A COUPLED-LOOP THERMOSIPHON WITH AN INTEGRATED FREEZING PREVENTIVE FOR A PASSIVE COOLING SYSTEM

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ABSTRACT

Experimental and theoretical studies were performed on a natural convection system consisting of three interconnected thermosiphon loops. The experimental set-up was devised as a pilot plant for the emergency cooling system of storage tanks, containing highly radioactive waste. Neither mechanical or electrical energy input nor any human control was provided. In order to guarantee safe operation of the cooling system at low ambient air temperatures it has been supplied with a hydraulic self-control unit, acting as a passive freezing preventive.

INTRODUCTION

Investigations and applications of single-loop thermosiphons have been widely discussed in literature, especially respecting stability criteria [1,2,3,4]. The authors of this paper performed experimental and theoretical studies on a natural convection system consisting of three coupled thermosiphon loops (shown schematically in Fig.1). This system was devised for the emergency cooling of highly radioactive waste (HAW), dissolved in pentamolar HNO_3. Assuming failure of the operational cooling circuit, the thermosiphon system should carry heat from the source (HAW, contained in a storage tank) to the sink (ambient air) without mechanical or electrical energy input and without human control. When the main cooling system gets ineffective, the temperature of the HAW rises due to self-heating within 20 hours from its storage temperature of $T_C = 65^\circ$ C to the boiling temperature of 107° C and starts evaporating. The vapor travels up to a heat exchanger, is condensed there and flows back via a pipe into the storage tank (primary loop). The secondary loop, containing hexamolar HNO_3 convects the heat from the condenser to an air-cooler, mounted in a natural draft tower. Its main objective is to build up another barrier against

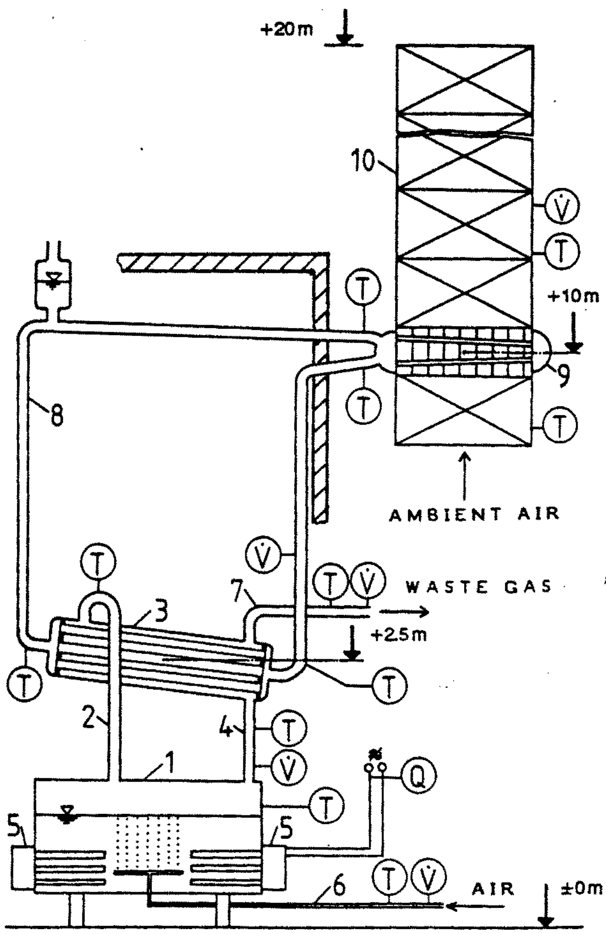


Fig.1: Test facility

carry-out of radioactivity in case of leakage. Thus a ternary loop, open to the environment and using ambient air as heat carrier, is activated. Buoyancy forces provide the only driving power for all three loops (Thermosiphon effect). In order to guarantee safe operation of the cooling system at ambient air temperatures down to $T_{A1} = -30^{\circ}$ C, a hydraulic self-control unit, acting as a freezing preventive was contrived and integrated into the secondary loop.

TEST FACILITY

The experimental set-up (see Fig.1) was devised as a pilot plant for the emergency cooling system of HAW-storage tanks. At the design point it should carry off a heat rate of 33 kW to the ambient. The primary loop comprises a storage tank (1), partly filled with 2000 litres of pentamolar HNO_3 under atmospheric pressure, a vapor pipe (2) to the condenser (3) and another pipe (4) for the condensate back flow. Electric heaters (5) with a maximum power input of 60 kW simulate the radioactive heat source of the HAW. Via pipe (6) air may be blown into the container, thus simulating the release of radiolytic gases; it leaves the primary loop through the exhaust gas pipe (7). The secondary loop (8) contains hexamolar HNO_3 (freezing temperature -42° C) which transports the heat to the air-cooler (9), mounted into a natural draft tower (10). Ambient air is sucked through the tower and conveys the heat to the atmosphere. This open flow system represents the ternary loop.

Instrumentation was provided for the automatic recording of all relevant parameters. Temperatures were measured by Pt-100 resistance thermometers (T), flow rates by magnetic inductive flow meters in the primary and secondary loop, by a resistance velocity meter in the ternary loop and by variable area flow meters in the ventilation pipe ($\dot{V}$). For further details reference [5] may be drawn up.

FREEZING PREVENTIVE

When pentamolar HNO_3 evaporates at 107° C the concentration of the vapor is only that of 0.5 molar nitric acid, which has a freezing point of about -2° C. Under winter conditions with temperatures far below 0° C and reduced heat rate (partial filling of the storage tank) the secondary loop will be cooled down below 0° C too and freezing of the weak acid condensate around the condenser tubes may occur. Thus obstruction of the primary loop and failure of the whole cooling system is to be suspected.

In order to cope with this situation, an anti-freezing system was contrived, that could be integrated easily into the secondary loop, a short distance off the air-cooler (not shown in Fig.1, see Fig.2). A fixed weir (1) is mounted into the outlet tube of the air-cooler, the trailing-edge of which ranges some centimetres higher than the inlet tube. Its main objective is to act as a fixed throttling or interrupting element to the flow of the coolant through the secondary loop in response to a reduction in the volume - or the mean coolant temperature, respectively. In order to set a safe cut-off temperature of e.g. $+5^{\circ}$ C, the secondary loop must be filled with hexamolar HNO_3 at this temperature up to the trailing-edge of weir (1), when the whole system is non-active.

Two accessory means were contrived to ensure proper operation of the

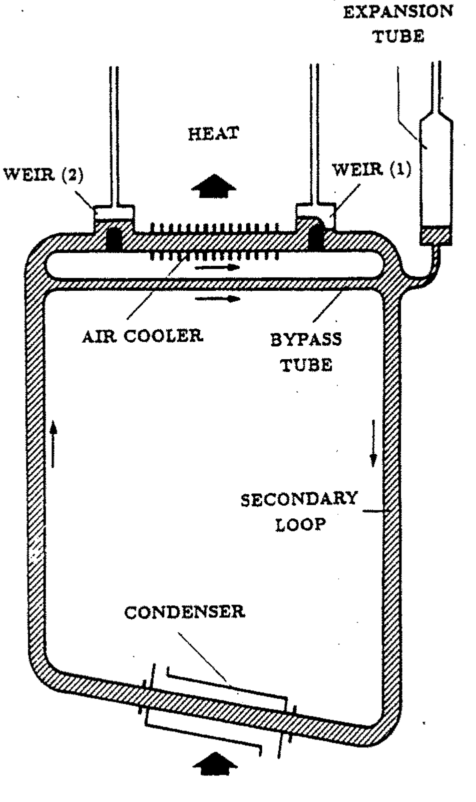


Fig.2: Freezing preventive

freezing preventive: (1) A by-pass pipe of high flow resistance reduces the temperature differences between rising and falling main due to the mixing effect of the lower sub-loop, which in addition serves as a welcome starting circuit, when cut-off condition exists at weir (1). (2) A second weir (2), mounted into the inlet tube of the air-cooler (its trailing-edge at the same level as the first weir), acts as a check valve, when there is no overflow at weir (1). The whole freezing preventive has been patented [6].

SCALING OF THE TEST FACILITY

As already mentioned, the pilot plant should carry off a heat rate of 33 kW, which is 10% of the operational plant's heat rate. To meet the requirements of the energy equation, it was only necessary to match the two heat exchangers (the condenser and the air-cooler) by reducing proportionally the number of tubes and the free flow areas in the condenser and in the draft tower of the pilot plant. In the storage tank the ratio of fluid surface area to heat rate was kept constant to provide equivalent conditions for the carry-out of aerosols. Concerning the geodetic heads in the primary loop no scaling requirements had to be considered because of the very high density difference between liquid and vapour phase which offers abundant driving force.
In contrast with this, scaling of the secondary loop, which couples source and sink loop, had to be done painstakingly. Due the limited height of the experimental building the geodetic head had to be reduced by 50% (7.5 m instead of 15 m in the operational plant). The momentum equation, formulated for the natural convection flow of the secondary loop fluid, equates driving force (wielded by the density difference in the rising and falling mains) with flow resistance due to acceleration and friction.

$$g\rho\oint[\beta(T-T_0)-1]\,dz=\rho\oint\frac{1}{A(s)}\,ds\,\frac{d\dot V}{dt}+ +\frac{\rho}{2}\left(\sum_i\frac{L_i}{D_i A_i^2}\xi_i+\sum_j\frac{K_j}{A_j^2}\right)\dot V^2$$

Loop properties:

z	vertical coord.	L	tube length
s	path coordinate	ξ	fric. coefficient
D	diam. of mains	K	obstacle resist.
A	cross-section (general)	i,j	summation variab.

Fluid properties:

ρ	density,	β	expansion coeffic.
T	temperature,	$\dot V$	flow rate
T_o	ref. temp.	g	gravitation const.

After transformation of this equation into dimensionless form, two paramount characteristic parameters show up which must have the same numerical values for both plants. Extension of the horizontal tube length and proper dimensioning of the tube diameter were the two means to meet these requirements. Whilst the first parameter (appearing as a factor at the acceleration term) could be matched by providing sufficient fluid mass in the horizontal tubes, the second parameter (the factor at the resistance term) was adjusted by choosing an appropriate diameter for the mains (see [5] for more details).

The natural draft tower - being mounted to the outer wall of the experimental hall - was provided with the same height (10 m above air-cooler) as that of the operational unit. Thus atmospheric perturbations could be expected to have the same influence on the sensitive thermosiphon characteristic of the ternary loop.

EXPERIMENTAL RESULTS

The test facility had to prove its reliability especially under starting conditions as well at high (+40° C) as at low (-30° C) ambient air temperatures. We shall, therefore, mainly present results of take-off runs. With exception of the first test (1) all the following ones were conducted with HNO_3 in the secondary loop. When sufficient tests with HNO_3 likewise in the primary loop had proven, that one could use boiling water in the primary loop as well - and this conservatively, due to its unfavourably lower saturation temperature - HNO_3 was replaced by the less dangerous substance water. Fig.3 shows in schematic form the location of flow meters ($\dot V$) and temperature probes (T) within the test facility.

(1) Starting operation at high ambient air temperatures

When the test facility was just assembled - being not yet equipped with the anti-freezing system - rather high summer temperatures of up to 36° C prevailed. At that time the primary and the secondary loops were filled with water instead of nitric acid for preliminary testing. Due to the lower evaporation temperature of water (100° C instead of 107° C) the driving temperature differences to the ambient air were thus reduced in both loops. As the thermal properties of water and HNO_3 compare well with respect to their total impact on free convection, the results of the "water-tests" may be regarded as conservative. Heat rate carried to the ambient was 47.5 kW (44% over-load). Figs. 4a and 4b show flow rates and system temperatures during take-off operation. When the container temperature has reached 95° C, circulation starts already because 600 l/h of moist air, blown through the container fluid, initiate condensation. Flow rates and temperatures adopt steady values roughly one hour after the primary fluid has reached its boiling point. Especially the flow rate $\dot{V}_S$ in the secondary loop exhibits a wavy behaviour during this take-off period, which is caused by the hot plug of fluid originally resting in the air-cooler and subsequently being shifted around with a circulation time of about 8 minutes. Neither self-induced oscillations nor any temperature overshoots are observed and all the steam is condensed. Thus the cooling-system has shown its reliability under summer conditions with a high safety margin.

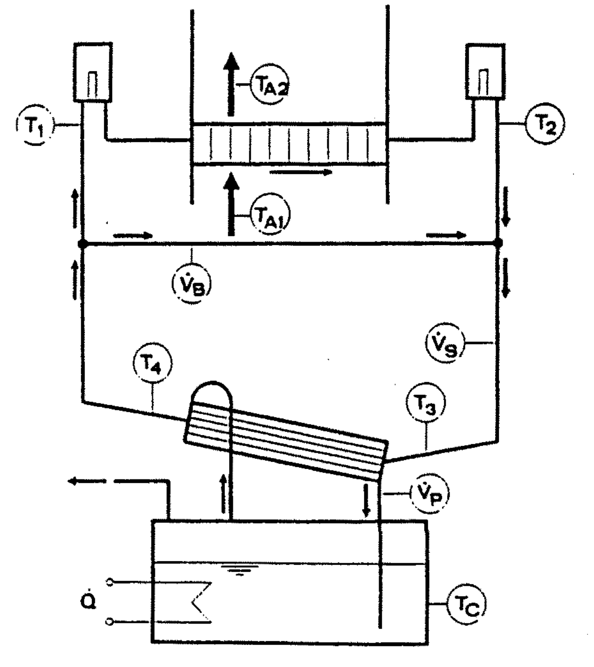


Fig.3: Location of flow meters and temperature probes

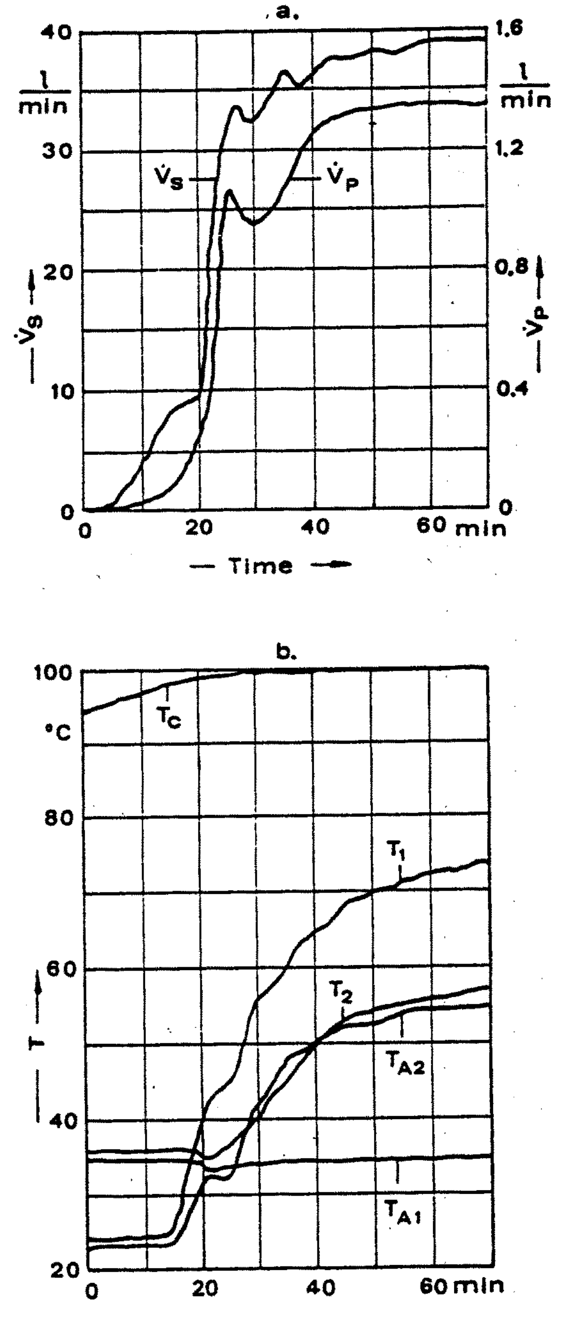


Fig.4: Starting operation at high ambient air temperatures.
Heat rate: $\dot{Q}$ = 47,5 kW (44% overload)

a): Flow rates in the primary ($\dot{V}_P$) and secondary loop ($\dot{V}_S$)

b): Temperatures in the three loops.

(2) Starting operation at low ambient air temperatures

On a cold day in January with ambient air temperatures of about - 16° C the freezing preventive could be tested. During these experiments the primary loop contained water and the secondary loop hexamolar HNO_3. Cut-off temperature was set at 5° C and heat rate at $\dot{Q}$ = 21 kW. Fig.5a shows total flow rate $\dot{V}_S$ and by-pass flow rate $\dot{V}_B$ of the secondary fluid. The difference between both is equal to the flow rate through the air-cooler. Obviously weir overflow and thus activation of the air-cooler starts at t = 30 min. on the time scale. Fig.5b displays pertinent temperatures of all three loops. Our special interest is concerned with the temperature T_3 in the falling main behind the by-pass junction or at the condenser inlet, respectively. This temperature rises to a peak of 11° C at t = 37 min. and then comes down rapidly to the prescribed value of 5° C. It is weir (1), assisted by the by-pass line, that throttles the overflow such as to keep the condenser inlet temperature constant, though the whole cooling systems has not yet met its steady state. Many test runs, performed with different heat rates, cut-off temperatures and ambient air temperatures, invariably revealed proper operation of the freezing preventive.

Benchmark tests:

(3) Uper limit of ambient air temperature set at T_{A1} = 40°

The condenser area was purposely oversized to enable higher heat rates to be imposed on the system (up to $\dot{Q}$ = 60 kW) instead of 33 kW nominally). Neglecting subcooling of the condensate, Fig.6a gives a rough picture of the transfer characteristics of the secondary loop during interaction with the oversized condenser (water, boiling at T_C=100° C in the container): If the ambient temperature T_{A1} rises at constant heat rate $\dot{Q}$, the temperature distribution of the secondary fluid is shifted upward. As long as there is a surplus in condenser area, the temperature difference $\Delta T = T_4 - T_{A1}$ between condenser outlet and ambient remains nearly constant. We now vary heat rate $\dot{Q}$ at a constant ambient temperature of $T_{A1} \approx$ 28° C up to $\dot{Q}$ =60 kW. The experimental results shown in Fig.6b reveal that at the nominal heat rate of $\dot{Q}$ =33 kW there is a $\Delta T \approx$ 30K and at $\dot{Q}$ =60 kW there is a $\Delta T \approx$ 46 K. With some more profound reasoning than outlined here we may thus argue, that our thermosiphon system will work properly up to T_{A1}=(28+16)K=44K.

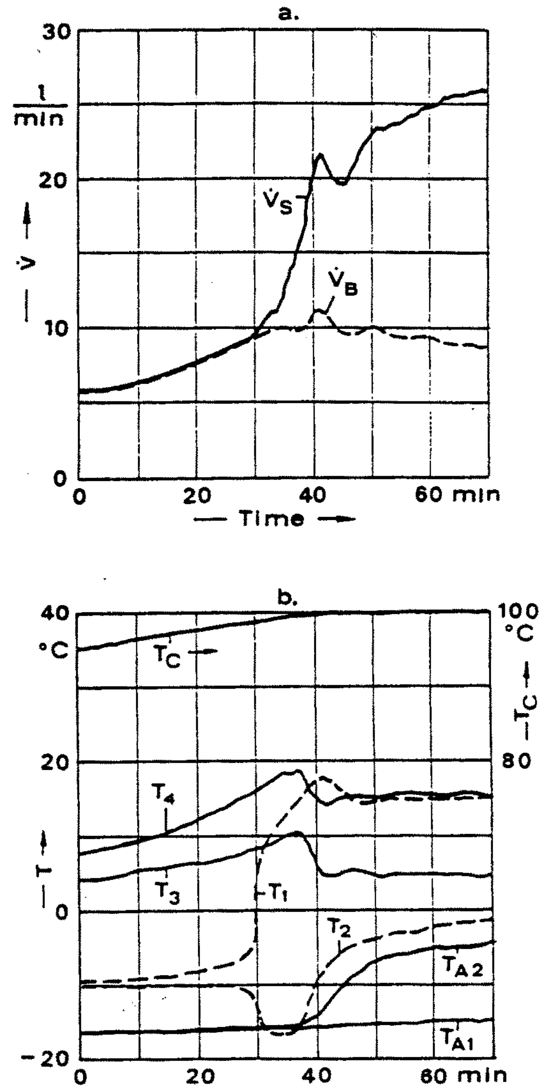
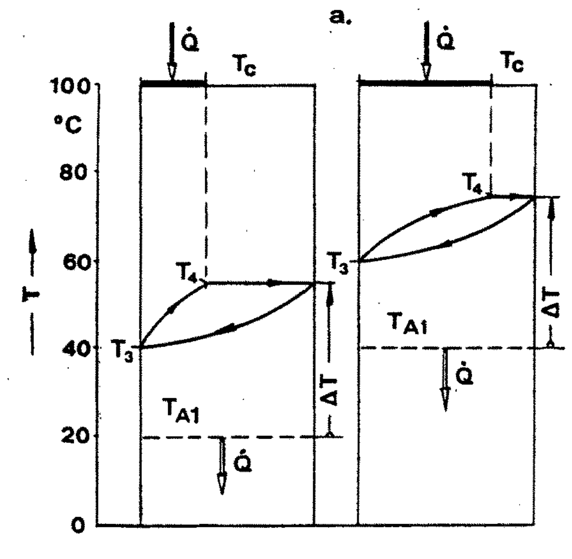


Fig.5: Starting operation at low ambient air temperatures.
Cut-off temperature: + 5°C.
Heat rate: $\dot{Q}$ = 21kW (64% load)

a): Flow rates in the secondary loop.
$\dot{V}_S$: total flow rate
$\dot{V}_B$: bypass flow rate

b): Temperatures in the three loops



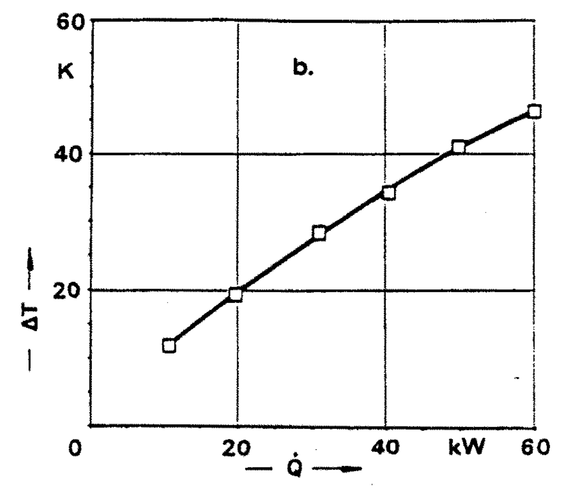


Fig.6a: Transfer characteristics of the secondary loop during interaction with an oversized condenser at different ambient air temperatures T_{A1} (parallel flow for ease of demonstration)

Fig.6b: Temperature difference ΔT between condenser outlet (T_4) and ambient (T_{A1}) versus heat rate

(4) Lower limit of ambient air temperature set at T_{A1}=-30° C

In order to proof that the system works properly down to T_{A1}-30° C, a cooling unit capable of carrying off 90 % ($\dot{Q}$=30 kW) of the full heat load at a temperature of -30° C, was installed.

At the very high cut-off temperature of 21° C a test in starting operation (Fig.7) proved the reliability of the emergency cooling system. The observed rise of the cut-off temperature from 21 to 24° C may obviously be attributed to the rather inhomogeneous temperature distribution in the secondary loop. This shift (to the conservative side) makes up for the higher density in the tube section between weir (1) and bypass inlet (see temperature T_2!)

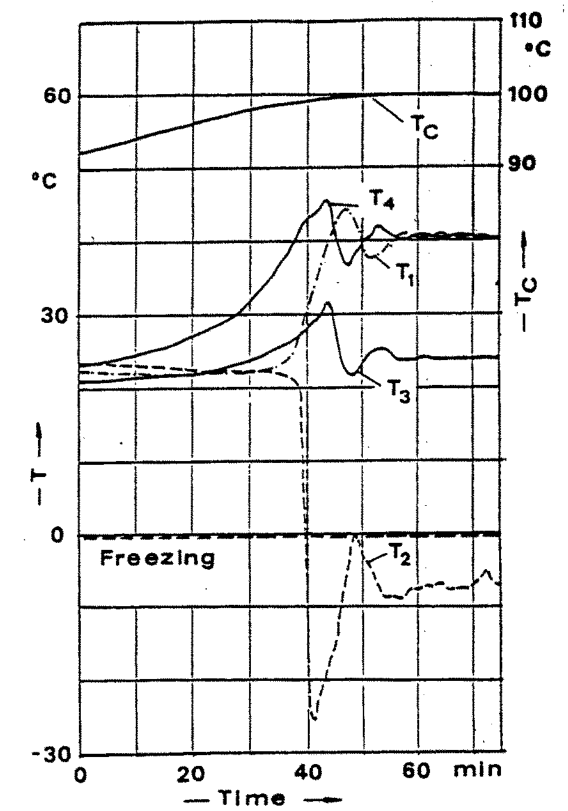


Fig.7: Temperatures during starting operation with an auxiliary cooling unit.
Cooling-rate : $\dot{Q}$=30 kW
Coolant temperature: T_{A1}=-30° C
Cut-off temperature: 21° C

(5) Stability tests

Instead of trying to find theoretical stability criteria for our complex thermosiphon system we imposed strong sinosidal oscillations of the heat rate $\dot{Q}$ on it with period times t_p ranging from 180 seconds to one hour. An amplification ratio R was defined by the relation

$$R = (\Delta\dot{V}_s/\dot{V}_m)/(\Delta\dot{Q}/\dot{Q}_m)$$

$\dot{V}_s$ and $\dot{Q}$ characterise flow rate and heat rate in the secondary loop. $\Delta\dot{V}_s$ and $\Delta\dot{Q}$ are amplitudes and $\dot{V}_m$ and $\dot{Q}_m$ are absolute values (measured or imposed) at the mean values around which the oscillations develop. The phase lag Θ is defined by

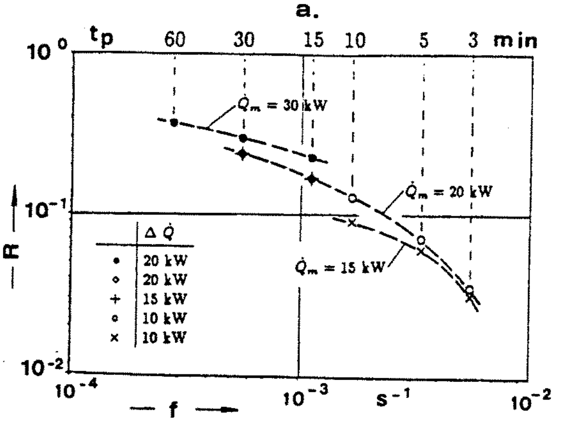
$$\Theta = -360 \text{ deg. } \Delta t/t_p$$

and the frequency by

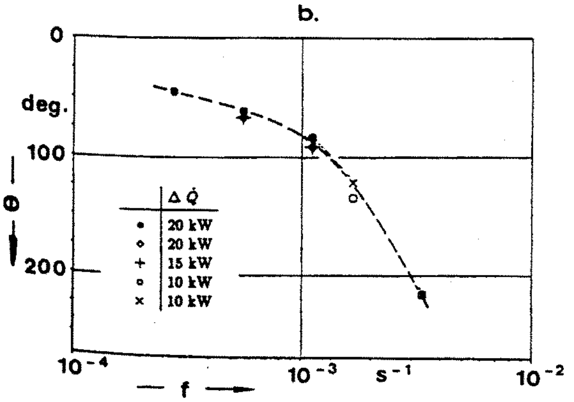
$$f = 1/t_p.$$

Fig.8a and Fig.8b indicate, that no self-induced oscillations appeared; the system revealed mere aperiodic behaviour.

a.



b.



ig.8a: Amplification ratio R versus frequency f

g.8b: Phase lag Θ versus frequency f

CONCLUSIONS

Investigation of the passive cooling system has demonstrated its excellent stability within a wide range of heating power (up to 80 % overload) and ambient air temperatures (-30° C to +40° C). Squally winds had no influence on the stability of the system because the circulation time of the secondary fluid is 6 to 9 minutes. Oscillations and flow reversal, as reported by other authors [2,3,4], were not encountered with our system. Freezing of the condenser on the primary side at low ambient air temperatures could be prevented by meanas of an auxiliary passive system which keeps the temperature of the fluid in the secondary loop above a prescribed level, independent of ambient air temperature and strength of heat source. The safe operation of the system was also demonstrated in several long-time runs up to 1000 hours. Consequently this passive cooling system may be regarded as a very reliable apparatus for the emergency or even operational cooling of radioactive solutions and other non-extinguishable heat sources.

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