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Lawrence Livermore Laboratory

PERFORMANCE TEST OF A BLADELESS TURBINE
FOR GEOTHERMAL APPLICATIONS

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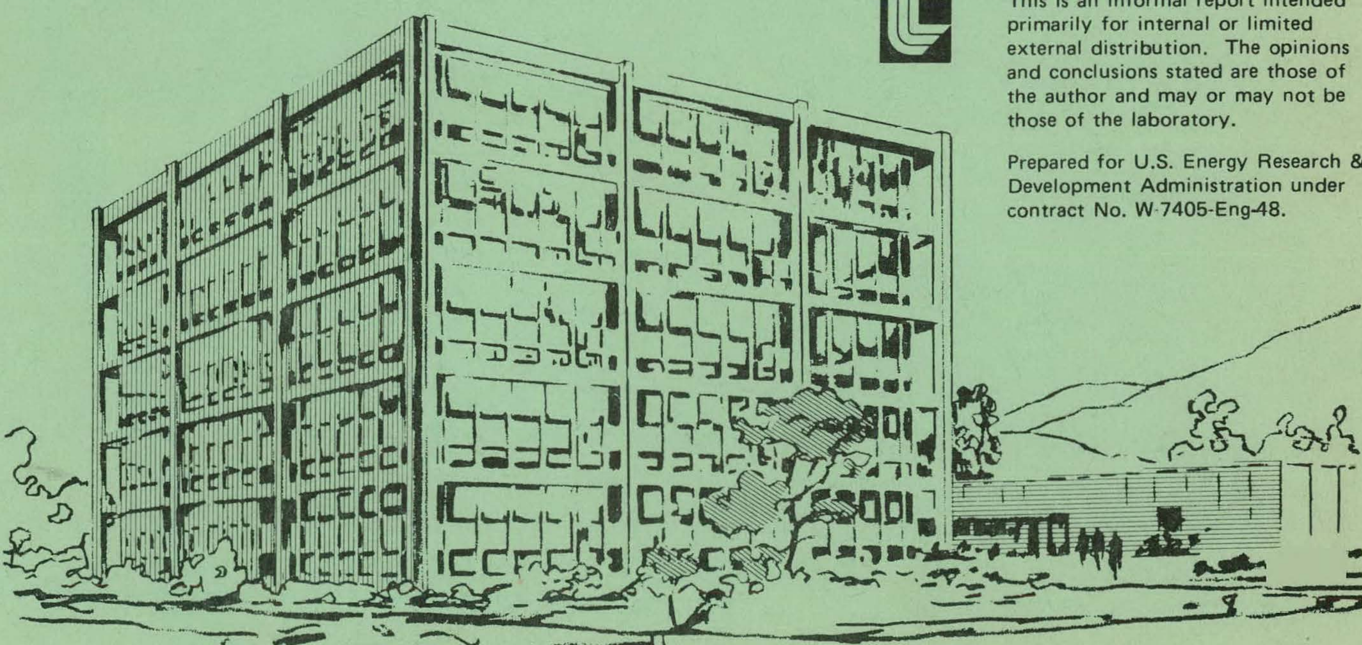
MASTER

March 24, 1976



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INTRODUCTION

The Geothermal Energy Program at the Lawrence Livermore Laboratory (LLL) emphasizes energy conversion for hydrothermal resources by means of the Total Flow process,[1] which has the potential for the greatest efficiency of utilization of the available energy, i.e., direct expansion of the wellhead fluid in a conversion machine. A part of the effort by the Energy Conversion Group is to evaluate potential candidate machines for application in the total flow process. The bladeless or Tesla turbine is one such machine (Fig. 1).

Generally, the wellhead condition of geothermal fluids is a mixture of liquid and vapor, with quality (vapor fraction) up to 40%. Pressure and temperature may be as high as 400 psia (2.75 MPa) and 450°F (505 K), respectively. In some cases, dissolved minerals comprise up to 30% by weight.

The Possell turbine that is discussed in this report (Fig. 2) has been operated in a geothermal environment at Cerro Prieto, Mexico. Its performance, however, was not evaluated, and this study performs that function.

The work described was performed at the Lawrence Livermore Laboratory in its Geothermal Test Facility.[2]

TEST DESCRIPTION

A schematic diagram of the test setup is shown in Fig. 3, and Fig. 4 depicts the turbine under test. Inlet pressure to the turbine nozzle varied from 159 to 410 psia, the vapor fraction from 6 to 15%. In all tests the exhaust pressure was 1 atmosphere.

Figure 5 is a temperature entropy diagram showing the state points of the fluid at various locations.

In operation, the hot water generator established state point 1 (Fig. 5), and throttling of the fluid through the system control valve to a prescribed pressure defined state points 2 and 2'. The eddy current brake and the mass flow rate were both adjusted to obtain a desired speed.

The losses in the bearings and the timing belt were measured separately by means of a dc motor driving through the eddy current brake (Fig. 6). These were added to the measured power to give the total output of the engine.

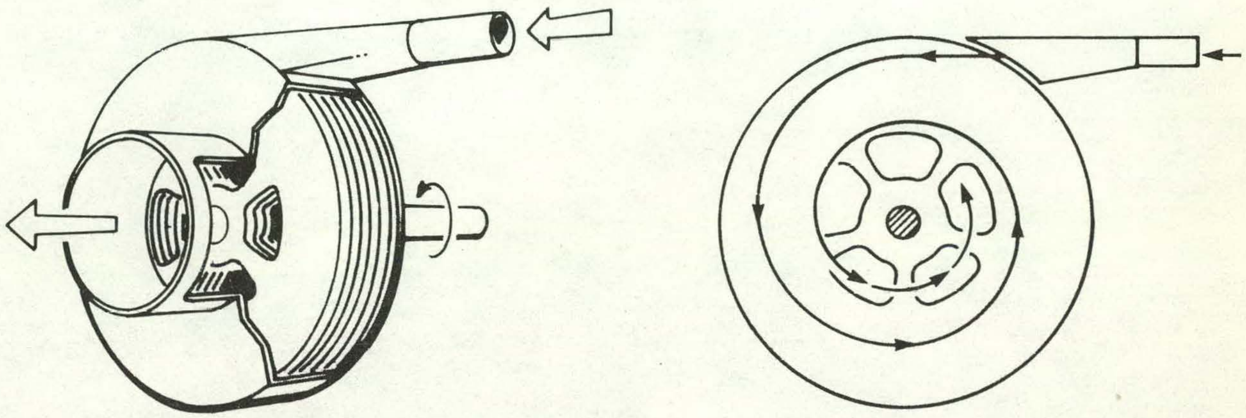

Fig. 1. A bladeless or Tesla turbine.

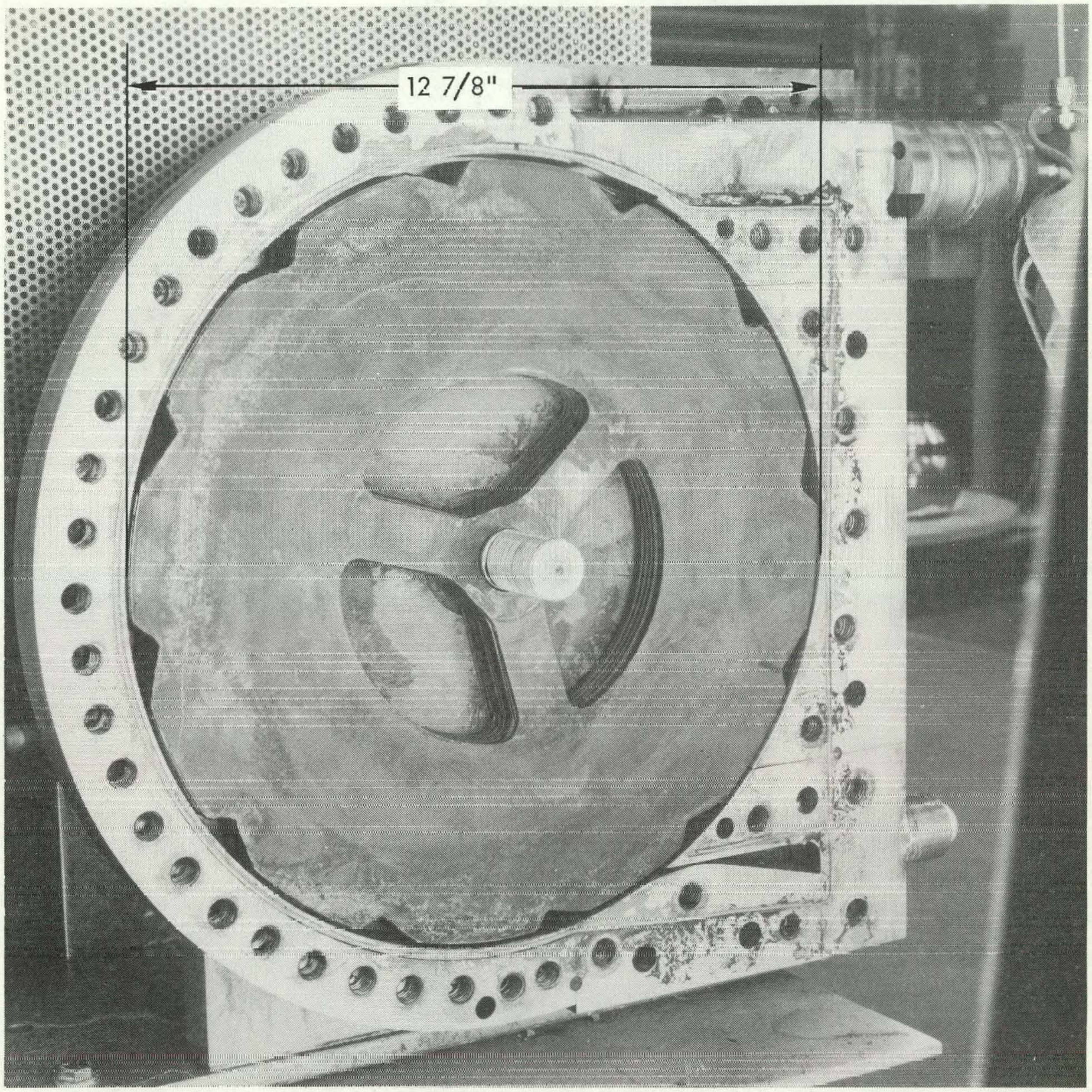

Fig. 2. Possell bladeless turbine.

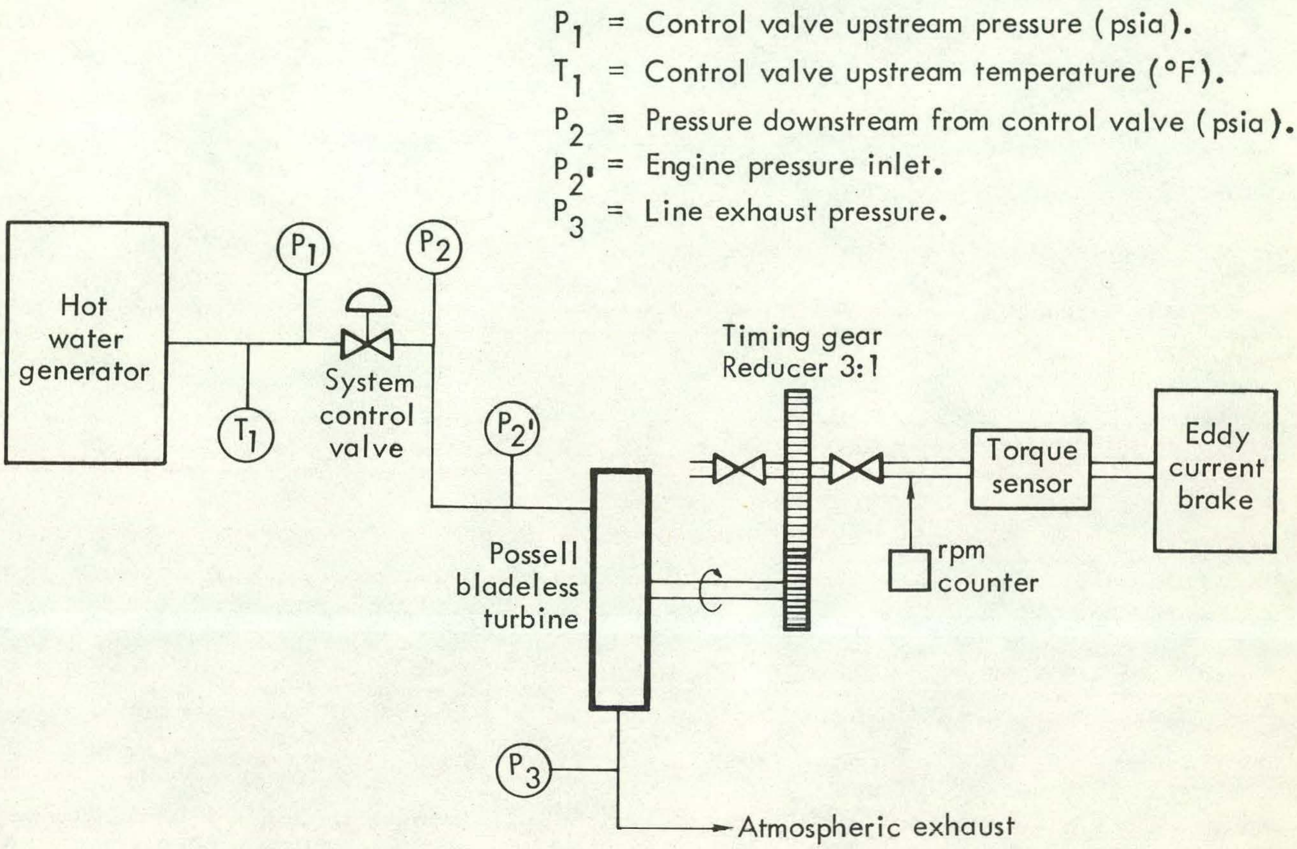

Fig. 3. Test setup of Possell bladeless turbine, flow schematic.

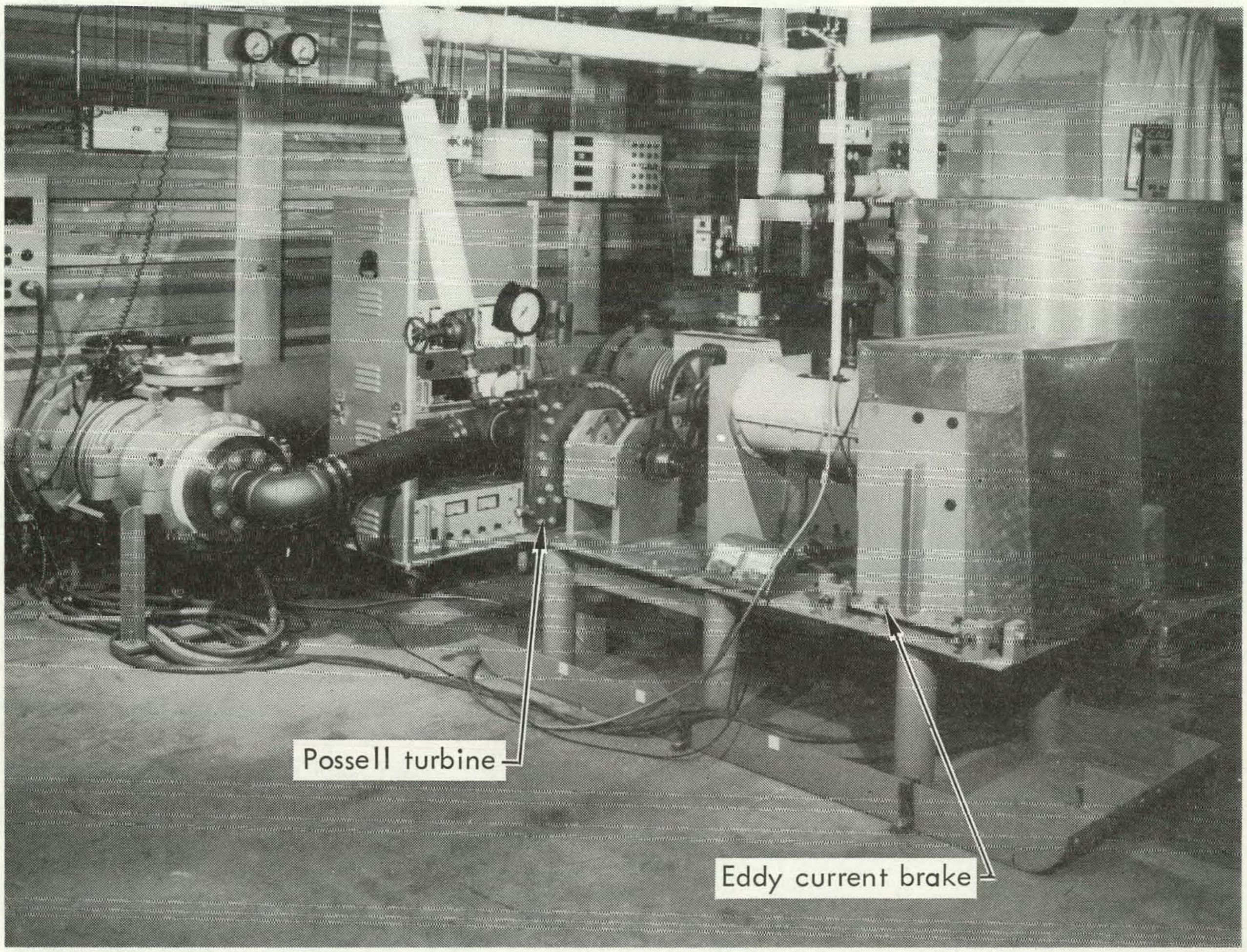

Fig. 4. The Possell bladeless turbine under test at Lawrence Livermore Laboratory.

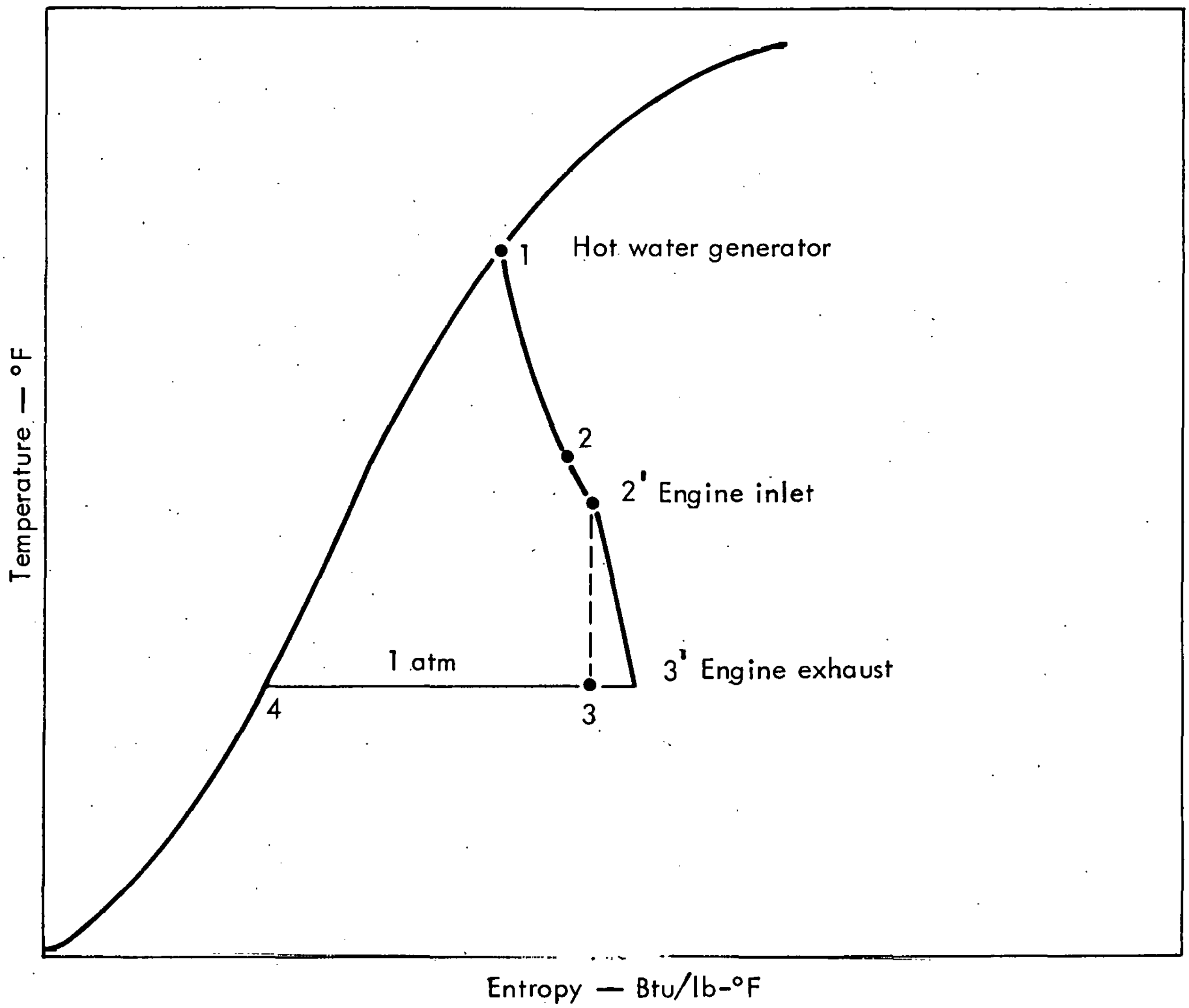

Fig. 5. Temperature-entropy diagram depicting the state points of the fluid at various locations.

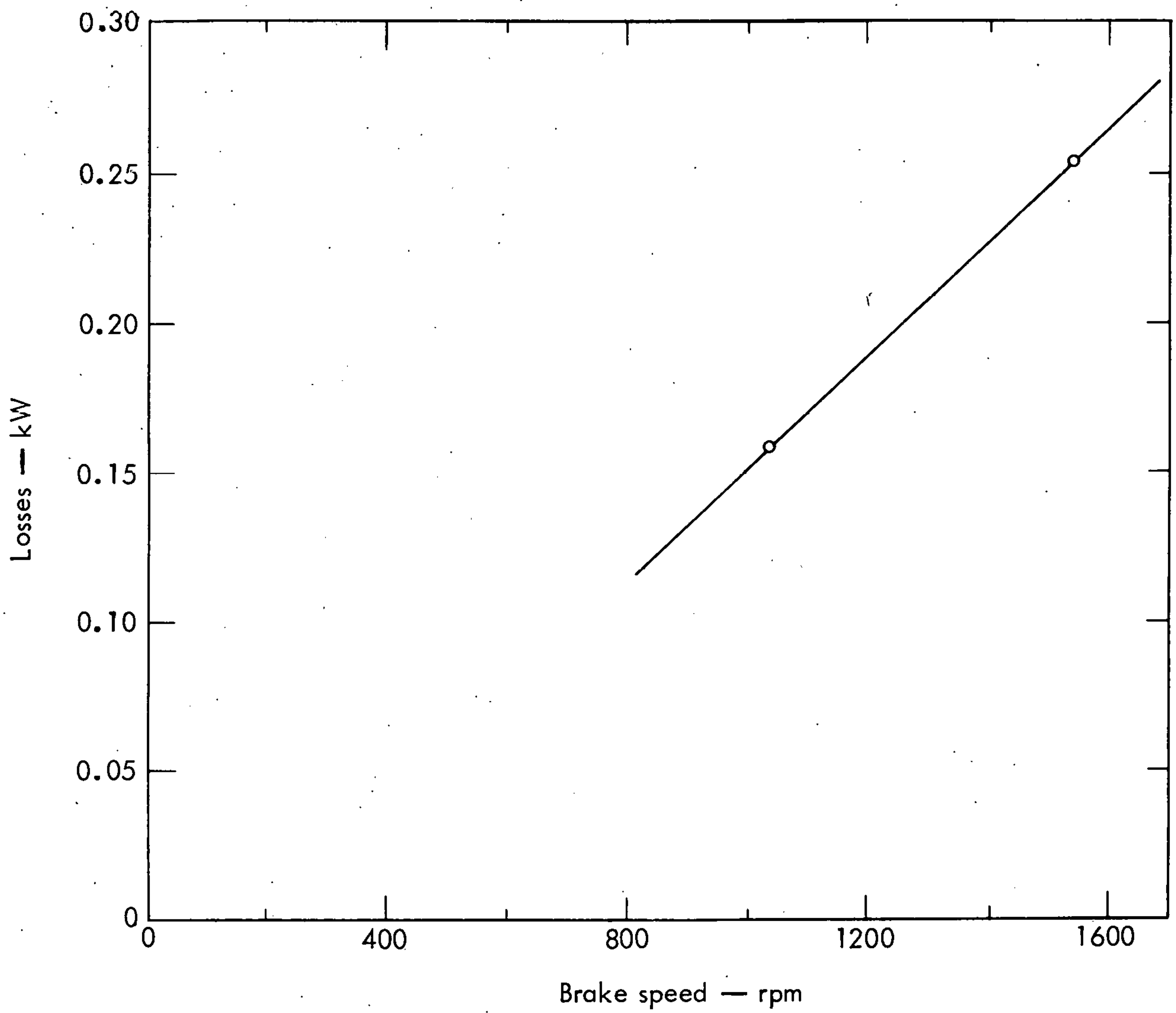

Fig. 6. Measured losses of the turbine -- includes timing belt and bearings (kW losses = 1.899×10^{-4} N – 0.0373).

TEST DATA AND CALCULATIONS

The reduced and calculated data are given in Tables 1, 2, 3, and 4 for 3000, 3500, 4000 and 4500 rpm, respectively. Each test point is identified by the test condition, run number, and date (e.g., 2-3/3-2 is for condition 2, run 3, on March 2, 1976). Included in the reduced data is the power output calculated from the measured values of speed and torque, flow rate, specific water rate, and the engine efficiency. The raw data are included in Tables 5 and 6.

In the calculations, the inlet enthalpy is defined by the state condition of the compressed liquid leaving the hot water generator. The inlet enthalpy to the turbine is assumed to be the same; that is, pressure drops through the gate value and orifice plate, and the pressure drops through the piping, are pure throttling processes.

Referring to Fig. 5, engine efficiency η_e is derived as follows

$$\eta_e = \frac{h_{2'} - h_{3'}}{h_{2'} - h_3}$$

$$= \frac{3412.5}{(\text{S.W.R.})\ (h_{2'} - h_3)}$$

$$\eta_e = \frac{(3412.5)\ (\text{kW})}{\dot{w}(h_{2'} - h_3)} \quad . \tag{1}$$

Here, $h_{2'}$ = enthalpy before turbine nozzle, BTU/lb

h_3 = enthalpy at turbine exhaust, assuming an isentropic expansion BTU/lb

$h_{3'}$ = enthalpy at turbine exhaust, BTU/lb

$\dot{w}$ = flow rate, lb/hr

kW = power, kilowatts

S.W.R. = specific water rate, lb/kWh

Table 1. Possell bladeless turbine performance at 3000 rpm nominal speed.

Run No./date	Speed (rpm)	Flow (lb/sec)	Power (kW)	$P_{2'}$ (psia)	P_3 (psia)	Water rate (lb/kWh)	Engine efficiency η_e	Specific speed N_s	Specific diam D_s
1-1/3-2	2994	0.419	0.648	214	14.22	2327	0.041	2.356	8.096
1-2/3-2	3024	0.356	0.703	189	14.18	1822	0.047	2.000	9.106
1-3/3-2	3057	0.426	1.121	224	14.14	1368	0.058	2.125	8.378
1-4/3-2	3006	0.493	1.362	274	14.25	1303	0.058	2.158	7.895
1-5/3-2	3006	0.573	1.576	304	14.26	1308	0.055	2.267	7.350
1-6/3-2	3012	0.475	1.246	250	14.17	1372	0.058	2.208	7.951
1-7/3-2	3012	0.586	1.578	300	14.18	1336	0.055	2.312	7.252
1-8/3-2	3012	0.669	1.840	340	14.28	1308	0.055	2.420	6.856
2-1/3-2	2940	0.288	0.543	159	14.14	1910	0.045	1.790	9.902
2-2/3-2	3000	0.413	1.100	220	14.15	1352	0.057	2.033	8.480
2-3/3-2	3012	0.554	1.555	300	14.20	1282	0.057	2.244	7.461
2-4/3-2	3012	0.666	1.887	344	14.24	1270	0.054	2.401	6.834
2-5/3-2	3000	0.871	2.093	430	14.29	1498	0.046	2.712	5.967
2-6/3-2	3012	0.935	1.960	444	14.33	1718	0.042	2.917	5.715

Table 2. Possell bladeless turbine performance at 3500 rpm nominal speed.

Run No./date	Speed (rpm)	Flow (lb/sec)	Power (kW)	P_2, (psia)	P_3 (psia)	Water rate (lb/kWh)	Engine efficiency η_e	Specific speed N_s	Specific diam D_s
5-1/3-5	3486	0.446	1.311	240	14.42	1224	0.059	2.350	8.232
5-2/3-5	3504	0.601	1.926	321	14.50	1123	0.060	2.606	7.221
5-3/3-5	3492	0.700	2.387	367	14.50	1055	0.062	2.747	6.732
5-4/3-5	3504	0.820	2.700	410	14.55	1094	0.059	2.936	6.259

Table 3. Possell bladeless turbine performance at 4000 rpm nominal speed.

Run No./date	Speed (rpm)	Flow (lb/sec)	Power (kW)	P_2, (psia)	P_3 (psia)	Water rate (lb/kWh)	Engine efficiency η_e	Specific speed N_s	Specific diam D_s
6-1/3-5	4014	0.794	2.876	410	14.58	993	0.065	3.301	6.364
6-2/3-5	4011	0.671	2.400	357	14.50	1007	0.066	3.110	6.841
6-3/3-5	4005	0.564	1.922	304	14.51	1056	0.064	2.899	7.432
6-4/3-5	3993	0.453	1.570	260	14.42	1039	0.068	2.680	8.211
6-5/3-5	3960	0.411	0.776	220	14.40	1908	0.040	2.632	8.536

Table 4. Possell bladeless turbine performance at 4500 rpm nominal speed.

Run No./date	Speed (rpm)	Flow (lb/sec)	Power (kW)	$P_{2'}$ (psia)	P_3 (psia)	Water rate (lb/kWh)	Engine efficiency η_e	Specific speed N_s	Specific diam D_s
4-1'/3-5	4506	0.811	2.736	394	14.60	1067	0.067	3.834	6.316
4-2/3-5	4488	0.587	1.875	304	14.51	1126	0.064	3.388	7.308
4-3/3-5	4500	0.464	1.383	250	14.39	1208	0.060	3.093	8.095

Table 5. Possell bladeless turbine test data, March 2, 1976.

Run	P_1 (psia)	T_1 (°F)	$P_{2'}$ (psia)	P_3 (psia)	h_1 (BTU/lb)	$\dot{w}$ (lb/sec)	Torque (in. lb)	Speed (rpm)	Power (kW)	Losses (kW)
1-1	826.4	471.0	214	14.22	454.2	0.419	42	998	0.496	0.152
1-2	826.4	471.0	189	14.18	454.2	0.356	46	1008	0.549	0.154
1-3	832.3	477.2	224	14.14	461.2	0.426	80	1019	0.965	0.156
1-4	828.8	480.9	274	14.25	465.6	0.493	102	1002	1.209	0.153
1-5	836.8	486.5	304	14.26	472.0	0.573	120	1002	1.423	0.153
1-6	810.1	476.4	250	14.17	460.4	0.475	92	1004	1.093	0.153
1-7	809.9	484.4	300	14.18	469.5	0.586	120	1004	1.425	0.153
1-8	811.4	484.7	340	14.28	470.0	0.669	142	1004	1.687	0.153
2-1	1021.3	478.8	159	14.14	463.1	0.289	34	980	0.394	0.149
2-2	1020.9	482.6	220	14.15	467.4	0.413	80	1000	0.947	0.153
2-3	1026.7	485.1	300	14.20	470.4	0.554	118	1004	1.402	0.153
2-4	1017.8	489.8	344	14.24	475.8	0.666	146	1004	1.734	0.153
2-5	1012.9	494.3	430	14.29	481.0	0.871	164	1000	1.940	0.153
2-6	1025.8	490.5	444	14.33	476.6	0.935	152	1004	1.806	0.153
3-1	1017.2	499.8	460	14.35	487.6	0.923	42	1509	0.750	0.249
3-2	1012.3	501.3	394	14.25	489.3	0.737	78	1035	1.388	0.248
3-3	1022.8	498.8	364	14.23	486.3	0.671	70	1035	1.238	0.247
3-4	1018.4	497.0	310	14.23	484.2	0.561	54	1035	0.955	0.247
3-5	1021.3	499.3	280	14.16	487.0	0.482	46	1035	0.823	0.250

Table 6. Possell bladeless turbine test data, March 5, 1976.

Run	P_1 (psia)	T_1 (°F)	$P_{2'}$ (psia)	P_3 (psia)	h_1 (BTU/lb)	$\dot{w}$ (lb/sec)	Torque (in. lb)	Speed (rpm)	Power (kW)	Losses (kW)
4-1	827.1	488.1	410	14.62	473.9	0.883	136	1506	2.4233	0.249
4-2	829.4	488.7	304	14.51	474.6	0.587	92	1496	1.6284	0.247
4-3	826.0	490.8	250	14.40	477.1	0.464	64	1500	1.1358	0.248
4-1'	819.5	491.2	394	14.60	477.5	0.811	140	1502	2.4880	0.248
5-1	826.5	293.4	240	14.42	480.0	0.446	82	1162	1.1274	0.183
5-2	828.1	496.4	321	14.49	483.6	0.601	126	1168	1.7413	0.184
5-3	827.2	498.0	367	14.49	485.6	0.700	160	1164	2.2035	0.184
5-4	825.8	499.3	410	14.55	487.0	0.820	182	1168	2.5151	0.184
6-1	828.4	500.2	410	14.58	488.1	0.794	168	1338	2.6597	0.217
6-2	828.4	500.2	357	14.49	488.1	0.671	138	1337	2.1830	0.217
6-3	822.3	497.7	304	14.51	485.1	0.564	108	1335	1.7059	0.216
6-4	825.5	493.2	260	14.42	479.8	0.453	86	1331	1.3543	0.215
6-5	819.2	488.5	220	14.40	474.4	0.411	36	1320	0.5622	0.213

We calculated the shaft power output by crediting the turbine with the losses incurred in the turbine bearings and the speed reducer. To determine this, the rotor and side plate were removed, and the assembly was driven by an electric motor at 1035 rpm (3105 turbine rpm) and 1540 rpm (4620 turbine rpm). Torque and speed measurements were taken and the power losses determined. Figure 6 shows these two points, with the points connected by an empirical line.

$$\text{kW losses} = 1.899 \times 10^{-4} N - 0.0373$$

For the shaft power output,

$$\text{kW} = \frac{TN}{28173} + \text{kW losses.} \qquad (2)$$

Here, T is torque, in.-lb

N is turbine speed, rpm

The specific water rate is defined as the flow needed to generate 1 kW of power. The conventional units for specific water rate are lb/kWh.

$$\text{S.W.R.} = \frac{\dot{w}}{(\text{kW})} \qquad (3)$$

Specific speed N_s and specific diameter D_s are parameters that indicate the characteristic performance of fluid machines.[3,4] Only two such parameters are needed to establish the performance state. Turbines and rotors with equal N_s and D_s values will have equal performance, i.e., efficiency, if there is geometric similarity. This means that optimum geometry can also be stated as a function of N_s and D_s. This information can be presented in a two-dimensional plot with lines of constant efficiency.

Considering isentropic expansion to exhaust conditions, for two-phase flow, with weight flow $\dot{w}$, the volume flow Q in cubic feet per second at the exhaust pressure would be $\dot{w}\bar{v}$. $\bar{v}$ is the specific volume of the working fluid at the exhaust pressure. The isentropic head $H = 778(h_{2'} - h_3)$.

$$N_s = \frac{NQ^{\frac{1}{2}}}{H^{\frac{3}{4}}} = \frac{N(\dot{w}\bar{v})^{\frac{1}{2}}}{[778(h_{2'} - h_3)]^{\frac{3}{4}}} \tag{4}$$

$$D_s = \frac{DH^{\frac{1}{4}}}{Q^{\frac{1}{2}}} = \frac{D[778(h_{2'} - h_3)^{\frac{1}{4}}}{(\dot{w}\bar{v})^{\frac{1}{2}}} . \tag{5}$$

In both definitions, the speed N is measured in rpm and the gravitational constant is excluded. It is more convenient to use these forms of specific diameter and specific speed, although it is less pure.

The Possell turbine is not a bladeless turbine, in the recognized definition of the Tesla turbine, since the presence of welded spacers interrupts the fluid flow. Using Balje's descriptions, it would be a combination of the bladeless and spotface drag turbine. Both are high energy input, low power output, low performance machines $1 < N_s < 5$ and $5 < D_s < 10$.

RESULTS

The test points for test condition 3 were all discarded. These test points were taken at the end of the first test day on March 2, nominally at a turbine speed of 4500 rpm. It was difficult to maintain equilibrium, and as a result the observed data are erratic, with no observable relevance to the other data. The test condition was repeated two days later on March 5. Test condition 4 is a repeat of test condition 3, and the repeated data were consistent, and these data are plotted. Test point 4-1 was also discarded. This was the first test run on March 5, and the pipes were cold. Test 4-1' is the repeat of test 4-1.

Figure 7 shows the flow rate as a function of shaft power output. The curves show the generally accepted characteristic of any Willans line. The lines for 4000 rpm and 4500 rpm seem to be juxtaposed, but either would be within our limits of measurement.

Figure 8 shows the specific water rate as a function of shaft power output. The specific water rate is discouraging, even for non-condensing operation. The minimum specific water rate was 993 lb/kWh, a factor of 5 greater than that observed for the Lysholm expander run under less favorable

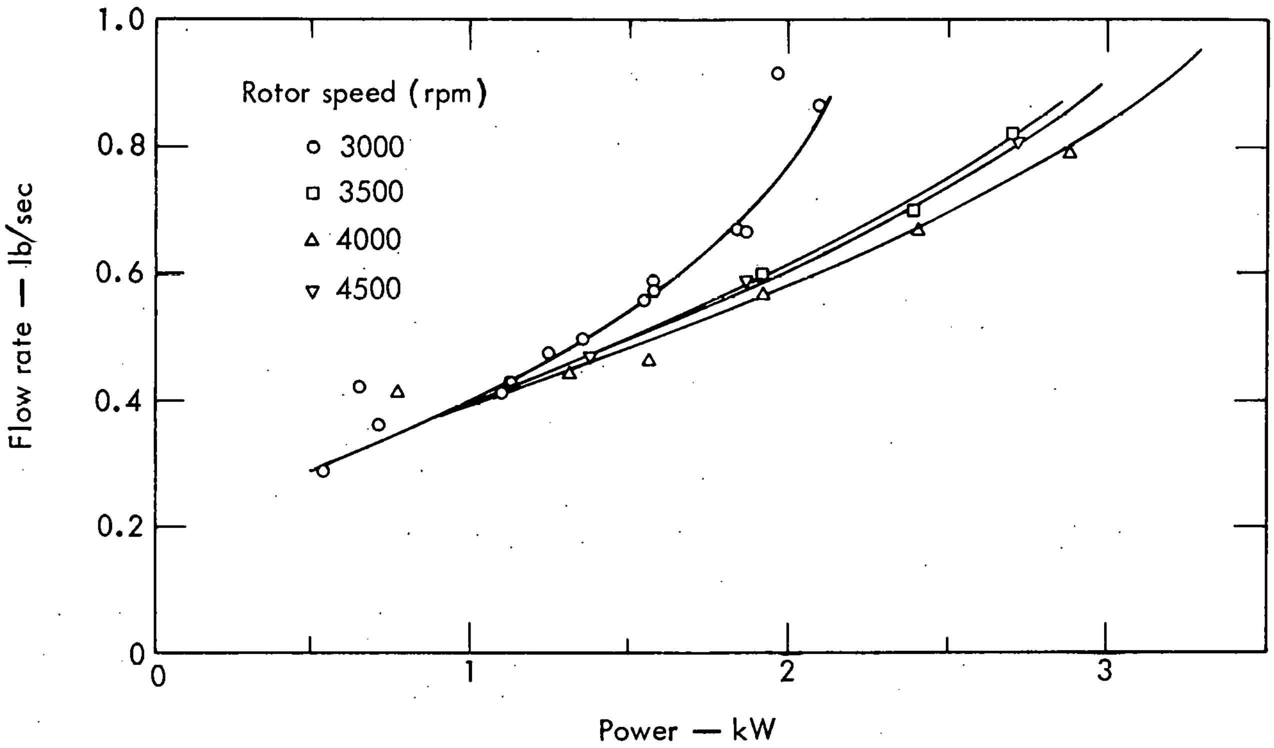

Fig. 7. Flow rate as a function of power. The curves show the generally accepted characteristic of any Willans line.

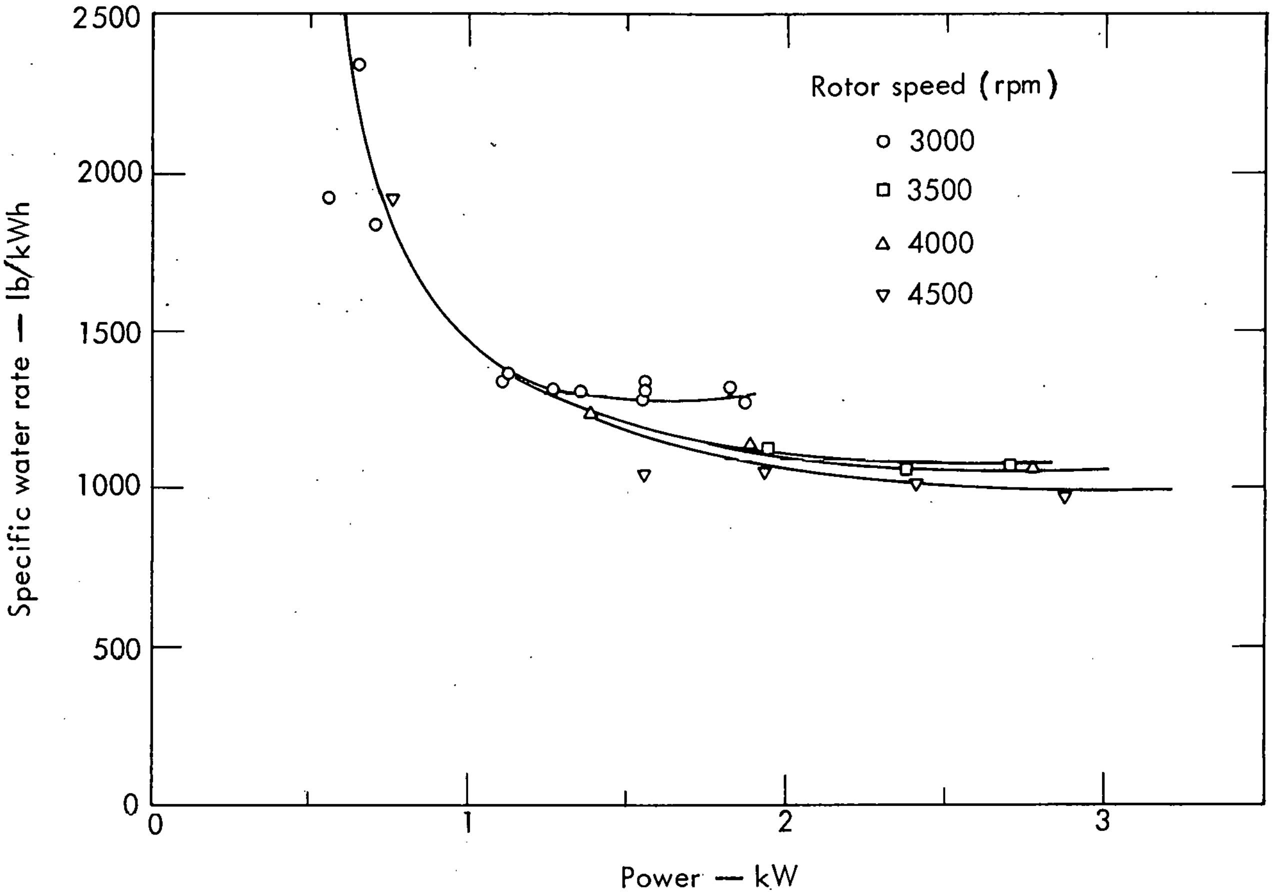

Fig. 8. Specific water rate as a function of shaft power output.

conditions,[5] and more than an order of magnitude above what generally would be acceptable. The results appear to be consistent.

Engine efficiencies are shown in Fig. 9. The maximum observed engine efficiency was 6.7% at 4500 rpm and 2.74 kW. A slightly higher engine efficiency was calculated at 4000 rpm and 1.57 kW, but it seems to be marginally consistent with other data. The engine efficiency seems to be relatively flat between 1.5 kW and 3 kW for 3500, 4000, and 4500 rpm.

In Fig. 10, the performance regime is plotted. The performance data fall where the data were predicted to be, overlapping the only other performance data we have, which are from Warren Rice[6] (see Fig. 11). It is only part of a performance contour, but only part was expected. The shape of the contours appear to be slightly different than the Rice data, and of course, the efficiencies are lower for the Possell turbine. There is a good argument that the performance regime is narrow, $2 < N_s < 4$ and $5 < D_s < 8$, and that the Possell turbine will not work well out of that regime. There is every reason to believe the performance data to be valid and no indication that they are faulty or invalid.

CONCLUSIONS

The following conclusions are based on the tests described in this report, and are not necessarily representative of all possible designs of the bladeless turbine concept.

- The performance characteristics are valid.
- The test conditions covered the operating range where optimum performance could be expected.
- The maximum engine efficiency observed was less than 7%.
- There is no indication that running at speeds higher than 4500 rpm will significantly improve performance.
- The Possell turbine, as tested, is not a viable candidate machine for the conversion of energy from geothermal fluids by the total flow process.

The machine that was tested consisted of seven blades, each blade being 0.062 in. thick, with a spacing of 0.084 in. between blades. In an analysis

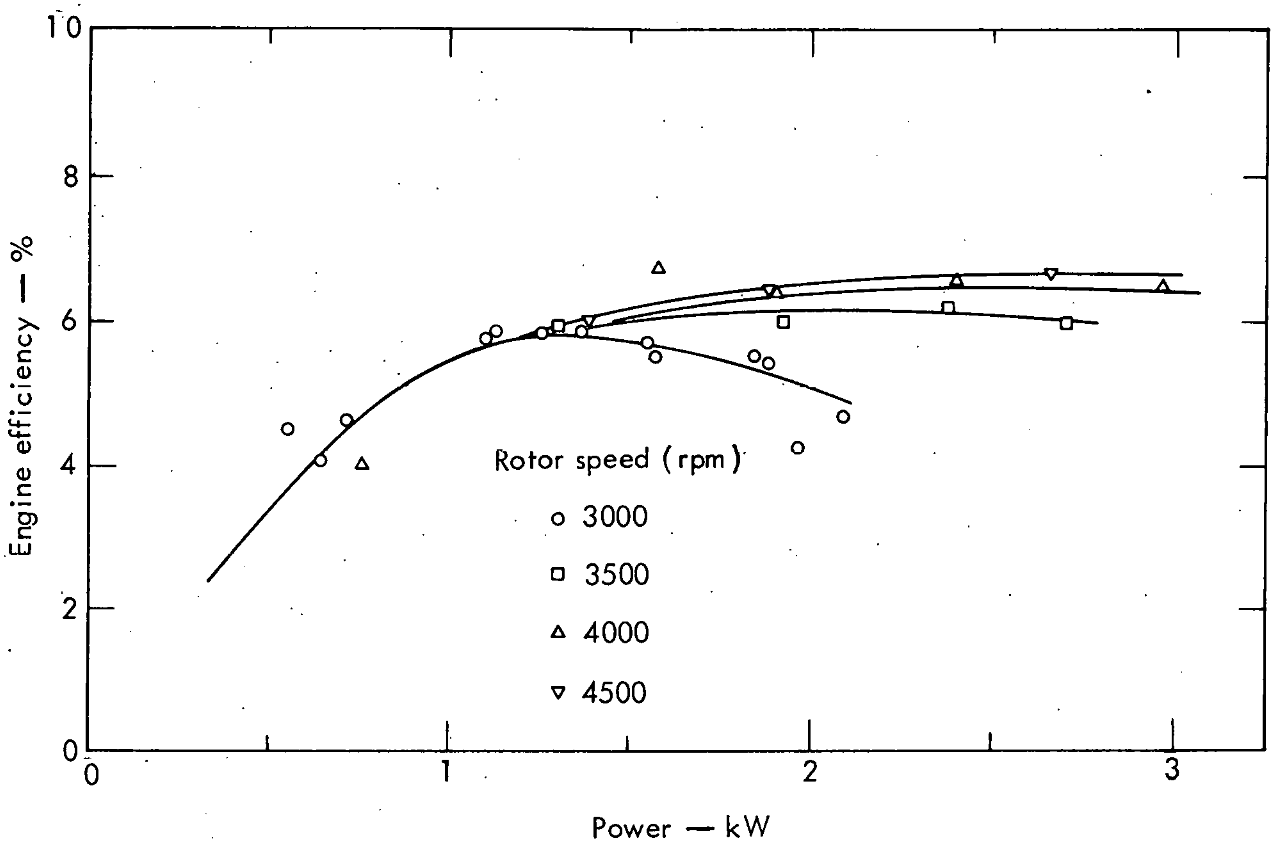

Fig. 9. Engine efficiency vs power at various rotor speeds.

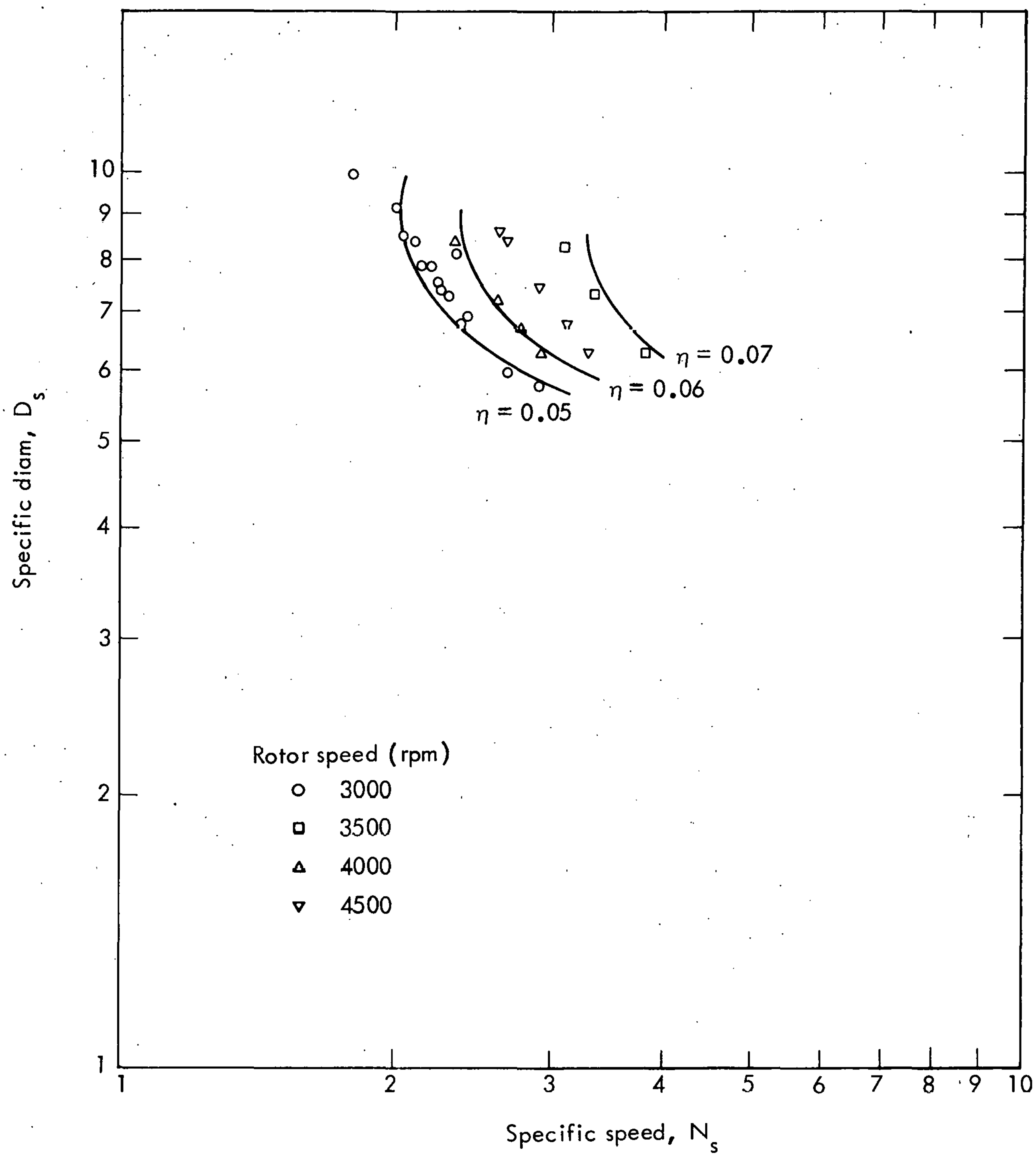

Fig. 10. Specific diameter vs specific speed as a function of engine efficiencies.

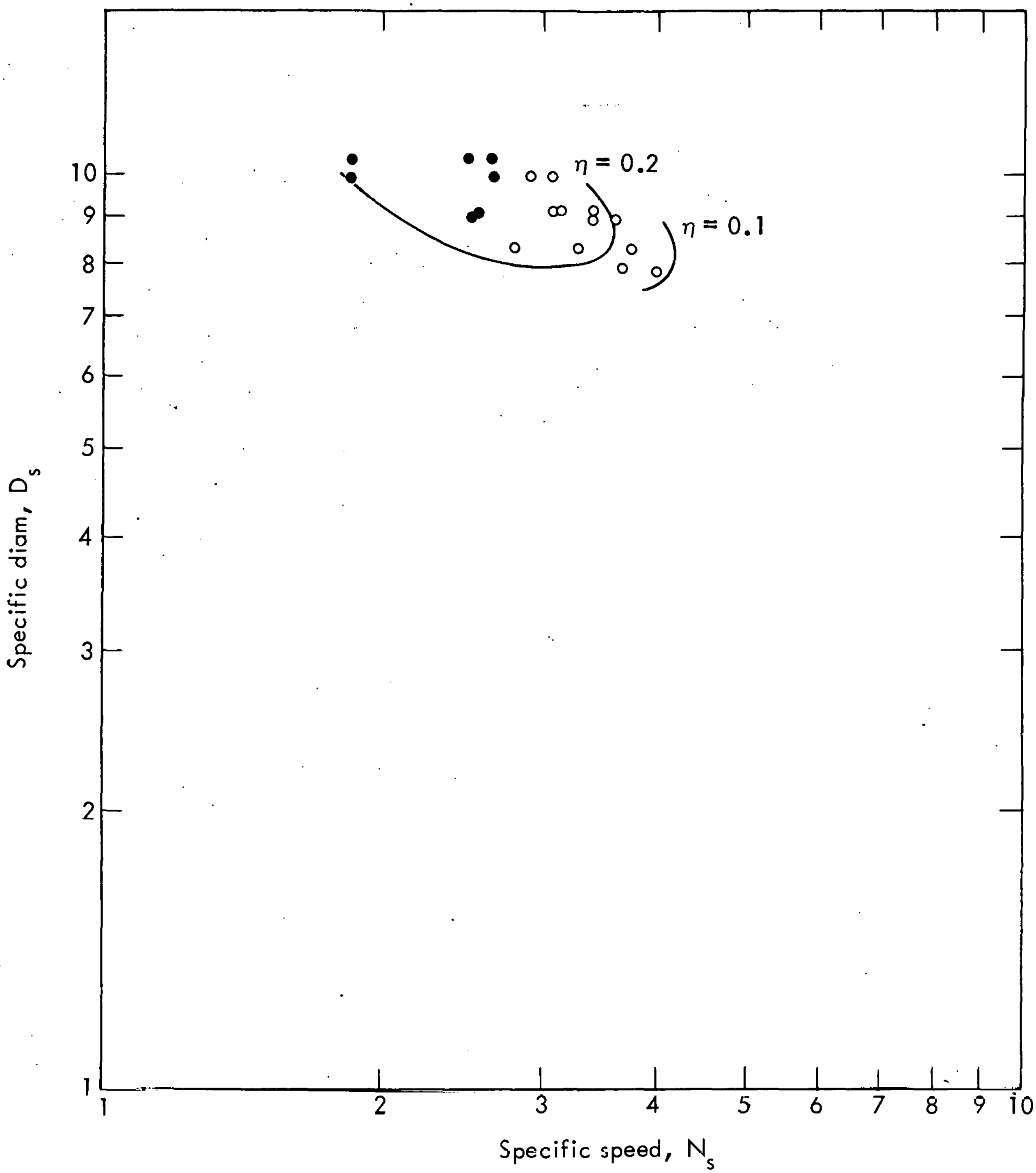

Fig. 11. Specific diameter vs specific speed as a function of engine efficiencies. Data are from W. Rice.

of a Tesla bladeless pump,[7] the blade spacing was found to be critical, with the optimum blade spacing being in the order of twice the thickness of the boundary layers. The Possell blade spacing was many times greater, leading to the conclusion that most of the fluid was passing through the rotor without transmitting its momentum to the blades.

Successful bladeless pumps have total admission and optimum blade spacing. We have every reason to believe that this statement would also be true for the bladeless turbine.

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