12 Oct 1978

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Abstract

From developed design procedures a number of electric, thermal storage unit configurations are compared. The comparison is based on specifications of low volume, high energy capacity and high power output suitable for residences in North America.

1. INTRODUCTION

Storage heating equipment has been used for off-peak heating in Europe for the past 25 years. The electric heating component of the load on the national grid is shifted from the day-time peak to the night-time valley. This provides a flatter daily load demand and is suitable for base load generation. In North America, with the growing base load generation, the regulation of the load has become increasingly important. Due to a large grid system, where power is shared by many utilities, a new problem arises. A utility agrees to have a reserve spinning generation capacity. However, to under-utilize generation capacity is expensive. An alternative solution is to have rapid control over part of the load which may be shed during peak periods of demand. Storage heating may be used for short term load shedding and for daily off-peak heating.

The European storage heating equipment is designed for off-peak usage with a typical discharge period of 16 hours. Also, the heating demand is small (5 kW) compared to that of northern regions of North America. The use of individual room storage radiators, although common in Europe, would be impractical here. Due to the larger room heating demand, individual units would cost more than asingle central unit. Due to cost and consumer demand only central storage units with a typical forced air distribution system are acceptable. Central storage units are produced in Europe; however, they are designed for low power outputs and long term storage periods only. The high power outputs required for North American units present some core design difficulties.

2. DESIGN

2.1 CORE DESCRIPTIONS

2.1.1 European Unit

A typical European storage furnace core is shown in Fig. 1. These furnaces are designed and return air plenums to be connected at the base of the unit. The core is mounted on a fan chamber at the base. Since the core is at a high temperature (900°C max.), it is desirable that the fan sucks air from the return air duct through the storage core and discharges it. The output temperature is regulated by a proportional control that mixes high temperatuere air from the core with air that bypasses the core to maintain the desired output air temperature. The air passage through the core consists of a number of parallel air passages from the bottom of the core to the top and then to the bottom again. This inverted U tube design provides a natural air lock to prevent convective losses. The core is built up from individual bricks and the air passages are formed by vertical grooves in the sides of the bricks. Electric elements are placed in horizontal grooves in the bricks. The element is typically an open wire spiral type and the element groove has sufficient internal surface area to conduct the radiated energy from the element into the brick mass. Due to low material conductivity there is a temperature gradient across the brick.

It is required to investigate design procedures for a storage furnace that would be suitable for North America. The design requirements are high power output (10 to 25 kW), long and short term storage (2 to 16 hours) and to supply the heating demand while the core is being charged, preferably without the use of an auxiliary furnace.

In addition to a modified European brick design two other core configurations were considered.

2.1.2 New Core Designs

One design consists of a tubular air passage as shown in Fig. 2. The use of tubes allows a thinner cross-section of material than is found in a brick design. The heat transfer is improved since air may flow over the interior and exterior tube surface to increase the available output power. This design, and the European design, may be modified to supply the heating load during the core charging period. This is accomplished by increasing the heating element capacity in the core, and placing it in such a manner that the incoming air to the core will be rapidly heated to the core charging temperature and passed through the core, without disturbing the charging heat transfer process.

Another design has the form of a packed bed as shown in Fig. 3. This design utilizes an exterior source to heat air which is passed through the bed at a low air velocity. The core pebbles have a natural large surface area to facilitate heat transfer and absorb heat energy quickly. During discharge the energy can also be released quickly so that high output power can be developed. Since the heating elements are external to the core, the supply of the heating load during the charge interval is readily accomplished.

2.2 CORE MATERIALS

It was found that the core material provided the limiting design factor. For high temperature (800^{o}) storage, refractory materials are the most suitable. Metals such as aluminum and iron have superior thermal qualities but suffer from higher cost, and oxidation.

High output power from a storage unit requires a high heat flux from the core material. Values of between 4 and 10 BTU/hr.ft^2 ^{o}F are typical. If there is a significant difference between the mean material temperature and the heat exchange surface temperature, the required heat exchange area will be proportionally larger. When there are temperature gradients in a material, it is termed thermally thick. The modelling equation is

$$\frac{Km}{Q\delta}\ (t - to) = Fo + \frac{x^2}{2} - X \qquad (1)$$

and $\quad Fo$ = Fourier number = $\frac{\alpha\theta}{\delta^2}$,

where Km = material conductivity (^{BTU}/ft.^{2} hr. ^{o}F),

Q = surface heat flux (^{BTU}/hr. ft.^{2}),

δ = material thickness (ft.),

t = temperature at time θ (^{o}F),

to = initial temperature (^{o}F),

θ = time (hrs.),

α = thermal diffusivity and

X = distance from the face(ft.).

For Fourier numbers greater than 0.5 the material may be assumed to be thermally thick and the surface temperature is approximately two-thirds of the mean material temperature. Thus, it is desirable to maximize the thermal conductivity even though temperature gradients cannot be avoided. Typical design conditions for a tubular air passage bed are θ = 6 hrs., δ = 0.133 ft., ρ = 160 lb. ft.^{3}, Cp = 0.24 BTU/lb. ^{o}F and K = 0.813 BTU/hr. ft. ^{o}F. Thus, α = 0.0212 and F_o = 7.19.

Since the Fourier number is much greater than 0.5, the unit material is thermally thick.

In a packed bed design the heat transfer coefficients are lower (2 to 3 BTU/hr. ft.^{2} ^{o}F) due to the large surface area. The Biot criterion is used to determine if a material is thermally thick. That is,

$$B_i = \frac{h}{kx}\ , \qquad (2)$$

where h = heat transfer coefficient (^{BTU}/hr.ft.^{2} ^{o}F),

k = thermal conductivity (^{BTU}/hr. ft. ^{o}F) and

x = thickness (ft.)

For small diameter particles (3/4 to 1 1/4") a Biot number of 0.1 or 1% temperature gradient may be obtained with high conductivity refractories. The thermal conductivities of various materials are given in Table 1 and Fig. 4. High density materials have high thermal conductivities due to the bulk density. Densities in excess of 160 lb./ft.^{3} are found to be acceptable. A material called Feolite, developed in Britain, has a density of 250 lb./ft.^{3}. Due to the high density of the core, units with large capacity are of a size suitable for residential applications. Thus, for heat-transfer considerations packed beds have superior properties.

The electrical conductivity of refractories increases exponentially with temperatures in excess of 1400^{o}F. Thus, for cores where the heating element is in contact with the core material, the maximum operating temperature is 1400^{o}F.

2.3 INSULATION MATERIALS

Commercially available insulating materials are satisfactory. The thermal conductivities of these materials generally increase exponentially with temperature as shown in Fig. 4. For the insulation of high temperature cores the insulation jacket thickness and cost will increase exponentially with temperature. If cost is a consideration, less costly but bulky materials may be used. If bulk is a consideration of more moment, such as when producing a storage furnace shell that must pass through a doorway, then the cost will be high. Typical insulation jacket design results are presented in Fig. 5.

2.4 DESIGN PROCEDURES

The required cross-sectional area for air flow through a core is dependent on the air temperature, the output power and the air velocity. The governing equation is

$$K = (3414 \times P / Cp\ (T_{IN}-T_{out}))/\rho \times Vel \times 60, \qquad (3)$$

where K = air passage cross-sectional area (ft.^{2}),

P = required power (kW),

C_p = heat capacity (^{BTU}/lb. ^oF),

T_{IN} = input air temperature (^oF),

T_{out} = output air temperature (^oF),

Vel = air velocity (^{ft}/min.) and

ρ = air density ($^{lb.}$/ft.^{3}).

The air mass flow rate and hence the required cross-sectional area vary exponentially with the output temperature. This is shown in Fig. 6. Thus, the output air temperature should be maximized to reduce the cross-sectional core area required for air passages and hence decrease the core volume.

2.4.1 Tubular Cores

The modelling of cores consisting of rectangular or cylindrical tubes is as follows. First considering the heat transfer during the discharge cycle, the series parallel arrangement of tubes (normally two series passes) may be modelled by one long set of parallel passages representing the sum length of the series passes. Using the air flow cross section from equation (3) the size of each tube may be determined. The tube must be of sufficient length to provide adequate heat transfer. From equation (3) the air velocity and input-output air temperatures are fixed; thus the heat transfer coefficient may be determined. It is given by

$$H = R \times K \times Nu / D, \qquad (4)$$

where $Nu = .023 \times Pr^{0.4} \times Re^{0.8}$ (Nusselts number),

$Re = \rho \times Vel \times D / N \times 60$ (Reynolds number),

K = conductivity of air (^{BTU}/ft. ^oF. hr.),

ρ = density of air ($^{lb.}$/ ft.^{3}),

N = viscosity of air ($^{lb.}$/ ft.) and

R = surface roughness factor.

The surface roughness for a refractory tube may be assumed to be 2.0.

The average brick and air temperatures required for heat transfer are calculated on the basis of a linear model with a constant temperature difference.

The equation of heat transfer is given by

$$Tc_{av} = T_{av} + 3414.4 \times P / (H \times As), \qquad (5)$$

where As = surface area of heat transfer (ft.^{2}),

T_{av} = average air temperature (^oF) and

Tc_{av} = average core temperature (^oF).

Knowing the required heat transfer surface area, the length of the parallel tubes may be calculated. As previously shown the average core material temperature is one third above the average wall temperature (Tc_{av}). For a given maximum core temperature the volume of core material from which the thickness may be calculated is given by

$$Vol = E \times 3414.4\ Cp\ (Tc_{max} - Tc_{min}) \times \rho c, \qquad (6)$$

where Vol = core volume (ft.^{3})

E = energy stored (kW),

Cp = material specific heat (^{BTU}/lb. ^oF),

Tc_{max}, Tc_{min} = average max./min. core temperature (^oF) &

ρc = core material density (lb./ft.^{3}).

At the high charging temperature the heat transfer is dominantly radiative. The basic equation of radiative heat transfer is given by

$$Q = F\sigma\ (T_1^4 - T_2^4), \qquad (7)$$

where Q = heat flux (^{BTU}/ ft.^{2}),

σ = Stefan Boltzman constant,

T_1, T_2 = absolute temperatures and

F = form factor.

For calculating the form factor it is assumed that the heating element surface will have cylindrical symmetry and that there is negligible end loss from the element. For the core design the input charging power is known and hence the heat flux Q is known. The maximum average core temperature is known and hence the problem is to calculate the required element temperature and size.

2.4.2 Packed Beds

The design procedure for a packed bed involves comprising between small pebbles for good heat transfer and large pebbles for low air flow pressure drop. Once the size has been determined, it becomes standard for units of all energy and power ratings.

During the discharge cycle air is forced through the bed from the bottom to the top. The heat transfer and associated air velocity calculations are based on the assumption of rapid heat transfer, so that the Biot number must not exceed 0.1. The maximum heat transfer rate is

$$H = 0.2\ Km\ /\ Dm \quad (BTU\ /ft.^2\ ^oF.\ hr.).$$

A knowledge of the rate of heat transfer (assumed to be mainly convective) from pebbles to air leads to the value of the maximum air flow. Using the correlation of Pei[1] the free tower velocity is given by

$$FTV = \frac{\mu a}{Dm\ \rho a}\ \frac{H\ Dm}{(0.016\ Ka\ Ar^{1/4}\ Q^{3 \cdot 76})}\ ft./min.), \quad (8)$$

where μa = viscosity of air (ft. min./lb.),

ka = conductivity of air (BTU/ft.^{2} ^{o}F. hr.),

ρa = density of air (lb./ft.^{3}),

Ar = Archimedes number and

Q = shape factor (1 for spheres and 6.7 crushed rock).

Results are shown in Fig. 7.

Substituting the free tower velocity into equation (3) gives the required bed cross section for any specified output air temperature.

The pressure drop across the core is given by the Ergun equation.

$$\Delta P = \frac{fp\ \rho_a\ L\ V^2\ (1 - V_f)}{Dp\ V_f^{\ 2}\ g_c}\ , \quad (9)$$

where L = core height (ft.) ,

Vf = void fraction ,

g = gravitational constant ($^{lb.ft}$/lb. s^2)

fp = friction factor and

Dp = particle diameter (ft.).

The friction factor depends on the physical shape and packing of the bed material. For spherical particles Ergun gives fp =(0.584 + 33.3/Re) as a lower limit and for crushed material Hollands gives fp =(1.27 + 210/Re) as an upper limit. Fig.8 shows the pressure drop per unit height of core for different air temperatures, particle sizes and air velocities. The rate of mass-flow of air and hence the air pressure drop can be reduced by increasing the output air temperature for a given power. However, this increases the required bed size for storage. A minimum air pressure drop of 0.15 to 0.2 inches of water is required to provide distribution of incoming air over the bed surface. This provides uniform heat flow across the bed. For this criterion a particle size, core height and diameter may be selected from Fig. 9.

A set of design curves may be produced as shown by Fig. 9, where a design for any bed diameter and particle size may be found by interpolation. For discharge time intervals other than 16 hours the x and y axis may be scaled proportionately.

During the charging process air is super heated and forced to flow from the top of the bed downwards. Since the air is at a high temperature, the air velocity will be low, with ensured rapid heat transfer. This results in a high temperature wave moving downwards through the bed. Charging the bed from the top down ensures that on discharge high temperature air will be available when the bed is only partly charged initially.

3. RESULTS

Table 1 presents a brief comparison of cores of 20 kW power output and storage times of 4,8 and 16 hrs. For installation in a residence it is desirable to have a maximum case size of 2.5 ft. to allow passage of the unit through doorways. Because of the weight involved the cores will be installed in the field.

An examination of Table 1 shows that the 4 and 8 hour storage units may be manufactured with a base size less than 2.5 feet. However, the 16 hour unit is too large. The packed bed has a size and cost advantage for 4 and 8 hour charges. For the 16 hour charge the rectangular tubed core is smaller and cheaper.

From these figures it may be seen that a packed bed core is suitable for short term storage and a tube core is suitable for long term storage. The cross over point is shown in Fig. 10.

4. ACKNOWLEDGEMENTS

The authors gratefully acknowledge the aid of the Canadian Electrical Association, which provided funds for this work through Contract No. RP76-21 and gave permission for this paper to be published.

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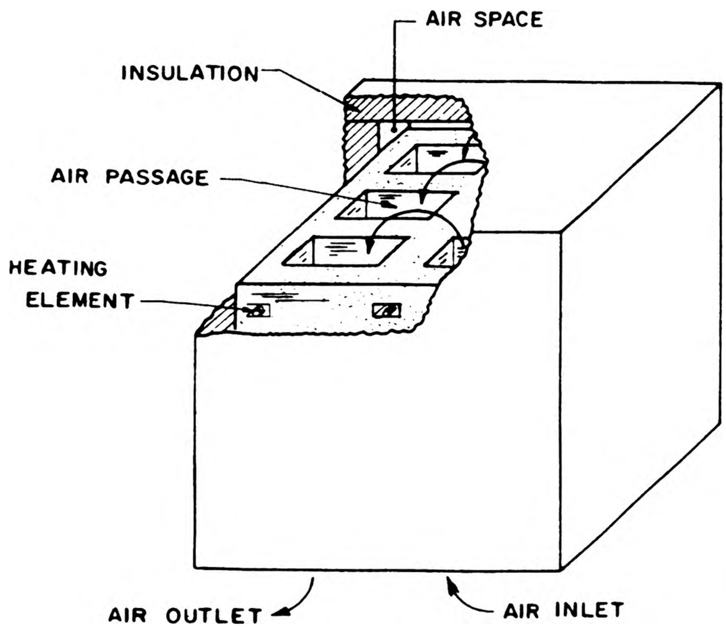

Fig. 1 Typical European Storage Furnace Core

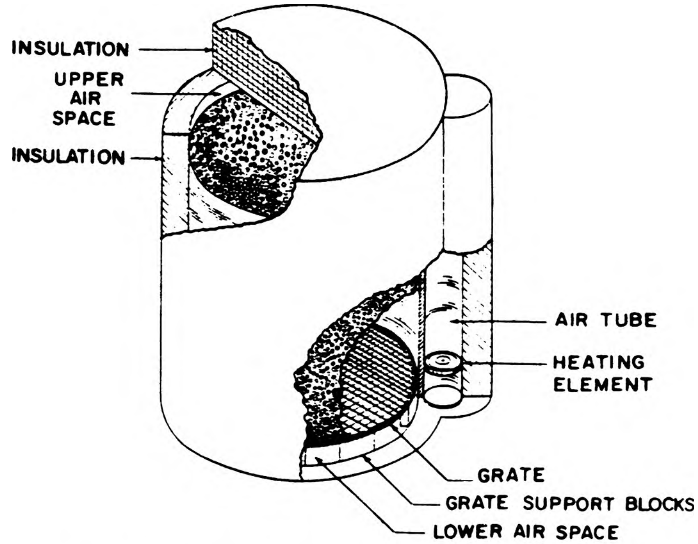

Fig. 3 Packed Bed

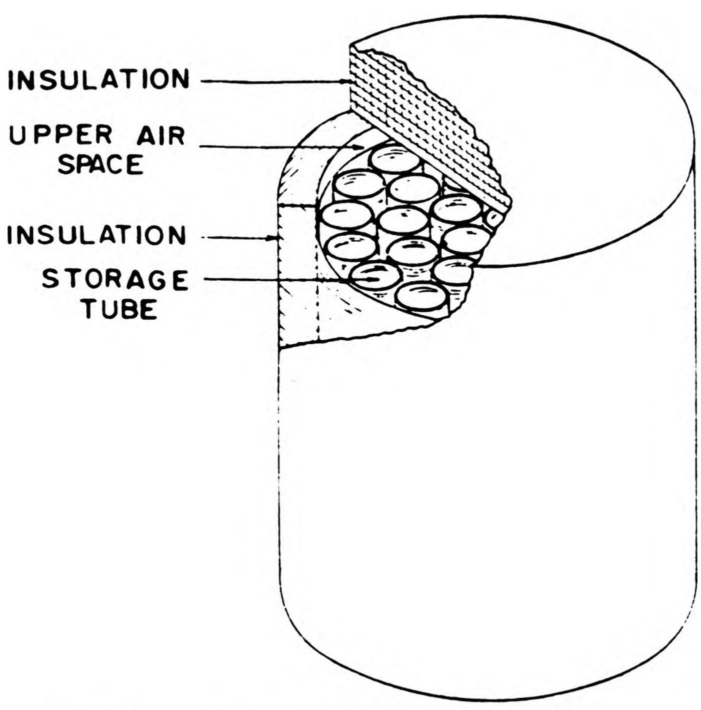

Fig. 2 Tubular Air Passage Core

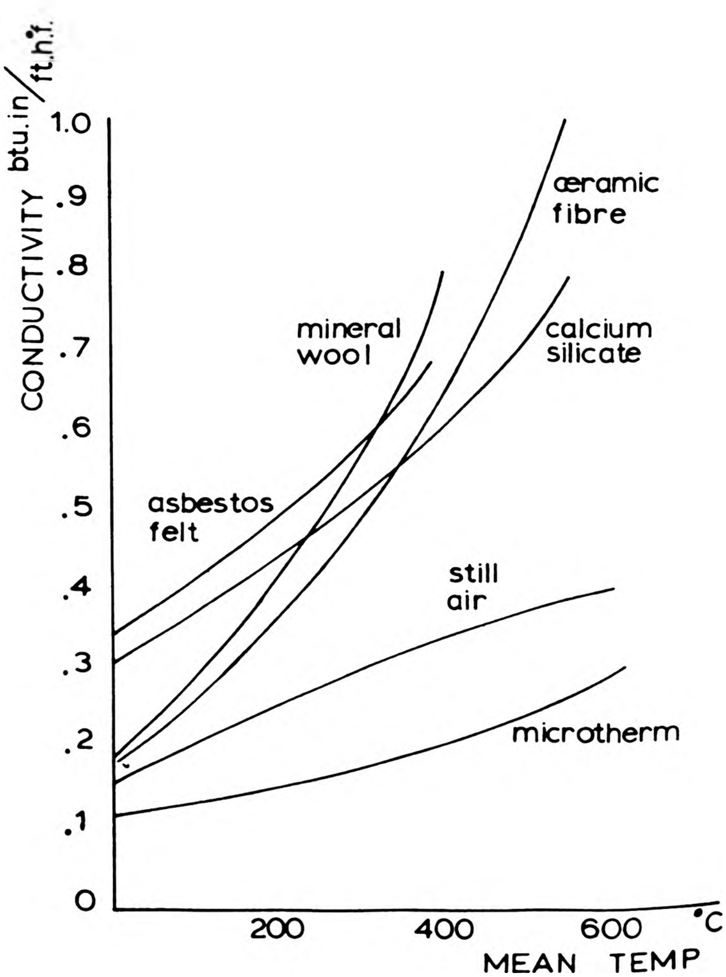

Fig. 4 Thermal Conductivities

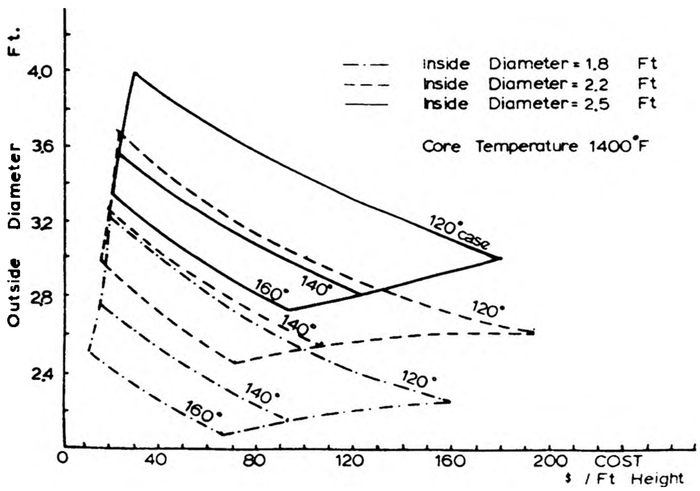

Fig. 5 Insulation Jacket Design

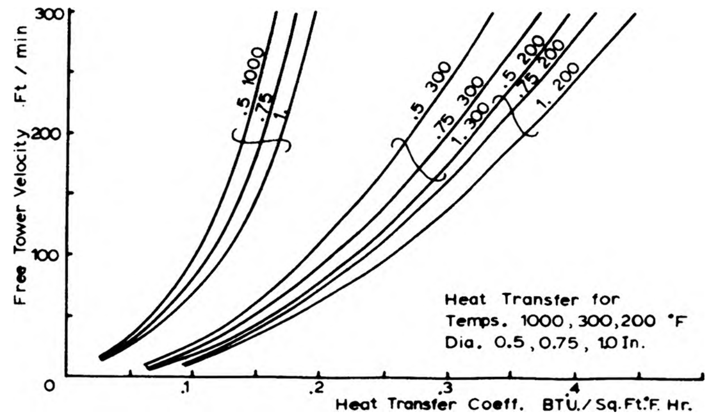

Fig. 7 Heat Transfer Rates

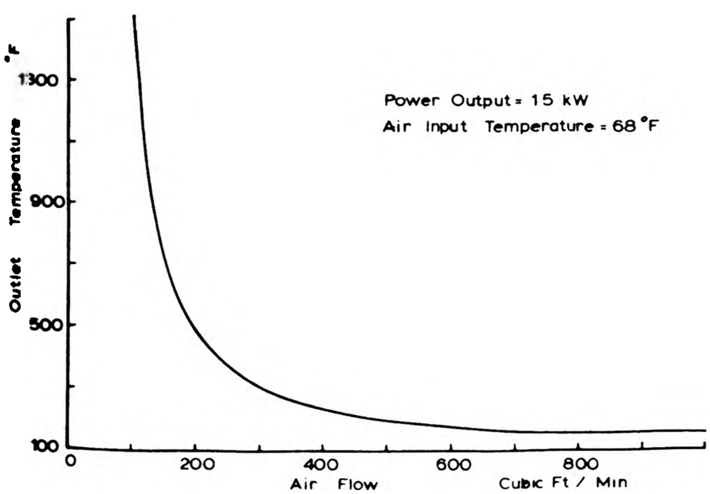

Fig. 6 Mass Flow Rates

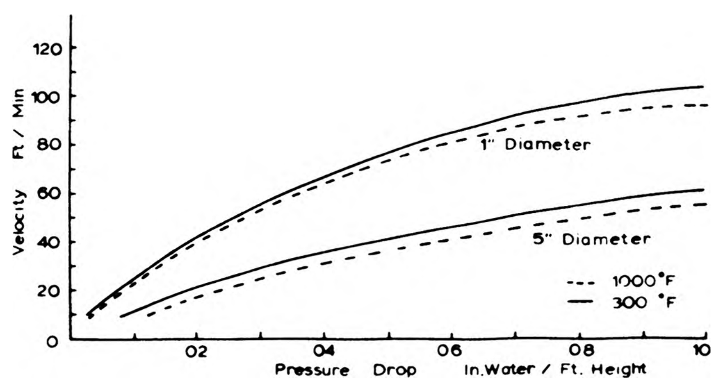

Fig. 8 Pressure Drops

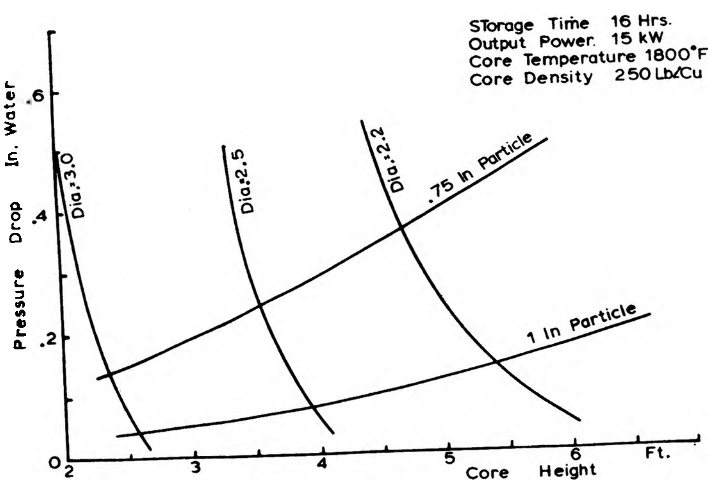

Fig. 9 Packed Bed Characteristics

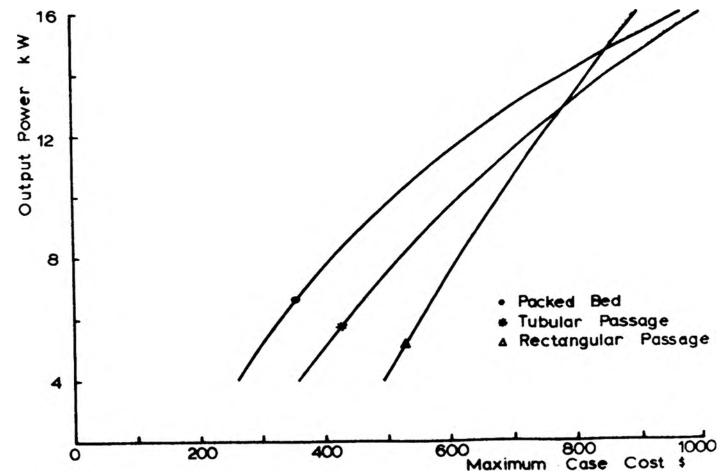

Fig. 10 Comparison of Cores

<u>Table I - Comparison of Units</u>

Power output: 20 kW

Dimensions in Feet

Time	Type	Core Base	Core Height	120°Case Base	120°Case Height	Max Cost	140° Case Base	140° Case Height	Max Cost
4	Tube-Tube	C*1.99	4.19	2.3	4.5	357	2.2	4.4	223
	Tube-Rect.	R*51 x 2.64	3.21	2.9 x 1.8	3.5	450	2.8 x 1.7	3.4	282
	Rect.-Rect.	S* 2.06	3.59	2.4	3.9	493	2.3	3.8	310
	Packed bed	C* 2.2	2.7	2.6	3.0	266	2.5	2.9	167
8	Tube-Tube	2.24	1.15	2.6	6.6	518	2.5	6.4	319
	Tube-Rect.	2.14 x 3.74	2.65	4.0 x 2.4	2.9	600	3.0 x 2.3	2.8	379
	Rect.-Rect.	2.24	4.25	2.5	4.6	614	2.4	4.5	387
	Packed bed	2.2	5.14	2.5	5.4	411	2.5	5.3	248
16	Tube-Tube	2.45	5.79	3.0	6.5	1000	2.8	6.3	719
	Tube-Rect.	2.71 x 2.71	5.01	3.0 x 3.0	5.3	864	2.9 x 2.9	5.2	548
	Rect.-Rect.	2.4	6.23	2.7	6.5	907	2.6	6.4	575
	Packed bed	2.5	5.68	3.1	6.0	978	2.9	5.9	650

* The base shapes are as follows:

C - Circular   R - Rectangular   S - Square