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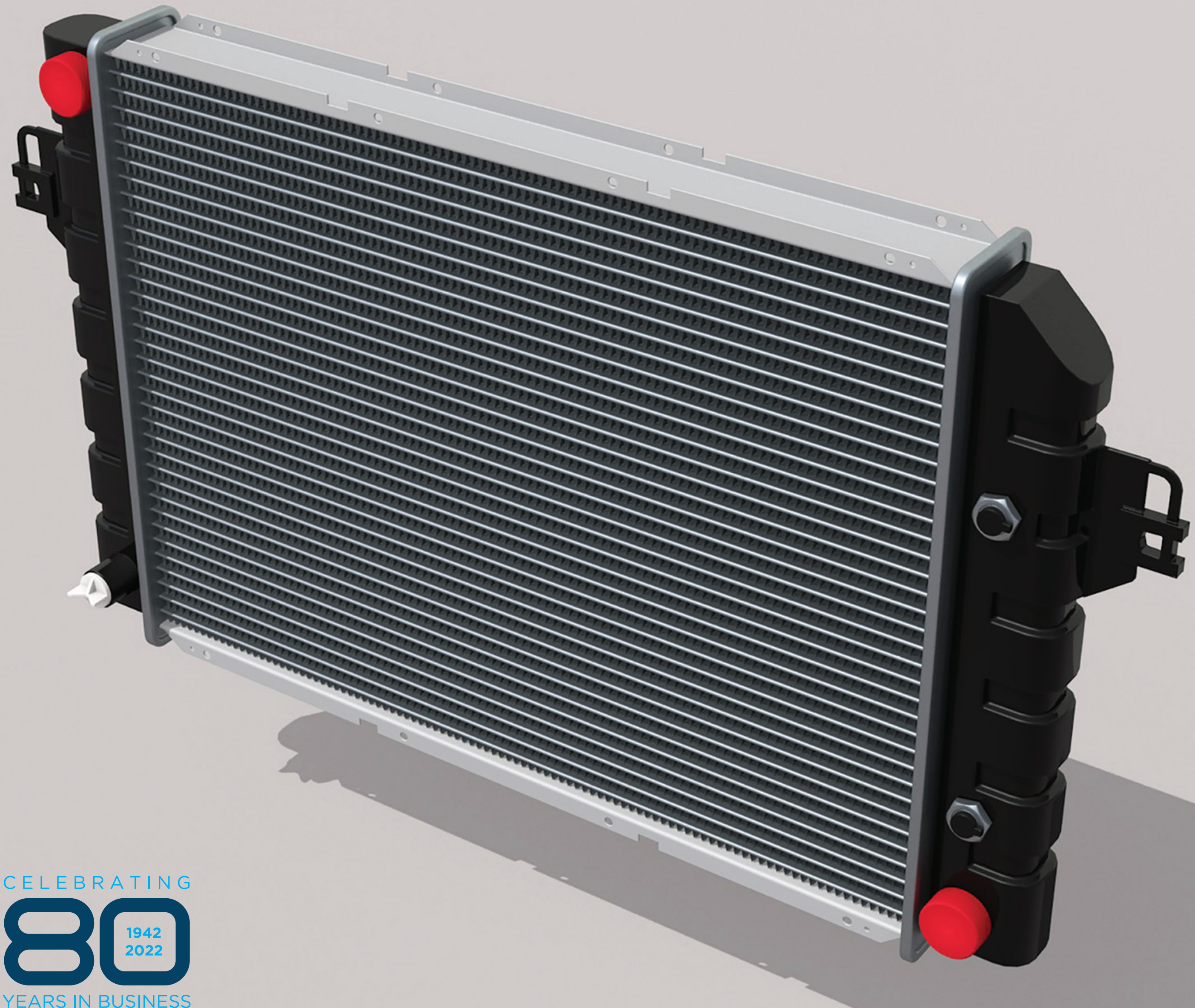
WHITE PAPER

A Study of the Heat Transfer Performance of Plain and Dimpled Radiator Tubes



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A Study of the Heat Transfer Performance of Plain and Dimpled Radiator Tubes.

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Synopsis

Details of an experimental facility for measuring and comparing the heat transfer characteristics of a plain and dimpled radiator core tube are presented. The experimental method is described and results from this exercise are produced in graphical form. The heat transfer process for the two tubes is discussed and recommendations made regarding future testing.

Notation

A	Heat Transfer Surface Area (m^2)
A_{TH}	Throughway Area Water Side (m^2)
C_p	Specific Heat at Constant Pressure (J/kgK)
D_h	Hydraulic Mean Diameter (m)
G	Mass Flow per Unit Area (kg/m^2s) (water side only)
h	Heat Transfer Coefficient for One Side of the Tube (W/m^2K)
k	Thermal Conductivity (W/mK)
m	Mass Flow Rate (kg/s)
Nu	Nusselt Number, hD_h/k
Pr	Prandtl Number, $Cp\mu/k$
Q	Heat Transfer Rate (W)
Re	Reynolds Number, GD_h/μ
T	Temperature (K)
U	Overall Heat Transfer Coefficient (W/m^2K)
ΔT_{lm}	Log Mean Temperature Difference
μ	Viscosity (Ns/m^2)

Subscripts

1	Inlet
2	Exhaust
a	Air
c	Cold Side
h	Hot Side

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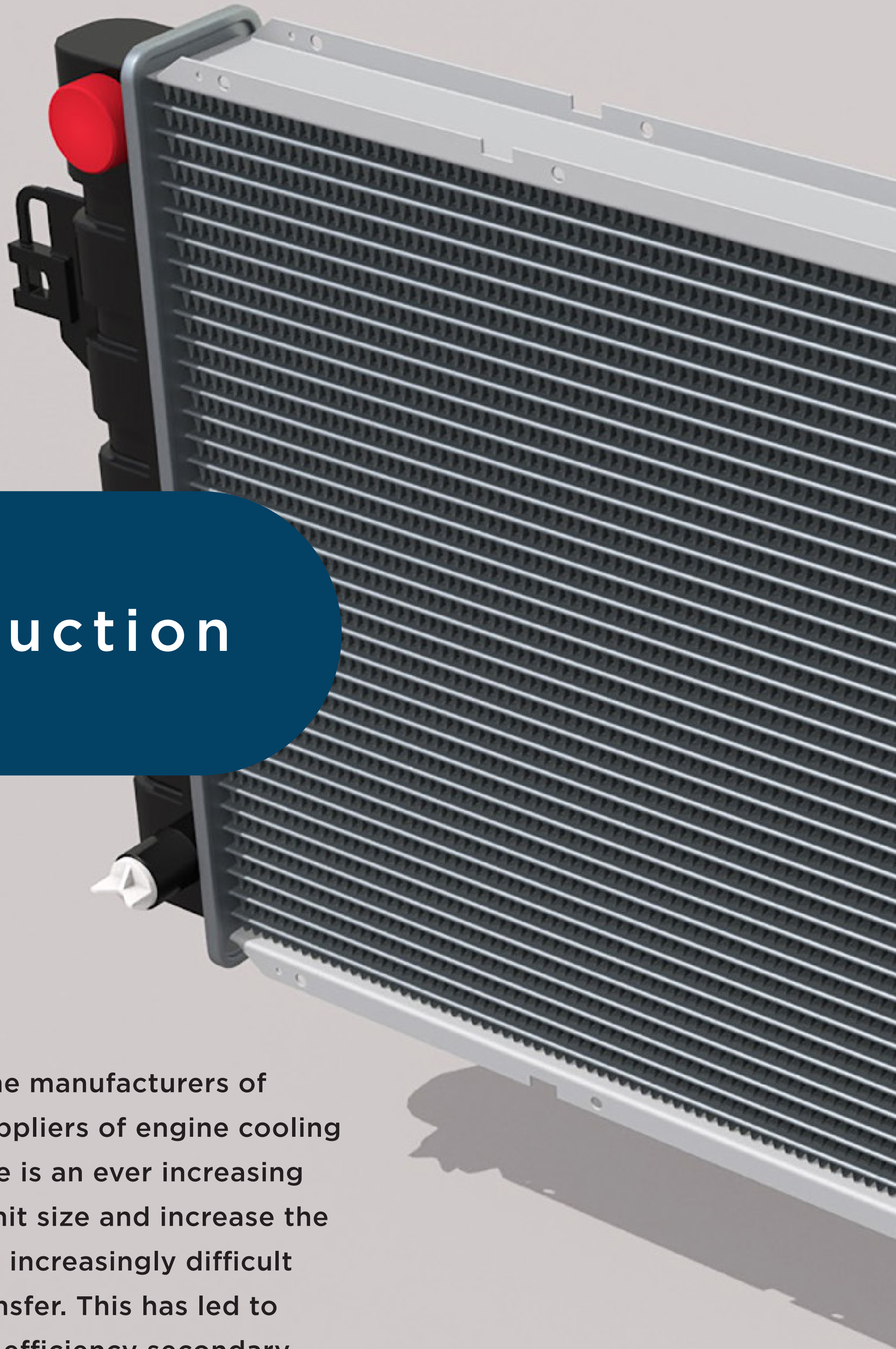


1.0 Introduction

The demands made by the manufacturers of motor vehicles on the suppliers of engine cooling systems is such that there is an ever increasing pressure to reduce the unit size and increase the duty of the radiator in an increasingly difficult environment for heat transfer. This has led to the development of high efficiency secondary surfaces on the air side with the adoption of designs and materials that are more common in the aerospace sector.

To facilitate this investigation a wind tunnel was constructed in the heat transfer laboratory in the Stephenson Building of the University of Newcastle upon Tyne and tests were performed to compare the heat transfer performance of the plain and dimpled tubes.

The dimpled tube that is the subject of this study is easy to manufacture in continuous strip, albeit with a set-up change and start-up scrap, production in low volumes can result in a reduced yield.

There may also be a requirement to have a plain (undimpled) length at each end of the tube to facilitate an improved join at the tube to the header tanks but this is not always necessary.

Various dimpling patterns are also available to meet customer specific requirements, where customers consider the dimpling makes a significant contribution to the heat transfer performance of the tube.

2.0 Theoretical Considerations

2.1 Basic Principles

The heat transfer rate for cross flow of air over a tube carrying a hot fluid can be described in two ways.

An energy balance of the hot fluid expressed in terms of the temperature change of the fluid as it flows through the tube can be equated to the product of the heat transfer area, a heat transfer coefficient and a temperature difference as shown below.

$$Q_h = m_h \, Cp_h \, (T_{h1} - T_{h2}) = U \, A_h \, \triangle T_{lm} \tag{1}$$

The log mean temperature difference is defined in the normal way

$$\triangle T_{lm} = \frac{(T_{h1} - T_{c1}) - (T_{h2} - T_{c2})}{\ln\left[\dfrac{T_{h1} - T_{c1}}{T_{h2} - T_{c2}}\right]} \tag{2}$$

The heat transfer from the hot fluid can be readily determined using equation 1. and once the air side heat transfer coefficient has been determined, equation 4. leads to the water side heat transfer coefficient.

which reduces to the following if the cold fluid temperature does not vary as in this case

$$\triangle T_{lm} = \frac{(T_{h1} - T_{h2})}{\ln \left[\frac{T_{h1} - T_a}{T_{h2} - T_a} \right]} \qquad (3)$$

The overall heat transfer coefficient U is made up of the heat transfer coefficients of the hot and cold sides of the tube as follows

$$\frac{1}{U} = \frac{1}{h_c} + \frac{1}{h_h} \qquad (4)$$

3.0 Experimental Facility

The experimental facility consisted essentially of a narrow working section measuring 280mm x 30mm. The tube was placed in the working section in a cross wise manner such that air drawn through the working section by an electrically driven fan flowed across the tube.

The air flow was measured using a pitot probe and regulated using a slide throttle in the fan exhaust. The Reynolds number of the air flow over the tube (based on the tube width) remained below the critical value of 500000 which marks the onset of turbulence for the plain tube for the entire flow range of the fan.

The maximum mid-stream velocity of the air was 27.2 m/s (62mph).

The tube was mounted in the working section and necessary connections made to the tube ends to supply it with a stream of hot water from a tank. The water was heated by a 3kW electrical heater that was controlled by a thermostatic unit that measured the temperature of the of the water at the pump delivery region in the tank.

3.1 The Wind Tunnel

An experimental facility for testing the tubes was designed around some existing equipment in the heat transfer laboratory.

The facility is shown schematically in figure 1.

The water was delivered to the test tube by a rotary pump and regulated by a gate valve. All pipework and connections were insulated with rubber foam pipe insulation.

The full list of equipment used is outlined in table 2.

A calibrated rotameter measured and indicated the water flow rate which was a maximum at 2kg/min for the plain tube and 1.2kg/min for the dimpled tube.

The water temperature was measured using chromal-allumel thermocouples mounted in the ends of the tube.

The output from the thermocouples was measured using a Cropco type P8 potentiometer with a resolution equivalent to 0.025°C.

3.2 Temperature Measurement

The classical method of temperature measurement using a potentiometer and ice point reference junction was adopted in this work as it was considered that electronic temperature indicators with floating reference points could not deliver the required accuracy.

Before the beginning of the experiment the temperature measuring system was calibrated as described below.

3.3 Calibration of the Temperature Measuring System

The thermocouples used to measure the fluid temperatures were calibrated against a platinum resistance thermometer in a fluidised bed, calibration bath. The accuracy of the resistance thermometer at 0.1% at 0°C was much greater than that of the thermocouples which in the uncalibrated form is +/- 2.5°C.

The potential error in absolute measurement using the thermocouples is larger than some of the measurements made during the experiment. This is offset to some degree by the processing of the results that is dealing with temperature differences and in all probability the errors would cancel out, especially as all the thermocouples came from the same initial batch of wire. Likewise, errors in the absolute value of the cold junction would cancel out.

The material that is used to construct the thermocouples is in fact, a metallic solution and it is possible for migration of the constituents to occur over time that significantly changes the calibration of the wire and lead to the quoted errors. It was necessary to calibrate the thermocouples to ensure that such a process had not occurred in the batch wire and that the thermocouples were producing, as far as is practical, the same output at a specified temperature. As the calibration included all the instrumentation, including the reference junction, the selector switch and the potentiometer, any errors in these items would also be accounted for.

The results of the calibration procedure are shown in figure 2.

3.4 Calibration of the Water Flow Meters

Two Rotameter type flow meters were used to measure the water flow rate into the tube. The standard calibration of these meters is normally at 20°C. These meters consist of an acrylic body with a tapered bore in which is fitted a brass float.

The displacement of the float caused by the fluid flow and the pressure drop across the float, is used as an indication of the fluid flow rate. The fluid flows in the annulus formed between the float and the bore. At temperatures higher than the calibration temperature, differential expansion between the float and the body reduce the size of the annulus, thus for a given flow rate of fluid the displacement of the float would be created and the meter would indicate a larger flow than the given flow. To overcome this problem, the meters were calibrated against a gravimetric system that weighed the delivery over a time period.

The outcome of this exercise is shown in figure 6.

3.5 Calibration of the Working Section Flow Field

The flow field in the working section was calibrated prior to testing to ensure that a uniform velocity field existed over the length of the tube and that the centre line velocity was a representative velocity to consider in the heat transfer analysis.

The calibration was performed by traversing a pitot-static tube across the flow path at several transverse locations. The encroachment of the end wall boundary layers, side and top and bottom is seen to be very small. The side wall boundary layer is considered to occupy a small portion of the length of the tube that its influence is negligible. The top and bottom boundary layers are so small that they will not interfere with the development of the boundary layer on the tube so they would also have little effect.

The results of this exercise are shown in figure 4.

4.0 Experimental Procedure

The experimental procedure adopted to establish the basic data was the same for both tubes and for obtaining the air side coefficient data and water side data.

Switch on water heaters in the water tank and the potentiometer

Place test tube in working section, connect up water & thermocouple

Check for leaks and rectify as necessary

Prepare the thermocouple reference junction

1 **Set-up equipment**

2 **Checks**

3 **Set-up for**

After starting up the process as below, the system was allowed to stabilise and after approximately, 5 minutes, the fluid flow rates checked.

Thermocouple outputs were measured, recorded and this step repeated several times until repeatable measurements were obtained.

Mixing in the tank

Check flow meter

Thermostat working

Zero the potentiometer and check output of the thermocouples

Circulate water through the by-pass loop

Circulate required amount of water through test tube

Switch on air fan and adjust throttle to produce required air flow

instrumentation **4** **Start-up**

5.0 Presentation of the Results

It is normal industrial practise to present the heat transfer coefficient in a non-dimensional form as a Nusselt number (hD /k). In this instance, the most appropriate presentation of the data is a plot of Log NuPr plotted against Log Reynolds number (GD/μ).

6.0 Analysis of the Results

6.1 The Heat Transferred and Overall Heat Transfer Coefficient

The energy transferred from the hot water to the cooling air is determined from the temperature measurements.

Using equation 1. on p.7 which is also used to determine the overall heat transfer coefficient.

6.2 The Air Side Heat Transfer Coefficient

The evaluation of the air side coefficient makes use of the relationship in equation 4 between the overall heat transfer coefficient and the heat transfer coefficients of each side. If one side of the heat exchanger is kept at a constant condition such that the heat transfer coefficient is constant whilst the heat transfer coefficient on the other side is allowed to vary it is possible to produce the graph shown in figure 5.

This graph shows the relationship between 1/U and 1/m

As the water side mass flow rate is increased the heat transfer coefficient increases until in the extreme, as m becomes very large, 1/h becomes very small compared to 1/h. The intercept of the curve with the y axis thus gives the value of 1/h .

Shown for plain and dimpled tubes in figures 6 and 7.

6.3 The Water Side Heat Transfer Coefficient

The water side heat transfer coefficient follows from the equation with a knowledge of the air side and overall heat transfer coefficients.

6.4 Check for Accuracy

The above analysis was checked for accuracy by performing an energy balance on the tube. The heat transferred from the water to the air must be conducted to the tube and this must be equal to the heat transferred from the tube to the air. It is possible then to set up two equations with the only unknown, the tube temperature.

If the results from the previous analysis is correct, it follows that both equations will produce the same wall tube temperature.

A second way of assessing the accuracy of the water side heat transfer coefficient is by repeating the experiments for a different air side condition and determining the water side heat transfer coefficient for the same water side conditions as in the initial analysis.

Both of these tests were performed on a sample of the experimental data and the outcome of this was considered to be satisfactory.

7.0 Discussion of the Results

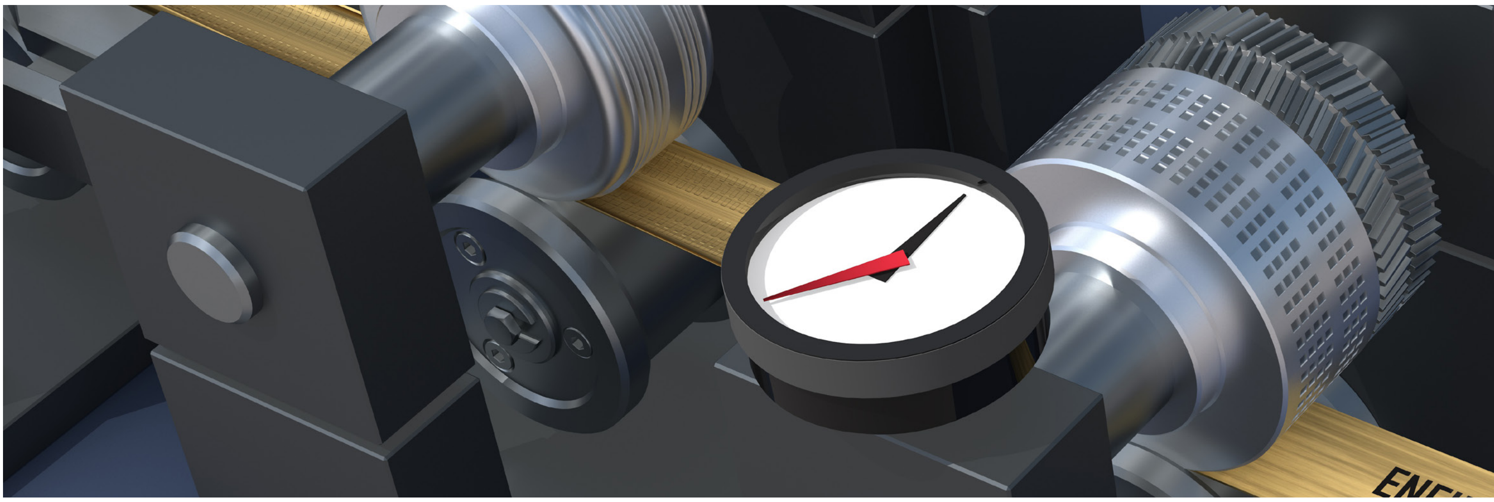
The heat transfer characteristics of the water side heat transfer for the two tubes are shown in figures 8 and 9. It can be seen that the dimpled tube exhibits a larger heat transfer rate than the plain tube and that both the water side and the air side heat transfer coefficients are larger than the plain tube.

The overall heat transfer coefficient in both cases however is dominated by the air side heat transfer coefficient as it is very much the smaller of the two coefficients and the improvement in the heat transfer of the dimpled tube is produced primarily by an increase in the air side heat transfer rate.

8.0 Conclusions & Recommendations

1 The effect of dimpling on the tube's primary surface area improves the overall heat transfer rate.

2 The improvement is brought about by increasing both the water side and air side heat transfer coefficients.

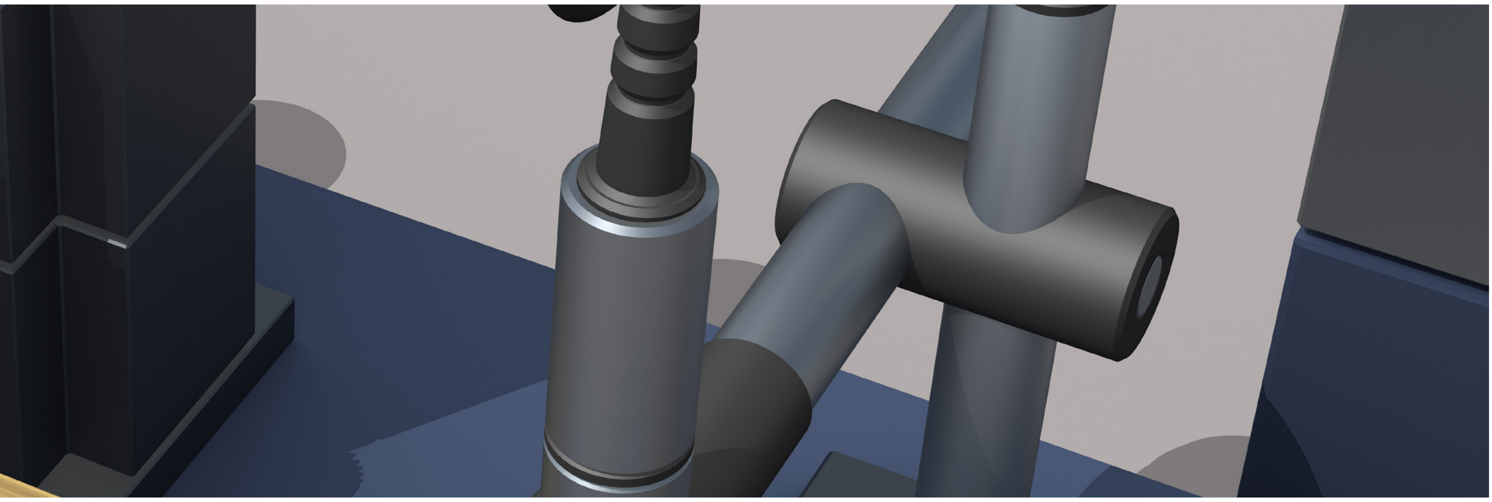


5 The dimpled tube has a much larger pressure drop than the plain tube and will therefore, require more pumping power to achieve the same water flow rates.

6 When applied to a radiator core it would seem that any improvement brought about by the dimpling is a result of changing the heat transfer characteristics of the air side primary surface.

3 The overall heat transfer rate is limited by the air side heat transfer coefficient. The water side coefficient is significantly larger than the air side coefficient such that an increase in the water side coefficient alone will have a negligible effect on the overall heat transfer coefficient.

4 The effect of partial dimpling of the tubes will produce a water side characteristic that lies between the two lines in figure 9. Taken on its own this will have little effect on the overall heat transfer coefficient.



7 The 'most probable' explanation for improvements in the radiator core is that the air side primary surface boundary layer is "tripped" by the dimples becoming more turbulent and displacing thermal energy from the surface by an increase in the diffusion of mass through the boundary layer.

8 Further studies could be designed to examine the air side characteristics of full and partial dimpling with the secondary surface included. Overall behaviour could be determined using traditional energy balance methods and also through detailed flow measurements in the radiator core with a laser anemometer.

Tables

Table 1. Heat Transfer Characteristics of the Tubes

Heat Transfer Characteristics	Characteristic	Value
	Tube Length	325mm
	Tube Width	18mm
	Throughway Area	$25.5.\ 10^{-6}\ m^2$
	Heat Transfer Area (Water Side)	$11.1.\ 10^{-3}\ m^2$
	Heat Transfer Area (Air Side)	$12.\ 10^{-3}\ m^2$
	Hydraulic Diameter	$2.756\ 10^{-3}\ m^2$
	Dimple Pattern	In-line 3mm transverse pitch 4mm axial pitch 4 dimples transversely

Table 2. List of Equipment Used in the Tests

Equipment	Equipment	Type	Details
	Cropico Potentiometer	P8	No 35830
	Cropico Selector Switch	SP2	No 28264
	Furness Controls Manometer	FC001	No FM 1777
	Platon Flow Meter	A10HS	Range 200-3000 cc/min
	Gap Meter		Range 20-300 cc/min

Figures

Figure 1. Schematic Diagram of the Test Facility

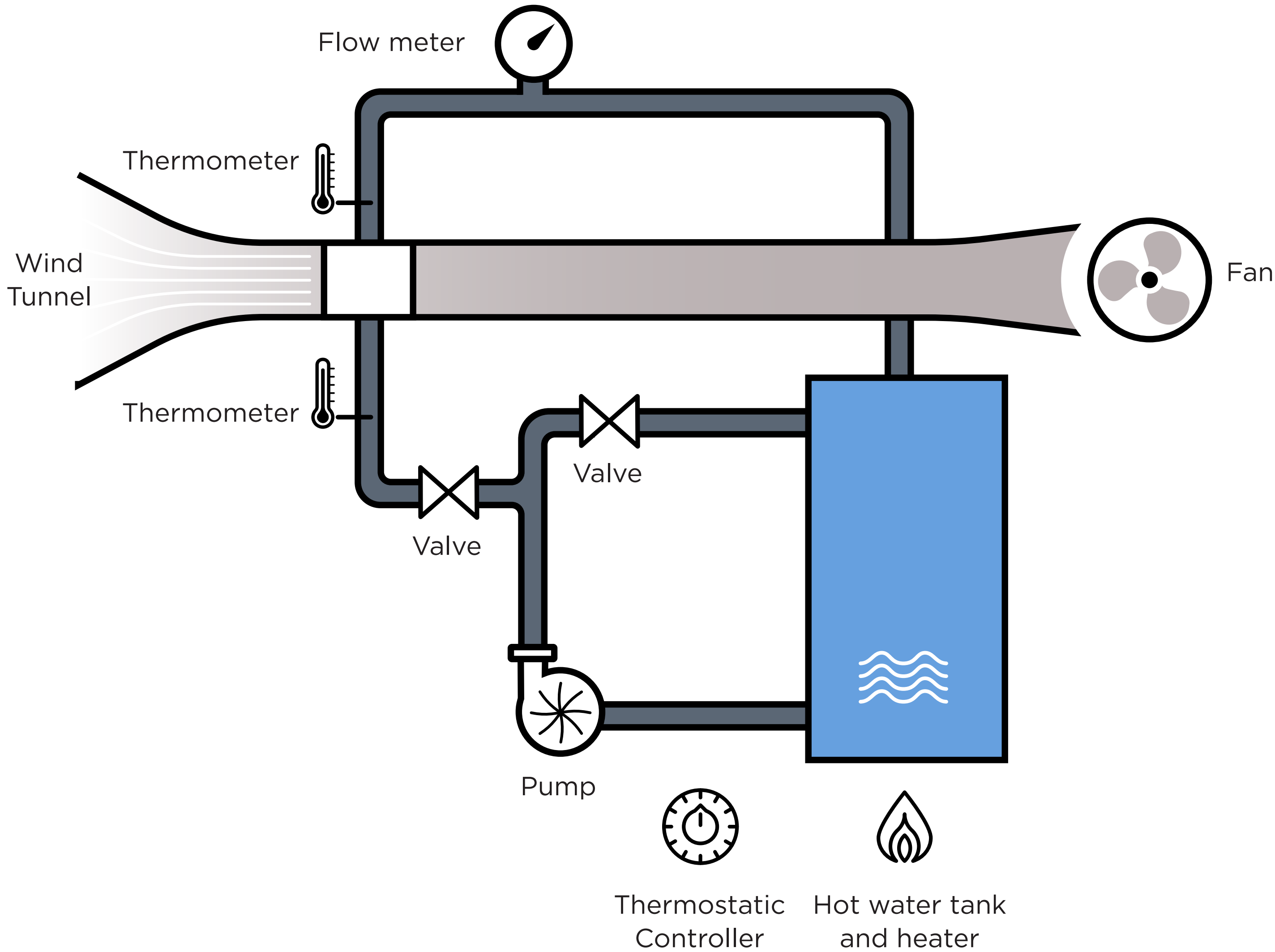


Figure 2. Calibration Curve of the Chromal-Allumal Thermocouples

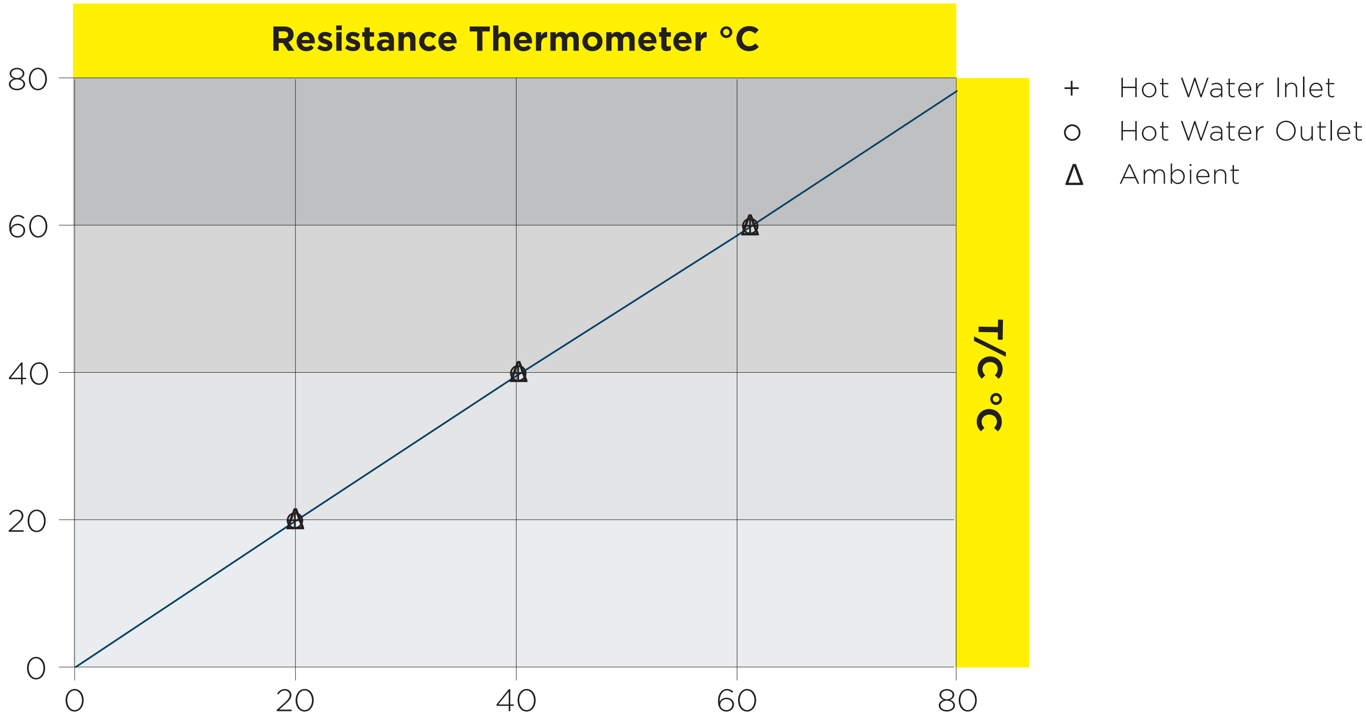


Figure 3a. Calibration Curve of the 3000 cc/min Water Flow Meter

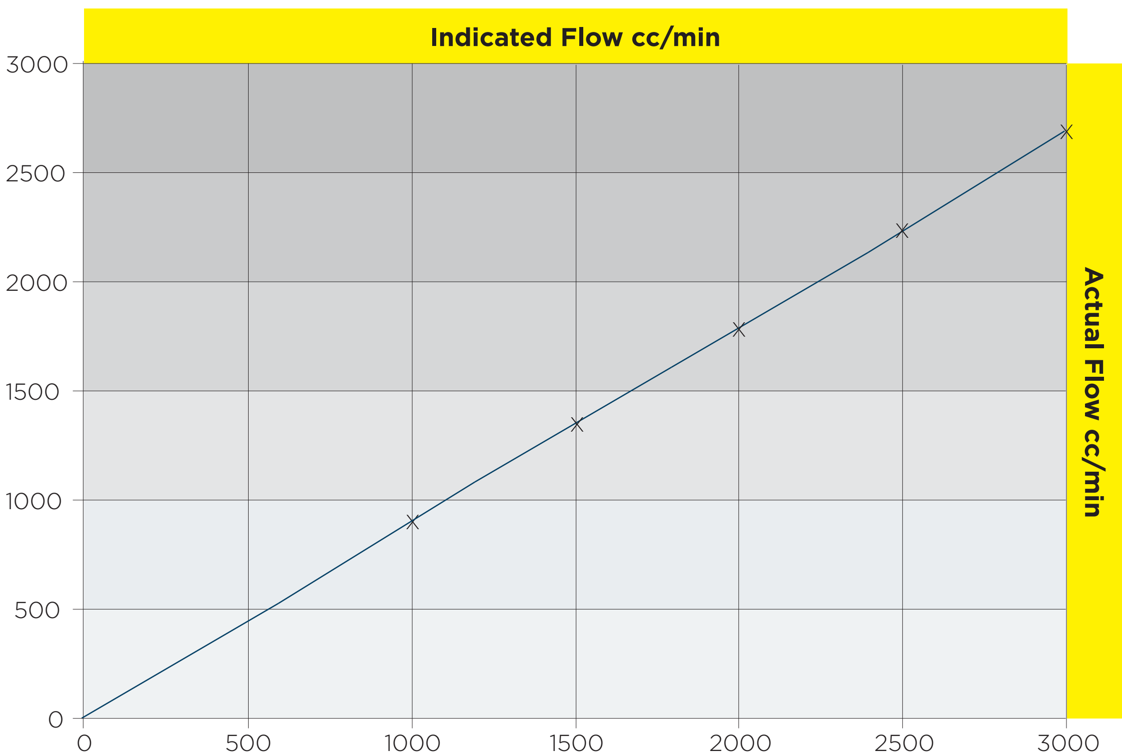


Figure 3b. Calibration Curve of the 300 cc/min Water Flow Meter

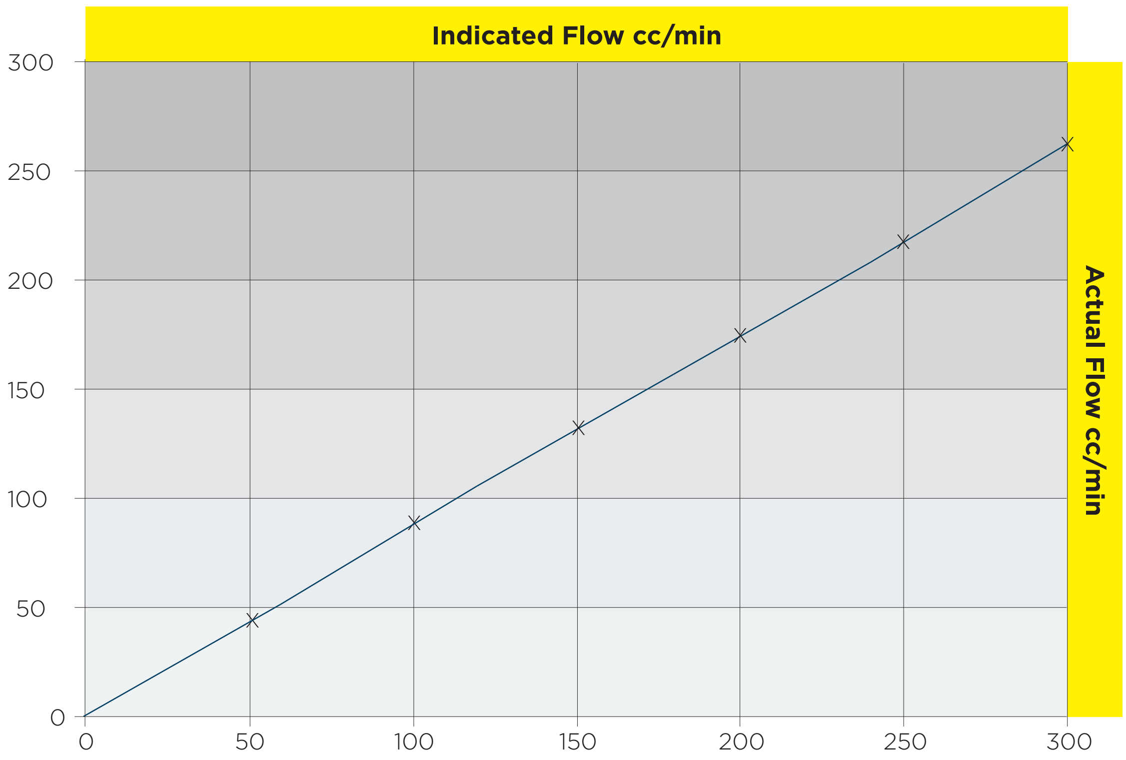


Figure 4. The Variation of the Velocity Field in the Test Section

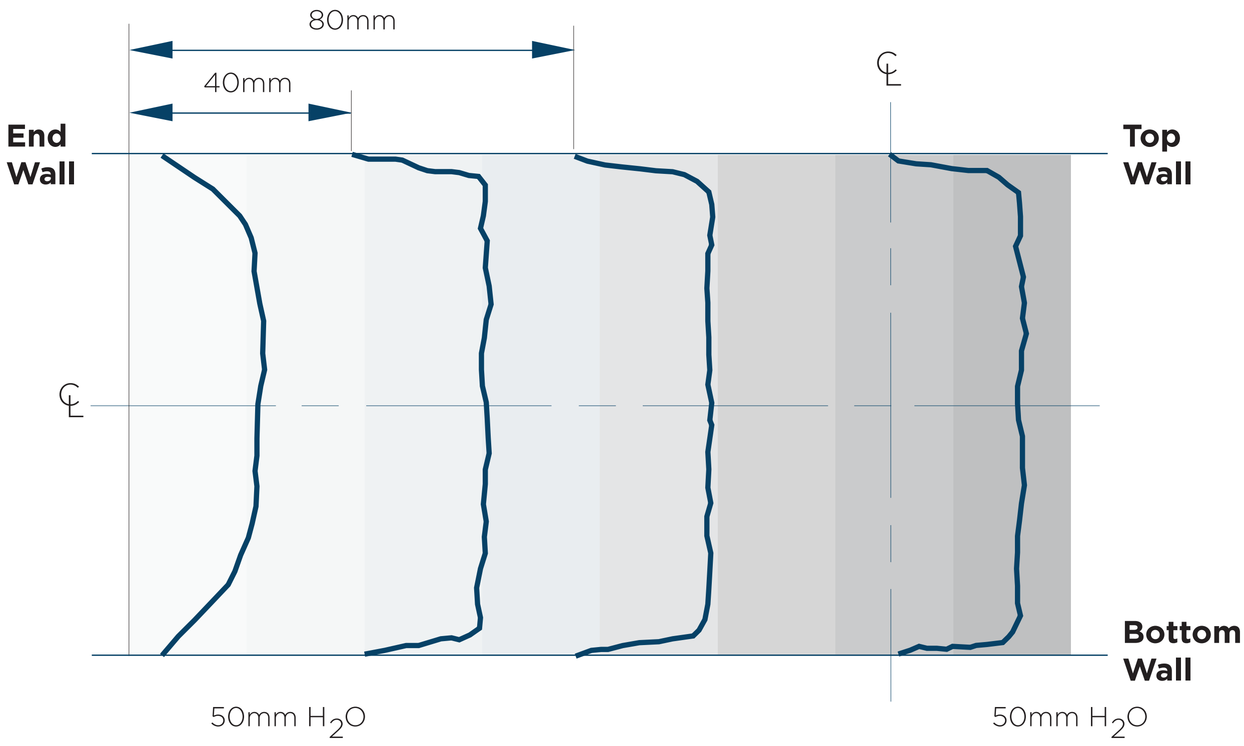


Figure 5. The Relationship between 1/U and $1/m_h$ (typical)

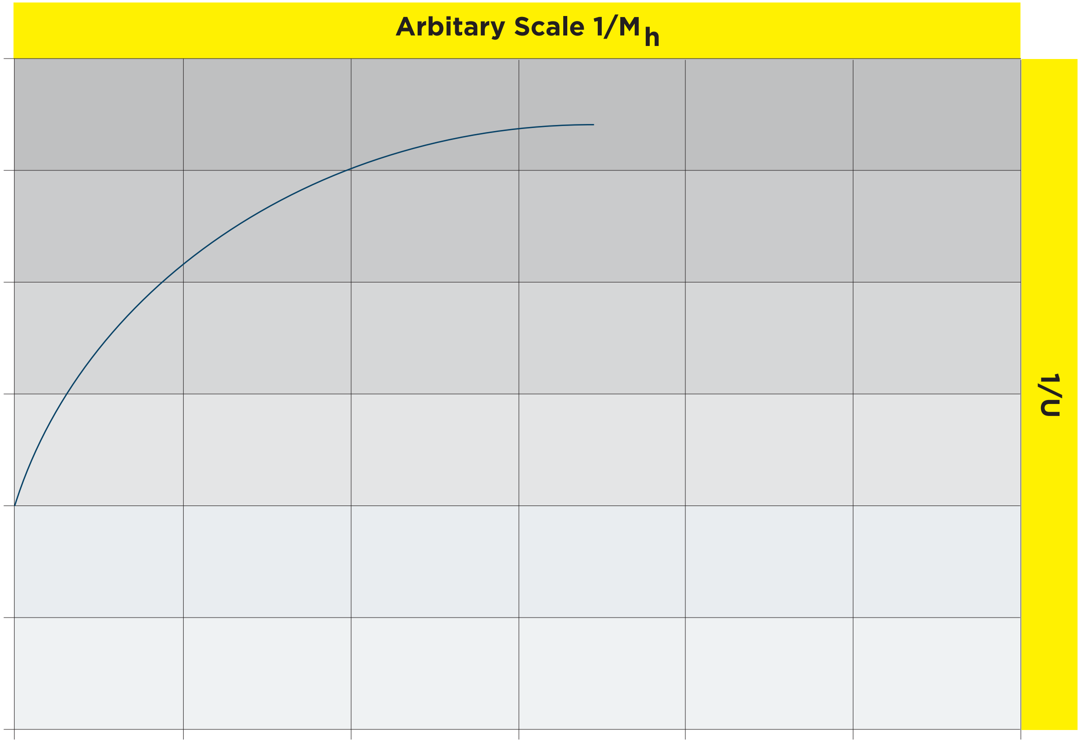


Figure 6. The Relationship between 1/U and $1/m_h$ for the plain tube

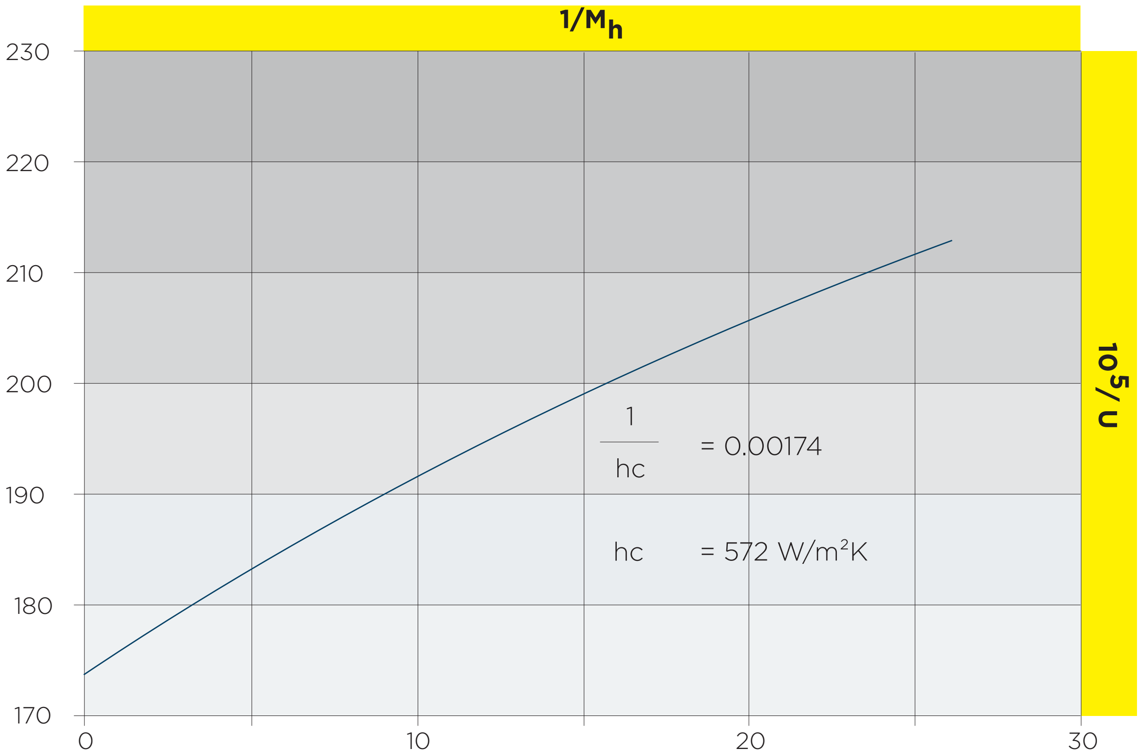


Figure 7. The Relationship between 1/U and 1/m_h for the dimpled tube

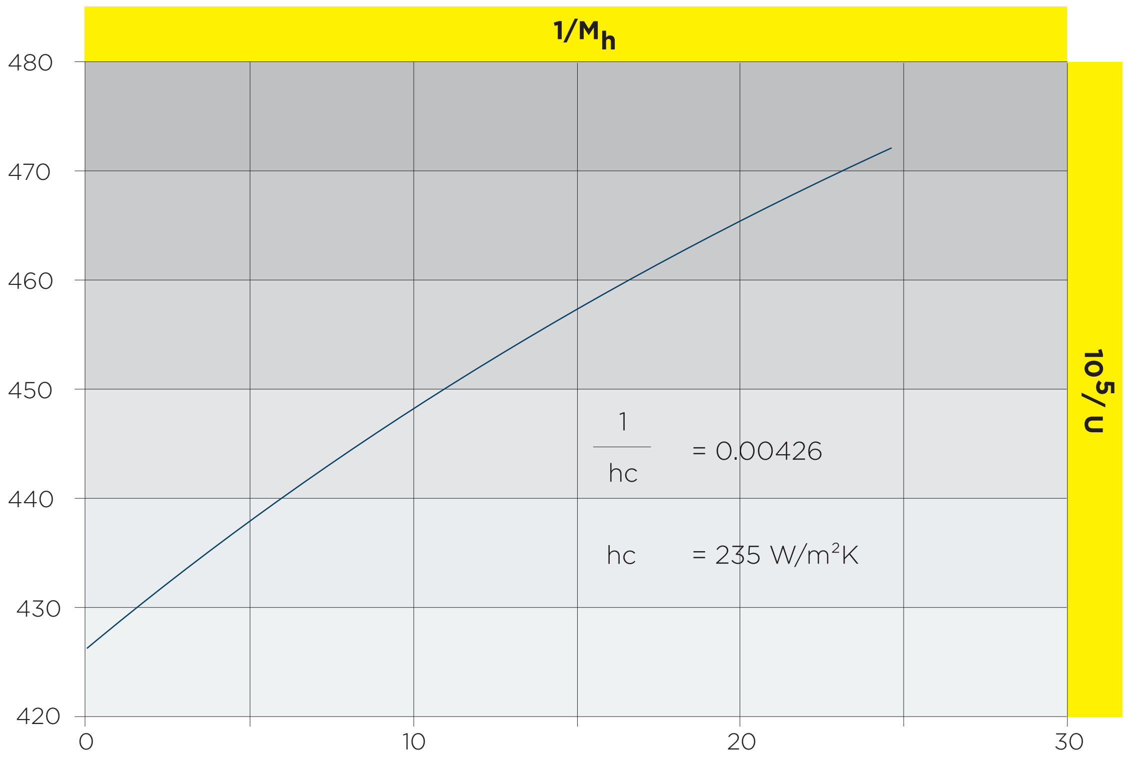


Figure 8. Heat Transfer Performance of the Plain and Dimpled Tubes

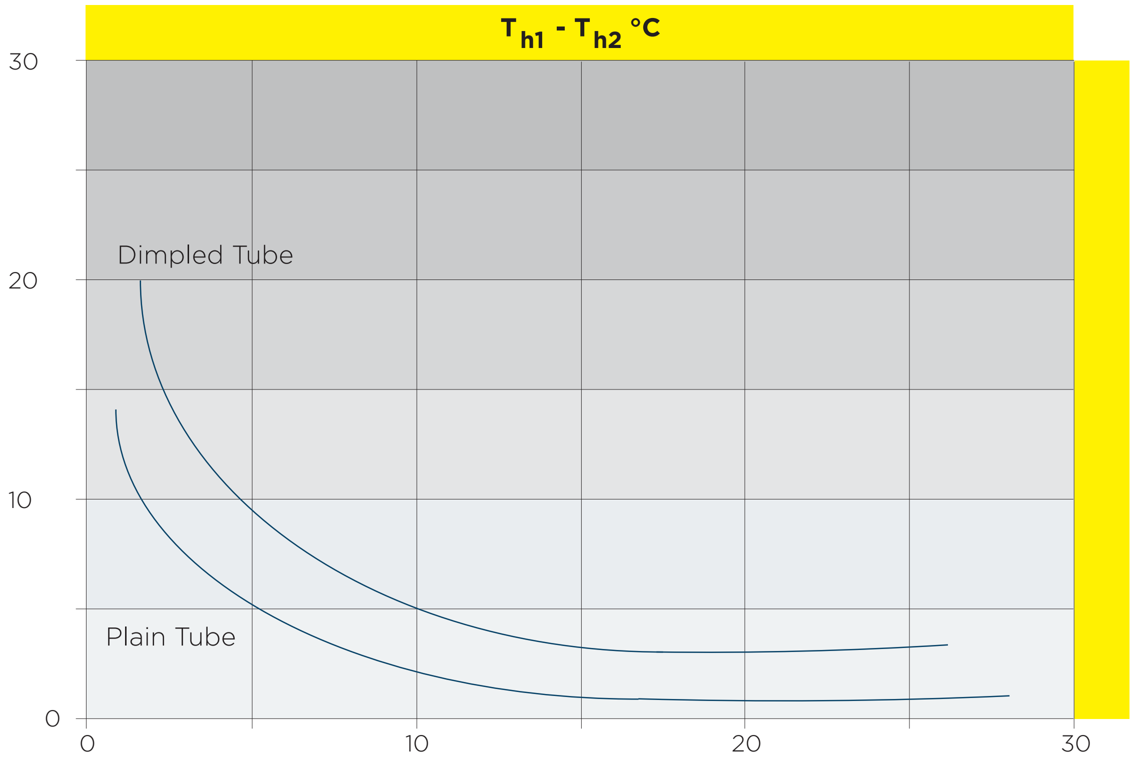


Figure 9. Heat Transfer Performance of the Plain and Dimpled Tubes

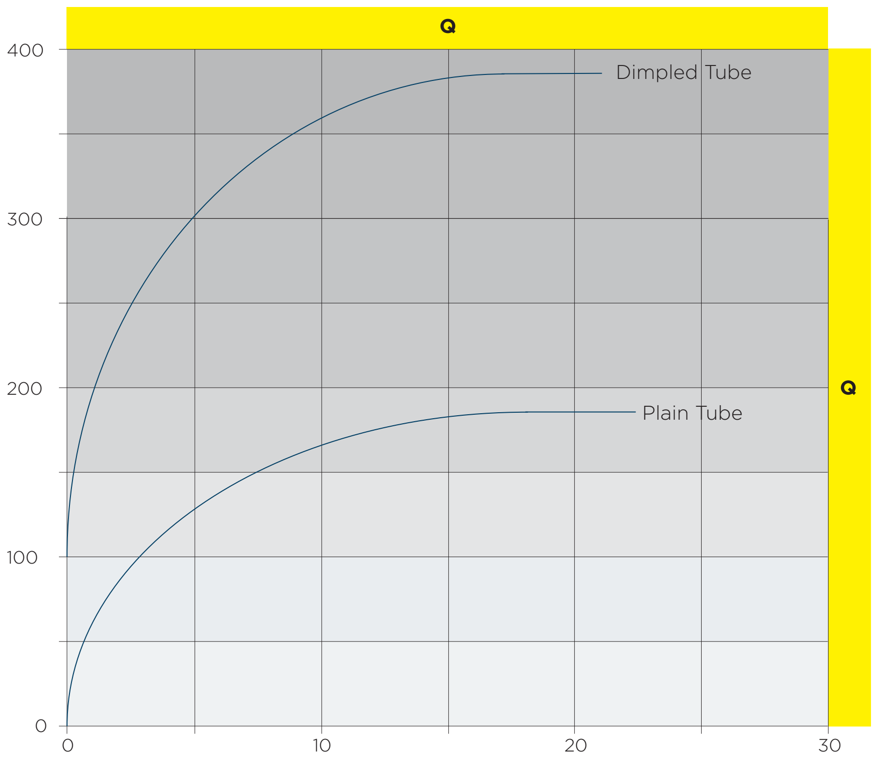


Figure 10. Log NuPr $^{-1/3}$ against Log Re for the Plain and Dimpled Tubes

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