# EXPERIMENTAL TECHNIQUE FOR EVALUATION OF HEAT TRANSFER COEFFICIENTS WITH THERMOCAMERA IN A PLEXIGLAS MODEL

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## ABSTRACT

This technique is developed to improve and verify the cooling, and to determine the heat transfer coefficients for gasturbine components

On a perspex model of the object an heaterfoil is mounted. The heatfoil modulates the heat load. The model is cooled with air from the air supply system on the hot side. The temperature is measured with a thermocamera and the cooling effectiveness can be calculated from the heat load and the difference between surface temperature and ambient.

The evaluation process is carried out in Matlab® and Excel®. The results are presented in field and average curves. It's possible to achieve a detailed information with inaccuracy often below 7 %.

The results show good agreement to earlier investigations and that the method is cost efficient. Another advantage is that new configurations can relative easily can be applied, which mean time saving.

## INTRODUCTION

Constant increasing energy prices and an increased awareness of environmental influence has put high demands on growing efficiency for power generation. The efficiency for a gasturbine is strongly depending on the gastemperature in to the turbine stage. Rising temperature put high demands on the construction and the material for parts exposed to the high temperature. A great deal of work is done with various kinds of calculation programs more or less advanced. All calculation of cooling efficiency demands some kind of validation. ALSTOM Power Finspong has at one's disposal a high-pressure test facility, for testing of cooled gasturbine components at near-engine conditions. One of the methods uses an infrared thermocamera to measure the surface temperature of the test object. To calibrate the measurement thermocouples are used. This test rig was put in operation in 1997, for more information concerning this test facility, see Ref 1.

More recently this technique has also been implemented in scaled perspex models at atmospheric conditions. This paper considers this last application.

## NOMENCLATURE

Latin letters

| | | |
|---|---|---|
| A | Area | [m$^2$] |
| Bi | Biotnumber | [-] |
| c | Velocity | [m/s] |
| Cp | Specific heatcapacity | [J/kg /K] |
| l | Length | [m] |
| $\dot{m}$ | Mass flow | [kg/s] |
| Ma | Mach number | [-] |
| n | Scale factor | [-] |
| Nu | Nusselt number | [-] |
| T | Temperature | [K] |
| q | Power | [W] |
| q | Power | [W/ m$^2$] |
| Re | Reynold number | [-] |

Greek letters

| | | |
|---|---|---|
| $a$ | Convektiv heattransfer | [W/ m$^2$/K] |
| $\Delta$ | Difference | |
| $l$ | Thermal conductivity | [W/m/k] |

Subscripts

| | |
|---|---|
| c | Characteristic |
| e | Engine |
| ins | Inside |
| m | Model |
| n | Pixel number |
| out | Outside |
| surf | Surface |

## LABORATORY FACILITIES

In the part of the Fluid dynamics laboratory were the atmospheric tests are carried out there is an air supply from the workshop, 6 bar @ 5 kg/s. and a internal low pressure system 0,6 bar @ 6 kg/s.

For massflow measurements normally a v-cone$^{\circledR}$ manufactured by Mc Crometer's or a standard flange, ISO 5167-1, is used. For pressure measurements PSI$^{\circledR}$ pressure measurement equipment is used. For temperature measurement thermocouple is used. The analog/digital converter system for data acquisition from the thermocouples is of type Datascan$^{\circledR}$ unit.

The device presented above is the laboratory standard device used in this type of measurements.

## SCHEMATIC DESCRIPTION

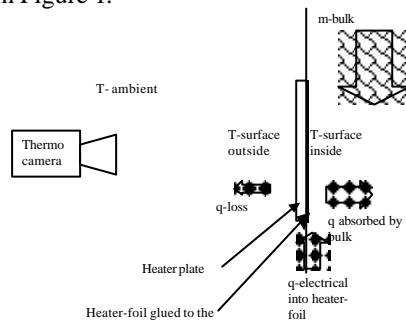The procedure can schematically be described as in Figure 1.



Figure 1

On a plate made of perspex or polycarbonat a heater-foil is glued. The heater plate is usually a part of a model that is going to be investigated, shown schematically above as a flat plate. A known electrical power is feed into the heater-foil. Some of the power will disappear to the surroundings through the heater plate. The rest will be taken up by the bulk flow. If the temperatures and the supplied power are known the heat transfer coefficient can easily be calculated. The temperature on the ambient surface is measured with a thermocamera, the bulkflow temperature with a thermocouple and the power with a powermeter.

## THEORETIC DESCRIPTION

### GEOMETRIC AND DYNAMIC SCALING

Geometrically the model is scaled with constant scale factor through the model:

$$n = \frac{l_{\text{model}}}{l_{\text{engine}}}$$

Equation 1

Dynamically the model is scaled with respect to Reynolds number and Mach number, both to be kept constant between model and engine.

$$\text{Re} = \frac{\dot{m}\, l_c}{\boldsymbol{m} A}$$

Equation 2

$$Ma = \frac{c_{lokal}}{c_{sound}}$$

Equation 3

### EQUATIONS FOR HEAT TRANSFER

The convection heattransfer coefficient on the inside is calculated from:

$$\boldsymbol{a}_{\text{sur, ins}} = \frac{q_{\text{absorbed by bulk}}}{T_{\text{surf, ins}} - T_{\text{bulk}}}$$

Equation 4

When the testing is done on a scaled model the heat transfer coefficient has to be calculated to a dimensionless form so the results can be transformed to engine conditions. The Nusselt number is then used as dimensionless heat transfer coefficient. Nusselt number is defined as:

$$Nu_{\text{surf, ins}} = \frac{\boldsymbol{a}_{\text{surf, ins}} \cdot l_c}{\boldsymbol{l}_{\text{bulk}}}$$

Equation 5

When surface enlargers and turbulators, for example ribs, are used inside the cannel, the ration of convective heat transfer on the surface to the heat conductivity in the material (the Biot number) has to be kept constant between the model and reality. The Biot number is defined as:

$$Bi = \frac{\boldsymbol{a}_{\text{surf bulk}} \cdot l_c}{\boldsymbol{l}_{\text{enlardger}}}$$

Equation 6

This is not so easy done because in most of the relevant gas turbine applications the results from keeping Biot number constant will give heat conductivity around 10 W/m/K at room temperature. These order materials like Hastelloy X (9.7 W/m/k at 20 $^{\text{o}}$C) to be used. There are large problems associated with the machinery of these materials. Therefore it can be a benefit if a more easy-machined material could be used.

Often aluminum is used which is easy-machined and could even be mould by hand. Aluminum has heat conductivity around 220 W/m/K, which is much too high.

If only the cannel surface is studied and if the enlargers heat conductivity is large the enlarger's

influence on the heat conduction can be said to be insignificant. This because heat conducted through the enlarger's is much higher then that conducted through the heater wall. In this case it can consequently be better to use aluminum. (The heaters then have been considered as they where put on the enlarger surface)

This simplification was studied in a work done by Persson, Ref. nr.1, and it shows that there was no major fault due to this.

The power to the heater-foil is regulated with a transformer, and measured by a wattmeter. It is recalculated to a power per surface value.

The temperature measured by the thermocamera is the temperature on the surface facing the ambient (see chapter Thermocamera). A mathematical model is needed to calculate the surface temperature on the heated side. That model has to include the influence of the heat transfer to the surroundings and the conductive resistance through the plate. The thermocamera will give the temperature in a known number of pixels.

The heat loss is calculated from the knowledge of the heattransfer coefficient on the outside. (See chapter Determination of heat transfer coefficient to the ambient) The power loss per pixel then can be calculated from:

$$q_{loss} = a_{surf, out} \cdot \left(T_{surf, out} - T_{ambient}\right)$$

Equation 7

Before the heat transfer coefficient can be calculated the temperature on the inside of the channel wall has to be calculated. Since the material, and there by the thermal conductivity, and its thickness is known the temperature can be calculated from:

$$T_{surf, bulk} = T_{surf, out} + \frac{q_{loss} \, l_{thickness}}{l_{heater \, plate}}$$

Equation 8

The heater plate heat conductivity coefficient is a material property and can be taken from material data.

One should be aware of the fact that the temperatures profile, to some extend, will even itself out because of heat transfer through the heater plate. However, the accuracy of the average value of the heat flow will not be affected by this phenomenon.

The heattransfer coefficient now could be calculated as:

$$a_{surf, out} = \frac{q_{electrical} - q_{loss}}{T_{surf, ins} - T_{bulk}}$$

Equation 9

The bulk temperature will rise due to heat load absorbed by the bulkflow. The actually temperature is calculated from:

$$T_{bulk,pixel \, n} =$$

$$= T_{bulk,pixel \, n-1} + \frac{Cp_{bulk,pixel \, n}}{\dot{m}_{pixel \, n} \left(q_{electrical} - q_{loss}\right)_{pixel \, n}}$$

Equation 10

I most cases this temperature rise will be quite small, because the massflow often is large.

Above equations can be put together with each other.

$$Nu =$$

$$\frac{\left(q_{el} - a_{surf, out} \cdot \left(T_{surf, out} - T_{ambient}\right)\right)}{\left(T_{surf, ins} - T_{bulk}\right) \cdot l_{bulk}} \cdot l_c$$

Equation 11

Usually this is not done because there are a benefit if each quantity can be studied alone. Faults that have been done could then be much more easily found. The calculations are done pixel by pixel.

## Determination of heat transfer coefficient to the ambient

To estimate the heat transfer coefficient to the ambient a layer of isolation is applied over the heaters, in the bulkflow channel. Measurements of the power input and the surface temperature are done. The heat transfer coefficient then is calculated as:

$$a_{surf, out} = q_{electrical} \Big/ \left(T_{surf, out} - T_{ambient}\right)$$

Equation 12

This method is useful when the bulkflow is low or when outlet of the model not is into the room. Because the bulkflow has a large impact on the room air so it will also impact the heat transfer coefficient. If above method not work it is better to mount a detachable heaterplate in front of the ordinary. Between them there will be isolation.
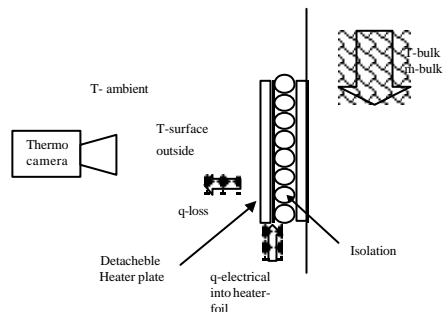


Figure 2

Normally a hood is mounted on the ambient side to ensure stable conditions. In many configurations the heat transfer coefficient vary over the surface. For example if there are

reinforcement on the ambient side of the heaterplate they will free convektion over the surface and there by the heattransfer coefficient. In these cases the heat transfer coefficient is evaluated pixel by pixel.

Normal value for the heat transfer coefficients is between 9 – 20 W/m²K

## DATA ACQUISITION

The temperatures are measured with thermocouple type K, and Datascan® unites. The data are stored in a text file by a Labview® application. This is a laboratory standard procedure.

The temperature field on the ambient side of the heaterplate is examined with a thermocamera.

The data from the thermocamera is stored in a matrix format in a chosen number of pixels.

## EVALUATION PROCESS

The thermocamera sotfware is running under O/S-9. Unfortunately there are no clear way to transport data from O/S-9 to Windows NT-4 (standard operativesystem at ALSTOM power Finspong). One possible way is to dump the data to a diskette and copy the file to a computer running under Windows NT-4.

The main postdata process is carried out in Matlab. Matlab has good visualization possibilities, and easy data treatment possibilities.

Normally the datafiles are named from actual date and time for the test. The program automatically reads them when datum and time are given. The textfile generated by Labview usually contain information about actual model configurations. Often a number of different heater plates are used and by that could the heaterfoil have different coordinates.

The program calculates each parameter individually, pixel by pixel and show them in data fields so they can be examined.

After that the data files have been read the program reads actual configuration information.

Then the temperature field that is read by the thermocamera is showed for the operator. This to be able to give opportunity to adjust the coordinates for the measurement. Each test could be adjusted, and this is often necessary because it is difficult to place the measured surface in to coordinate system when measurements are carried out. Instead there are some defined positions marked (with a different temperature) on the surface. The temperature field is then adjusted after these positions. If the model contain some curved surfaces the temperature field has to be transformed to actually shape. The camera produces a flat field.

The operator chooses which coordinates that are going to be evaluated. The power input to the bulk and its conditions then is calculated

The results are presented in field for each property and tree dimension pictures. The result can also be recalculated to engine conditions. An out data file is produced that often are opened in Excel where mean values etc. can be analyzed, and different configurations can be compared. Often there are some reinforcement made to the model and in these regions can not temperature measurements be carried out. This reinforcement has instead been used to find the right positions for the measured surface, see above. If a large area shall be examined it has to be divided into smaller areas if high resolution is required. Therefor there is a possibility in the program to put different areas together. Massflow and other parameters then are normalized.

## ACCURACY

A study of the uncertainty of Nusselt number gives an uncertainty below 5 % for the example in chapter Burner chamber. A case study of the most parameter to find maximal error is shown below. The values are taken from a normal case from the example below, in an actual case this has to be done for the case explicit. The temperature difference between the bulk and the surface is the most important parameter for the uncertainty, and in this case study set to 30°C.

$$\left|\frac{\Delta Nu}{Nu}\right| = \left|\frac{\Delta q_{el}}{\left(q_{el} - a_{surf} \cdot \left(\overline{T}_{surf} - T_{amb}\right)\right)}\right| +$$

$$\left|\frac{\Delta\left(a_s \cdot \left(\overline{T}_{sur} - T_{amb}\right)\right)}{\left(q_{el} - a_{surf} \cdot \left(\overline{T}_{surf} - T_{amb}\right)\right)}\right| + \left|\frac{\Delta l_c}{l_c}\right| +$$

$$\left|\frac{\Delta(\overline{T}_{surf} - T_{bulk})}{\overline{T}_{surf} - T_{bulk}}\right| + \left|\frac{\Delta l_c}{l_c}\right|$$

Equation 13

$$\left|\frac{\Delta Nu}{Nu}\right| = \left|\frac{0.005 \cdot 2000}{(4000 - 9 \cdot (50 - 20))}\right| +$$

$$\left|\frac{12 \cdot 1}{(4000 - 9 \cdot (30 - 20))}\right| + \left|\frac{0.001}{0.011}\right| +$$

$$\left|\frac{1}{50 - 20}\right| + \left|\frac{1.6 \cdot 10^{-4}}{257 \cdot 10^{-4}}\right|$$

$$\left|\frac{\Delta Nu}{Nu}\right| < 0.005 + 0.003 +$$

$$0.001 + 0.035 + 0.006 = 0.051$$

One should be aware of the fact that the temperature profiles, to some extend, will even itself out because of heat transfer through the perspex plate. However, the accuracy of the

average value of the heat flow will not be affected by this phenomenon.

## TWO EXAMPLES

Two examples are introduced to give an idea of the method possibilities.

### BURNER CHAMBER LINER TEST

With purpose too achieve a more uniform cooling and achieve a lower pressure loss, a model test of a burner chamber liner was set up. The burner liner is annular and made of steel, it is simulated in a plane model made of perspex and polycarbonat including 60 of 360 degrees. Scaled from engine conditions (~ 450 C cooling air 19 bar 60 kg/s) to model condition (~60 C 1,5 bar 2 kg/s). Geometrical scaled 1:2. The heatflux is simulated with heaterfoils glued on the inside of the linerwall. On the coolingside different cooling configurations are tested, such impingement cooling and ribs etc.
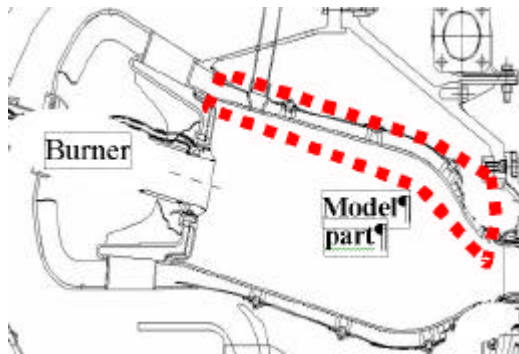


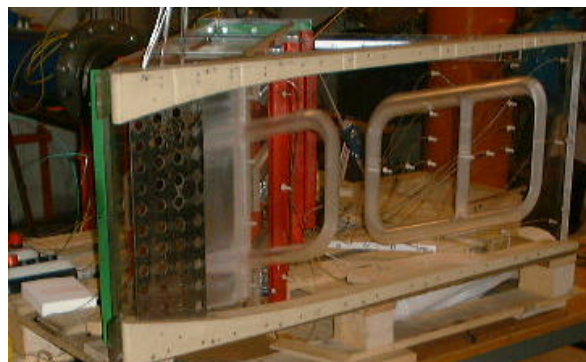Figure 3 Engine test section location



Figure 4 The model with dismounted heaterplate

As seen in Figure 4 the heaterplate area is large, 1000 by 450 mm, this will give large pressure forces on the plate. Therefore had the heaterplate to be reinforced with ribs on the ambient side. These ribs will influence the free convection on the surface, and therefor has the convection coefficient to be determined over the model height. It was found to be between 8 to 18 $w/m^2/K$.

In the start one important part was to determine the scaling parameters to be able to

rescale back to engine conditions. The scaling to engine conditions is done with respect to constant Nusselt number and as Nusselt number is a function of Reynolds and Prantel numbers. It can be written as:

$$ Nu = f\left( \mathrm{Re}^{\exp \mathrm{Re}}\, \mathrm{Pr}^{\exp \mathrm{Pr}} \left( \frac{T_{bulk}}{T_{wall}} \right)^{\exp T} \right) $$

Equation 14

There are also some parallel investigations to find out how or if the temperature quotient bulk wall affect the Nusseltnumber. Equation 14 can be rewritten if Nusselt engine to model is constant as:

$$ \alpha_e = \alpha_m n \frac{l_e}{l_m} * $$

$$ \left( \frac{\dot{m}_e n m_m}{\dot{m}_m m_e} \right)^{\exp \mathrm{Re}} \left( \frac{\mathrm{Pr}_e}{\mathrm{Pr}_m} \right)^{\exp \mathrm{Pr}} \left( \frac{T_{bulk,e}}{T_{wall,e}} \middle/ \frac{T_{bulk,m}}{T_{wall,m}} \right)^{\exp T} $$

Equation 15

Because of complicated geometry and the use of both impingement and rib cooling is no use in present the Nusseltnumber, since the characteristic length couldn't be distinct. For the smooth channel the Reynolds exponent show good agreement for different Reynoldsnumber for a higher Reynolds exponent than for a ribbed.

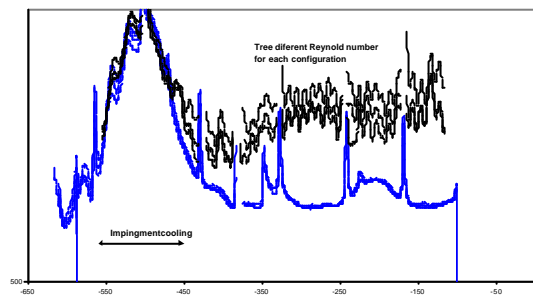

Figure 5 Alfa number along the chamberline,"higer" Reynold exponent

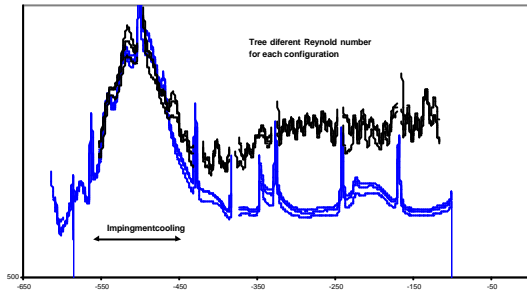For the impingement part is it the same Reynolds exponent as for the ribbed channel.

Figure 6 Alfa number along the chamberline,"lower" Reynold exponent

The high peeks seen in Figure 5 Figure 6 are from the vertical reinforcement ribs. The temperature read by the thermocamera will be to low there with will result in to high heattransfer coefficient. The horizontal reinforcement ribs are exempted in the average value.

The test is continued with different configurations where smaller parts are studied at a time. Below is one thermocamera picture from one off these tests. It's the impingement part that is shown, the actual geometry of the impingement plate is not shown here.
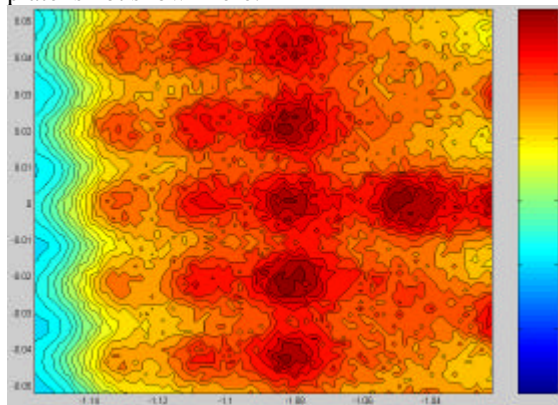


Figure 7 Impingement part of burner chamber liner wall

BLADE COOLING

One experimental investigation has been performed to verify and improve the cooling effectiveness on a turbine blade.

One test with smooth pipe was done to get a reference for the enhancement factor, A see Equation 16.

$$Nu = A \cdot \mathrm{Re}^{0.8}$$

Equation 16

Different rib height and rib distributions were investigated. The enhancement factor was found for different configurations.
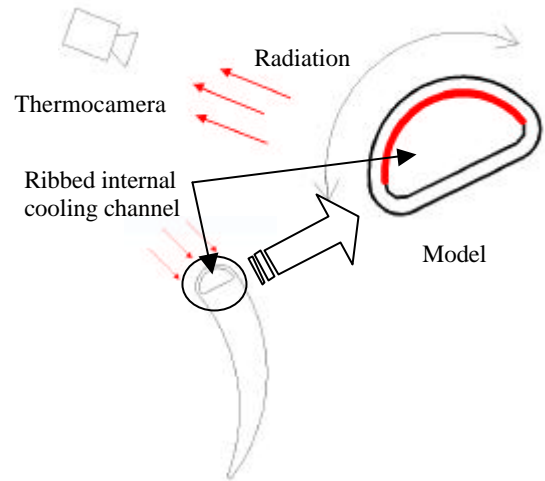


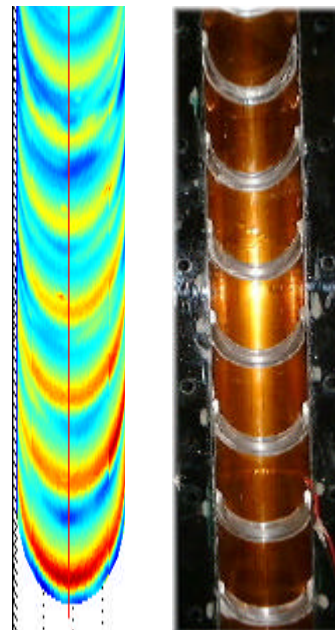Figure 8 Blade cooling example



Figure 9 Temperature distribution along cooling channel wall and picture of the inside of the heaterplate supplied with ribs.

The temperature picture in Figure 9 is in fact six different measurements. The test object is mounted on a traversing table so the camera is standing still and the model is moving between the measurements. Since the model is curved the post data process must recalculate the positions for each pixel. The measurements are then put together and a three-dimension picture is generated. This is useful when checking the temperature pictures so that each picture fit in to each other.

**EQUIPMENT**

TEMPERATURE MEASUREMENT

Thermocouple

Thermocouples of type K has a standard uncertainty of 0.4 % relative to temperature in [°C]. That means for the thermocouples (T/E K)

assembled on the heaterplate an uncertainty of 0.2 °C at a temperature of 50 °C. At air temperature (T/E K) has an uncertainty of 0.08 °C at 20 °C.

### Thermocamera

The thermocamera used in the laboratory is: Thermovision 900 system, Ref 3, AGEMA 900 SW/ST, detector type 2xInSb, serial scanning, Sterling cycle cooling. Spectral response 2.0 – 5.6 micron. Accuracy is ± 1°C or ±1%. Software ERIKA 3.1.

Lenses:
$10^{\circ}$, Field of view $10^{\circ}$ x $5^{\circ}$, minimum focus 1.0 m
$5^{\circ}$, Close up lens minimum focus 1.5 m

Photography 1 Thermocamera and control computer

The camera was installed with a band pass filter measuring the radiation in the spectrum from 3.6 μm to 4.1 μm. With intention to minimize the influence from surrounding incoming reflections the surface is painted galt black, and the measured area is supplied with a hood too insure no in falling radiation.

The camera-input parameter's for calculating the surface temperatures is emissity, object distance relative humidity and atmospheric temperature and reflected ambient temperature.

All object parameters exempt from the emissivity are given as input to the camera software. On the surface thermocouples are glued. These thermocouples are attached to a copperplate (approximately 1 $cm^2$) to ensure a uniform temperature over a small part of the surface. The copperplate is painted with the same paint as the heated plate.
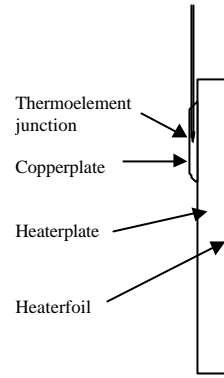


Figure 10 Mounting of thermocouple on the heaterplate

In the camera software an area is defined around the thermocouple, and an average temperature is calculated. This average temperature is then compared with the thermocouple temperature. The software then recalculates the emisivity. This emisitivity will contain the emisivity and any correlation's of the other given values.

## FUTURE DEVELOPMENT

Since the results are good and the technique is useful, and there are more test object waiting to be tested, there gone be future development in this technique. One of these developments is in the camera software. The camera is running under the operation system OS-9 and there fore data transfer online to for example Matlab is not possible. With a new program running under Windows NT data acquisition online at high frequency while be possible.

## REFERENS

Ref 1

A new test facility operating at near-engine conditions for testing of cooled GT components, Christer S Hjalmarsson, Jens F Jensen, Frank G Rubensdörffer, Ulf R Rådeklint, ABB STAL (ALSTOM Power Finspong) Department of Research & development, s-612 82 Finspong Sweden.

Ref 2

Experimental Investigation of Impingement Cooling with Turbulators or Surface Enlarging Elements
Johan Persson, Thesis work, Dep. of Heat and Power Engineering Lund Institute of Technology, ABB Alstom power Dep. of Rescharch and development

Ref 3

Agema Infrared Systems AB
Rinkebyvägen 19 Box 3
182 11 Danderyd, Sweden