REPORT NO.: C002702-1

MASTER

Quarterly Progress Report
to the
U.S. Energy Research and Development Administration

SOLAR POWERED DESICCANT AIR CONDITIONING
SYSTEM - PHASE II
QUARTERLY PROGRESS REPORT

by
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August 1975

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COMPUTER MODEL DEVELOPMENT

The heat transfer subroutine has been written and integrated into the computer model. Debugging is incomplete so there are no results as yet.

COMPONENT (AND SYSTEM) DESIGN

A major analytical effort has been made to quantify the required heat exchange areas and silica gel bed pressure drop, since these two items are very important to the design and did not receive attention in our past contract.

The basic cooling system design is a nominal two-ton system thought representative of the cooling load for a well designed solar house. We are aware that the usual solar air conditioner size (ARKLA) is a nominal three tons, but such a unit needs a solar collector temperature of 180°F-200°F and thus cannot operate as much of the day as the silica gel unit can.

The design system, based on Figure 12 of the last phase final report, has been updated and is presented as Figure 1. Bed size has been raised 20% to compensate for an anticipated decrease in dynamic performance as a result of a shift to Davison 03 silica gel, which is of 3-8 mesh as opposed to 8-10 mesh size for the material used in the experiments from which the mathematical model was derived. The lower characteristic pressure drop of the larger 03 material is necessary to make the fan size reasonable. Even so the bed must be made large in diameter (6 ft) so that the resultant low path length (2.87 in. total) induces only a small pressure drop (0.9 in. H₂O) making the fan power only about 200 watts for each of the two fans (at 50% efficiency).

Heat exchanger design has been difficult because the desired temperature approach is close (2.5°F) and with the large diameter beds the air velocity is low. Tube-fin concepts were discarded in favor of finned tubes because these break up the developing laminar film as the air leaves one tube and passes to the next. In this way heat transfer coefficients on the order of $h = 6 \text{ btu/hr ft}^2\text{°F}$ were projected, based on the fin surface area, and a bank of 1-in. $\times$ 3/4-in. finned tubes three rows deep is planned at each heat exchanger location. The pressure drop past these finned tubes is estimated at less than 0.4 in. H₂O, making total bed and heat exchanger fan power about 300 watts each for the adsorption and desorption air circuit. To this is to be added the power for the main heat rejection fan, which will also

run about 300 watts, and that of the pumps and the motors rotating beds and the heat regeneration heat exchangers. The electrical total power consumption sums to about 1000 watts. This is a savings of about 2/3 of the electrical power used in a contemporary high-efficiency (8 btu/watt-hr) mechanical air conditioner.

The design is now being carefully recalculated using the best information available. The completed analytical design will comprise an important portion of the final report; and will also serve as the basis for the experimental test bed design.

TEST BED DESIGN

The test bed is planned to be 1/4 the capacity of the full size bed, an appropriate size for the dehumidification load in a well-insulated residence. The rotating wheels will thus be about 3 ft in diameter, a size which can be fabricated without too much difficulty. The design task has not yet begun on the test bed because the pressure drop and heat exchange parameters have been found to be so important that final information was needed to proceed. As a result this task is behind schedule and may interfere with completion of the program on schedule. The budget, however, is not expected to be affected.

Preliminary discussions have been held with the Choice-Vend Division of Seeburg Corporation at Windsor Locks, Connecticut regarding the manufacture of the experimental model. Their interest has been indicated and further, more detailed discussions will be conducted. Other sources for the manufacture of the experimental model will also be investigated if warranted.

FUTURE PLANS

In the next quarter the computer model work should be completed. The final component (system) design should be made and the experimental bed design complete and procurement underway. Test installation design and procurement will have begun.

Cooling Capacity: 24,000 btu/hr (25% latent)
 Thermal C.O.P.: 0.6
 Bed Thickness: .7183 inches each
 Rotation Speed: 1.5 rev/hr
 (20 min Half-Cycle Time)
 Bed Diameter: 6 ft

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 CEM
 25 July 1975

FIGURE 1: SOLAR DESICCANT AIR CONDITIONING WITH SILICA GEL.