IMPROVEMENT TO THE DESIGN OF A SOLID ABSORPTION SOLAR REFRIGERATOR

by

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Abstract
The paper presents the highlights of the design of an existing solid absorption solar refrigerator, using $CaCl_2$ stabilised with $CaSO_4$ as the absorbent. The performances are also discussed. The sources of poor performance were identified as high thermal and pressure inertia of the system, high heat loss coefficient, and low absorbent thermal conductivity. Based on these, measures taken in the design of a new model refrigerator, to improve the refrigerator performance are then presented.

Introduction
Significant work has been done in the recent past to show experimentally, that solid absorption refrigeration, with solar energy as the heat source, is technically feasible [l, 2, 3, 4]. The absorbent-refrigerant pair in [1] was $NaCl_2$-NH_3. In [2, 3] it was $CaCl_2$-NH_3, while in [4], activated carbon methanol pair was used. The overall or net useful coefficient of performance obtained in [2, 3, 4] ranged from 0.02 to 0.12. The above COP range is significantly below the theoretical estimates. For $CaCl_2NH_3$ systems, the theoretical COP is approximately 0.5, while for activated carbon-methanol it is about 0.8. Consequently, one of the current objectives among researchers is the improvement of the overall efficiency of the system.
Results reported in [6] indicate that cooling could not be obtained during the rainy season at Nsukka, Nigeria. Thus since the solar refrigerator is being developed for the cooling and storage of meat, foods, vaccine and other drugs in rural areas where electricity is not available, and for use throughout the year, interest is being focused also on the development of field prototypes as well as on the coupling of auxiliary heating systems to the solar powered refrigerator. A re-design of the solar powered solid absorption refrigerator reported in [2] and [6] was therefore undertaken with the aim of improving the efficiency, developing a field prototype and providing auxiliary heating to the solar heat source.

Refrigerator Model - 1
Description
Solid absorption refrigerators are inherently intermittent. Consequently the solar collector, refrigerant absorber and generator were combined in a single collector/generator/absorber unit. This unit consisted of a flat plate solar collector with a corrugated steel collector plate. Six collector tubes, having a perforated co-axial galvanised inner steel tube, were bonded to the plate at 152 mm pitch. The collector plate and tubes were coated with black oil paint. The $CaCl_2$ absorbent, specially stabilised with $CaSO_4$ [7], and called Nsukkanut, was prepared in 5 10 mm granules and introduced into the annular space between the tubes at a parking density of 0.61 kg/1. Mass of granules per tube was 1.25 kg; The perforated inner tube distributed NH_3 from the inlet header to the granules during absorption, and collected the gas from the granules to the outlet header during generation. The collector double glazing consisted of an inner 3 mm thick glass sheet and an outer 4 mm thick clear PVC sheet. The exposed glazing area was 1.41 m^2. Rear plate insulation consisted of 36mm of dry grass (k_{eff} = 0.2 W/mk) and 73 mm of styroloam chips. The assembly was contained in a 2023 x 950 x 221 mm mild steel casing of G-16 thickness. Removable upper and lower end cover plates were provided to facilitate collector plate cooling by natural convection after the generation phase. The condenser was a stagnant evaporative rectangular tank with 0.6753 m free volume, and 60 mm thick walls and 4m of tubing. The walls were

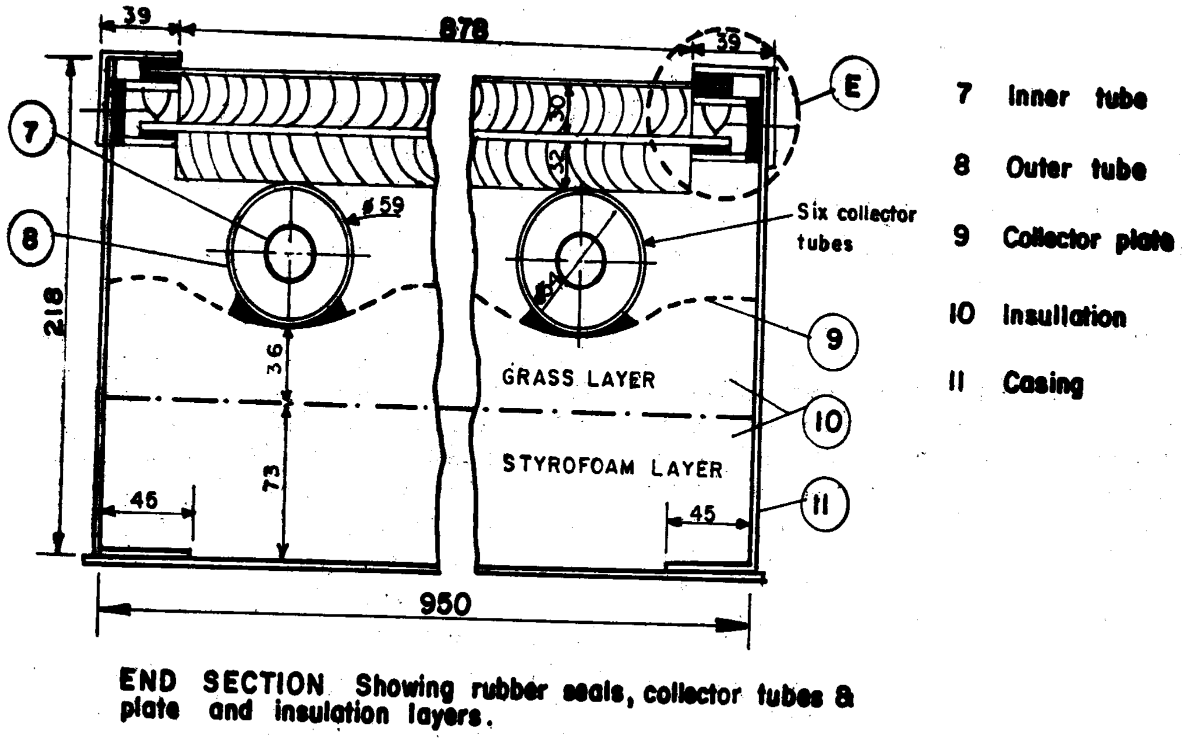

Figure 1: Cross-Section of Model-1 Refrigerator.

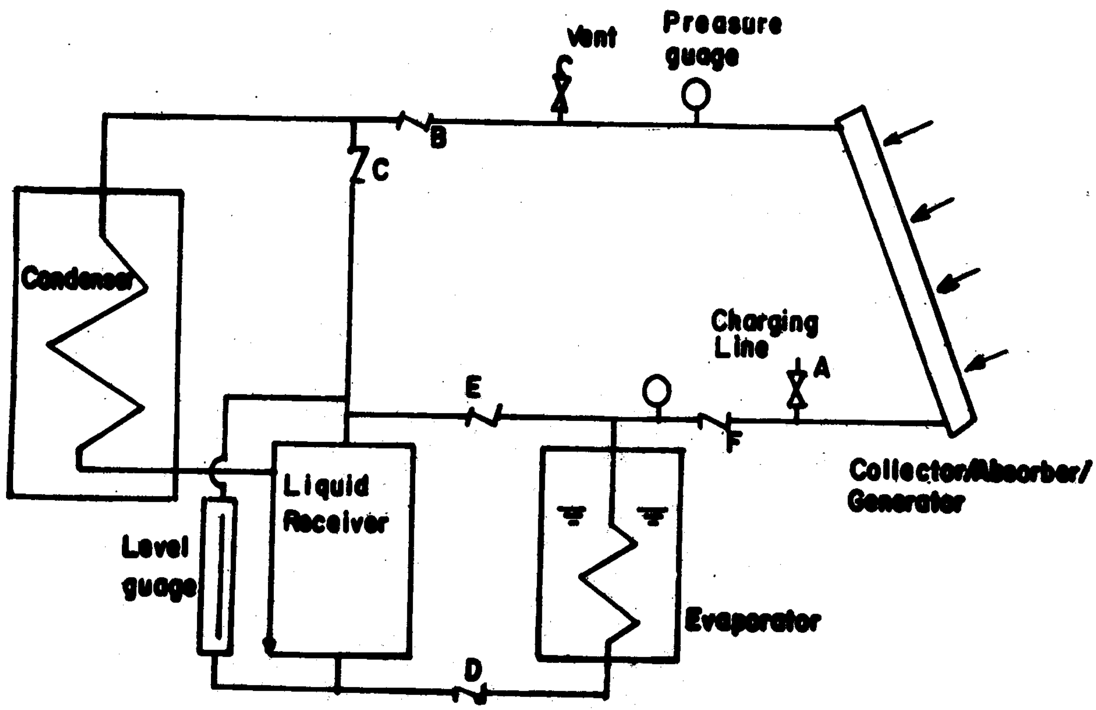

Figure 2. Sketch of Refrigerator.

cast in reinforced sandcrete. A vertical cylindrical liquid receiver of 200 mm i.d x 206 mm o.d x 300 mm high received the condensate. A separate evaporator having 1.5m long spirally coiled tube inside a cylindrical evaporator vessel of 200 mm i.d x 300mm high x 1.5 mm thick, was coupled to the receiver. Condenser and evaporator tubing were of 18 mm i.d x 21mm o.d. All constructions were of steel. Figure .1 shows a cross-section of the collector/generator/absorber while figure 2 shows a sketch of the refrigerator layout. Other details of the system construction may be found in [2].

Performance of Model - 1

Performance tests of the Model -1 Nsukkanut refrigerator were reported in [2, 6 and 8]. Figure 3 shows the condensate, collector plate temperature, NH_3 saturation temperature corresponding to the system pressure, 'and other temperatures, during generation for one of the test runs. It is seen that in a generation period of 8 hours, it took 3 hours, or 375% of generation time, before condensation could start (onset of constant NH_3 saturation temperature). In other tests about 50 - 60% of the generation time was used in this way. The initial NH_3 generated was utilised for the pressurisation of the system, and the larger the free volume of the system the longer this process takes. The above results indicated that the free volume of the system, 181, was excessive.

The rate at which NH_3 is being generated during the pressurisation and heat up period, and hence the length of this period, depends on the balance between the heat collected from the sun and the heat required to raise the sensible energy of the collector plate, tube and. granules. This is so provided the granule temperature is not less than the equilibrium reaction temperature at the system pressure (see Figure 4). The heat capacity of the collector plate, tube and absorbent is therefore a controlling parameter. Figure 5 from reference [3] shows the effect of mass of steel in absorber tubes and collector plate per unit mass of desorbable ammonia, on the COP. With a collector plate and tube mass of 53 kg for the design under consideration, the value of this ratio was 16, which is within the suggested range of 10-20 in reference [3]. However, a reduction in the ratio, should improve the thermal response of the refrigerator.

Table 1 shows a comparison of the COPs of the Nsukkanut refrigerator [6] with those of other reads for solid absorption systems.

It is seen that the peak performance of the Nsukkaout refrigerator though reasonable is up to a factor of two less than those of the other units. The reason is obvious from column 5 of the table which shows the solar energy collection efficiency as being up to a factor of two less than those obtained for other refrigerator.

Other Sources of Inefficiency

Mention has already been made of the high thermal and pressure inertia of the Nusukkanut refrigerator and their contribution to the poor perfom1ance of the unit. Other sources of poor performance include the absence of a selective surface for the collector plate, air leakages and low absorber thermal diffusivity. If the collector is not properly sealed a natural convection current of air will be set up through the collector, during generation and significant heat losses can occur as a result. Finally, it is known that calcium chloride has a very low thermal conductivity (0.12 - 0.23 W/mk). The low thermal conductivity, and consequently the low thermal diffusivity greatly reduce the rate of transport of beat to or from the granule mass during generation or absorption.

Table 1: Comparison of Refrigerator Performances

Reference	system	Available Overall COP	Actual Overall COP	Collector Efficiency % -	Number Glazings	Plates surface
6	$CaCl_2NH_3$	0.014-0.175	0.008-0.0053	2.9 – 15.1	2	Black oil paint
3	$CaCl_2NH_3$	0.098	-	28	2	Black Coating
3	$CaCl_2NH_3$	0.129^+	-	-	2	Selective (Maxorb)
4	Activated carbon/Methanol	0.165^+	0.12	20.8^+	1	Selective

+ Figure deduced from data in the reference

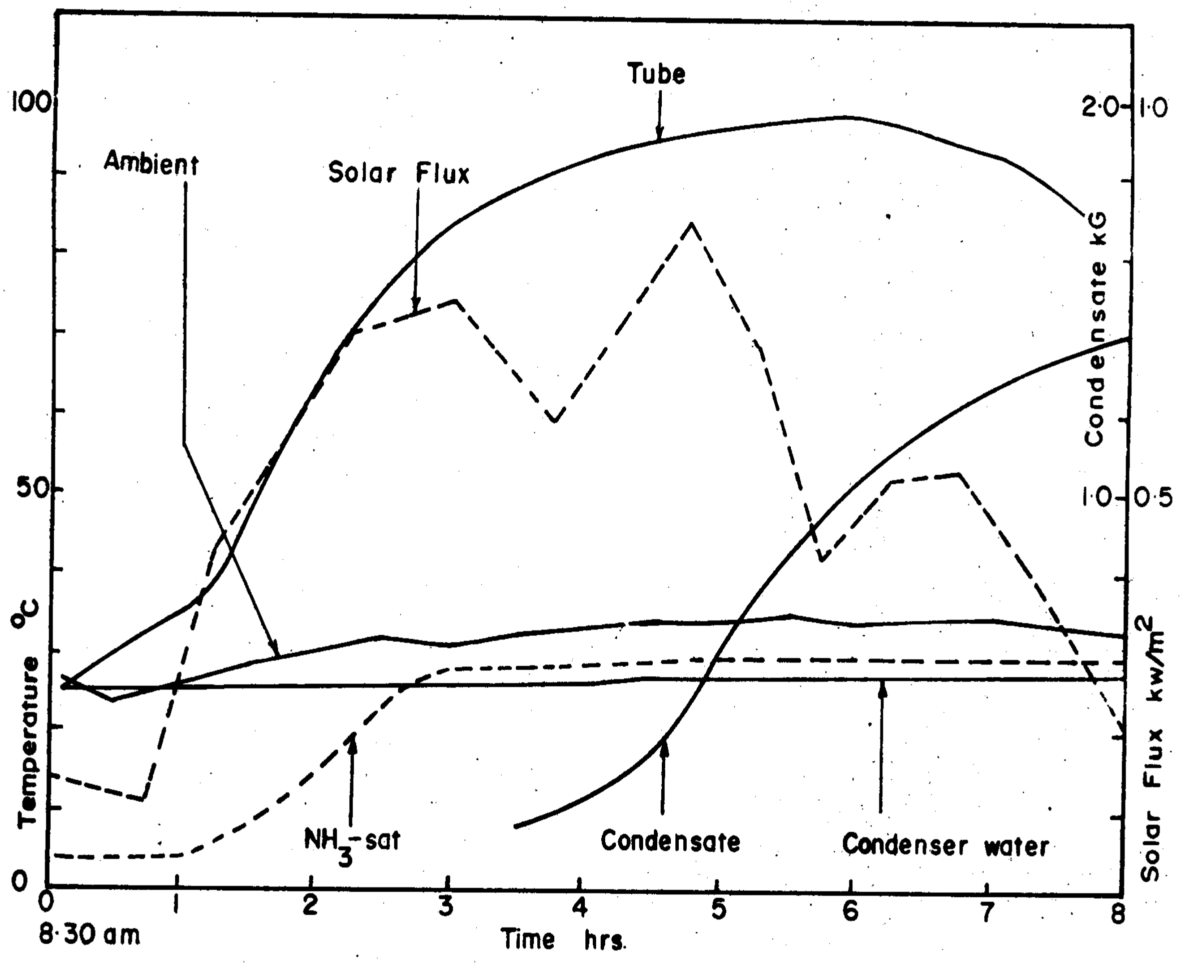

Figure 3. Temperature Profiles During Generation in Sunny Dry Season, Run #31.

These lead, not only to a reduction in the granule heat up rate, and ammonia generation and absorption rates, but also to a reduction in collector efficiency during generation.

New Model Solar Refrigerator

Bearing in mind the sources of poor performance of the Model-1 Nsukkanut refrigerator, the new model has been designed with the following features:

Reduced Thermal and Pressure Inertia

Figure 6 shows a section of the new collector/generator/absorber. The materials of the collector plate and outer tube have been changed to aluminum. The inner tube material remains steel due to non availability of aluminum tube of appropriate size. Outer tube is 6mm o.d x 125 mm i.d. With a collector plate area of $1.85m^2$ and thickness of 0.7mm, the collector material heat capacity per unit actual area becomes $14kJ/m^2$, using the same spacing of 15201m as for the Model- refrigerator. This compares with 16.3 kJ/012 for the Model-1 refrigerator. It is noted that the heat capacity surface density calculated for the new model includes the fittings and piping manifold needed for the auxiliary heater, which did not exist in the old model. If these are deleted, then the heat capacity density for the new model becomes approximately $13{:}3kJ/m^2K$ which represents a reduction of approximately 18.4%.

The corresponding surface densities of granules, with the granules inside the tube estimated at the same parking density of 0.61 kg/l, are 3.24and $4.97kglm^2$- actual Collector area for the new and old models respectively. These lower surface densities for collector heat capacity and granule mass, should help the attainment of higher granule temperatures and more complete desorption of NH3 during generation, since collector thermal inertia is reduced.

The equivalent steel mass of collector plate and tubes, for the same heat capacity, is 45.13 kg. Total

granule mass is 6 kg. Collector plate and tube mass per kg of disorbable NH_3 is·10.2. This compares with. 16 in the old design, and judging from figure 5, should lead to some increases in the COP.
With respect of the pressure inertia, much of the free volume is contained in the liquid receiver and the unnecessarily long piping in the old design. The receiver volume should be just enough to accommodate the total mass of liquid refrigerant in the system at condenser conditions. This calls for a receiver volume of 3 x $10^3 m^3$, a reduction by factor of 3 from the old design. Since the new model is intended to be movable, the piping will be compact. These measures will greatly reduce the pressure inertia.

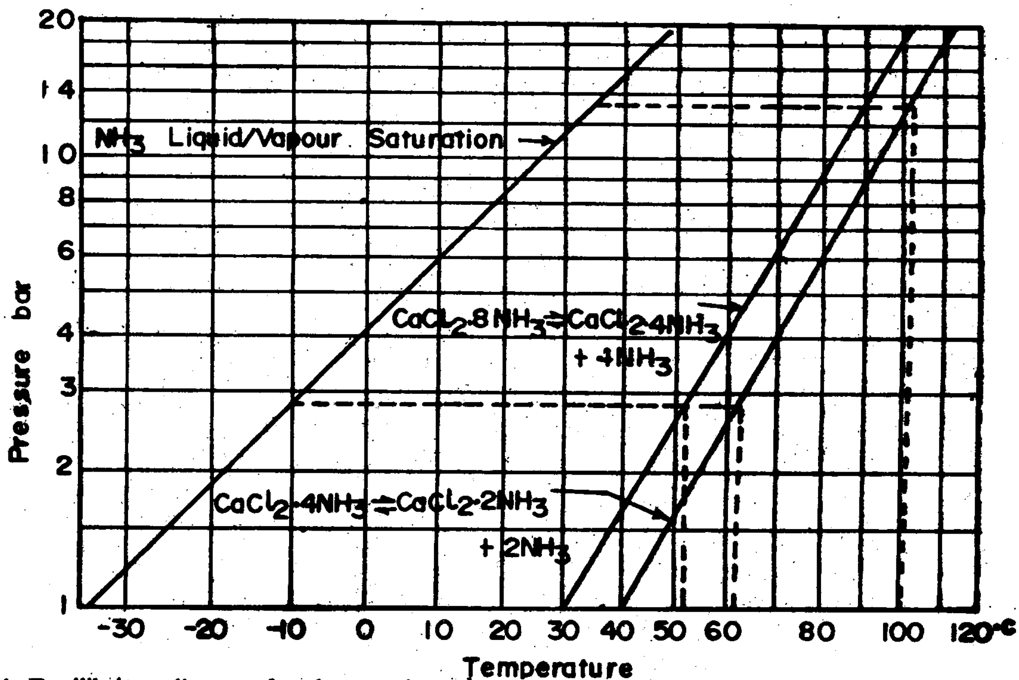

Figure 4: Equilibrium diagram for the reaction of Calcium–Chloride and Ammonia (10).

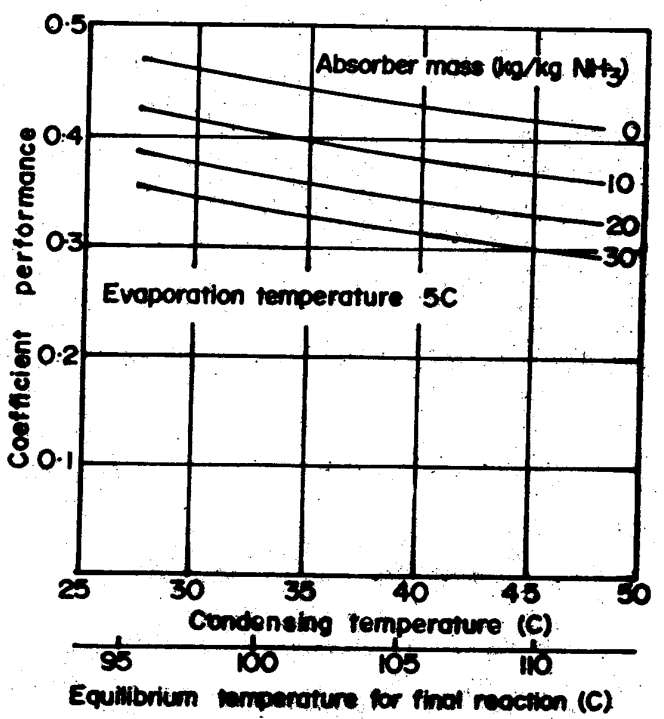

Figure 5: Coefficient of performance of the calcium chloride/ammonia cycle with different absorber mass ratio (mass of steel in absorber tubes and collector plates per unit mass of desorbable ammonia. For a practical design the mass ratio is about 10–20 [30]

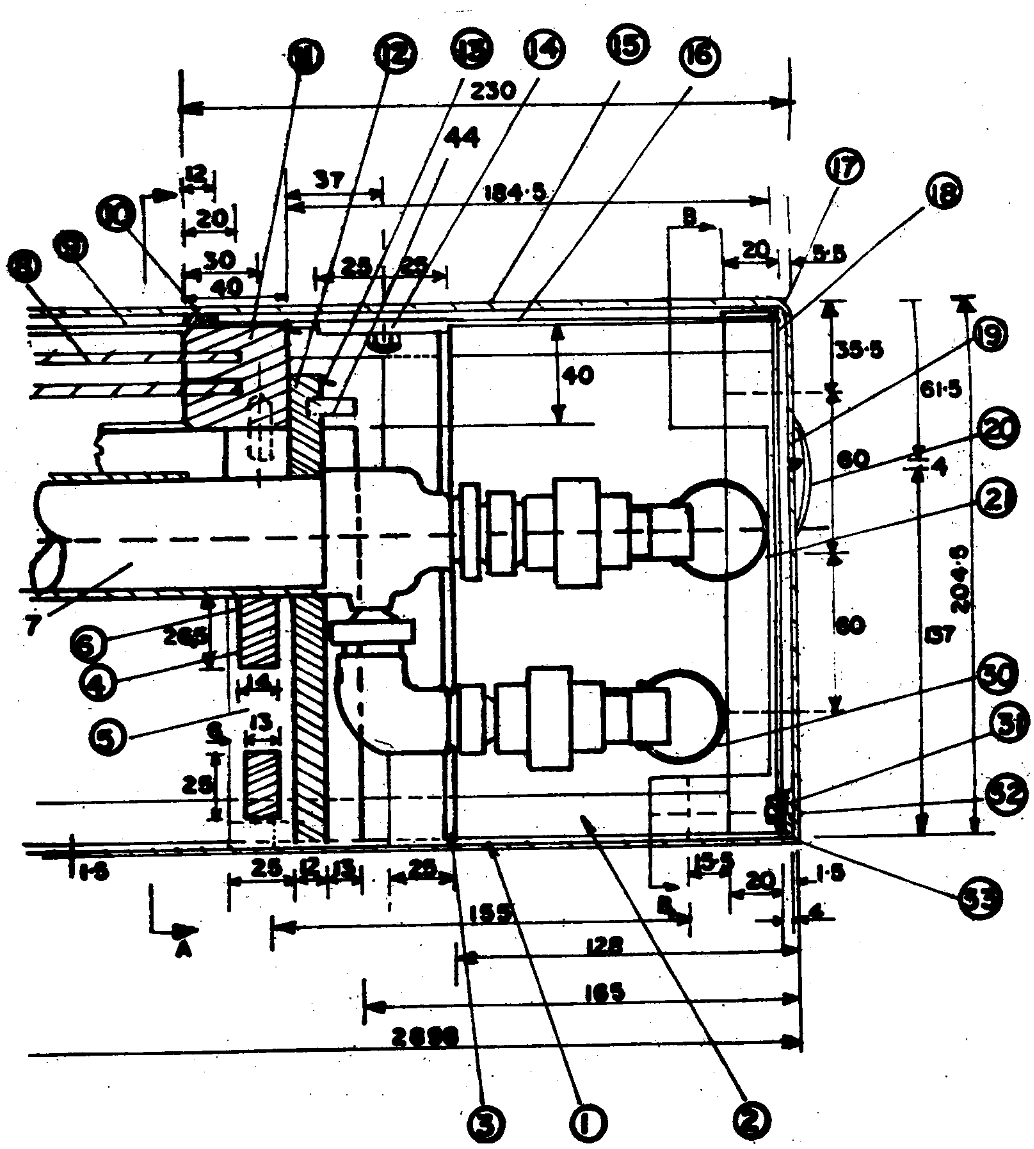

Figure 6: Longitudinal Cross-Section of new Collector (Scale x1/2mm)

Collector Plate Surface Heat Loss Coefficient

An inspection of the old model showed that there were air gaps, through which natural circulation of air could be set up over the collector plate during generation. This obviously would increase the plate surface heat loss coefficient. Measures have been taken in the new design to reduce this to a minimum. It is recognised that a selective surface for the collector plate is a necessary provision for the temperature range at which the plate is expected to work, if the heat loss coefficient is to be reasonably low. Contact has been made with the British Manufacturers of Maxorb, a popular selective coating, but foreign exchange restrictions limit the possibility of obtaining supplies from them. At the same time, a. project was -initiated in the Department of Mechanical Engineering, at the Masters level, with the aim of developing in house, the technology of selective film production. That project is still at the early stages. It therefore looks likely that the new model refrigerator will, for now, be without it selective surface.

Absorbent Thermal Properties

To increase the effective thermal conductivity of Nsukkanut, a new mix was produced which has the potentials of having an increased thermal conductivity. An experimental programme was also set up to test the new Nsukkanut.

Analytical Basis for Thermal Property Test

The experiment is based on a transient solution of one dimensional radial conduction equation in cylindrical co-ordinates, as illustrated in figure 7, and due to Beck et al [l l]. The equation

$$pc\frac{\delta T}{\delta t} = \frac{1}{r}\frac{\delta}{\delta r}\left(kr\frac{\delta T}{\delta t}\right) \qquad (1)$$

Was solved with the following boundary conditions

$$k\frac{\delta T}{\delta r_{r=r_1}}) = q(t) \qquad (2)$$

$$k\frac{\delta T}{\delta r_{r=r_2}}) = 0 \qquad (3)$$

If the uniform granule temperature at start (t = 0) is T_1 and at the end of the experiment (t =∞) is T_1 then equation (1) can be integrated with the boundary conditions to obtain that

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$$k = r_1 Q" [0.5 - \frac{r22^{In(r2/r1)}}{r22^{-r_{12}}}] \int_{t=0}^{\infty}(T_{r2} - T_{r1})dt \qquad (4)$$

$$pc = \frac{2r1Q"}{(T_f - T_1)(r22^{-r}12)} \qquad (5)$$

a = k/pc

k = effective thermal conductivity of specimen

p = effective density of specimen parking

c = specific heat of specimen parking

a = effective thermal diffusivity of parking

r_1 = inside. radius of specimen

r_2 = outside radius of specimen

T_{rl} = Specimen temperature at rl

T_{r2} = Specimen temperature at *r*

T_f = Uniform specimen temperature at t = ∞

t = time

$Q" = \int_{t=o}^{\infty} q(t)dt$

The experiment involves obtaining T_t, T_f and the transient trace of (T_{r2} - T_{r1}) and, by graphical integration, obtaining:

$$\int_{t=o}^{\infty}(T_{r2} - T_{r1})dt$$

After a finite time, (T_{r2} - T_{r1}) tends to zero, and the experiment is stopped so that the upper limit of the integral is not at infinity.

The quantity Q was obtained by passing a known constant current through the resistance heater located at the inner core. The resistance of the heater and the heating time were measured. Q" was then determined using the above values and the surface area of the specimen at r_1

Figure 8 shows a sketch of the experimental set up. It consisted essentially of two concentric steel tubes, with the specimen granules. 3-5 mm in size, packed in the annular space between the tubes, The heater consisted of an electric resistance wire wound in spiral grooves on an aluminium core and located inside the inner tube. The wire was insulated from the core and the tube. Temperature measurents were with copper-constantan thermocouples, three of which were located equidistantly on the outer surface of each tube. The tube length of 300mm and outer diameter of 59.4 mm were chosen to give a length/diameter ratio of approximately 5 so as to minimise axial conduction. Thermocouple circuitry was arranged so as to give the millivolt traces of T_{rI}.

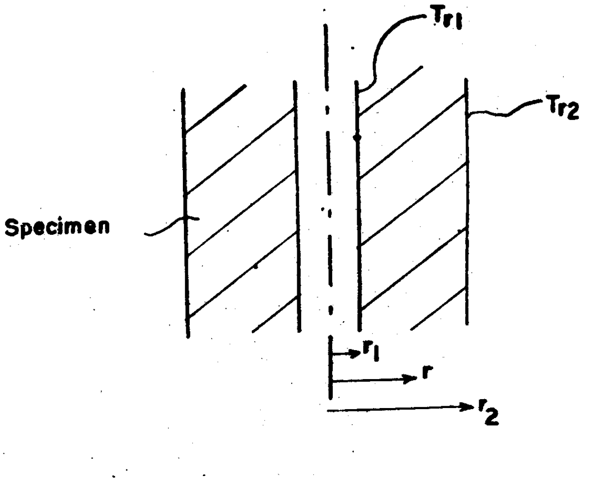

Figure 7: Model for Conductivity Experiment.

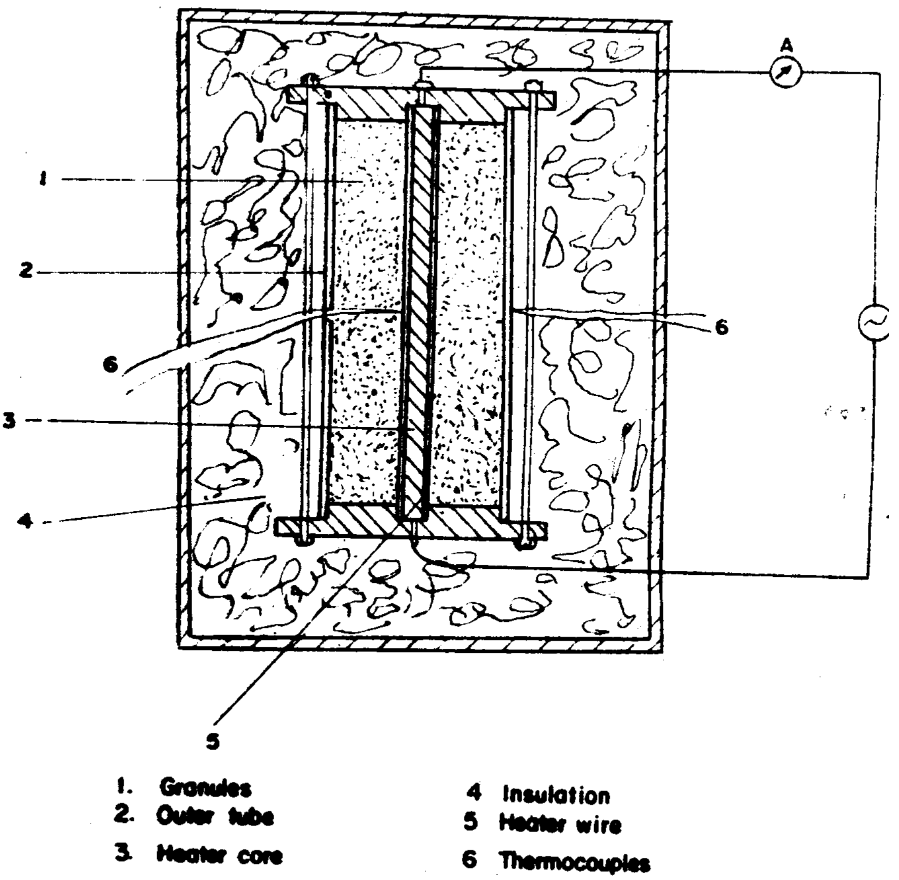

Figure 8: Conductivity Test Apparatus.

T_{r2} and (T_{r2} -T_{r1}) on a chart recorder (Yokogawa Electric Works YEW Type 3(61).

Thermal Properties Test Results

Figures 9 shows typical chart traces of the millivolt outputs. The trace I_B - O_B should tend to zero at large time. The deviation from zero reflects the non-absoluteness of thermal insulation. Data reduction to obtain Q" took into account the heat absorbed by the apparatus containing the specimen. Also, since the outer surface thermocouple was located on the outer surface of the outer tube and not on the outer surface of the specimen, allowance had to be made in the data reduction for the contribution of the outer tube material to the effective properties obtained. The specimen properties are thus given by

$$K_{sp} = \frac{\mathrm{kln(r2r1)}}{\mathrm{ln(r3/r1)}-(\mathrm{k_{eff}}/k_s)\mathrm{ln(r3/r2)}} \qquad (6)$$

$C_{sp} = ((m_{sp} + msC_{eff} - m_sC_s)/m_{sp}$ (7)

Where

K_{eff} = effective thermal conductivity of specimen and steel tube as given by equation (4) with r_2 replaced by r_3.

k_s = thermal conductivity of outer steel tube

k_{sp} = thermal conductivity of specimen

m_s == mass of outer steel tube

m_{sp} = mass of specimen

C_s = specific heat of outer steel tube

C_{sp} = specific heat of specimen

C_{eff} = effective specific heat of specimen and steel tube combined:

$2\pi r_1 LQ''/(M_{sp} + M_s)(T_f - T_1)$.

Table 2 below gives the results obtained for the improved Nsukkanut

Test No.	K W.mk	C kJ/KgK	α_2 m^2/s
1	0.147	7.80	3.48×10^{-3}
2	0.193	3.81	9.34×10^{-3}

The results obtained so far are only indicative. Results for the ordinary Nsukkanut used as a reference were not reliable. However published data on thermal conductivity for $CaCl_2$ powder from references (9) and (10) are 0.0865 W/mk and 0.12 - 0.23 W/mk respectively. By considering these data with those given in Table 2 for granular packing of porous new-Nsukkanut pellets, there is reason to expect that the improved Nsukkanut would have better thermal conductivities. Further tests are being carried out to confirm this trend.

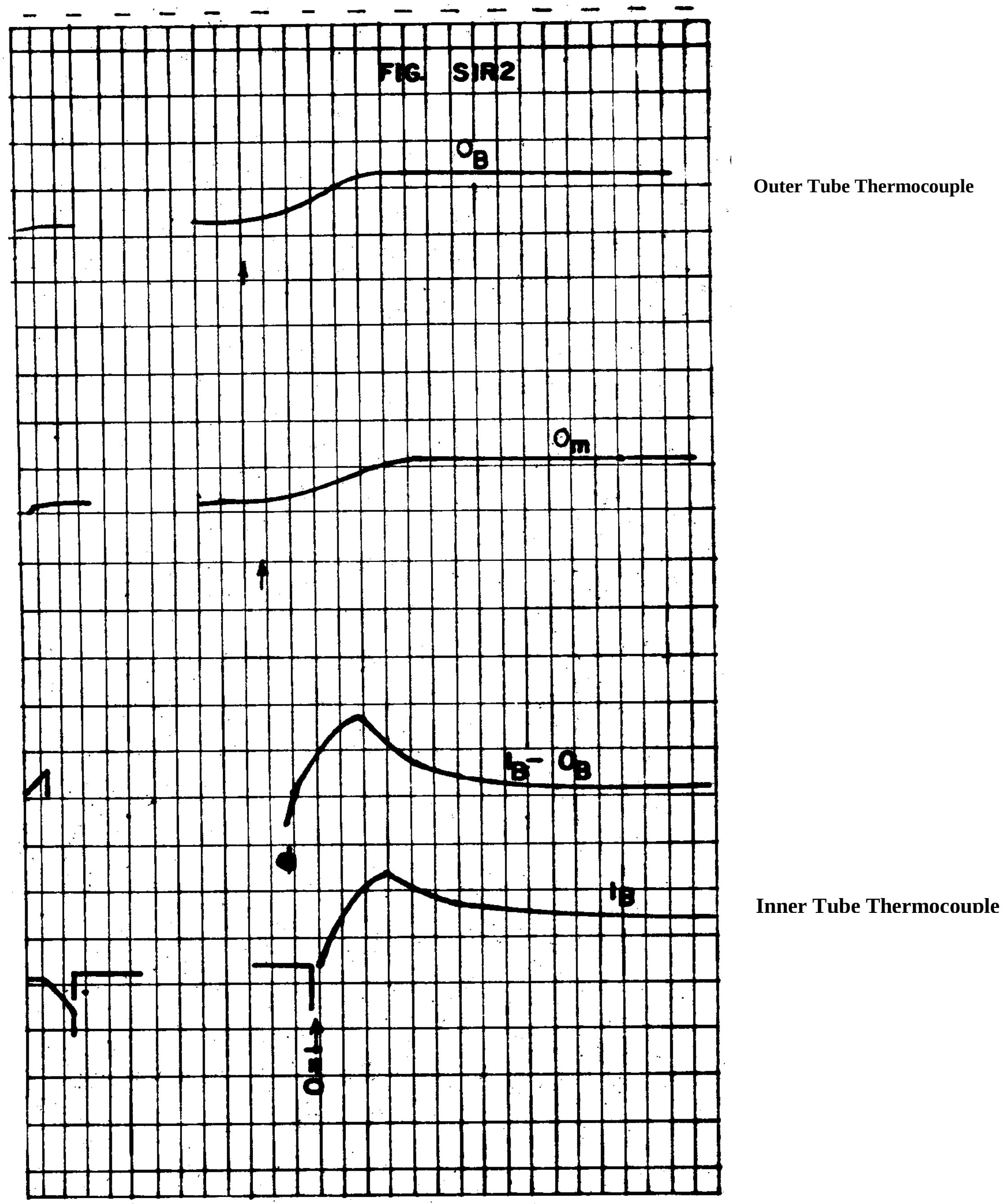

Figure 9: thermocouple trances for Conductivity Tests

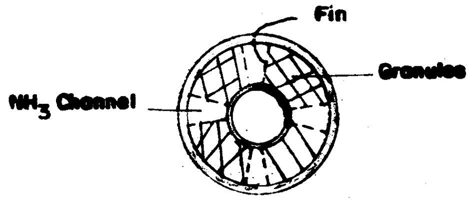

Figure 10: Cross-Section of new Collector Tube.

Auxillary Heat Source

A gas fired beater has been designed for coupling to the refrigerator, through an inlet and an outlet hot gas headers. The combustion products from the inlet header flow through the inner pipe of the collector tube, 85 shown in figure 6. Consequently the inner tube is no longer available for NH_3 distribution. The annular space between the inner and outer pipes of the collector tube has been provided with longitudinal passages separated by radial fins, as shown in figure 10. The fins also permit more effective heat transfer to the granules.

Gas-Solid Interface Areas

The reaction rate during absorption of' the refrigerant by the solid absorbent, depends on the contact area between gas and solid, and should increase as the area increases. The absorbent granule size for the new-model refrigerator has therefore been-reduced to 3-6 mm from 5 - 10 mm in the old model in order to achieve increased interface area.

Conclusion

The limitations and sources of poor performance of Model-l Nsukkanut refrigerator have been identified and discussed. Provisions in the design of the new model refrigerator, with a view to improving the performance and permitting all season and all weather operation for the unit have been presented. An experimental programme to test the thermal properties of the improved Nsukkanut absorbent pellets has also been discussed in some detail. The new refrigerator is expected to produce 4 kg of ice per cycle. It is also expected that improved features incorporated in the new design will lead to a higher performance refrigerator.

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