A SMALL ABSORPTION POWER GENERATOR USING LOW GRADE GEOTHERMAL HEAT

by

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Introduction

The absorption power generator is similar to a steam power plant where water vapor is generated under pressure in a boiler and then expanded in a prime mover such as a steam turbine or a reciprocating expansion engine. All of the exhaust water vapor from the steam power plant prime mover is collapsed to liquid by condensation in a condenser and then pumped back to the boiler. The absorption power generator works similarly except that the temperature of the heating medium for the boiler can be much lower. This is because of a low boiling point liquid, such as ammonia, is used to make pressurized vapor instead of water. Instead of condensing the low pressure ammonia vapor from the prime mover in a condenser, it is absorbed by water containing spray in an absorber vessel. Unused heat is rejected from the process at ambient air temperatures in a high pressure condenser vessel connected to the boiler. In the absorption power generator, the boiler is replaced by a special vertical tube evaporator.

This invention was originally described in U.S. Pat. 4,622,820 granted in 1986. It has not yet been tested. It is possible that this invention could produce electric power from heating mediums at temperatures 200°F or less at acceptable cost. It could possibly work well with low grade geothermal heat.

General Operation of the Invention

Figure 1 is an isometric sketch and Figure 2 is a process flow diagram of a small absorption power generating plant using geothermal heat for generating electric power. Table A presents estimated steady state flow conditions and comparative flow rates of the fluids shown in figure 2 based on evaporator vapor 1 flow of 1 lb/hour.

There is a downhole heat exchanger in a geothermal well and this heat exchanger receives heat from geofluids circulating through the well casing. Heating liquid 7 transports heat from the heat exchanger to the tops of vertical tubes 31 inside evaporator vessel 30. The heating liquid then flows downward very slowly inside the vertical tubes, heating the weak solution, both fluids being at the same temperature. Spent heating fluid 12 is a richer aqua-ammonia solution than weak solution 4. The extra ammonia diffuses very rapidly upward to liquid surface 32, above the tops of the vertical tubes. Evaporator vapor 1 separates from this boiling surface and mixes with a spray of reflux condensate 13. This mixture then exits from the vapor space of the evaporator as rectified vapor 2. Heat for vaporizing evaporator vapor 1 and reflux condensate 13 comes from vertical tubes 31 and flows upward through the highly agitated weak solution in the shell space of the evaporator.

Part of rectified vapor 2 flows to expansion engine 40 where it produces shaft power q_{27}, and then exits from the expansion engine as exhaust vapor 3. Exhaust vapor 3 flows into the top of absorber vessel 50, where it is absorbed by the water in a spray of weak solution 4. Weak solution 4 flows from the bottom of evaporator vessel 30 into absorber vessel 50, driven by the pressure difference between the two vessels.

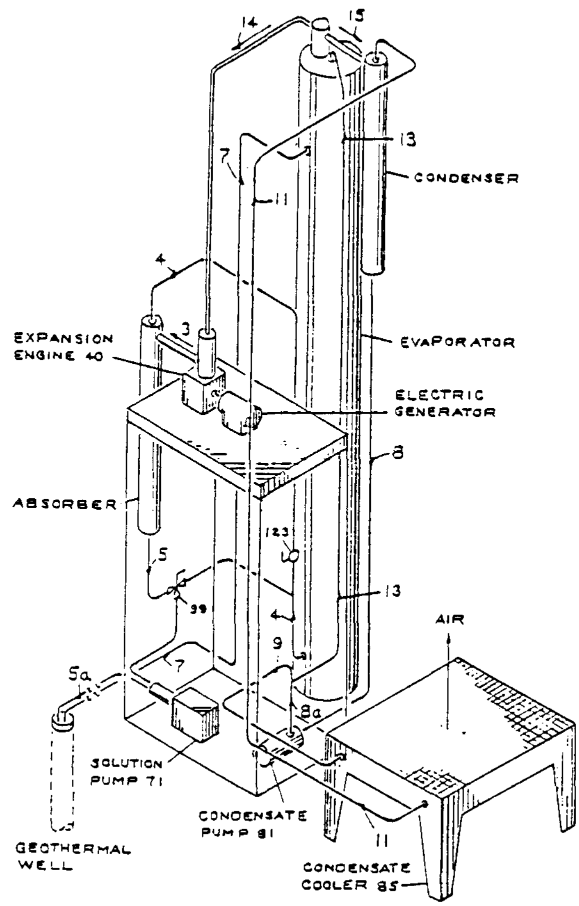


Figure 1. Absorption power generating plant.

Strong solution 5 which collects in the bottom of absorber vessel 50 and then flows to solution pump 71. From solution pump 71, strong solution 5a finally returns to the downhole heat exchanger in the geothermal well.

The other part of rectified vapor 2 flows to condenser vessel 60, where it is condensed on a spray of cool condensate 11. Condensate 8 collects in the bottom of condenser vessel 60 and then flows to condensate pump 81. Condensate 8a from condensate pump 81 is then divided into two parts, pumped condensate 9 and reflux condensate 13. Pumped condensate 9 goes to condensate cooler 85 and then, as cool condensate 11, to the spray nozzle in the top of condenser vessel 60. Reflux condensate 13 flows to the spray nozzle in the top of evaporator vessel 30.

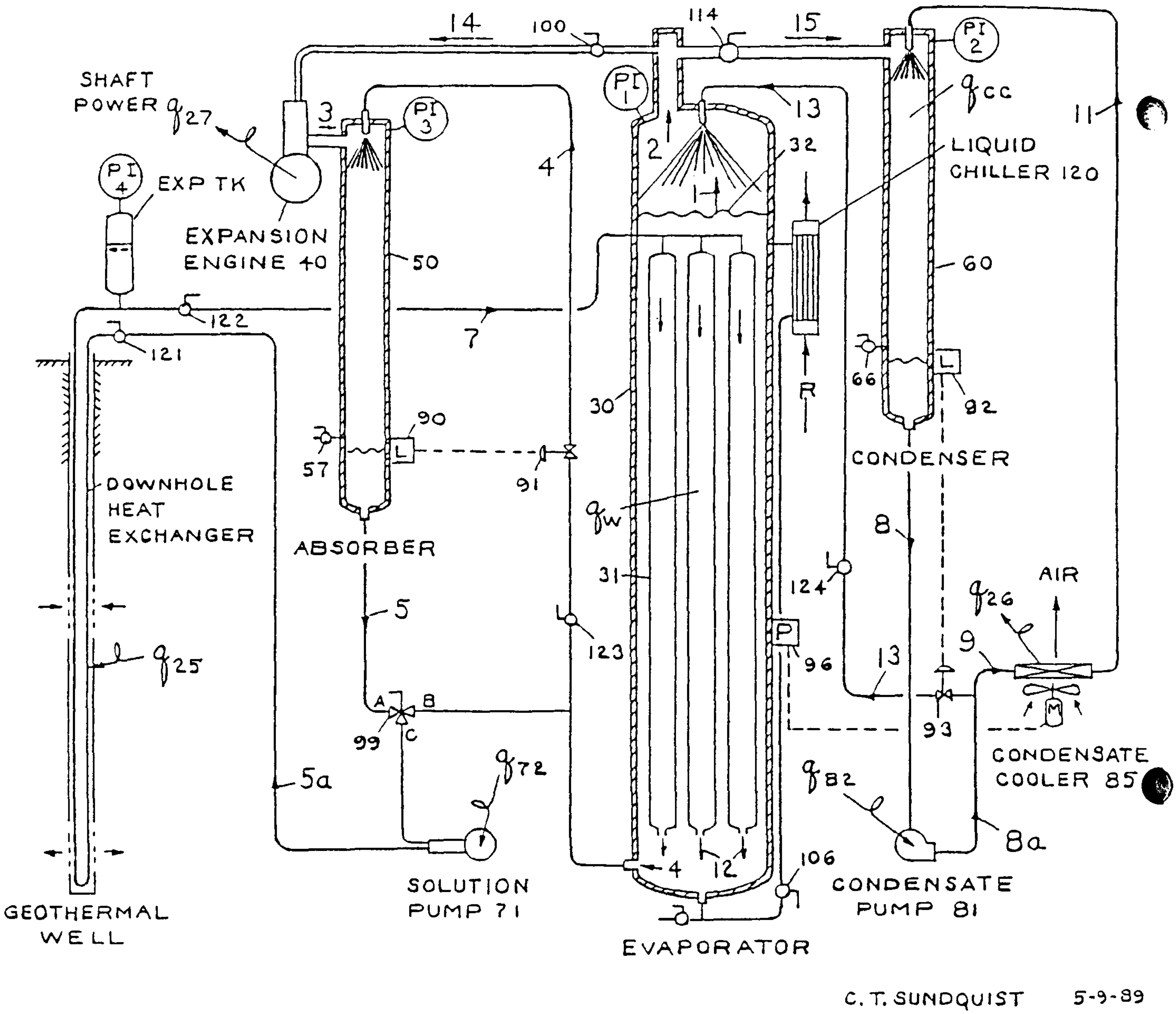


Figure 2. Process flow diagram of a small power generating plant.

The Vertical Tube Evaporator Condenser Combination

The combination of a vertical tube evaporator and a condenser like that shown in Figure 2 is unusual. There is no technical literature that explains how these two items of process equipment work together. Tests conducted on a similar evaporator condenser combination revealed that a steady state temperature distribution, like that shown in Figure 3, established itself naturally in the evaporator. The temperature difference at the top of the evaporator between the heating liquid, inside the vertical tubes, and the shell space solution was from 30°F to 50°F during these tests. The temperatures of the heating fluid and shell space solution at the bottom of the vertical tubes always converged, at steady state conditions. This converged temperature stayed constant as long as there were no changes in the heat entering and leaving rates to the apparatus as a whole. This converged temperature, however, was very sensitive to changes in the heat flow rate to or from the apparatus. For instance, if the temperature of the cool distillate going to the spray nozzle in the condenser increased slightly, the converged temperature quickly increased and then locked in again. Some sort of rapid thermal communication between the condenser and the bottom of the evaporator appeared to exist.

Subsequent analytical study has led to the theory that this convergent temperature behavior must be a manifestation of the Second Law of Thermodynamics. It is believed that this effect should work the same way for the absorption power generator. By making calculations like those made to prepare Table A and from those making further calculations to obtain data points of total entropy change, the diagram in Figure 4 was developed. In this diagram, the ordinate represents total entropy change while the abscissa represents the temperature of weak solution 4. The t_7 curves sloping downward and to the right represent evaporator heat-up curves where the temperature of heating fluid 7 is held constant and t_4 is allowed to rise. The t_7 curves indicate that the temperature of weak solution 4, in the bottom of the evaporator, can rise a limited amount before the apparatus total entropy change becomes zero. At this point, the temperature of weak solution 4 can rise no further. The Second Law of Thermodynamics prohibits the total energy change of any thermodynamic process from being less than zero.

TABLE A

One Pound Fluid Conditions (t_7 = 150°F)

Fluid	t	x	h(Btu/lb)	p(psia)	m(PPH)
1		1.00			1.00
2	+120F	1.00	+559	290	3.00
3	+11F	1.00	+479	39	1.00
4	-10F	.50	-153		8.03
5	+55F	.55	-83		9.03
5a	+56.1F	.55	-81.8		9.03
7	+150F	.55	+27		9.03
8	+120F	1.00	+102		30.60
8a	+120.3F	1.00	+120.3		30.60
9	+120.3F	1.00	+102.3		28.60
11	+90F	1.00	+70		28.60
12	-10F	.55	-150		9.03
13	+120.3F	1.00	+102.3		2.00
14	+120F	1.00	+559	290	1.00
15	+120F	1.00	+559	290	2.00

x = ammonia fraction

It should be noted that in the flow diagram of Figure 2, liquid chiller 120 is connected to evaporator vessel 30 so that it can be used to chill the lower part of the evaporator vessel by thermosiphon circulation. An ammonia refrigeration condensing unit can be connected to this chiller to provide the necessary chilling effect. Liquid chiller 120 would be used during the start-up and possibly to force temperature t_4 down if for some reason it should rise.

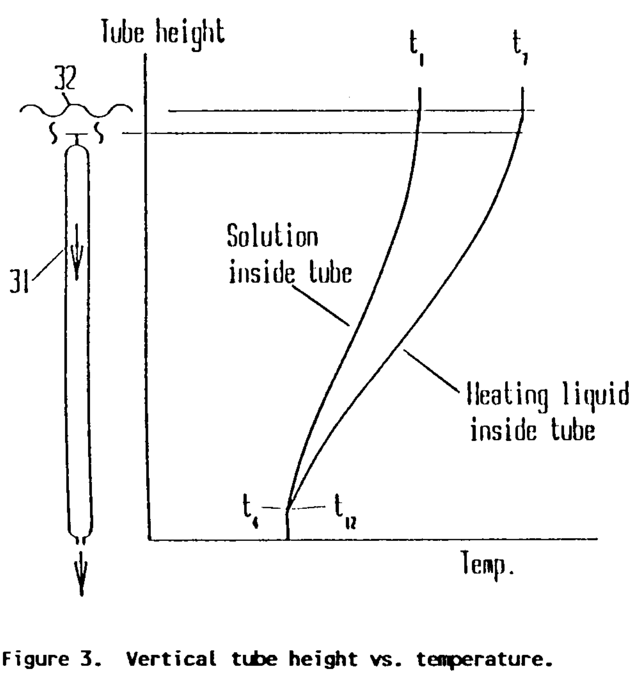
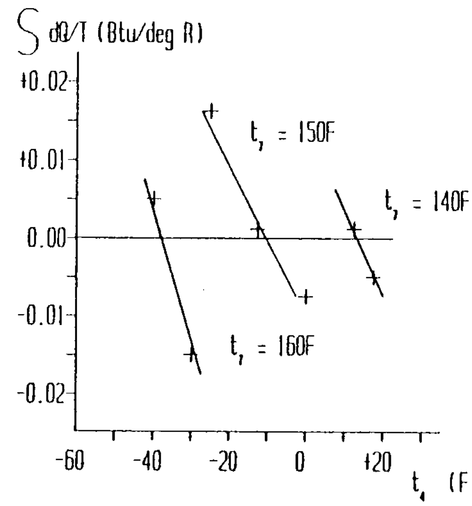


Figure 3. Vertical tube height vs. temperature.

Estimated Performance of the Invention

Tables A, B, C, D and E present the results of calculations of the estimated performance of a small example of the invention. The evaporator has eighty-five 2 in. O.D. vertical tubes, 20 ft high. Each tube is estimated to transfer heat at the rate of 13,000 BTUH. For eighty-five tubes, the total evaporator heat transfer rate Q_w is estimated to be 1,105,000 BTUH. Dividing Q_w by q_w, from Table B, yields a scaling factor F_s = 691. F_s is used to calculate the data in Table D. No allowance has been provided in the calculations for heat transmission through the walls of vessels, piping, pumps, or the expansion engine. In Tables A and B, the mechanical efficiency of expansion engine 40 is estimated to be 80%, that of the solution pump 71 to be 80%, and that of condensate pump 81 to be 70%.



$$\int \frac{dQ}{T} \simeq \frac{q_{25}(\ln T_7 - \ln T_{5a})}{T_7 - T_{5a}} - \frac{q_{26}(\ln T_8 - \ln T_{11})}{T_8 - T_{11}} - \frac{q_{27}(\ln T_3 - \ln T_{14})}{T_3 - T_{14}} + \frac{2q_{72}}{T_5 - T_{5a}} + \frac{2q_{82}}{T_1 - T_{14}}$$

Figure 4. Total entropy change vs. weak solution temperature t_4.

The estimated performance of the above small example of the invention is based on a summer outdoor air temperature of approximately 80°F and an evaporator weak solution ammonia fraction of 0.50. If the outdoor air temperature is decreased to 60°F and the evaporator weak solution ammonia fraction is decreased to 0.40, the estimated net overall thermal efficiency will increase from 6% to 8%. This will result in a substantial increase in the power output from the apparatus. It should, therefore, be kept in mind that the figures presented in Tables A, B, C, D and E are not necessarily the best that can be expected from this invention.

TABLE B

One Pound Energy Flow Rates (t_7 = 150°F)

Input heat, q_{25}	= 982 BTUH
Rejected heat, q_{26}	= 922 BTUH
Shaft power, q_{27}	= 80 BTUH
Solution pump heat, q_{72}	= 11 BTUH
Condensate pump heat, q_{82}	= 9 BTUH
Vertical tube heat transfer rate, $q_w = m_7(h_7 - h_{12})$	= 1598 BTUH

TABLE C

Efficiencies

Carnot Cycle efficiency = $(T_7 - T_{11})/T_7$	= 0.098
Thermal efficiency = q_{27}/q_{25}	= 0.082
Net Overall efficiency = $(q_{27} - q_{72} - q_{82})/q_{25}$	= 0.061

TABLE D

Total Input-Output Energy Flow Rates

$Q_{25} = F_s \times q_{25}$	= 679,000 BTUH
$Q_{26} = F_s \times q_{26}$	= 637,000 BTUH
$P_{27} = F_s \times q_{27}/2547$	= 21.6 HP
$P_{72} = F_s \times q_{72}/2547$	= 3.0 HP
$P_{82} = F_s \times q_{82}/2547$	= 2.4 HP

TABLE E

Estimated Net Available Electric Power

Power output from electric generator = P_{gen}

$P_{gen} = P_{27} \times$ generator efficiency = 21.6 x .85 = 18.4 HP

Power to motors = P_{ml}

Solution pump motor =

P_{72}/motor efficiency = 3.0/.82 = 3.7 HP

Condensate pump motor =

P_{82}/motor efficiency = 2.4/.805 = 3.0 HP

Condensate cooler fan motor =

3.00/motor efficiency = 3.00/.82 = 3.7 HP

P_{ml} = 10.4 HP

Net available electric power,

$P_{net} = P_{gen} - P_{ml}$ = 8.0 HP (6.0 KW)

Outline Specifications for Major Components

Evaporator:
- ASME pressure vessel (300 psi WP)
- 24 in. Schedule 20 seamless steel pipe, 25 ft high
- 85 vertical tubes, 24 in. O.D. x 0.035 in. wall, 20 ft tall, carbon steel
- Total weight: 4,000 lbs.

Absorber vessel 50:
- 8 in. Schedule 40 seamless steel pipe, 8 ft high
- Total weight: 250 lbs.

Condenser vessel 60:
- Same as absorber vessel 50

Expansion engine 40:
- Type: reciprocating
- Throughput: 691 PPH, saturated ammonia vapor at 120°F
- Inlet pressure: 290 psia
- Outlet pressure: 39 psia
- Efficiency: 80%

Electric generator:
- Input power: 22 HP
- Efficiency: 85%
- 110V, 60Hz, single phase

Solution pump 71:
- Type: reciprocating power pump
- Pump efficiency: 80%
- Fluid: 15.4 GPM, 55% rich aqua-ammonia at 55°F
- Inlet pressure: 40 psia
- Outlet pressure: 311 psia
- NPSH: 5 feet

Condensate pump 81:
- Type: seal-less centrifugal
- Pump efficiency: 70%
- Fluid: 75 GPM liquid ammonia at 120°F
- Inlet pressure: 293 psia
- Outlet pressure: 326 psia
- NPSH: 12 feet

Condensate cooler 85:
- Type: fan air cooled coil
- Inlet air temperature: 75°F
- Liquid cooled: 75 GPM ammonia from 120°F to 90°F
- Fan motor: 3 HP maximum, variable speed
- WP: 326 psia
- Liquid pressure drop: 2 psi

Start-up Procedure

In the shutdown condition, the apparatus is valved off into four separate pressure spaces--the evaporator, the condenser, the absorber, and the downhole heat exchanger. This is done to prevent ammonia vapor from moving in an uncontrolled manner from one pressure space to another. It should be noted that if the downhole heat exchanger temperature, in the example presented, rises to 200°F, the pressure in the exchanger can rise to 370 psia.

Referring to Figure 2, the step-by-step start-up procedure is as follows:

Step 1

Cooling valve 106 is opened and cold refrigerant is fed into liquid chiller 120. Weak solution, from the top part of evaporator vessel 30, then flows by thermosiphon circulation to the bottom of evaporator vessel 30, cooling the contents of the vessel slowly from the bottom upward. When the bottom of the evaporator has been chilled to below operating temperature, valve 106 is closed and the flow of cold refrigerant to liquid chiller 120 is stopped.

Step 2

Downhole heat exchanger valve 122 is slowly opened to equalize the pressure in the downhole heat exchanger with that in the evaorator. Then downhole heat exchanger valve 121 is opened and solution pump 71 is started. Weak solution 4, from the bottom of evaporator vessel 30, then starts flowing into the downhole heat exchanger. Also, heating fluid 7 commences to slowly heat the contents of evaporator vessel 30 from the top downward.

Step 3

Condenser pump 81 is started and cool condensate 11 starts to flow to the spray nozzle in the top of condenser vessel 60. At this point, a blanket can be thrown over condensate cooler 85 to minimize the cooling effect.

Step 4

When the pressure in the vapor space of evaporator vessel 30 has increased to the same level as in condenser vessel 60, condenser valve 114 is opened, allowing condenser vapor 15 to flow from the evaporator to the condenser.

Step 5

When the pressure in the combined evaporator and condenser vapor spaces has risen to operating level, the blanket is removed and the fan motor of condensate cooler 85 is activated.

Step 6

The flow of weak solution 4, from the bottom of evaporator vessel 30, is then switched to the spray nozzle in the top of absorber vessel 50. This is done using three-way start-up valve 99 and two-way valve 123. This will cause the pressure in absorber vessel 50 to decrease to a lower level.

Step 7

Power valve 100 is gradually opened and expansion engine 40 begins to turn over. Power valve 100 is gradually opened until the power output of expansion engine 40 has reached operating level.

Controls

In order to keep total shaft power Q_{27} at a maximum rate, it is necessary that the temperature of weak solution 4 be prevented from increasing. Considering the previous discussion regarding Figure 4, it is apparent that this can be achieved by holding the total entropy change for the apparatus to zero. This can be done if all other temperatures are held constant and total energy flow rates Q_{25}, Q_{26}, Q_{27}, Q_{72} and Q_{82} are held constant.

To accomplish these objectives, the following things should be done:

1. The flow rate through solution pump 71 should be held constant. A change in this flow rate can result in an increase in the total entropy change and, as a result, an increase in temperature t_4.

2. Means should be provided to hold total heat rejection rate Q_{26} constant while the temperature of the air entering condensate cooler 85 varies. A way to do this is indicated in Figure 2, where pressure controller 96 operates the variable-speed fan motor of condensate cooler 85 to hold the pressure constant in evaporator vessel 30.

Besides the automatic pressure controller system for evaporator vessel 30, the only other automatic controls necessary are the liquid level controllers for absorber vessel 50 and condenser vessel 60.

References

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6. Mark's Mechanical Engineers' Handbook, 6th Edition, Section 4 - Heat.

7. ASHRAE, 1972 Handbook of Fundamentals, Chapter 2, Heat Transfer.

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