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Air cooling of concrete by means of embedded cooling pipes - Part I: Laboratory tests of heat transfer coefficients

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ABSTRACT

Embedded cooling pipes can be used to reduce the temperature rise in massive structures as a measure against thermal cracking. When air is used as a cooling medium, relatively large diameters with profiles causing friction losses along the pipe are preferred. In this paper, heat transfer coefficients for two different types of cooling pipes have been determined for different pipe flows in combination with various temperature levels. This paper relates to the first part of the investigation dealing with the laboratory tests of heat transfer coefficients. The second part, dealing with application in design, is presented in "Air cooling of concrete by means of embedded cooling pipes - Part II: Applications in design" [1].

RÉSUMÉ

Afin de prévenir la fissuration thermique, des tuyaux de refroidissement incorporés peuvent servir à réduire la montée de la température dans des constructions massives. Si l'on utilise l'air en tant que moyen de refroidissement, il est préférable d'incorporer des tuyaux de diamètre relativement large dont les profils causent des pertes par friction. Dans cet article, on détermine les coefficients de transfert thermique pour deux types de tuyaux, pour des flux différents dans les tuyaux combinés avec des niveaux différents de température. Cet article décrit la première partie d'une étude expérimentale et traite des essais en laboratoire des coefficients de transfert thermique. La deuxième partie, qui traite de l'application des résultats dans la préparation des projets, sera présentée en « Air cooling of concrete by means of embedded cooling pipes - Part II: Applications in design » [1].

1. INTRODUCTION

Generally, the temperature rise in a concrete structure due to the hydration process and thermal flow causes temperature differences which, due to restraint conditions present in the structure, produce thermal stresses [2-5]. During the first hours after casting, the mean temperature ascends. Larger thermal expansion in the core, compared to the surface, generates tensile stresses in the surface layer. Afterwards, as the mean temperature decreases the stresses are reversed so that the surfaces are under compression. This is valid for the cases where surface cracks are of main concern.

There are, however, many applications where the risk of through cracking is of greater importance. Through cracks develop as a result of restraint between the hardening concrete structure and the foundation and/or previously cast adjoining structures. Through cracking may also occur if the mean temperatures rising in different parts within the cast structure differ considerably.

In both cases of early age cracking referred to above, the thermal movement due to hydration is the key factor in the development of cracks in concrete structures.

At the Luleå University of Technology, studies of thermal and mechanical properties of concrete in early ages have been carried out during the last decades. Methods for the evaluation of material properties of concrete have been established and tools for numerical analysis of temperature, moisture and temperature stresses have been developed [2, 6]. Research is currently aiming at establishing material properties of new concretes such as high performance concrete (HPC) [7-9], energetically modified concrete (EMC) [10], concretes of various three powder mixes, etc. These properties are then used in theoretical modelling. The modelling is focused on combined effects of moisture and temperature changes on stresses in early age concrete by means of calculations in 2D and 3D models.

This paper deals in particular with laboratory tests related to a new method of reducing the risk for thermal cracking in concrete structures by air cooling. Pipes are placed in the structure and air is circulated through them during the first few days after casting in order to reduce the temperature differences between core and surface. Thus, the risk of surface cracking is prevented. The method has many advantages over the traditional use of water as a cooling medium, particularly in tall, prismatic structures such as columns and thick walls.

Previously, a number of temperature recordings were made on the building sites of the Tjörn Bridge (1980) and of the Igelsta bridge (1992 and 1993) in Sweden, where this air-cooling technique was used [12-15]. The results from the measurements have been used for comparison with calculations of the temperature development in the air-cooled sections and a good agreement has been achieved [1, 16].

In order to get reliable data to carry out the calculations, heat transfer coefficients of the cooling pipes were measured in experiments. Two types of pipes have been studied in the laboratory tests, and one of them was used at the Igelsta bridge as a cooling pipe. This paper describes the experimental part of this investigation and the evaluation of the heat transfer coefficients.

To make full use of the advantages of a cooling system, prediction of the effects concerning temperature and temperature related stresses have to be examined carefully for each specific case. Designing an air cooling system to reduce the temperature rise in massive concrete structure raises two important questions:

– Will the demands concerning temperature and stresses be met?

– What arrangement of cooling pipes, fans, etc. must be implemented in order to ensure the necessary cooling effect?

The required cooling effect can be expressed as the air flow rate in the system of cooling pipes. Design of fan capacity and other equipment connected with the cooling system follows current methods of ventilation engineering and mass flow in pipes.

When studying a concrete structure from the point of view of early age cracking, with or without embedded cooling pipes, the main issue is, of course, whether or not cracks will actually occur. Thus, it is necessary to relate developed stresses to the concrete strength. This is further discussed in [2, 11, 17]. Besides appropriate knowledge about mechanical properties of the concrete, the temperature development in critical cross sections has to be calculated to enable proper thermal stress analysis. Simulation of the temperature development in the structure (see [6]) has to consider the thermal properties of the concrete and of the form-work used. A complete analysis of the temperature flow in a general three dimensional non-stationary case of a newly cast structure, *e.g.* a bridge column, will soon becomes rather complicated. It is therefore necessary to simplify important parameters related to the geometry of the studied structure as well as the variation in time of the surrounding temperature, etc.

2. MEASUREMENTS AND METHODS

Designers and contractors can choose between many different types and dimensions of cooling pipes for their cooling systems. In the experiments presented in this paper, two different types of pipe were used:

1. A thin metal pipe normally used for ventilation systems in buildings. This ventilation pipe (Spiro type) was used in the columns of the Igelsta bridge in Södertälje, Sweden.
2. Sheet metal ducts normally used for post-tensioned cables, here called strand tubes. There is a vast range of tube dimensions available for post-tensioning purposes on the market. In the laboratory tests, the most frequently used dimensions in Sweden were chosen. An advantage of strand tubes over the Spiro type is that they can resist higher hydraulic pressure during the pouring operation. This is due to the shape of the strand tube, see Fig. 1.

The laboratory tests were performed to determine the heat transfer coefficient of the two types of cooling pipes mentioned above. At the time of the tests, the moisture conditions in terms of relative humidity, *RH*, in the laboratory was 60 ± 4%. Knowledge about the moisture condition of a cooling medium with relatively low heat capacity, such as air, is essential, as the moisture content in air strongly affects the capacity to store and transport the energy. The specific heat capacity of dry ambient air (c_p) is 1.00 kJ/kg K, which is small compared to water with a specific heat capacity of 4.18 kJ/kg K. Thus, the ambient air can be assumed to have higher or lower heat transfer capacity depending on the moisture content. The variation of heat transfer coefficients at different humidity levels has not been studied here.

2.1 Experimental equipment

When determining the heat transfer coefficient for cooling pipes, careful measurements of the velocity and temperature profiles have to be made. The two types of pipes studied are greatly different in geometry and shape of inner surfaces. The ventilation pipe has a smooth inside, while the strand tube has a more distinct surface profile, see Fig. 1.

Just by considering this difference in geometry, it is possible to guess that the pipe with the larger perimeter

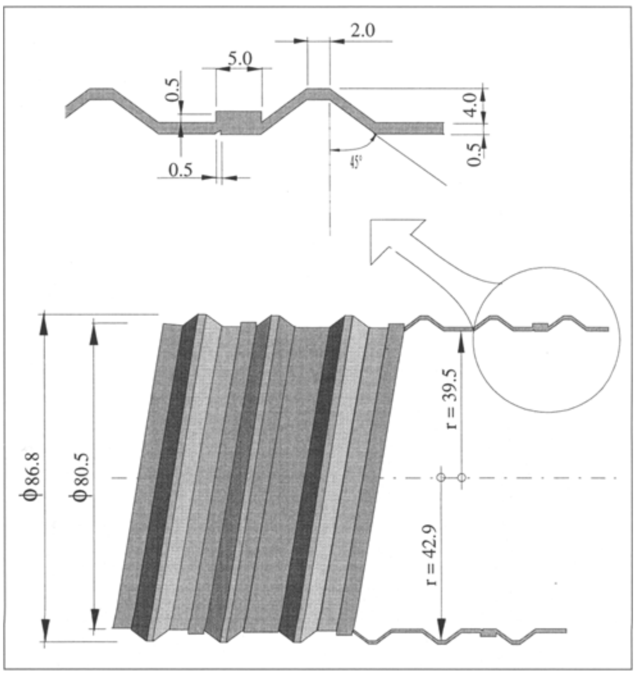

Fig. 1 – Studied strand tube and its dimensions (measures in mm).

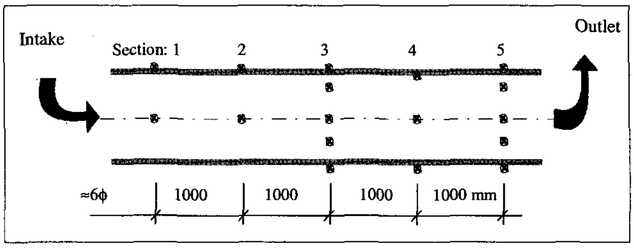

Fig. 2 – Sketch of position for temperature recording inside the cooling pipe.

in contact with its surroundings would have the higher heat transfer capacity, but a rough surface profile will generate higher friction and therefore a more irregular velocity and temperature profile.

2.2 Experimental set-up

Controlled conditions are necessary. The case with constant surrounding temperature can be described analytically which therefore makes it possible to determine the heat transfer coefficient from measured temperature differences between inlet and outlet of the control volumes according to equation (1), see Fig. 2. Hence, the pipe was placed in a large basin filled with hot water. The water basin provided the possibility to create sufficiently large temperature differences along the pipe. The large volume of the basin was suitable to maintaining a stable and constant temperature of the surroundings during the tests.

The cooling pipe was placed and clamped to the bottom of the basin with the volume of 16.5 m^3. The test set-up is shown in Fig. 3. The strand tube had a mean diameter of 90 mm and the Spiro pipe a mean diameter of 125 mm. As mentioned previously, the tests were performed under stationary conditions with regard to the surrounding temperature conditions. Temperature profile in the water was measured for each temperature level.

The temperature gauges were placed along the pipe at a distance of approximately 1.0 m, see Fig. 2. When positioning the temperature gauges and the mounting wire inside the cooling pipe, it is very important to use gauges as slender as possible. In this way, unwanted friction losses are avoided. This can be checked by simultaneous measurements of the static pressure along parts of the pipe where there is no interference from mounted equipment. If the static pressure loss differs too much between the two studied sections, the result is not characteristic for the pipe and should therefore be rejected, or the test should be repeated after the temperature recording devices have been adjusted. In the tests, no evidence of such pressure loss were found in any of the measurements of the static pressure.

While the basin was filled with hot water, the pump circulated the water to mix it properly. When the water level was high enough above the cooling pipe, the water supply was turned off while the pump was left on to circulate. Temperatures were recorded by the PC-logger at points shown in Fig. 2 as well as at the intake, the outlet and the basin itself. The temperatures were recorded at four to five different air velocities for both types of cooling pipes. The flow span was 3 - 7 m/s for the strand tube and 2 - 12 m/s for the ventilation pipe. The tests were performed at temperature levels between 25 - 35 °C of the surrounding water.

Measurements of the air flow were made by a handheld micro-manometer at the outlet of the cooling system. Before each test was started, the temperature in the basin had been checked and the fan was adjusted to the desired flow. Thereafter the fan was left on for some minutes, in order to develop a steady flow before new measurements of the velocity and temperature were made.

The risk of variations in the temperature and the flow in the cooling pipe due to density differences in the cooling-air must also be considered. Differences could occur if the surrounding temperature distribution of the

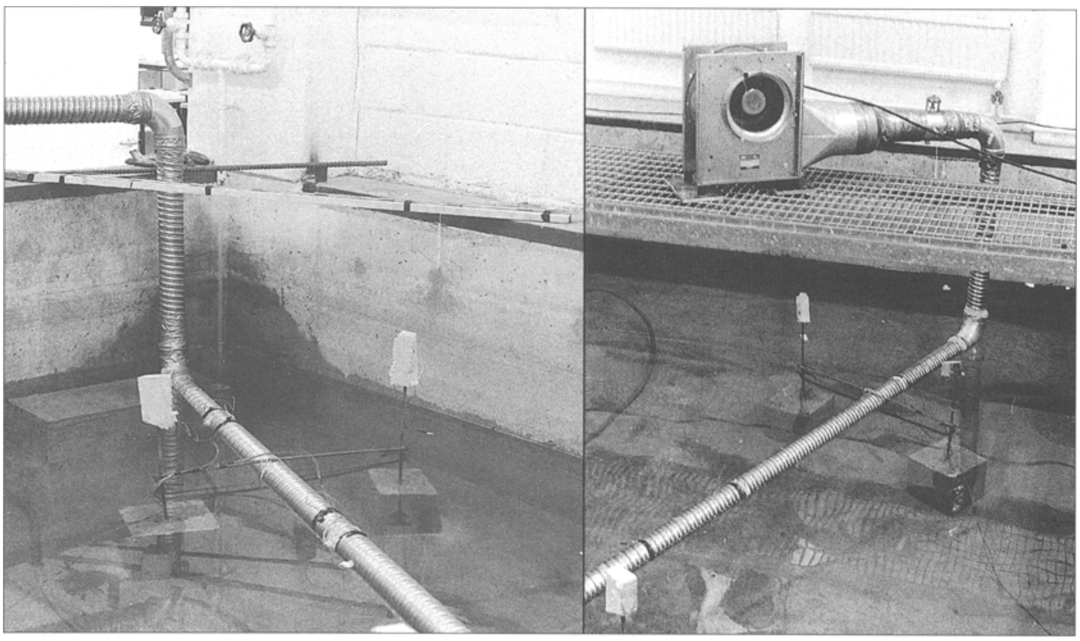

Fig. 3 – Basin with mounted cooling pipe (strand tube).

water in these kinds of tests is not homogeneous. Therefore, the temperature-gauges should be mounted cross-wise in each section. In the performed tests the temperature recording devices were mounted in only one direction – horizontally or vertically – in order not to introduce unnecessary disturbances in the flow. No inhomogeneities were measured in the temperature field in any of the studied cases.

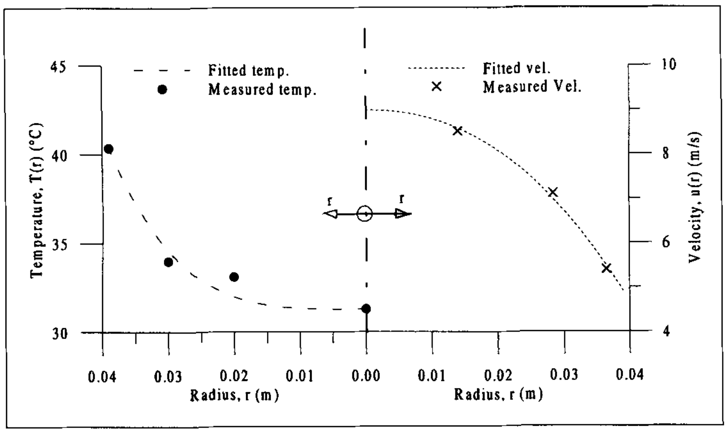

Fig. 4 – Some examples of the evaluated temperature and velocity profiles for the studied strand tube.

3. EXPERIMENTAL RESULTS AND EVALUATION

After the temperature recordings for one temperature level and several different flows had been performed, the water temperature was lowered in the basin. The water was circulated by the pump for approximately one to two hours until a new temperature profile was measured in the basin.

3.1 Evaluation of experiments

The tests on the cooling pipes were performed at three levels of water temperature for each type of cooling pipe and for four and five different flows, respectively. Afterwards, the basin was emptied and measurements of the static pressure along the cooling pipe were carried out.

Since the ambient temperature, T_a, round the pipes is kept constant and since the pipe-wall temperature, T_w, can be approximated with T_a, the following expression may be used according to classic theory on internal flow (see [18]) where $T_w \approx T_a$;

$$\bar{h} = -\ln\left(\frac{T_w - T_{m,o}}{T_w T_{m,i}}\right) \cdot \frac{\dot{m} \cdot c_p}{\Omega \cdot L} \tag{1}$$

The rate of mass flow, $\dot{m}$, in equation (1) is calculated as

$$\dot{m} = \rho \cdot u_m \cdot A \tag{2}$$

The bulk temperatures in equation (1) are calculated as

$$T_m(x) = \frac{2}{u_m R^2} \int_0^R u(r,x) \cdot T(r,x) \cdot r \cdot dr \tag{3}$$

Using equations (1) - (3), applied on the measured values of temperature and velocity profiles, it is possible to derive the heat transfer coefficient, $\bar{h}$. For the calculation of $\bar{h}$, the measured temperatures and velocities were fitted to equations chosen as

$$u(r) = \psi \cdot R^2 \cdot \left(1 - \left(\frac{r}{R}\right)^{\xi}\right) \tag{4}$$

$$T(r) = \theta_1 \cdot \left(\frac{r}{R}\right)^{\varphi} + \theta_2 \tag{5}$$

The measurements of temperatures and velocities were carried out at different flows through the pipe and with

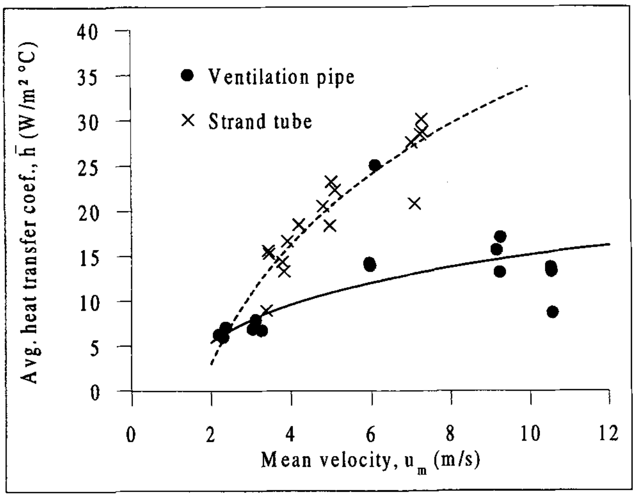

Fig. 5 – Average heat transfer coefficients, $\bar{h}$, as a function of mean velocity u_m. Evaluation from measurements of two types of cooling pipes.

different basin water temperatures. The evaluated results of temperature and velocity profiles are presented for one measurement in Fig. 4. For each studied case of ambient temperature and mean velocity, the average heat transfer coefficient, $\bar{h}$, was calculated according to equation (1) for the two types of pipes tested. The value of $\bar{h}$ varies with the mean velocity, u_m, according to Fig. 5. No significant variation with ambient temperature was found.

Another more common way to present heat transfer properties of pipes is by the use of dimensionless numbers. It can be shown that, in the case of smooth circular tubes, the Nusselt's number, Nu, must be a function of the type (see [16])

$$Nu = f(Re, Pr) \tag{6}$$

The function in equation (6) is determined experimentally and is usually expressed in the form

$$Nu = a \cdot Re^b \cdot Pr^c \tag{7}$$

For comparison, in the case of forced convection in smooth circular tubes and $T_w < T_m$, the following set of parameters are commonly used; $a = 0.0023$, $b = 0.80$ and $c = 0.30$.

Table 1 - Parameters according to equation (7)			
	a	b	c
Ventilation pipe	0.1755	0.5439	0.33
Strand tube	0.0070	0.9070	0.33

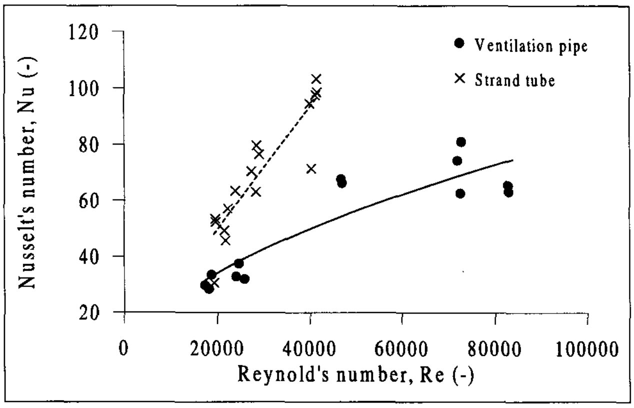

Fig. 6 – Relation between *Nu* and *Re* for the tested pipes.

In the present study, parameters were obtained for the two types of pipes by fitting the formula (7) to the experimental results, see Table 1 and Fig. 6. The results in Fig. 6 are presented in dimensionless Nusselt's number as a function of Reynold's number for tested pipes.

Theoretically, the use of non-dimensional numbers implies that the curves are independent of the pipe diameter. However, it must be noted that only one dimension of each pipe has been tested so that the independence of the pipe diameter has not been completely verified.

Clearly, the strand pipe offers greater heat transfer than the ventilation pipe for a given *Re*, or mean velocity, u_m. The greater efficiency of these pipes is probably due to their irregular shape, see Fig. 1, which entails higher turbulence and also presents a higher efficient surface area to the ambient medium. However, the shape of the pipe generates higher friction losses, which in turn means that for a given fan capacity the mass flow will be lower in the strand pipe than in the ventilation pipe. The obtained mass flow, or mean velocity, u_m, for a given configuration of pipes and fan capacity can be calculated by standard techniques when the mean friction effect is known for the pipe used.

4. DISCUSSION AND CONCLUSIONS

The experiments were easy to perform with sufficient accuracy with respect to the recorded temperatures. They can be carried out without using sophisticated equipment and are therefore relatively inexpensive. The only requirements to observe are to achieve stable and stationary conditions during the tests.

These experiments encourage performance of more tests using different diameters and new types of tubes other than those studied here. Comparative calculations based on experimental results evaluated and presented here are in good agreement with *in situ* measurements of temperatures, see [1]. Since all other factors were well known in the comparison, the only outstanding parameter influencing the results was the heat transfer coefficient. This indirect justification means that the heat transfer coefficients were correctly evaluated in this paper, and they can probably be used in prediction of *in situ* castings with high accuracy.

In conclusion, this paper describes an easy-to-use test method designed to establish the relationship between the heat transfer coefficient and the flow in cooling pipes.

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NOTATION

T_a	ambient temperature, (°C)
T_w	pipe-wall temperature, (°C)
$T_{m,i}$, $T_{m,o}$	in- and outlet bulk temperature of the fluid, (°C)
Ω	perimeter of the cooling pipe, (m)
L	distance between in- and outlet, (m)
$\dot{m}$	rate of mass flow through the pipe, (kg/s)
c_p	specific heat, (J/kg °C)
$\bar{h}$	average convection heat transfer coefficient, (W/m^2 °C)
x	co-ordinate along the length axis of the pipe, (m)
r	radial co-ordinate, (m)
R, A	radius of the pipe, (m) and pipe area, (m^2)
$u(r,x)$, $T(r,x)$	velocity profile, (m/s), and temperature profile, (°C)
u_m	the mean velocity, (m/s)
θ_1, θ_2, φ	fitting parameters obtained from the measured values of temperature in the pipe, (-)
ψ, ξ	fitting parameters obtained from the measured values of velocity in the pipe, (-)
Re, *Pr*	Reynold's number and Prandtl's number, (-)
a, *b* and *c*	fitting parameters, (-)
r	density of the fluid, (kg/m^3).

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