ADVANCED SOLAR RECEIVERS

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Even though more small cavity solar receivers have been designed, fabricated and tested in recent times, the perennial problem of low thermal efficiency has not gone away. Most solar receivers have not been as efficient as the analysis that went into their design predicted. Energy losses have been 5 to 50 percent greater than anticipated. Perhaps this should have been expected since little optimization was done in the formative periods of system design due to the relatively low cost of the receiver compared to the entire system cost. Recent system designs, however, have paid greater attention to receiver efficiency as a route to greater system efficiency recognizing too that there is a higher probability of good improvement per design dollar here than with already optimized subsystems.

Receiver losses result from all three modes of heat transfer, radiation, convection and conduction. Table 1 indicates how these might be distributed and shows where future improvements could be expected. These are of course temperature dependent, the numbers shown are for a receiver with a cavity temperature about 870°C (1600°F).

At this meeting last year, I showed data which indicated that even though there was not a lot of active receiver development in progress, the prospect of producing a very efficient i.e. greater than 90 percent, small cavity solar receiver was good. As new data became available, that analysis has been kept reasonably up to date and today I can give you a progress report.

The basic thesis that a highly efficient cavity receiver is practical is still a good one. And how we get to that goal is a little clearer. Test runs at the JPL Parabolic Dish Test Site on a Brayton cycle receiver built for us by Garrett AiResearch and a series of tests run for us by Sanders Associates at their Merrimack, New Hampshire test facility have given us better numerical insight into exactly how the losses from small cavity receivers are distributed.

Figure 1 is a cut-away drawing of the Sander's receiver. It was mounted on a test stand and preheated air (T_2) supplied at about 0.25 kg/sec (0.56 lb/sec), a rate typical of the Brayton engines under consideration. The small numbers on the figure indicate thermocouples and the q numbers bracket various zones of the receiver from which heat losses were measured. Table 2 summarizes the results obtained. Several facts are immediately evident from the numbers. The most obvious is that about two-thirds of all losses are in the window frame area q_M. This should easily be reduced by redesigning the insulation inside the cavity and adding external insulation in the frame area. Additional heat could be retained by better insulation systems in the flange area (q_{A3}) and in the outlet duct area (q_C). The overall lesson to be learned here is that conduction losses are not negligible and should receive adequate attention.

To estimate the overall efficiency of the receiver, it was allowed to reach thermal equilibrium and overall losses established from the temperature drop of the air stream. Typical temperature data is shown in Figure 2 and the losses calculated in Table 3. These data allow an overall receiver efficiency to be estimated. Thus, for a 75 kW capacity receiver operating at about 870°C (1600°F), when an 8 percent window loss of 6 kW due to Fresnel reflection is added to the 6.58 kW thermal loss, the overall efficiency is about 83 percent. This value agrees well with previous measurements and suggests that a highly efficient receiver is more likely to be windowless.

The disadvantage of not having a window is cavity convection. While considerable work has been done on this problem for the very large central receivers, not much confirmatory evidence exists for small cavities. This needs to be done especially since it is affected by so many variables such as wind speed and direction, cavity configuration, attitude, temperature, mounting geometry, and others. The highly efficient receiver must have these well under control.

Another major loss mechanism, usually the largest, is radiation out the aperture. But even though it is a large loss, very little work has been done recently. I think this is, at least in part, due to the misperception that there is not much you can do about it. But many routes exist to reduce this loss.

The most obvious of these is to reduce the size of the aperture. Very good systems engineering is essential to balance concentrator performance against costs for a minimal focal plane spot diameter. This also allows for the optimal spillage allowance to be established. Other techniques such as using terminal concentrators should be evaluated.

Within the cavity, several techniques are available to reduce reradiation. Figure 3 illustrates a number of these including overall cavity size ratio, cavity wall configuration, heat exchanger placement, thermal characteristics of the casvity components especially using absorbers and reflectors in an optimal fashion, using secondary heat exchangers as preheaters while cooling cavity elements, and others. Good radiation management is essential and should result in significant performance improvements.

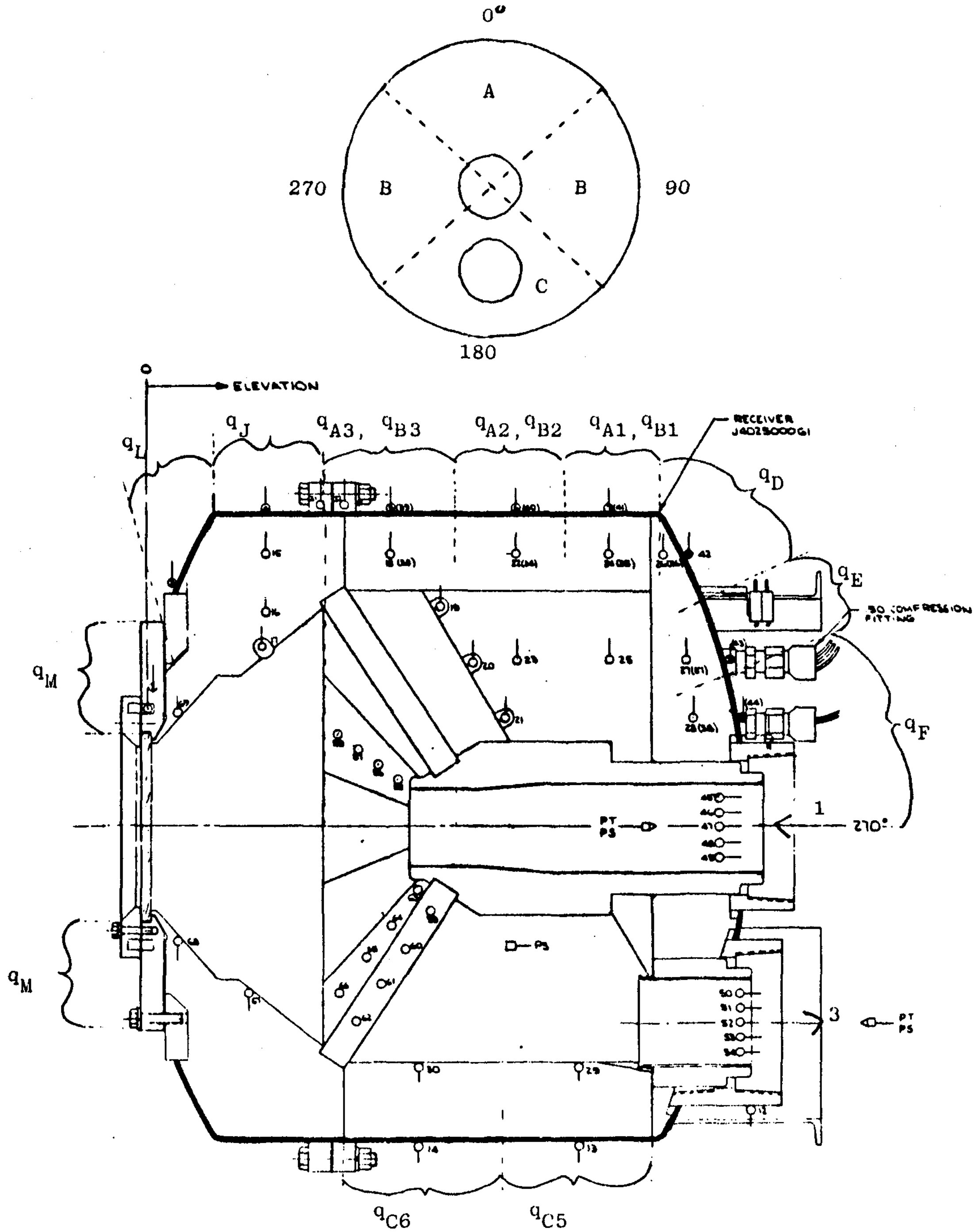
In summary, if careful attention is paid to the overall thermal systems design especially to conductive losses about the window and areas of relatively thin insulation; and if the cavity design is carefully managed to insure a small, minimally reradiating aperture, the goal of a very high efficiency cavity receiver is a realistic one.

EXPECTATIONS

	PREVIOUS LOSS-WATTS	IMPROVED
RADIATION	6000	1000 - 2000
CAVITY CONVECTION	2500	1000
CONDUCTION	2000	500
EXTERNAL CONVECTION	750	500
REFLECTION	500	200
EXTERNAL RADIATION	250	200
	12000	3400 - 4400
EFFICIENCY	85%	94 - 95%

Table 1: Receiver Losses

FIGURE 1: COMPOSITE HEAT FLOW ANALYSIS
FOR SAGT-1A RECEIVER



ONE DIMENSIONAL STEADY STATE COMPOSITE CONDUCTION MODEL

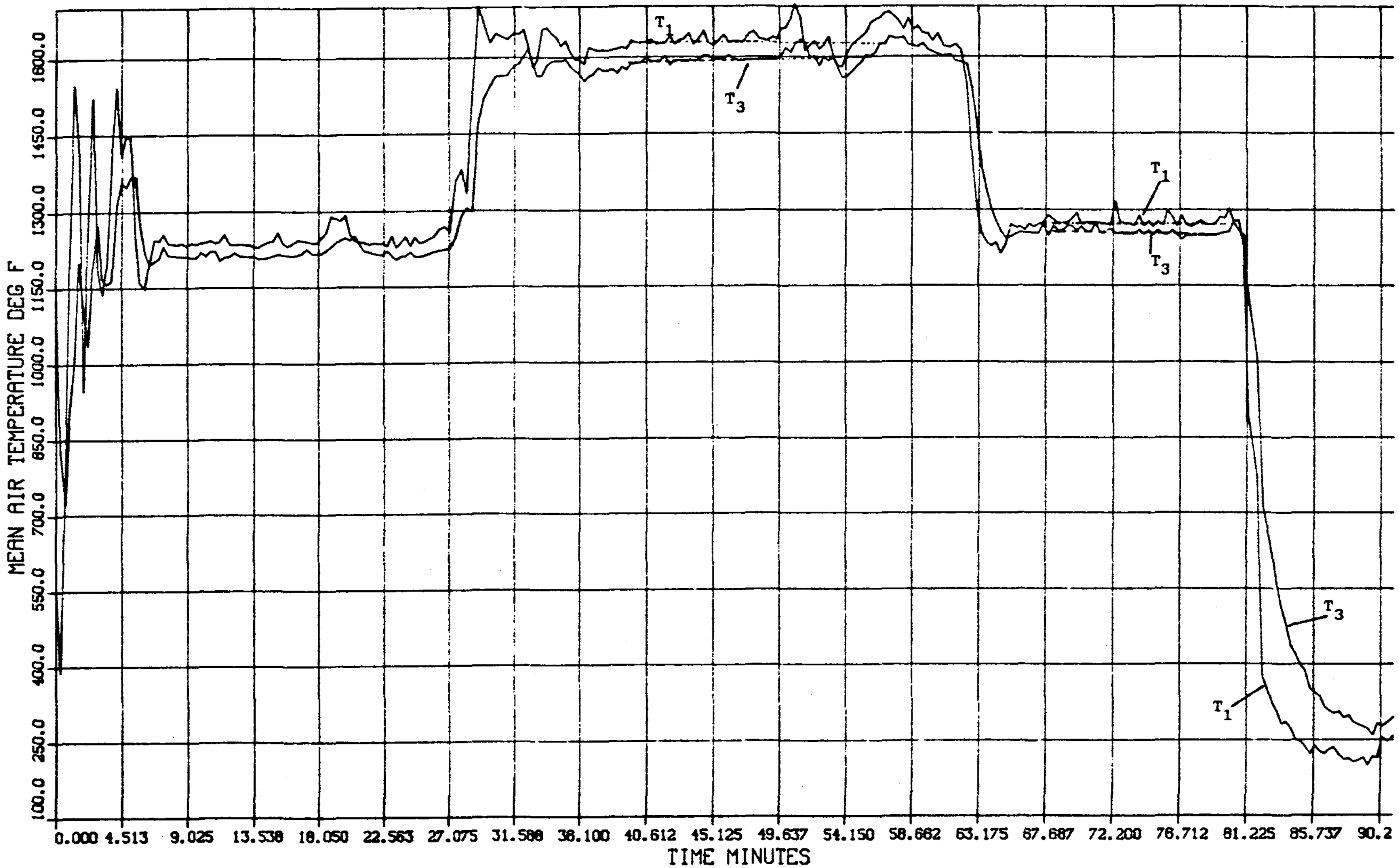
	Test 1	Test 2	Test 3	Test 4	Test 5	Location of TC Nodes
T_1	1271	1360	1478	1580	1648	46,47,48
T_2	1221	1323	1442	1542	1602	51, 53
q_{A1}	1.77	3.12	2.40	4.74	2.69	24, 7
q_{A2}	20.35	25.64	25.59	41.29	33.75	22,6
q_{A3}	62.08	71.13	87.47	97.09	105.59	18, 5
q_{B1}	0.07	0.30	0.51	1.40	1.61	35, 41
q_{B2}	19.80	23.67	25.06	40.50	32.80	34, 40
q_{B3}	69.98	77.35	87.81	104.4	107.81	33, 39
q_{C4}	34.03	38.43	43.48	212.24	229.35	29, 13
q_{C5}	37.87	42.42	47.15	51.83	260.36	30,14
q_{D-}	2.24	38.99	59.25	33.49	39.90	26,36,42,8
q_E	2.29	6.64	6.83	6.73	9.49	27,37,9,43
q_F	1.63	2.45	1.96	13.69	18.42	28, 38,10,44
q_J	93.44	104.57	123.07	139.98	137.38	15, 4
q_L	33.93	38.57	49.72	51.80	55.00	17,3
q_M	1613.71	1837.26	2099.77	2354.45	2605.20	69,2
Q_{SUM}	2083.0 (0.6110)	2413.0 (0.7078)	2773.3 (0.8135)	3300 (0.9680)	3781.7 (1.1093)	

TABLE 2: q VALUES IN BTU/HR, Q_{SUM} in KWt,
T IN $^{\circ}$F, $T_{\infty} = 80^{\circ}$F

$$Q_{SUM} = q_A + 2q_B + q_C + q_D + q_E + q_F + q_J + q_L + q_M$$

FIGURE. 2

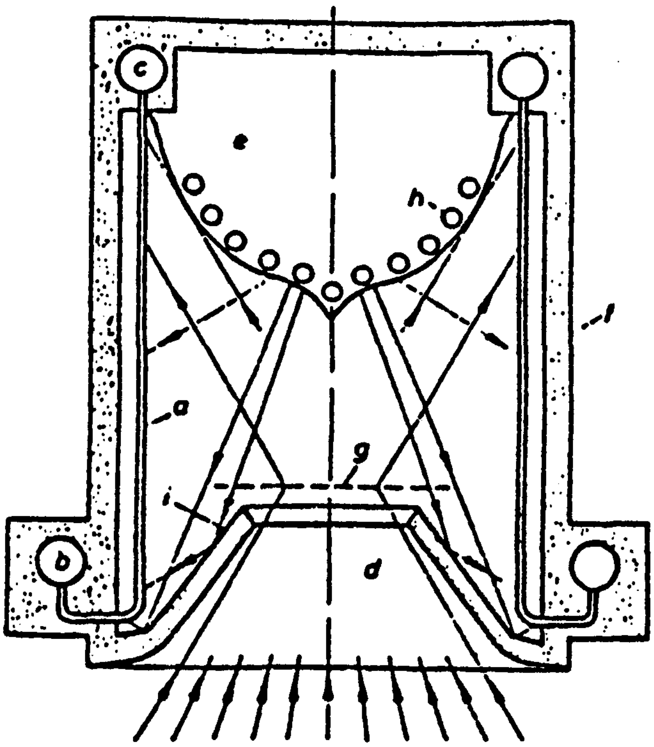
INLET AND EXHAUST AIR TEMPERATURE TRANSIENT



TIME AVERAGED ENTHALPY DROP OVER RECEIVER

Time (min.)	T_1 Ave^{o}F	ΔT ^{o}F	Mass Flow Rate	q Kw_t
34 to 56	1276	16.0	0.572	2.62
67 to 80	1631	38.5	0.564	6.58

TABLE 3 - AVERAGED OVER PERIODS SHOWN IN
FIGURE 2



a receiver tubing
b inlet header
c outlet header
d window
e radiation distribution cone
f receiver cage with insulation
g focal plane area
h cooling tubes
i reflective wall

Receiver.

Figure 3: Cavity Shape Optimization