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CONVECTIVE HEAT TRANSFER COEFFICIENTS IN TURNING

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Abstract. *Methodology for determination of convective heat transfer coefficients for specific conditions existing in turning was developed. Experiments on workpiece cooling conducted on a lathe provided reference temperature data for a model of a cylindrical workpiece, which was solved for temperature using a Control-Volume Finite Difference code. An iterative optimization procedure minimized the difference between computer prediction and temperature measurements. Heat transfer coefficients were obtained for various convective boundary conditions existing on a workpiece when cooling in air and in coolant. Cooling characteristics calculated using these heat transfer coefficients showed good agreement with the experiment. Presented approach can be used to obtain the convective heat transfer coefficients for studies on modelling thermal behaviour of a workpiece in other conditions.*

Keywords*: Turning, workpiece, convective cooling, heat transfer coefficients.*

1. INTRODUCTION

In recent years there has been a growing interest in modelling of metal cutting process. It has been accompanied also by the interest in modelling of thermal behaviour of the workpiece. Thermal expansion affects the machining accuracy, hence the latter interest can be attributed to the demand for higher accuracy and to the effort to improve the ability of its prediction.

To predict the machining error caused by the thermal expansion of the workpiece, it is necessary to know the convective boundary conditions. These are expressed by the convective heat transfer coefficients. The few earlier studies which dealt with the thermal expansion of the workpiece used values of these coefficients from published empirical correlations, assuming their validity for this case. Although much has been published on the stationary cylinder in the crossflow, little has been reported on the rotating cylinder in a quiescent or in turbulent air (e.g. Mills, 1999). Similarly, there are many works on the subject of cooling by jets (e.g. Pelletier, 1984, Goldstein and Franchett, 1988, Journeaux, 1990). However, the literature search did not reveal any convection data for water cooling in conditions corresponding to turning.

Cooling of a workpiece on a lathe is affected not only by the usual factors such as exposure of its surfaces and their velocities, but also by the presence of the chuck and the jet of the cutting fluid splashing on the workpiece while moving with the tool.

The spinning chuck is introducing the turbulence of the air, the effect of which extends nonuniformly over the length of the workpiece. The chuck is also influencing directly the thermal state of the workpiece through the contacting surfaces of the jaws; depending on the conditions, the chuck can act as a heat sink or heat source.

The cooling effect of the surface area under the impinging coolant jet is not uniform and therefore the heat transfer coefficient in this zone should be treated as a distribution. The initial conditions of cooling at the beginning of the pass, when the jet flashes also on the side face of the workpiece, poses additional challenge in determination of heat transfer coefficients.

The conditions existing in turning require that the values of convective heat transfer coefficients needed for modelling of thermal behaviour of the workpiece are determined for these specific conditions. This is the objective of the study reported in this paper.

2. PROBLEM FORMULATION

A cylindrical workpiece is rotating on a lathe, cooling from a pre-heated state. Subject of investigation are convective boundary conditions describing the process of cooling.

This situation is represented by a solid cylinder with initial non-uniform temperature field, for which the governing partial differential equation of the transient temperature field is given by the general conduction equation for the axi-symmetric case. The unknown convective heat transfer coefficient to be determined is contained in the expressions of the boundary conditions on radial and axial coordinates, which apply to the governing equation.

Mathematical formulation follows a method described by S. V. Patankar (1980), in which the discretization is based on energy balance on a control volume. The partial differential equation governing the case under consideration is given by:

$$\rho c \frac{\partial T}{\partial t} = \frac{k}{r}\frac{\partial}{\partial r}\left(r \frac{\partial T}{\partial r} \right) + k \frac{\partial^2 T}{\partial z^2} + S \qquad (1)$$

which is the conduction equation for the axi-symmetric case, where: *r, z* - radial and axial coordinates of the cylinder; *c, k,* ρ, *t* - specific heat, thermal conductivity, density and time, respectively. It should be noted that the volumetric source term, *S*, is zero in the physical problem being considered here. However, it is retained in Eq. (1) for the sake of generality, and, more importantly, for flexibility in the implementation of a finite volume solution method. With the initial condition: $T = T(r, z)$, the following boundary conditions apply to the governing equation:

$$-k \frac{\partial T}{\partial r} = h(T - T_\infty) \qquad \text{at} \quad r = R$$

$$-k \frac{\partial T}{\partial z} = h(T_\infty - T) \qquad \text{at} \quad z = 0$$

$$-k \frac{\partial T}{\partial z} = h(T - T_\infty) \qquad \text{at} \quad z = L$$

where h is the unknown convective heat transfer coefficient to be determined for different cooling conditions, which may exist in different zones.

In numerical formulation derivatives in the governing equation are replaced by finite differences (CVFD). The implicit discretized form of Eq. (1) is:

$$a_P T_P = a_E T_E + a_W T_W + a_N T_N + a_S T_S + b \qquad (2)$$

where a_E, a_W, a_N, a_S represent the conductance between a particular node P and its neighbors on the East, West, North and South. The constant b is defined as:

$$b = S_C \Delta V + a_P^0 \ T_P^0$$

where

$$a_P^0 = \frac{\rho c \, \Delta V}{\Delta t}$$

which appears in the coefficient a_P

$$a_P = a_E + a_W + a_N + a_S + a_P^0 - S_P \Delta V$$

Convective boundary conditions are introduced through the linearized "source term", $\bar{S}$, which is expressed as the average value of S over the control volume, $\bar{S} = S_C + S_P T_P$, where:

$$S_C = \frac{T_\infty hA}{\Delta V} \quad \text{and} \quad S_P = \frac{hA}{\Delta V}$$

A,V and T_∞ are convective area, volume and ambient temperature, respectively.

After calculating the coefficients for each node, the associated equation is obtained. Assembling all the discretized equations together results in a set of linear simultaneous algebraic equations. To solve this set of equations a line Gauss-Seidel iterative procedure has been used (Patankar, 1980). The computed nodal values of temperature are denoted as T_{calc}. The details of the formulation can be found in the work by Arenson (1997).

The workpiece was represented by a grid corresponding to its physical dimensions (Fig. 1) and was assigned typical material properties of steel: density, ρ = 7800 kg/m^3; specific heat, c = 473 j/kg^0C, thermal conductivity, k = 43 W/m^0C. The density of grid was increased for areas of higher temperature gradient. Selected nodes match the location of the thermocouples in the experiment.

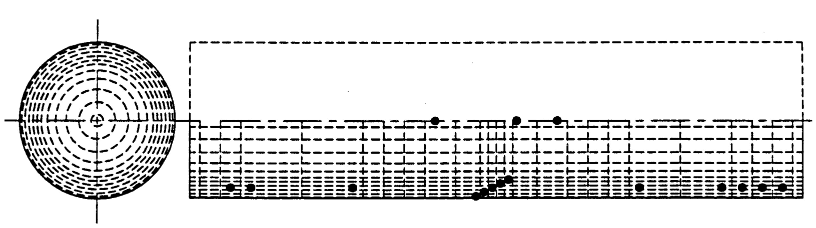


Figure 1 - Discretization of the model; dots indicate nodes corresponding to the thermocouple location. Denser grid was used for impinging coolant jet.

3. SOLUTION PROCEDURE

As the objective is to determine the convective heat transfer coefficient, an overall iterative procedure which optimizes the value of h to obtain the minimum difference between the computed and experimental temperature fields was developed. Based on such h a good model will provide the cooling curves matching closely the corresponding experimental curves over the whole time domain of cooling. The difference was defined as:

$$DIFF = \frac{\left|T_{calc} - T_{exp}\right|}{m \cdot n} = \frac{1}{m \cdot n} \sum_{i=1}^{m} \sum_{j=1}^{n} \left|T_{calc_{(i,j)}} - T_{exp_{(i,j)}}\right| \tag{3}$$

where m is the number of thermocouples used in experiments and n is the number of measurements per thermocouple. To find the minimum difference, the Matlab f_{min} routine, implementing a standard minimization algorithm, used Eq. (3) as its objective function. The *DIFF* was calculated by the Fortran code developed in this work, which solved the thermal field and compared it to the experimental one. Following the guidelines suggested by Kirsch (1996), all readings from all thermocouples and all corresponding nodes were used to calculate the average of the absolute difference between those fields.

The total problem of finding convective boundary conditions is divided into the following sub-problems: cooling in air – workpiece insulated and noninsulated from the chuck, and cooling by an impinging jet with workpiece noninsulated from the chuck.

4. EXPERIMENT

The experiments to determine the cooling behaviour of a workpiece were performed on a lathe. The experimental setup is shown schematically on Fig. 2. A cylindrical steel workpiece of 75 *mm* diameter and 300 *mm* length was pre-heated and then spun. The thermocouples strategically located on various depth along the workpiece, measured the temperature while the workpiece was cooling. The thermocouple signals were individually amplified by the chip-based amplification circuit located inside the spindle, and then carried through the slip-ring assembly to the data acquisition unit and a personal computer. The workpiece was clamped in the chuck and supported at the tailstock by a center. In the tests with insulated chuck, the workpiece was clamped through an insulating ring.

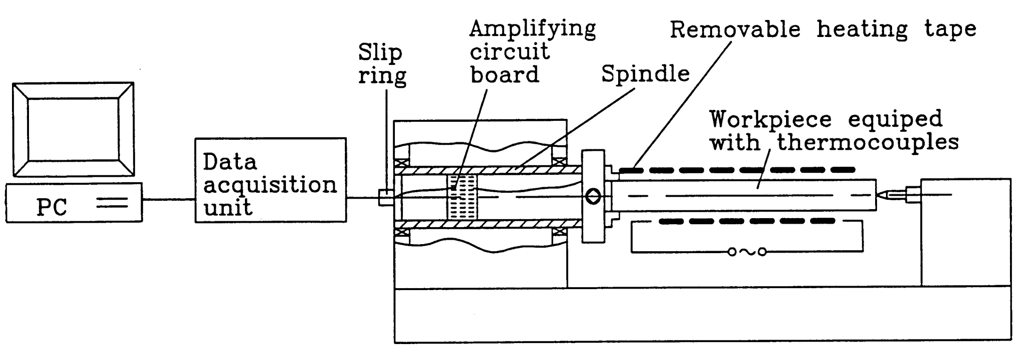


Figure 2 - Schematic of the experimental setup.

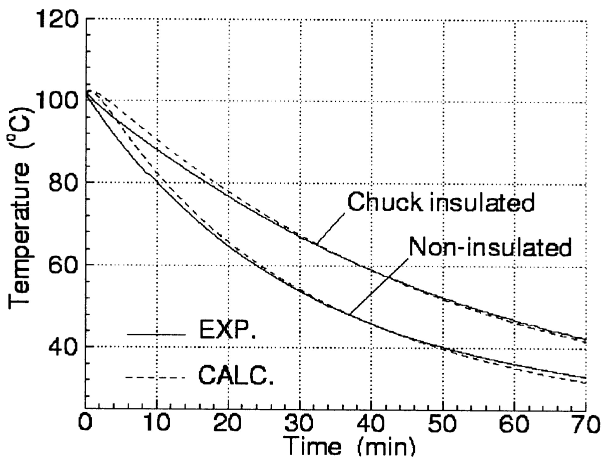


Figure 3 - Experimental and calculated cooling curves showing the effect of the chuck; cooling in air, surface velocity v =137 *m/min.*

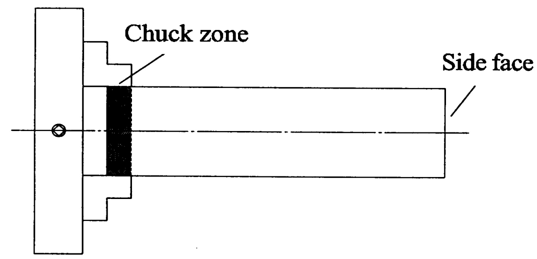


Figure 4 - Chuck zone shown on workpiece clamped in the chuck; tailstock center now shown.

The workpiece was heated by a flexible electric heating tape wound around its entire length. Once the predetermined temperature of 65^{o}-130^{o}C was reached, the tape was unwound and the spinning begun. The tests were stopped when the temperature dropped below 40^{o}C. Temperature was recorded at scanning intervals of 20 *sec.* for tests in air, and 2 *sec.* in coolant tests. The effect of two surface velocities was investigated : 137 *m/min* and 88 *m/min*.

The tests with coolant were carried out by an impinging jet from a nozzle, covering about 1/3 of the workpiece length. The coolant flow rate was 4,4 ℓ */min.* With the nozzle diameter d = 12.7 *mm* and kinematic viscosity, v = 9.82 x 10^{-7} $m^2/sec.$ based on water at 21^{o}C, the Reynolds number of the jet was Re_j = 7500.

The results were validated in cutting experiments conducted in air and with coolant. All experiments are described in detail in Reports by Caron *et al*. (1993), Caron and Arenson (1994) and Beauchemin and Agnerian (1995).

5. RESULTS AND DISCUSSION

5.1 Convective heat transfer coefficients when cooling in air

Results for both cases of workpiece insulated from the chuck, and non-insulated are presented on Fig. 3. The experimental and calculated cooling curves are superimposed, indicating a good match. The curves are for the thermocouple and corresponding node on the axis located at the distance of approximately l/3 length from the chuck. Optimization procedure provided the heat transfer coefficient value of h = 22.6 $W/m^2\ ^oC$ with the minimum average temperature difference less than 1^{o}C for the case of cooling with the insulated chuck. Side faces were assigned heat transfer coefficient value changing linearly from 6 $W/m^2\ ^oC$ at the axis of the workpiece up to a value found on the circumference. The value of 6 $W/m^2\ ^oC$ corresponds to natural convection (Holman, 1997).

The effect of chuck on acceleration of cooling is clearly visible on Fig. 3. The chuck acts as fins do, pulling heat towards its cooler surface spinning in turbulent air with the surface velocity higher than that of the workpiece. This effect was accounted for by the introduction of the equivalent thermal contact coefficient to the 25 *mm* wide “chuck zone”

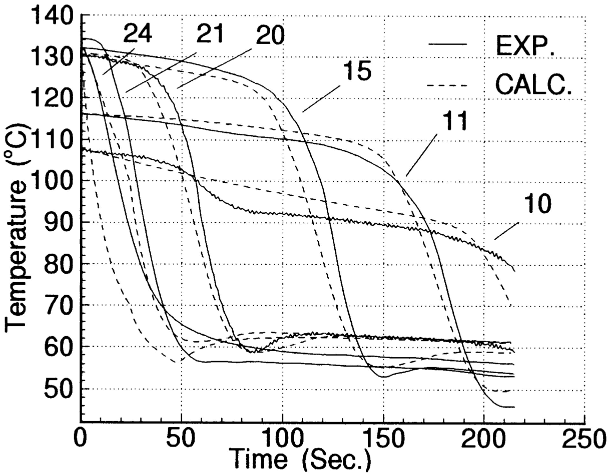


Figure 5 - Cooling with impinging jet moving along the workpiece with feed of 0.12 *mm/rev* (1.16 *mm/s*).

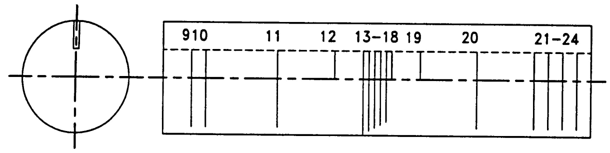


Figure 6 - Identification of thermocouples. The chuck is on the left. The coolant jet is moving from the right to the left.

shown on (Fig. 4). It represents the heat transfer through the contact interface between the workpiece and the jaws as well as the heat transfer from the workpiece surface between the jaws. The optimization procedure found that a value of 240 $W/m^2\,^oC$ is appropriate for this coefficient for the conditions of the experiment.

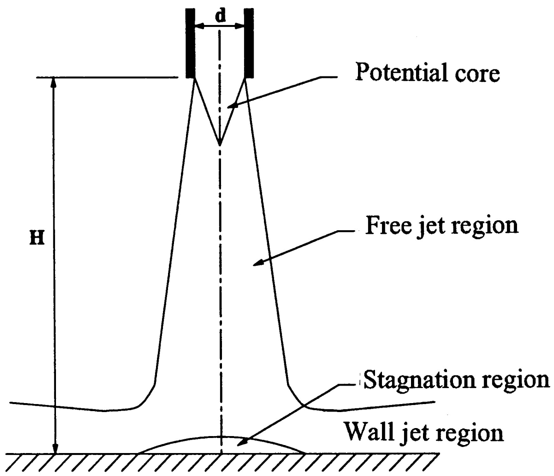
The heat sink effect of the chuck is less pronounced in case of a coolant, as the effect of the latter is dominant. Experiments have shown that in such a case the chuck can act even as a heat source.

In addition to surface velocity of 137*m/min*, the effect of lower velocity of 88 *m/min* on cooling in air with the chuck insulated was investigated. As expected, the workpiece cools slower at lower speed; the heat transfer coefficient obtained for this case is 16.8 $W/m^2\,^oC$.

5.2 Convective heat transfer coefficient under a coolant jet

During the turning operation the coolant jet is travelling with the tool along the workpiece. This situation was represented by an experiment in which the pre-heated spinning workpiece was cooling while the coolant supply nozzle was moving from the tailstock to the chuck. The impinging jet zone of 1/3 workpiece length was thus covering consecutive regions of the surface, the remaining areas exposed to air. The thermocouples provide the record of the temperature change as the coolant jet passes by (Fig. 5). Their location and numbers are shown in Fig. 6. As can be seen, the curves recorded from the thermocouples at the tailstock side exhibit steep slope of fast cooling immediately (TC 24) or shortly after the beginning of the pass (TC 21). The temperature plots from the subsequent thermocouples start with longer and longer slowly decreasing curves, at the end of which the steep slope occurs. This pattern corresponds to initial slow cooling in the air, and then, when the coolant jet arrives in the vicinity of the particular thermocouple, the fast cooling takes place. Once the jet passes the thermocouple, cooling in the air resumes and the temperature slopes decline again slowly.

One can notice that there is a difference in the transition from the fast to slow cooling between the first (TC 24) and consecutive thermocouples. The record for TC 24 shows a slow monotonic transition, while TC 20 and TC 15 display a dip at the end of fast



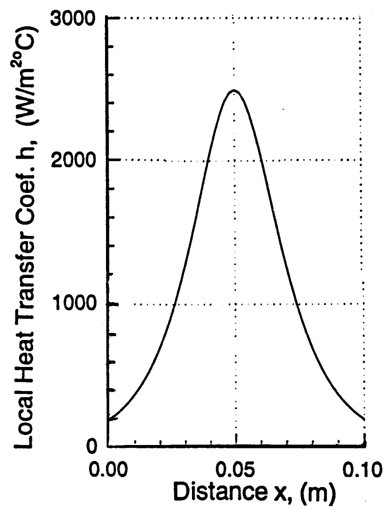


Figure 7 - Characteristic features of an impinging jet.

Figure 8 - Distribution curve of the heat transfer coefficient under a jet.

cooling, from which the temperature recovers and then continues at air cooling speed. This interesting behaviour could be explained by the heat transfer to local cooler skin-deep areas from the warmer core, unaffected yet by the coolant. TC 24 does not exhibit this dip because the core is cooled through the side face, which is flushed by the coolant at the beginning of the pass.

Cooling with a moving jet is a case where four cooling zones co-exist : the coolant zone, air on its side(s), air on the side face(s) and the chuck zone.

With the knowledge (from the previous experiments) of the heat transfer coefficients in all but the coolant zone, the optimization procedure was applied to find the value of the heat transfer coefficient in the zone of impinging jet.

The characteristic features of an impinging jet are shown on Fig. 7. The diameter of the nozzle, its distance from the surface and the angle of the jet axis with respect to the surface, are the geometric parameters affecting the cooling effectiveness of the jet. In general, the maximum value of the heat transfer coefficient is at the intersection of the jet axis with the surface, perpendicular to each other. From there, the heat transfer coefficient decreases sidewise. The cooling curves obtained from experiments with a stationary jet indicate the agreement with this.

A hyperbolic cosine function was found to produce the best match with the experimental results. As a function of axial coordinate z the distribution of the heat transfer coefficient is :

$$h(z) = \frac{h_{max}}{cosh\left(\frac{c_1\left(z - \ell/2\right)}{\ell}\right)}$$

Bell-shaped distribution (Fig. 8) was scaled to fit the width of the coolant zone ℓ = 9.5 *cm*. Maximum heat transfer coefficient was found to be h_{max} = 2500 $W/m^2\,^{o}C$ and the coefficient c_1 = 5.5 for the conditions of the experiment.

The values of the heat transfer coefficients for the other zones used in the optimization procedure were those found in previous computations:

$$h_{air} = 22.6\ W/m^2\ {}^oC,$$
$$h_{chuck} = 240\ W/m^2\ {}^oC$$

and side faces $h_{sf} = 6 - 22.6\ W/m^2\ {}^oC$ varying linearly from $r = 0$ to $r = R$ (value at $r = 0$ was assumed).

The moving jet zone was modelled by shifting it along the workpiece at the feed velocity. The calculation time-step was equal to the time of crossing the control volume corresponding to the actual velocity of the jet. The boundary conditions (the zones) were shifting continuously with each time-step.

As can be seen from Fig. 6 the computed curves provided by the model follow quite closely the experimental ones. To portray the continuous character of the change from the air to the full cooling conditions of impinging jet a transitional values of *h* were required. The length of the coolant zone and the fine mesh are critical for the accuracy of modelling. Special treatment was applied to the beginning of the pass, where the jet is partially impinging on the side face and partially on the cylindrical surface ; it was necessary to reduce the value of h_{max} of the jet at that time.

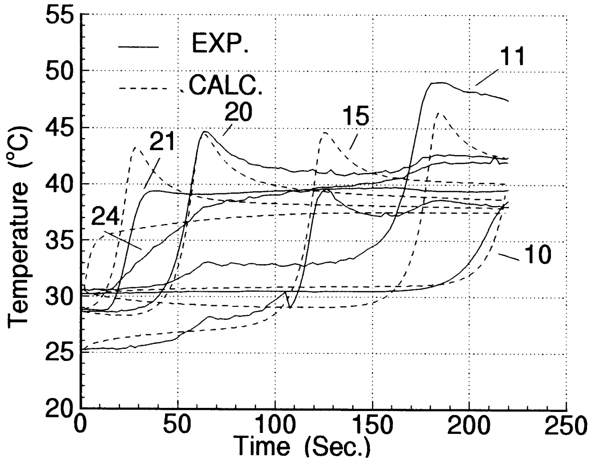
The dip in the temperature at the end of the passing coolant zone and the subsequent recovery portrayed by the computed curves are indicative of the capability of the modelling procedure presented in this work.

5.3 Validation of the results

The validation of the heat transfer coefficients determined for various cooling zones was conducted by using them to calculate the temperature of the workpiece during cutting, both in air and with coolant. The closeness of curves computed for the location of thermocouples, to those obtained experimentally from the thermocouples, is an indication of the validity of the *h* values obtained.

To calculate the temperature of the workpiece with all convective boundaries already known the same iterative optimization procedure described in section 3 was used, this time to find the strength of the heat source which would provide the minimum difference between the computed and experimental temperature fields. This value was then verified by comparing it to the values calculated using specific energy and cutting conditions, according to Shaw (1986) and Boothroyd and Knight (1989), and considering the amount of heat entering the workpiece. The heat source was modelled as a thin ring of $(f \cdot d)$ cross-section moving along the workpiece, which is justified by the relatively high rotational velocity (Caron *et al*. 1993).

As can be seen from Fig. 9, which represents temperatures when cutting in air, the agreement between the computed and experimental curves is reasonable except TC 24 at the beginning of the cut which exhibits in the experiment slower initial rise of the temperature but continues rising due to the internal conduction, and TC 21, which just starts exhibiting a peak. TC 15 half way along the workpiece shows lower peak temperature recorded, in reality however, this is similar temperature rise from the initial temperature, as TC 20. Computed curves show consistent increase of the temperature of consecutive peaks and a remarkable agreement in slopes of temperature rise when the heat source approaches the thermocouple. The cooling in the air is slow and, as the experiment shows, it is subsequently counterbalanced by heat conducted from the regions cut afterwards (TC 20 and TC 15).



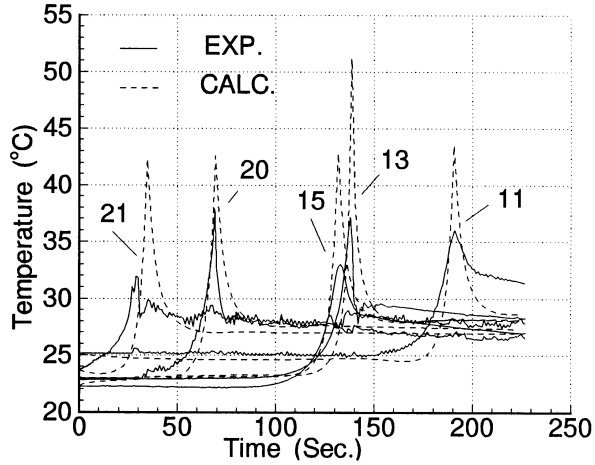


Figure 9 - Workpiece temperature during cutting in air; d = 2*mm*, f = 0.12 *mm/rev*. v = 137 *m/min*. TC numbers as in Fig. 6.

Figure 10 - Workpiece temperature during cutting with impinging coolant jet. Cutting conditions as in Fig. 9.

It is interesting to note that the highest initial temperature is at the chuck end of the workpiece (TC 10 and TC 11), slightly lower at the tailstock (TC 24, TC 21, TC 20) and the lowest is in the middle of the workpiece length (TC 15). Higher temperature at the chuck could be explained by the heat remaining from the previous cut which ended there, or by a warmer chuck. The tailstock end might have been affected by the friction heat from the center.

Cutting with coolant (Fig. 10) produced interesting results in the form of sharp peaks from the lower temperature base, indicating distinctive effect of fast heating by the approaching heat source (cutting tool) and nearly as fast cooling, but to much lower workpiece temperature than in the case of air. Uniformly higher computed peaks are an indication that the response time of the thermocouples was too slow to record the true temperature when the tool was passing by. The highest peak of the computed temperature represents location of TC 13, which is on the very surface of the workpiece.

6. CONCLUSIONS

1. The convective heat transfer coefficients for cooling of a cylindrical workpiece in air and in water-based coolant were obtained for specific conditions in turning through a numerical iterative optimization procedure which uses as a reference the temperature measurements from the cooling experiments on a lathe.
2. To describe the cooling conditions in turning, separate values of the heat transfer coefficient are required for various cooling zones of the workpiece. A hyperbolic cosine function for the distribution of heat transfer coefficient values was found to best represent the cooling by an impinging coolant jet.
3. The effect of the chuck can be described by the equivalent thermal contact coefficient, which accounts for both heat transfer through the contact interface with the jaws and convection from the surfaces between them.
4. Presented methodology could be used to obtain the convective heat transfer coefficient for modelling of thermal behaviour of a workpiece also in other conditions in turning.

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