department of Deep Earth's Heat Utilization has been newly organized by the USSR Ministry of Gas Industry whose charge was to build up hot water drilling projects and develop methods of their utilization all over the country. A number of newly born craft boards, such as North Caucasian Board and Kamchatka Board have recently been set up to deal with hot springs utilization. The problem of hot water utilization is especially important for the Kamchatka region, as this vast and distant area with severe climatic conditions has no mineral resources of its own. Moreover, fuel imports costs are too high.

Therefore it is reasonable to build up a turbo unit with heat transfer fluid of low boiling temperature using hot water of the Kamchatka's springs of about 85-130°C. The heat transfer fluid of the first unit was Freon-12, of domestic high-grade production. The main problems encountered were: 1) Creation of a pioneer Freon turbo unit of industrial importance and study of combined operation of the unit as a whole. 2) Practical testing of the possibility of assembling and operation of the Freon turbo-unit in the conditions of operating steam-power plant. 3) Building-up of geopower plant with a Freon turbo-unit. Solution of all these problems would provide some valuable evidence on the problem of binary water-freon cycles. The problem of utilization of binary water-freon and water-ammonium cycles is being investigated in the Soviet Union, USA, England and Poland. The British Center in Marchwood reported a comprehensive programme of building-up of water-freon single-shaft turbo-units of 2000 MW. This programme provides for an experimental water-freon plant of about 2000 kW with four experimental turbines. A design of a freon-power plant was developed by the Institute of VNII Refrigeratory Industry, the project worked out by Thermal Physics Institute, Siberian Branch, Ac. of Sciences. The construction was realized by several industrial enterprises of the Ministry of Chemical Machines and Ministry of Heavy Machine Industry, Transport and Power Machine Industry.

The testing of the turbo unit was conducted at the machine-hall of the Shatur HEPP-5 (Figure 1).

The results of the tests are as follows.

1) Operability of the project of freon-cycle plant and the basic equipment of the plant: centrifugal turbine, system of controls, heat transfer apparatus and sealed freon rotary pumps.

2) Testing of the unit was not completed, as expected by the programme of testing. No duty tests were made because of the lack of cooling water of required temperature, which did not permit one to conclude whether this unit meets engineering requirements. The temperature of the incoming water in condenser was 14-20 °C. Peak power of the terminals was 340 kW. The diagrams of η_{ad} and « *Ga* » discharge of the Shatur testing programme are listed below.

3) The analysis of the obtained data allows one to conclude that turbine power and nominal operation power approaches the designed one. The accuracy of coincidence of virtual and designed data must be checked in the test for the nominal operation of the geopower plant.

4) Some work was carried out to make some of the assemblies more reliable to ensure its normal

* Thermal Physics Institute, Siberian Branch, USSR Academy of Sciences, Novosibirsk, USSR.

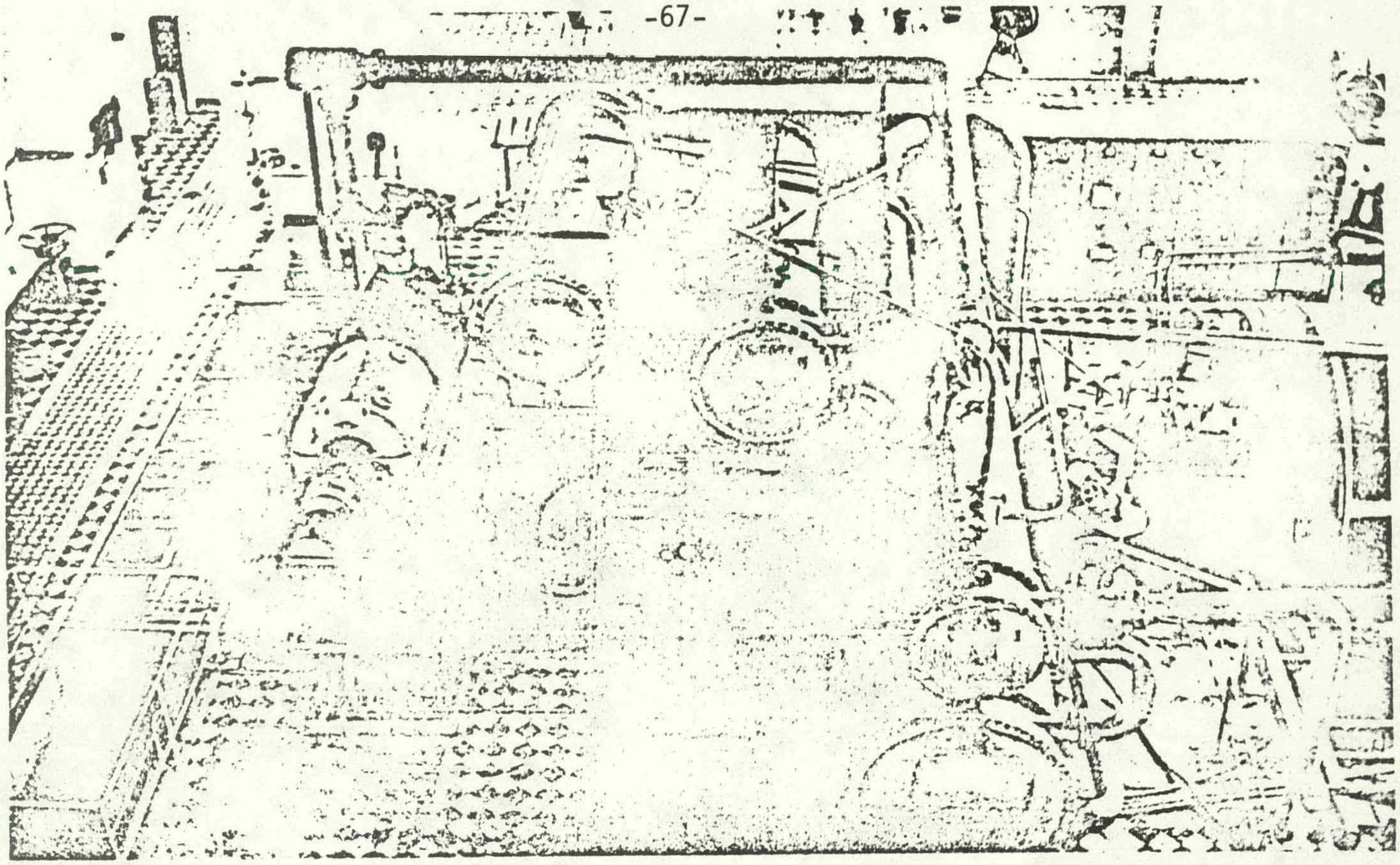


FIG. 1. — *Machine hall of the Shatur HEP-5.*

operation; some recommendations were offered as to its later improvements.

5) The testing time of about 200 h was insufficient for estimation of mechanical motor-source and unit durability.

6) The results obtained and conclusions about the tests allowed one to adopt the decision of shipping the UEГ 90/65 unit to the experimental geopower plant of the Thermal Physics Institute of the Siberian Branch, USSR Ac. Sci., in the middle Paratun'ka springs of the Kamchatka, for subsequent testing.

Design of the plant was completed by a staff of research workers from Novosibirsk Branch of GIPRONII of the Siberian Branch of the Academy of Sciences of USSR. Construction was performed by the « Kamchatka-Selstroy Thrust ». Assembling of the plant was carried out by « Dalenergomontahz ». The project realization was carried out under the guidance and direct participation of the Thermal Physics Institute.

The pattern of the project was designed to satisfy two purposes: to be the experimental laboratory for testing various methods of hot water utilization and to operate as a part of the engineering project of the large-scale hot-house-green-house State farming (Figure 2).

The sources of hot water (wells 300-600 m deep and Ø 127-200 mm) are situated at 1.5 km from the geopower plant; a hot water pipeline system 1732 m long, Ø 377 m is laid from the source. Double row of supports is designed for the second pipeline systems

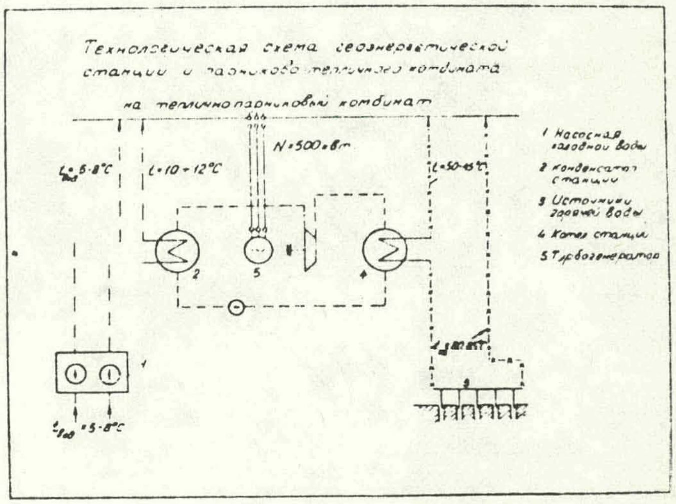


FIG. 2. — *Technological scheme of the geothermal power plant and hot-green house station.*
1. *Pumping station for cold water* - 2. *Condenser* - 3. *Hot water source* - 4. *Heat exchanger* - 5. *Turbo generator.*

for the hot-house-green-house State farms. Operation of this hot water pipeline provides some valuable information for designing hot water supply systems in the north of our country. The engineering utilization of the hot water (45°C) cooled in turbo unit is effected by a system of soil-heating of the hot-green houses, while circulating water heated in condensers is applied for the plant watering, for Paratun'ka river water temperature of 5-7°C cannot be used for these purposes.

A view of the experimental station on the Paratun'ka river is shown in Figure 3; in Figure 4 a view is drawn of the turbo unit and mechanical hall. The pumping station of circulating water is in a separate house to serve simultaneously for watering of the green-house State farm. The water-supply station is situated at the river bank. Maximum hot water discharge to provide for the plant needs is 280 m^3/h, while cool water discharge is 1500 m^3/h.

The tests were carried out in conditions other than provided for by the design. Hot water temperature = 81°C against 90°C (given); cooling water temperature = 6-8°C against 5°C (given).

Testing procedures synchronized with the period of the hot water fields, which placed additional difficulties.

During the tests, all the systems and units operated satisfactorily ensuring normal operation of the unit. Compaction of the freon system was reliable with leaks being within the calculated standards. Systems of control and protection of the turbo unit ensured its normal operation. Stuffing box of the unit ensured sealing of the turbine both in operation and stop. Grease leak through the stuff did not exceed 3 droplets per minute The system of pressure maintenance in the boiler was reliable, though pressure variations were a bit higher than expected. We failed to achieve the operation of automatic control system of the feed-pumps because of the

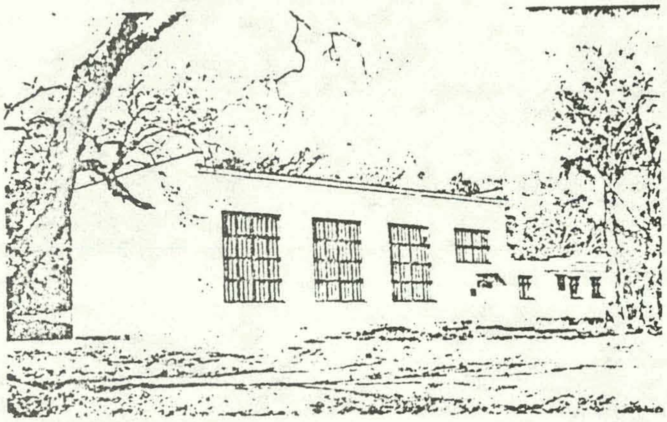


FIG. 3. — *General view of the Paratun'ka river geothermal power plant.*

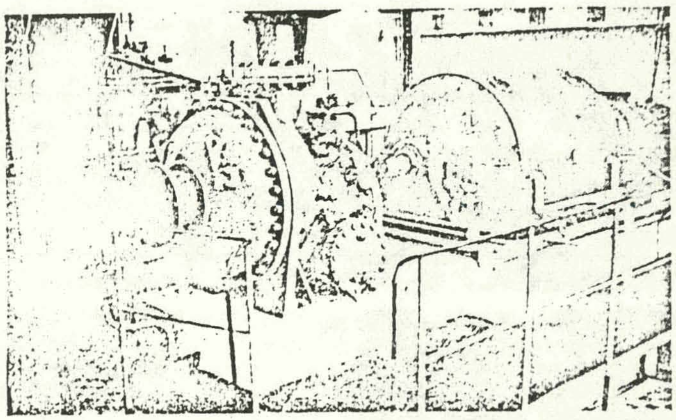


FIG. 4. — *Turbo-unit in the machine hall.*

The construction had been completed by September, 1967.

The start-up of the unit was effected by November 4, 1967.

Testing of the unit was carried out by the Thermal Physics Institute together with VNII Refrigeratory Machines in collaboration with the Kaluga Turbine Plant and Experimental Plant of Hydraulic Machines.

On starting up the plant in June-August, 1968, a programme of testing the plant was initiated.

The object of a series of testing of the UEF 90/05 unit at the experimental geothermal plant of the Thermal Physics Institute was: 1) to check up the operability of the unit at nominal regime under conditions of operation, 2) to obtain external characteristics of the unit, i.e. correlation between the power of terminals and hot water discharge, 3) to confirm the agreement between the results obtained and the data calculated, as anticipated from the project.

lack of reliable freon gage. The turbo unit is easy to operate, the start up and stopping of the units is quite simple. Power supply system ensured synchronization of the starting diesel and turned out to be quite reliable under load.

After 120 hours of operation, the unit was checked up. The check indicated that machinery starting up the turbine was in a satisfactory state, the operating wheel was in a perfect state. High deterioration of the unit nozzle was marked. The latter had begun earlier at the time of Shatur test programme. Deterioration of the vane over its thickness was 3-4.5 mm. The nature of deterioration indicated that it was due to the shocking action of the droplets of unevaporated freon, possibly due to some other rigid foreign bodies (rust, sand or slag). Fluid droplets and solid particles are ejected together with the gas to the wheel. Gas penetrates to the discharge site of the turbine while solid particles and droplets are thrown away by the centrifugal force to the nozzle. This results in determination and deflection of the existing rim of the nozzle.

The main external characteristic of the geothermal unit is its hot water discharge per unit power.

In Figure 5 is shown an experimental curve of the turbo unit power as a function of hot water discharge. The same diagram drawn in Figure 5 shows the curves of capacity when hot water temperature is between 81°C and 90°C. The unit was designed for operation at hot water temperature of about 90°C. However, real temperature of the Paratun'ka wells was between 81 and 81.5°C. Such temperature lowering resulted in higher hot water discharge. The peak attained at the terminals of the turbogenerator was to be 680 kW for the given conditions of tests. Plotted also is power self-discharge. This power is made up of the power of two

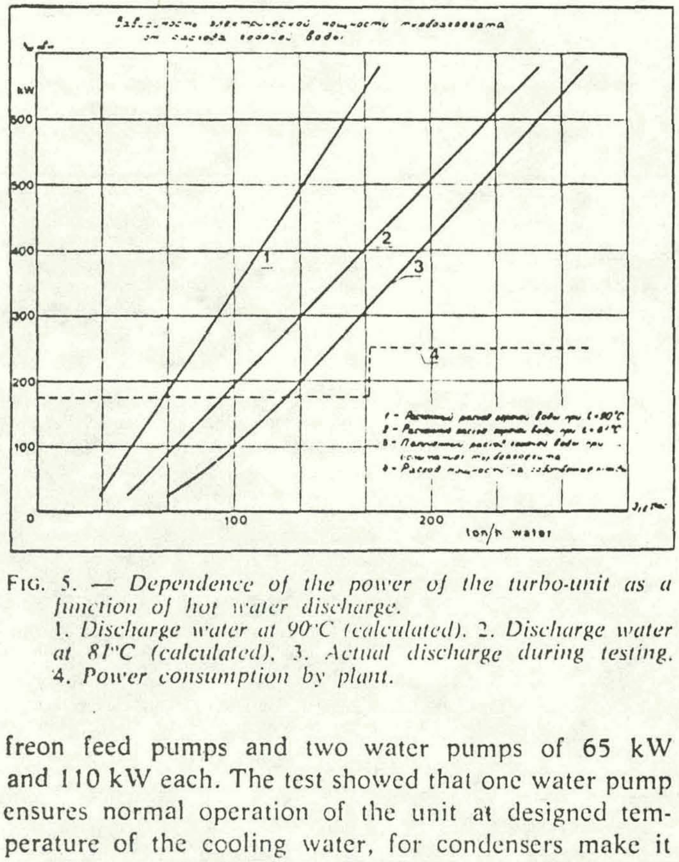


FIG. 5. — *Dependence of the power of the turbo-unit as a function of hot water discharge.*
1. *Discharge water at 90°C (calculated).* 2. *Discharge water at 81°C (calculated).* 3. *Actual discharge during testing.* 4. *Power consumption by plant.*

freon feed pumps and two water pumps of 65 kW and 110 kW each. The test showed that one water pump ensures normal operation of the unit at designed temperature of the cooling water, for condensers make it possible to maintain the designed pressure and temperature of condensation. The tests of the series chemical pumps have demonstrated that their virtual capacity did not exceed 30%. When more elaborated pumps of normal efficiency were applied, the self-discharge power would reduce itself by twice the amount.

For one cooling water pump and two freon pumps to operate, the self-discharge power is to be 240 kW, i.e. about the designed value. The jump in the self-discharge power shown in the diagram is due to the setting into operation of the second freon pump. At given conditions of the test the unit withstood external load up to 450 kW.

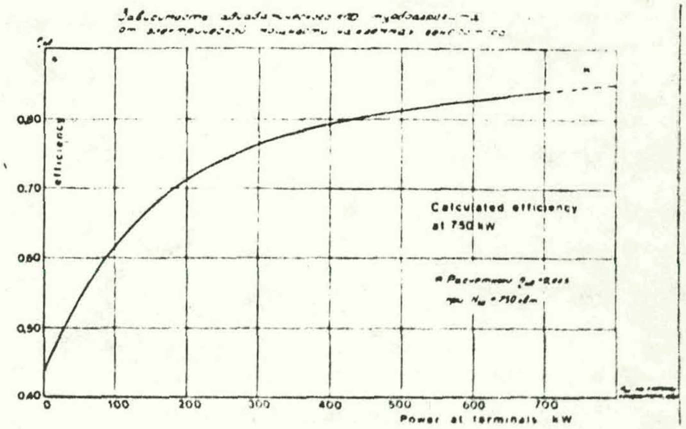


FIG. 6. — *Adiabatic efficiency of the turbo-unit as a function of electric power at the generator terminals.*

Figure 6 shows the change in the efficiency of the stage of freon turbine depending on the power at the terminals with a new nozzle device. As expected, the efficiency increases monotonously with power and 680 kW load attains 0.82. This adiabatic value of the stage efficiency is lower than the designed (η_{ad} = 0.84) and higher when we deal with the heated freon vapours. Thus the efficiency value obtained for the turbine stage, being the first of the experimental freon turbine may be regarded as satisfactory if one takes into account total lack of somewhat experimental data on shaping and designing of turbine stages operating with freon vapours.

In Figure 7 are drawn the curves of the change in the efficiency and discharge as a function of power for the turbine operating with new and deteriorated nozzles. Listed also are the experimental results of the tests shown in the control board of the Shatur HEP.

FIG. 7. — *Comparative data of the turbo-unit in operation under different conditions.*

η_i and Ga - new unit η_i' and Ga' - used unit

η_i'' and Ga'' - test in Kamchatka

Water discharge in operation with new and deteriorated nozzles is almost the same. The discharge reported during the plant tests is extremely high. This may be explained by the fact that at the time of plant tests the unit was operating at the temperature of condensation $t_k = +32°C$ ($t_k = 15°C$, designed) and, hence, much lower in temperature difference as compared with the designed one.

From comparison of the curves of efficiency it may be concluded that they do not differ from each other. The efficiency values for plant tests are also much lower in connection with higher temperature of condensation.

When a deteriorated nozzle was used for turbine operation, the curve of efficiency has shown itself at maximum for the power equal to $N = 430$ kW. In this case the efficiency of $\eta_{rel} = 0.805$. The shift in the region of highest efficiency in this case may be explained by smaller straight cross-section of the nozzle and smaller flow ejection angle due to the bend in the exit rims of the nozzle vanes.

From the spring of 1970 the experimental geopower plant has been operating in the normal operational conditions to meet the requirements of the hot-green houses of the State farms and the village of this region in power, hot water and irrigated water supply. At the same time the tests of all thermal and engineering systems complex of the river Paratun'ka will be carried out and hot water utilization as well.

REFERENCES

ARONSON D. 1965 — Binary cycle for power generation. *Proc. amer. Power Conf.* 23.

BADYLKES I. S. 1962 — Working substance and processes of refrigeratory machines. *Gostorgizdat.*

KANAEV A. A., KOPP I. Z., KUTATELADZE S. S., MORGALIN G. P., MOSKVICHEVA V. N., ROZENFELD L. M. 1968 — Water-freon power plants of high productivity. *8th Int. Energy Conf.*.

KUTATELADZE S. S., ROZENFELD L. M. 1965a — Principles of energetical utilization of geothermal sources. *Izvestya SO AN SSSR*, 1.

KUTATELADZE S. S., ROSENFELD L. M. 1965b — Problems of geothermal power utilization. *Vestnik AN SSSR*, 10.

KUTATELADZE S. S., ROSENFELD L. M. 1965c — Working fluid for geothermal power plant. *License N 941517/24-6 February* 3.

KUTATELADZE S. S., ROSENFELD L. M., VYMORKOV 1965 — Geothermal power plant. *License N 941517/24-6 of February* 3.

1965 — 200 MW turbine generators using steam/freon cycle. *Electr. Rev.*.

1966 — Geothermal study and utilization of the earth's heat. *Izd. Nauk.*

APPENDIX B

October 22, 1973

LOC 73-9, Revision 1

MEMORANDUM

TO: J. N. Shearer

FROM: L. F. Wouters

SUBJECT: Shallow Pond Test Facility - Technical Issues

It may be useful to briefly review some of the technical aspects of a Shallow Pond Solar Power System, to which a small test facility would be addressed. As Arnold Clark recently pointed out, there are two broad demonstration issues:

1. Economic viability: Can it really be built cheaply ? Are the means of embodiment sufficiently stable and reliable ? How long will the plastic last ?

2. Technical viability: Will it work from a thermal standpoint ? "Of course the water will get hot" - Really ? How hot ? Will it heat at a rate within range of expectations and will it attain a corresponding equilibrium temperature ? Will there be in fact, a perceptible and recoverable excess of thermal energy over that lost by leakage through the plastic-layered blanket ? Can we show that we know how to predict these features, for larger "extrapolations" ?

The issue of technical viability may be further embroidered by recognizing that plastic sheet materials have not been used in this particular configuration or application. There are a number of quantitative uncertainties with respect to thermal performance in a shallow pond scheme. For example, the appropriate parameters for infra-red emissivity are only roughly known. We thus need realistic "test bench" numbers from which to determine such parameters in general, for these materials, geometries and thermal conditions.

The central technical argument can be developed from the following simple, first-order time-dependent description. Consider just the requirement of energy conservation: The Solar flux input to a representative Pond area is shared between specific heat (as a heating rate), thermal leakage through the blanket, and power removed as work or heat or "something":

$$\eta_c \phi = (\rho C_p) d \frac{du}{dt} + L + W$$

As an initial case, I assume a linear leakage term:

$$L = \alpha u$$

and put all the Solar flux into heating the water ($W = 0$):

$$\eta_c \phi = (\rho C_p) d \frac{du}{dt} + \alpha u$$

η_c expresses the photon capture efficiency of the pond configuration; d is the pond depth, and α is effectively a classical thermal transfer coefficient. (E.g., BTU/sq ft hr °F or cals/cm^2 sec °C.)

This is a traditional problem:

$$\underline{T = T_\circ + \frac{\eta_c \phi}{\alpha} (1 - e^{-at})}$$

with: $u = T - T_\circ$ ($T_\circ$ being the initial temeprature)

$a = \frac{\alpha}{(\rho C_p) d}$ (defining a characteristic time 1/a)

The water temperature behaves as in Fig. 1. The particular analytic case shown here is a 1 cm deep water layer, with three spaced plastic sheets; the corresponding leakage coefficient value is $\alpha = 6.0 \times 10^{-5}$ cals/cm^2 sec °C, and the capture efficiency is $\eta_c = .57$.

Next, consider the steady state situation:

$$\eta_c \phi = \alpha u + W$$

In terms of extracted heat:

$$W = \eta_c \phi - \alpha(T_s - T_\circ)$$

Obviously, in a "successful" pond scheme, $\alpha u < \eta_c \phi$. In particular, unless we take out a quantity W, T_s will reach the boiling point, and W will then appear as latent heat. So in a real test, we will have to "turn on the cooling", so to speak, at a safe point, as also shown in Fig. 1. But it may be useful to also run a "destruction test" just to see how bad that might get.

From a test Pond standpoint, there are then these noteworthy features which deserve quantitative experimental evaluation:

The initial heating rate: $\left(\frac{dT}{dt}\right)_o = \frac{\eta_c \phi}{(\rho C_p)d} = .010\ °C/sec$

The equilibrium temperature: $T_2 = \frac{\eta_c \phi}{\alpha} + T_o = 190\ °C$

The thermal time constant: $\tau = \frac{\rho C_p}{\alpha} d \simeq 4$ hours

The water depth behavior (d),

The truncation to a steady-state situation by introducing a real, competitive, thermal sink W,

The Solar flux or photon capture efficiency η_c: This is a somewhat complicated term; it is dominated by the Fresnel reflection loss and its dependence on Solar meridian angle. Experimentally, it is thus sensitive to the sheet materials, to the number of layers, and to the daily excursion of the Sun. It should be relatively temperature-independent. A verification of its analytic model is always in order. The number used above is taken from the "Solar Shallow Pond Analysis", and it corresponds to Equinoctial High Noon in Arizona.

The α dependence: This is another complicated term. It includes conduction, convection and radiation transfer. Again the number used here is taken from the Pond Analysis. The assumption of a linear leakage (with temperature) is surprisingly good, as shown in Fig. 2. The Pond Analysis suggests a slightly concave behavior, as also shown there. This is intuitively reasonable, in the light of the known non-linear behavior of convection and radiation. The radiation characterization for the plastic sheets, as used in the Analysis, is that of a "mediocre black-body". That also needs numbers. Note this comparison (at 95°C):

Blanket conduction	3.5×10^{-4} cal/cm^2 sec
Convection	15.5×10^{-4}
Radiation	18×10^{-4}
Earth conduction	3.5×10^{-4}
Total leakage	4×10^{-3} cal/cm^2 sec
Solar input	14×10^{-3}

There are several ways to test α: Number of sheets, steady-state temperature (literally-plot the curve by changing W), change the filling gas (experimentally permissible). It should not be sensitive to sheet spacing.

I should not claim, even by inference, that some small test ponds will give us all the precise numbers one will ever need. I suspect that there will be puzzles to unravel here, and then again different puzzles when one scales up to pilot size. But this discussion does indicate that they have a quantitative purpose beyond the mere qualitative demonstration of heating water.

There's one nice thing about these "Shallow Pond-lets" from an experimental standpoint: Their time constant is short enough that you don't have to stand around all month to get some data.

P.S.: Don't be deceived by the "time constant" above. (4 hours) That is not the thermal response time in a power system application. The thermodynamic loop is so sensitive to thermal excursions, that the effective system time constant is of the order of 15 minutes.

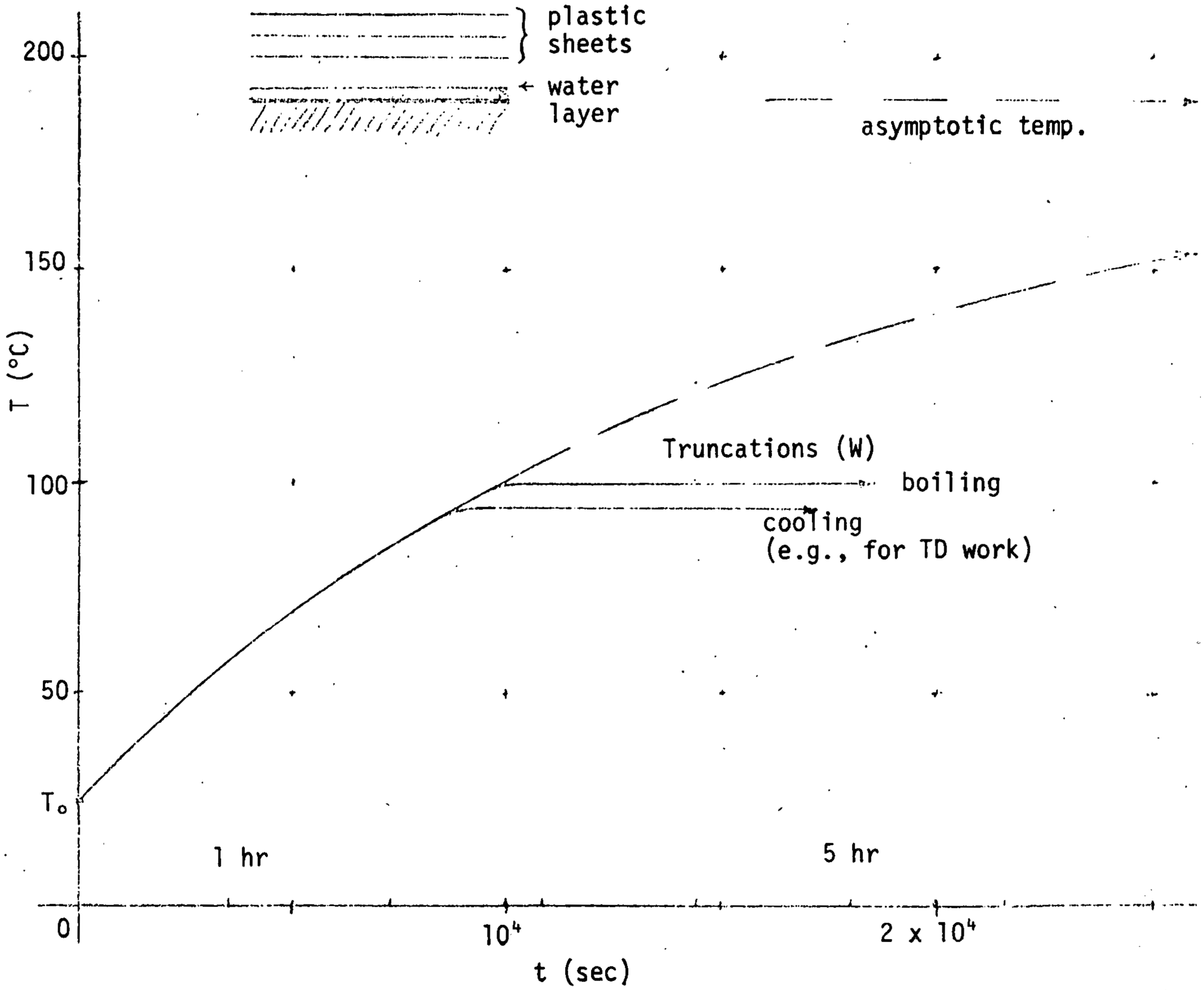


Fig. 1 Pond heating function
(1 cm water layer, constant Sun)

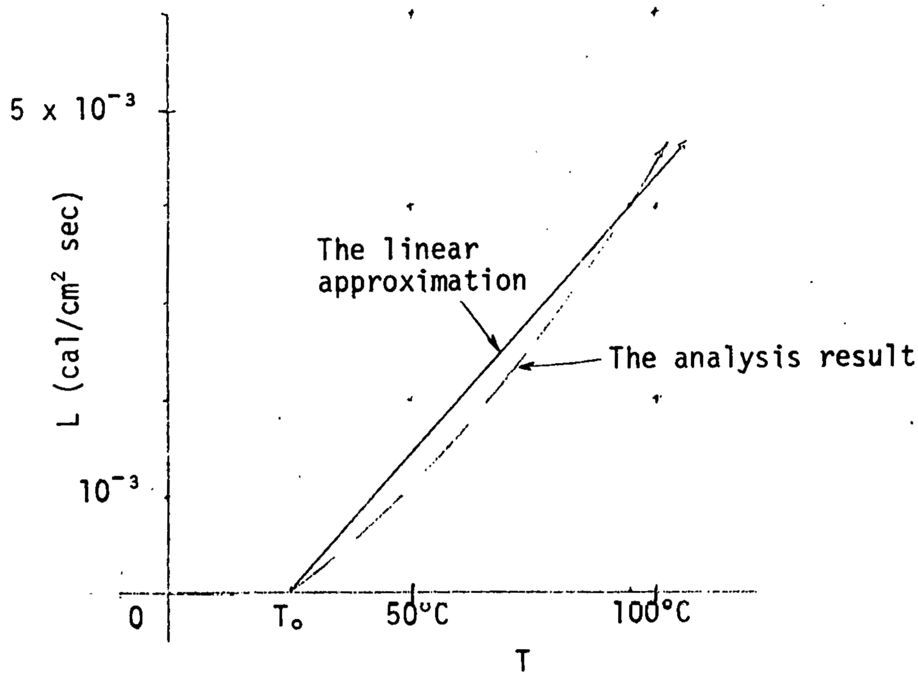


Fig. 2 The Leakage Function

$L = \alpha(T - T_\circ)$

Distribution:

J. E. Carothers
L. S. Germain
A. Holzer
H. B. McFarlane
J. N. Shearer
G. C. Werth
L. F. Wouters (5)
Energy Group (5)
Solar Group (5)
T.I.D. (5)

External Distribution:

C. Backus, Arizona State University, Tempe, Arizona
Prof. H. Buchberg, U.C.L.A., Los Angeles, California
F. Edlin, Sun City, Arizona
Prof. S. S. Penner, U.C., San Diego, La Jolla, California