Journal of Materials Processing Technology, 41 (1994) 125–142
Elsevier

A study of the heat-transfer coefficient as a function of temperature and pressure

Z. Malinowski[1], J.G. Lenard and M.E. Davies[2]

Department of Mechanical Engineering, University of Waterloo, Waterloo, Ontario, Canada N2L 3G1

(Received November 1, 1992, accepted in revised form February 15, 1993)

Industrial Summary

The accuracy of thermal–mechanical models of the hot/warm forging process depends on the proper description of the boundary conditions. At the hot workpiece–cold die interface this requires knowledge of the heat-transfer coefficient.

A method for the determination of the heat-transfer coefficients in bulk metalforming processes is presented. The technique consists of two steps. The first involves measuring the temperature distributions within two dies, one of which simulates the cold forming tool and the other the hot workpiece. The dies are brought into contact under closely controlled conditions. The second step makes use of the finite-element simulation of the resulting heat-transfer problem. The interfacial heat-transfer coefficient is treated as an unknown variable and is determined using a non-linear optimization technique, forcing the measured and the computed temperature distributions to be as close as possible. Examples of the interfacial heat-transfer coefficients as functions of the time of contact are given for different surface temperatures and interfacial pressures for 303 stainless steel. An empirical relationship is then developed, giving the coefficient of heat transfer as a function of time, temperature and interfacial pressure. The predictive capability of the relationship is substantiated by comparing its output to data obtained with the original values of the coefficient. The empirical relationships may be of use in the planning of hot-forming processes when the contact times are in excess of half of a second.

Notation

A	parameter defined by eqns. (23) or (27)
B	parameter defined by eqns. (24) or (28)

Correspondence to: Professor J.G. Lenard, Department of Mechanical Engineering, University of Waterloo, Waterloo, Ontario, Canada N2L 3G1.
[1] Permanent address: University of Mining and Metallurgy, Al. Mickiewicza 30, 30-059 Kraków, Poland.
[2] Present address: Long Manufacturing Company Ltd., Oakville, Canada, L6K 3E4.

0924-0136/94/$07.00 © 1994 Elsevier Science B.V. All rights reserved.
SSDI 0924-0136(93)E0061-K

B_a boundary of an element in contact with the atmosphere
B_c boundary of a cold die element in contact with the frame
B_h boundary of a hot die element in contact with the frame
B_p boundary of a cold and a hot die element in contact with one another
c_p specific heat (J/kg K)
h cold/hot die interface heat-transfer coefficient (W/K m^2)
h_a hot die/air or cold die/air interface pseudo heat-transfer coefficient (W/K m^2)
h_c cold die/frame interface pseudo heat-transfer coefficient (W/K m^2)
h_h hot die/frame interface pseudo heat-transfer coefficient (W/K m^2)
$\dot{h}$ rate of change of the hot die/cold die interface heat-transfer coefficient (W/K s m^2)
$J(\dot{h})$ integral defined by eqn. (12)
k thermal conductivity (W/mK)
l_r, l_z direction cosines of the outward normal to the boundary curve
P normal pressure (MPa)
$\bar{p}$ non-dimensional quantity defined by eqn. (26)
Q parameter defined by eqns. (25) or (29)
q_a heat flux at the hot die/air or cold die/air interface (W/m^2)
q_c heat flux at the cold die/frame interface (W/m^2)
q_h heat flux at the hot die/frame interface (W/m^2)
q_p heat flux at the hot die/cold die interface (W/m^2)
r, θ, z cylindrical co-ordinates
S surface of an element
t, t_0 time and initial time, respectively (s)
T temperature (°C)
T_a ambient temperature (°C)
T_d cold die surface temperature at the hot/cold die interface (°C)
T_i average temperature of thermocouples # 5, 6 and 7 (°C)
T_m average temperature measured by the thermocouples # 5, 6 and 7 (°C)
T_0 initial temperature (°C)
T_w hot die surface temperature at the hot/cold die interface (°C)
$\bar{T}$ non-dimensional quantity defined by eqn. (18)
ρ density (kg/m^3)
α linear expansion coefficient (m/m)
Δt time increment (s)

1. Introduction

Heat transfer plays an important role in metal-forming processes, where both the workpiece and tool behaviour are affected strongly by the temperature fields. The heat flux at the workpiece/tool interface is commonly assumed to be governed by the interface heat-transfer coefficient. The boundary conditions at the interface are usually formulated in terms of the heat-transfer coefficient leading to numerical solutions to a great number of practical, industrial problems. The solutions, of course, involve the integration of the quasi-harmonic heat-transfer equation.

While the solution algorithms are now well established [1–5], relatively little work has been reported regarding procedures that lead to an estimate of the interface heat-transfer coefficient in bulk-forming processes. There are essentially two approaches by which the heat-transfer coefficient may be determined. One of these is the attempt to choose h such that calculated and measured temperature distributions will agree closely. The other is to use the experimentally established time–temperature profiles to estimate the temperatures of the two contacting surfaces and use the definition of the heat-transfer coefficient as the ratio of the heat flux and the temperature difference of the surfaces. Naturally, both of the methods have limitations. In the former, success depends on the quality, accuracy and rigour of both the measurements that are to match the predictions of a model and those of the model itself. The latter is also dependent on the measurements in addition to the technique of determining the surface temperatures and, hence, their difference.

Among that which employed the first method is the work of Chen et al. [6]. The authors measured the time–temperature profiles during the hot rolling of aluminum strips, using four thermocouples, two of which were embedded in the strip with the two others being located in grooves machined on the surfaces. The heat-transfer coefficient at the roll/strip contact surface was then inferred by matching the surface temperatures, calculated by a thermal–mechanical model of the process, to the data collected by the thermocouples located in the grooves. The conclusion emerging from the study indicated that h is not constant along the arc of contact and that it is a function of the roll pressure distribution, the interface oxide layer, the surface roughness and, if a lubricant is used, the surface chemistry. The heat-transfer coefficient was found to vary from a low of $10\,000\,\mathrm{W/K\,m^2}$ to $54\,000\,\mathrm{W/K\,m^2}$ for a variety of reductions. A similar approach was described by Sellars [7], who wrote that ..." a trial-and-error procedure of fitting experimental cooling data enabled values the heat-transfer coefficient to be determined as a function of reheating temperatures and lubrication conditions". Dadras and Wells [8] also determined h by a trial-and-error procedure, using the predictions of a two-dimensional finite-difference model.

Semiatin et al. [9] developed a technique by which the heat-transfer coefficient can be estimated based on the measurement of the die temperature brought into contact with the deforming workpiece. A one-dimensional analysis, developed by Klafs [10], and a finite-difference model were used to derive calibration curves in terms of temperature differences of the die and the workpiece, from which the heat-transfer coefficients were determined. The technique proposed and described in [9] led to constant values of the heat-transfer coefficient for the given initial conditions. The technique was used to develop further data, applicable for the process of hot forging, in [11]. Tool-steel dies were used in experiments in which aluminium alloy rings were compressed. The heat-transfer coefficients at the interfaces were determined for a large range of parameters: temperatures up to 420° C and pressures up to 150 MPa were employed.

A more direct experimental technique was followed by Karagiozis [12] and Pietrzyk and Lenard [13, 14]. The method involved the hot rolling of slabs, instrumented with several thermocouples, monitoring their output during the pass and inferring the surface temperatures of the slabs in the contact zone by extrapolation. The original definition of the heat-transfer coefficient in terms of the heat flux and the difference of the temperatures of the contacting surfaces was then used to determine the magnitude of h.

As mentioned already, it is realized that both approaches have drawbacks. It is also understood, however, that accurae values of the heat-transfer coefficient – one of the boundary conditions for the evaluation of variables during hot forming – are critical if accurate and consistent modelling of a metal-forming process is required. In the present work the search for h is continued. The purpose is to develop a procedure which will lead to the interface heat-transfer coefficient and which would use as few assumptions as absolutely necessary.

An objective technique for the determination of the heat-transfer coefficient at the contact surface of a pair of dies is then presented in this study. The time–temperature profiles are measured in the experimental hot die/cold die set and a finite-element method is used to simulate the dynamic heat-transfer problem. The heat-transfer coefficient is determined by enforcing the equality of the measured and computed values of temperatures at particular locations in the die. Finally, an empirical relationship giving the heat-transfer coefficient as a function of time, pressure and temperature is developed. The relationship is shown to give values of h which, when used to calculate the distribution of the temperatures, agree with the measurements reasonably well.

2. Experimental procedure

The schematic set-up of the experiment is given in Fig. 1. As shown, the two stainless steel 303 dies, chosen for their low oxidation rate, of 25.4 mm diameter each, are instrumented with four Type K thermocouples of 1.6 mm outside diameter, with INCONEL sheaths and exposed beads, located 2, 4, 10 and 25 mm from the contact surface and embedded to a depth of 10 mm. The dies are connected to the water-cooled heat exchangers of a servohydraulic testing system. One of the dies is heated to a pre-selected temperature in a split, openable furnace. When the desired temperature is reached, the furnace is removed and the cold die is brought into contact with the hot die under closely controlled conditions. The velocity of approach is selected at 0.83 mm/s. After contact is made, the control of the closed-loop testing system is switched from "STROKE" to "LOAD" automatically and the load is increased, at a rate of 110 MPa/s, to a pre-determined level and kept there until the experiment is completed. During the approach of the dies and throughout the test the output

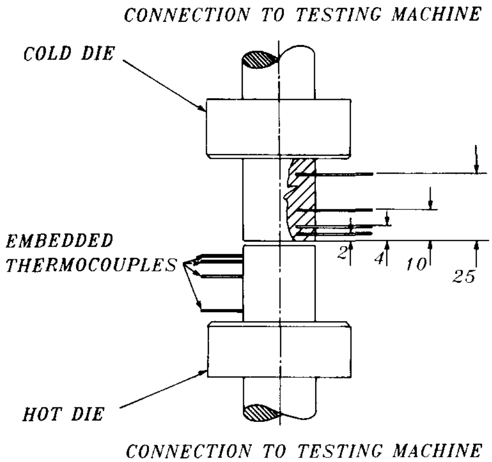


Fig. 1. Schematic diagram of the experiments.

of the eight thermocouples is monitored using a computer-based data acquisition system. The initial temperature of the hot die is varied from 300 to 900°C, whilst the interfacial pressures are varied from 30 to 90 MPa. The schedule of the experiments is as follows. First, the 300 °C tests are conducted, at 30, 50, 70 and 90 MPa. Then the temperature is increased to 500°C and the above sequence is repeated. The experiments at 700 and 900°C follow, with the low pressures first, the higher pressures later. The same dies are used throughout the program; the contact surfaces are left unchanged. The initial surface roughness of the cold die was 0.42 μm centreline average and that of the hot die was 0.61 μm.

3. Formulation of the finite-element method

The experiments were designed to allow the 3-D heat-transfer problem near the interface to be formulated in a cylindrical co-ordinate system. The finite-element representation of the two dies is presented in Fig. 2, with the locations of thermocouples marked by solid squares. A network of 10 × 38 4-node linear elements in the r- and z-directions, respectively, is employed in the discretization. At the contact surface a very fine mesh is used with the size of elements in the z-direction equal to 0.05 mm. The interface is assumed to have no heat capacity of its own, the heat flow through it being given by

$$q_p = h(T_w - T_d) \tag{1}$$

where T_w and T_d are the surface temperatures of the hot and the cold die, respectively. Both dies lose heat by natural convection and radiation to the

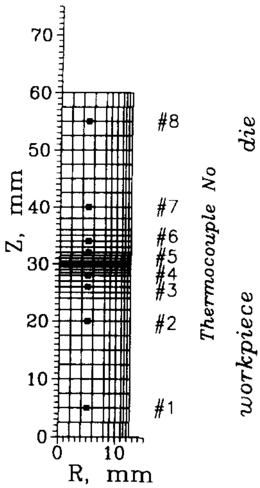


Fig. 2. Finite-element discretization of the workpiece and die used in the experiments, indicating the locations of thermocouples.

atmosphere. To account for these on a boundary surface which is in contact with the atmosphere, the heat flux is defined by

$$q_a = h_a(T - T_a) \tag{2}$$

where T_a is the temperature of the surroundings and h_a represents a pseudo heat-transfer coefficient.

The dies were connected to the frame of the testing machine by water-cooled heat exchangers, without any thermal insulation. The heat fluxes between the cold die and frame and the hot die and frame are then expressed as

$$q_h = h_h(T - T_0) \tag{3}$$

$$q_c = h_c(T - T_a) \tag{4}$$

where T_0 is the initial temperature of the hot die and h_h and h_c represent the pseudo heat-transfer coefficients at the hot die/heat exchanger and cold die/heat exchanger interfaces, respectively.

Having the heat fluxes on the boundary surface defined by eqns. (1)–(4), the distribution of the temperatures of the die set may be determined by solving the quasi-linear parabolic equation

$$\frac{\partial}{\partial r}\left(kr\frac{\partial T}{\partial r}\right) + \frac{\partial}{\partial z}\left(kr\frac{\partial T}{\partial z}\right) - r\rho c_p\frac{\partial T}{\partial t} = 0 \quad \text{on } S \tag{5}$$

subject to the boundary conditions:

$$kr\left(\frac{\partial T}{\partial r}l_r+\frac{\partial T}{\partial z}l_z\right)=rq_{\mathrm{p}} \quad \text{on } B_{\mathrm{p}} \tag{6}$$

$$kr\left(\frac{\partial T}{\partial r}l_r+\frac{\partial T}{\partial z}l_z\right)=rq_{\mathrm{h}} \quad \text{on } B_{\mathrm{h}} \tag{7}$$

$$kr\left(\frac{\partial T}{\partial r}l_r+\frac{\partial T}{\partial z}l_z\right)=rq_{\mathrm{c}} \quad \text{on } B_{\mathrm{c}} \tag{8}$$

$$kr\left(\frac{\partial T}{\partial r}l_r+\frac{\partial T}{\partial z}l_z\right)=rq_{\mathrm{a}} \quad \text{on } B_{\mathrm{a}} \tag{9}$$

and the initial conditions:

$$T(r, z, t=0)=T_0(r, z) \tag{10}$$

The linear discretization of eqn. (5) is accomplished using the standard Galerkin's method [2, 4, 5] which yields a set of linear algebraic equations, the solution of which leads to a discrete temperature field, assuming that the heat-transfer coefficient h is known. However, in this study the heat-transfer coefficient itself is an unknown quantity to be determined. Using a linear interpolation over a period of time $t-t_0$, h may be defined as

$$h(t)=h(t-t_0)+\dot{h}(t-t_0) \tag{11}$$

where $\dot{h}$ represents the rate of change of the heat-transfer coefficient for a given period of time $t-t_0$. The rate of change of the heat transfer is determined in the following manner. An integral over the time period $t-t_0$ is formulated, the integrand of which contains the square of the difference of two average temperatures. The first of these is the arithmetic average of the temperatures calculated from eqn. (11), using an initial value of the heat-transfer coefficient for $t=t_0$ and the current value of $\dot{h}$, at the locations of the thermocouples # 5, 6 and 7. The second one is the average of the temperatures measured by the thermocouples # 5, 6 and 7. Then, the actual value of the rate of change of the heat-transfer coefficient is computed from the minimum condition of the error norm

$$J(\dot{h})=\int_{t_0}^{t}(T_{\mathrm{i}}-T_{\mathrm{m}})^2\,\mathrm{d}t \tag{12}$$

where T_{i} designates the computed average temperature and T_{m} represents the average of the temperatures measured by the thermocouples. The golden section algorithm [15] is employed to minimize integral (12) with respect to the rate of change of the heat-transfer coefficient. The resulting value of $\dot{h}$ is then used in eqn. (11) to update the current value of the heat-transfer coefficient.

4. Results and discussion

The technique, given above, has been used to calculate the interfacial heat-transfer coefficient for different normal pressures and initial temperatures of the hot die. The tests were conducted using pressures of 30, 50, 70 and 90 MPa and temperatures of 300, 500, 700 and 900°C for each pressure. These conditions make up a set of 16 experiments.

4.1. Material properties

The material properties, necessary to solve the heat-transfer problem have been expressed as functions of temperature [16]:

$$k=6.447+22.05\bar{T}-3.69\bar{T}^2 \tag{13}$$

$$c_p=277.564+1251.8\bar{T}-2149.34\bar{T}^2+1405.83\bar{T}^3 \quad \text{for } T\leqslant 575°\text{C} \tag{14}$$

$$c_p=1398.68-1923.24\bar{T}+1548.26\bar{T}^2-391.48\bar{T}^3 \quad \text{and } T>575°\text{C} \tag{15}$$

$$\rho=\frac{7897}{(1+\alpha)^3} \tag{16}$$

where

$$\alpha=-0.00358+0.009472\bar{T}+0.01031\bar{T}^2-0.002978\bar{T}^3 \tag{17}$$

and

$$\bar{T}=\frac{T+273}{1000} \tag{18}$$

To complete the boundary conditions, the pseudo heat-transfer coefficients h_a, h_c and h_h need to be specified. Measurement of temperatures during the cooling of the hot die in the atmosphere and numerical tests gave the following expression:

$$h_a=\left(1.2-0.52\frac{T}{1000}\right)\frac{T^4-T_a^4}{T-T_a}5.675\times10^{-8} \tag{19}$$

for the coefficient of heat-transfer on the cylindrical surfaces, in contact with the environment. Using a trial-and-error method, it has been found that the coefficients h_c and h_h, introduced to account for the transfer of heat between the cold die and the heat exchanger and between the hot die and the heat exchanger, may be approximated by

$$h_c=300\left(1+\frac{t}{50}\right) \tag{20}$$

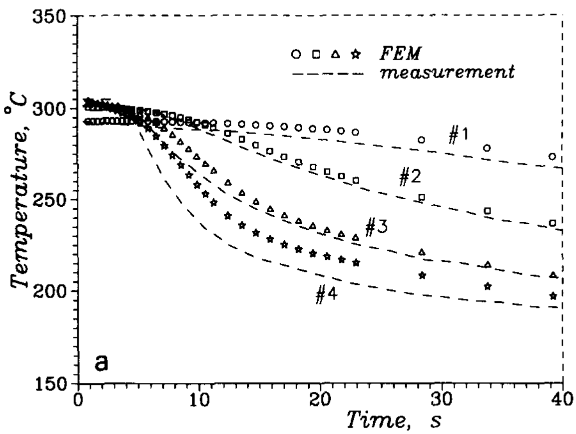
$$h_h=300 \tag{21}$$

respectively, where t is the time of contact. It is pointed out here that these heat-transfer coefficients, defined by eqns. (19)–(21), may not hold for experimental conditions different from those described in this study.

Having determined the boundary conditions and the material properties, computations have been performed to calculate the interfacial heat-transfer coefficient.

4.2. Temperature distribution

In Figs. 3 and 4 measured and computed temperatures for two typical experiments are compared, the dashed lines indicating the measurements while the symbols depict the calculations of the present method. The initial temperature of the hot die in the experiment shown in Fig. 3 was 300°C; in Fig. 4 900°C. The



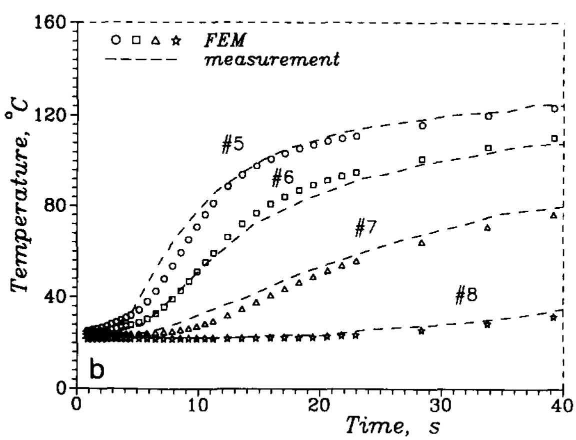


Fig. 3. Measured and computed temperature distributions versus time at the thermocouple locations for an interface pressure of 30 MPa.

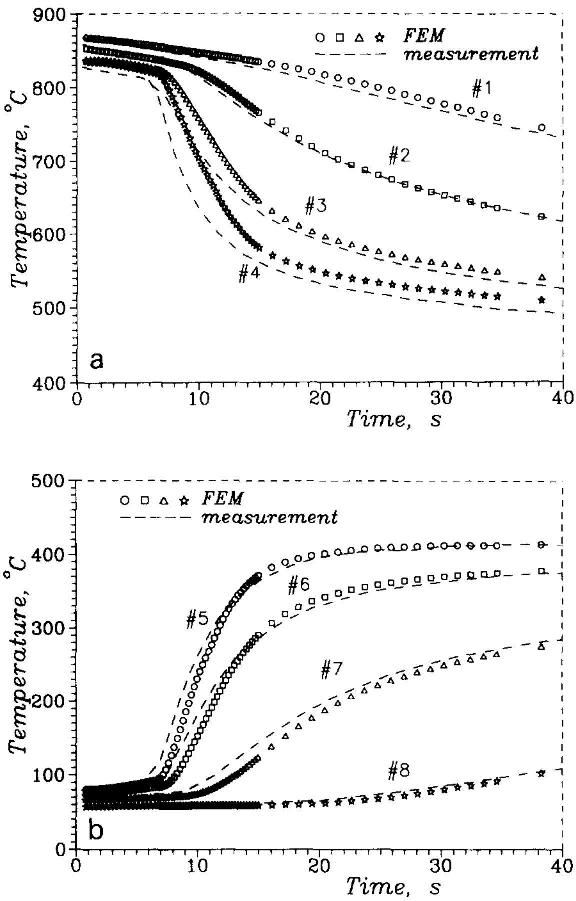


Fig. 4. As for Fig. 3 for an interface pressure of 90 MPa.

interfacial pressures in the two tests were 30 and 90 MPa, respectively. In both of the figures, (a) shows the time–temperature profile of the hot die whilst (b) gives the same for the cold die. The data acquisition system was activated as the two dies were approaching one-another and, as expected, the measurements show slow heating or cooling by all thermocouples. Contact of the dies was indicated by the sharp temperature gradients, especially of thermocouples # 4 and 5, closest to the contact surface. The full magnitude of the interfacial pressure was reached in approximately 0.5–0.8 s.

It may be concluded at this point that the measurements and the calculated time–temperature profiles agree reasonably well. The differences are observed to be small; the largest percentage difference – approximately 5% – is found during the cooling of the hot die from an initial temperature of 300°C, as indicated by the thermocouple closest to the surface. The differences elsewhere are considerably lower. The values of the heat-transfer coefficients, determined by the above technique and used to generate the temperature fields, may therefore be accepted with high confidence.

In order to compute the interfacial heat-transfer coefficient the measurements yielded by thermocouples # 5, 6 and 7 are used in the integral (12). The measurements yielded by the rest of thermocouples are used to check whether the boundary conditions determined by pseudo heat-transfer coefficients h_a, h_c and h_h are satisfied properly.

4.3. The heat-transfer coefficient

Figures 5–8 show the complete set of the results of the finite-element computations, depicting the heat-transfer coefficient as a function of the time of contact – beginning when the dies just begin to touch – for various conditions. The data are given by the following symbols: circles for 30 MPa, squares for 50 MPa, triangles for 70 MPa and stars for 90 MPa. The initial temperature of the hot die is 300°C in Fig. 5 and 500, 700 and 900°C, respectively, in the next three figures. Curves fitted to the data by careful non-linear regression analysis, to be discussed below, are given also in the figures, these being indicated by the solid lines.

4.3.1. Calculations of the heat–transfer coefficient

The heat-transfer coefficients, as computed by the inverse technique described above, vary in a broad range from 50 to 20 000 W/K m². For the first 5 s h increases at a reasonably fast rate but its magnitude is not very large, indicating the combined effects of the developing mechanisms that cause those changes. These include: the rate of rise of the pressure, reaching its full

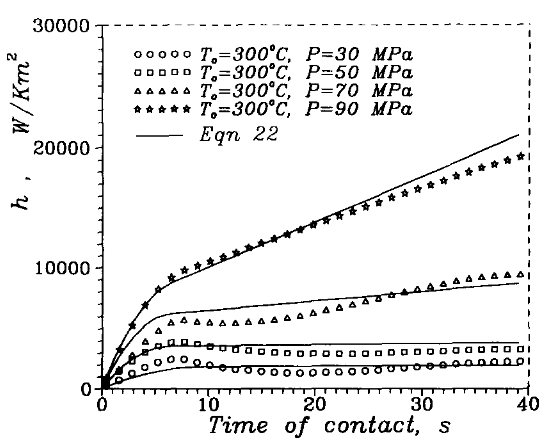


Fig. 5. The interfacial heat transfer coefficient h versus the time of contact for an initial workpiece temperature of 300°C and various normal pressures.

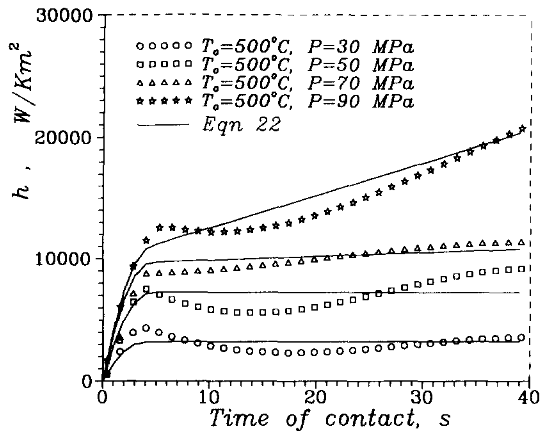


Fig. 6. As for Fig. 5 for an initial workpiece temperature of 500°C.

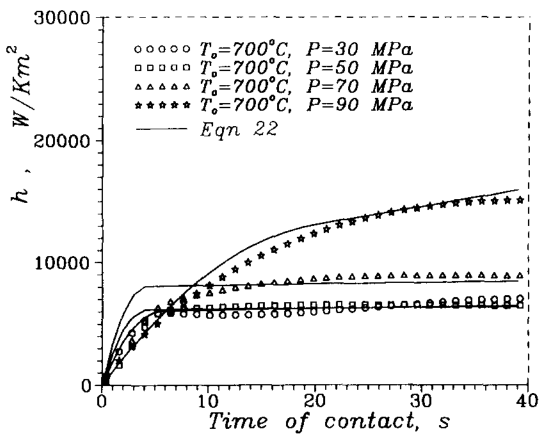


Fig. 7. As for Fig. 5 for an initial workpiece temperature of 700°C.

magnitude typically in less than 0.8 s; and the corresponding increase of the heat flux and the decrease of the difference of the temperatures of the two contacting surfaces, the ratio of which defines the coefficient of heat transfer. After approximately 5 s of contact a much slower rise of the coefficient, or steady-state conditions, are observed. The slow rise of h appears to be connected to higher interfacial pressures, regardless of the temperatures. The heat-transfer coefficient then remains constant at pressures below 70 MPa. The exception appears to be indicated by the data for the experiments in which the initial hot-die temperature was 900°C. Fast initial rise of h is still noted, but the rate of rise of the heat-transfer coefficient at the higher pressures is lower than before. The steady-state condition is achieved noticeably later.

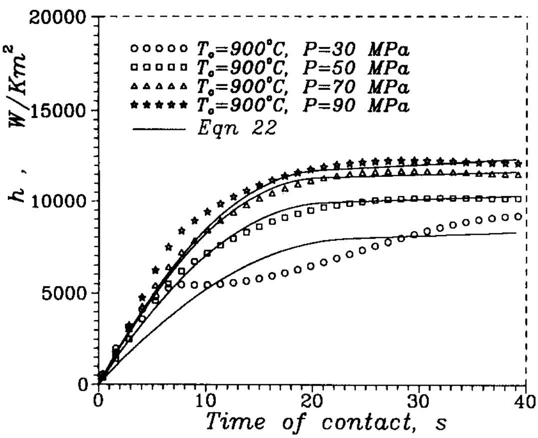


Fig. 8. As for Fig. 5 for an initial workpiece temperature of 900°C.

The heat-transfer coefficient appears to be dependent strongly on the interfacial pressures at all of the test temperatures, confirming the conclusions of [9, 11]. It does not appear to be as strongly dependent on the temperature, however. Steady-state conditions are reached in most of the experiments, indicating that the ratio of the heat flux and the interfacial temperature difference remains reasonably constant. The transfer of heat is not expected to be affected by the flattening of the asperities as this phenomenon must happen at the beginning of the test. Beyond the first few seconds of contact the true area is expected to remain unchanged.

4.3.2. Non-linear regression analysis

The experiments conducted were designed to simulate the bulk forming of metals, using a cold die, slow upset forging or cogging being the processes that are probably most closely simulated by the tests. In order to ease the choice of the interfacial heat-transfer coefficient when mathematical modelling of these processes is contemplated, empirical relations were developed, giving h as a function of time and three parameters, A, B and Q, each of which is expressed as a function of the non-dimensionalized pressure and temperature. Equation (22), below, may then be used to estimate the coefficient of heat transfer:

$$h = 1000\,[(A-2BQ)t + Bt^2] \quad \text{for } 0 < t \leqslant Q$$

$$h = 1000\,[At - BQ^2] \quad \text{for } Q < t < 40 \tag{22}$$

The predictions of eqn. (22) are valid for pressures in the range of 30–90 MPa and temperatures from 300 to 900°C. The parameters A, B and Q are defined as follows:

$$A=[-0.99861+1.39288\bar{p}+4.16282\bar{T}-1.93378\bar{p}^2-7.58453\bar{T}^2+4.85524\bar{T}^3]^2 \tag{23}$$

$$B=[-1.40191+1.50341\bar{p}+5.51342\bar{T}-0.89550\bar{p}^2-5.71863\bar{T}^2+1.19936\bar{T}^3]^2 \tag{24}$$

$$Q=[18.81105-24.37649\bar{p}-55.55330\bar{T}+35.30695\bar{p}^2+45.07317\bar{T}^2 +40.67235\bar{T}^3-16.59242\bar{p}^3-42.08732\bar{T}^4]^2+4 \tag{25}$$

where

$$\bar{p}=P/100 \tag{26}$$

except for 900°C at all pressures, and for 700°C at 90 MPa, for which the parameters A, B, Q are given by

$$A=[-3.70828+4.44037\bar{p}+5.59705\bar{T}-3.75139\bar{p}^2-3.58267\bar{T}^2-1.62445\bar{T}^3]^2 \tag{27}$$

$$B=[-0.27807+6.04270\bar{p}+0.16841\bar{T}-4.73161\bar{p}^2+4.00936\bar{T}^2-12.2651\bar{T}^3]^2 \tag{28}$$

$$Q=[-2.02052+14.04297\bar{p}-0.00530\bar{T}+1.26445\bar{p}^2-5.54230\bar{T}^2+2.02646\bar{T}^3 +0.59486\bar{p}^3-0.04034\bar{T}^4]^2+20 \tag{29}$$

The heat-transfer coefficients, as computed by eqn. (22), are given also in Figs. 5–8. It is observed that the chosen form of the empirical relations does not

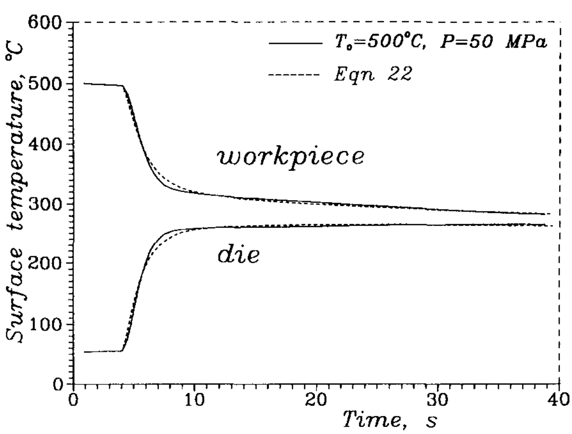


Fig. 9. Comparison of the surface temperatures of the workpiece and die computed using the calculated heat-transfer coefficients and eqn. (22).

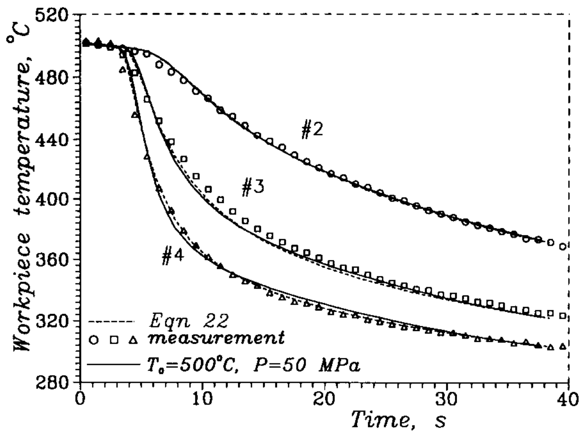


Fig. 10. Comparison of workpiece temperatures at the thermocouple locations computed using the calculated heat-transfer coefficients and eqn. (22).

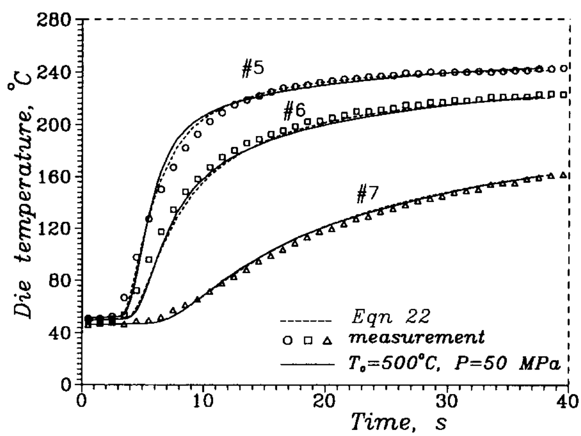


Fig. 11. Comparison of die temperatures at the thermocouple locations computed using the calculated heat-transfer coefficients and eqn. (22).

follow the data yielded by the finite-element method very closely, the differences reaching magnitudes of 10–20%. The data, indicated by the points, show first a rise, followed by a plateau and then a slight drop before reaching steady-state behaviour. The fitted curves indicate the rise and the steady-state conditions only. More complicated functions could have been chosen to model the data more closely; also, it would have been possible to model the individual curves. The objective here, however, was one of simplicity and the desire to produce a universal relationship, at least for the pressure and temperature ranges of the experiments.

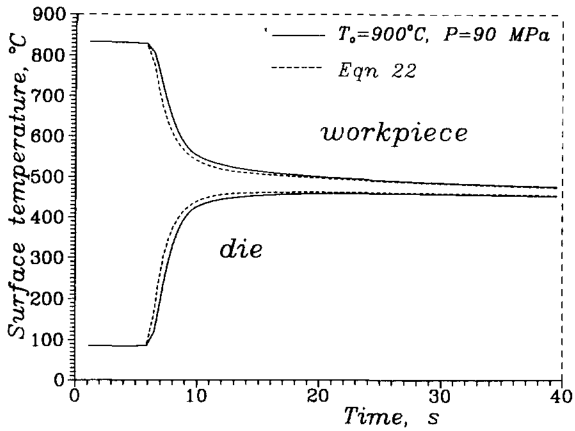


Fig. 12. Comparison of the surface temperatures of the workpiece and die computed using the calculated heat-transfer coefficients and eqn. (22).

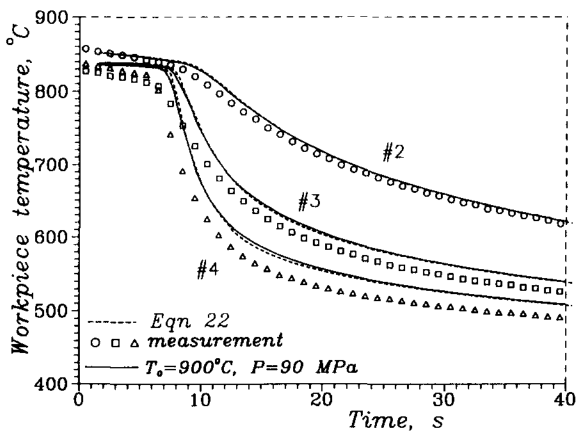


Fig. 13. Comparison of the workpiece temperatures at the thermocouple locations computed using the calculated heat-transfer coefficients and eqn. (22).

In order to show that the values of the heat-transfer coefficient, as predicted by eqn. (22), are valid and useful, they were re-substituted in the finite-element model of the process and the temperature distributions thus obtained were compared to the original data. These results, shown in Figs. 9–14, show convincing evidence that the empirical equation is capable of good predictions. Two typical cases are given in the above figures. The first set is for a pressure of 50 MPa and a temperature of 500°C. The surface temperatures of the hot and the cold dies are plotted as a function of time in Fig. 9, the solid lines indicating

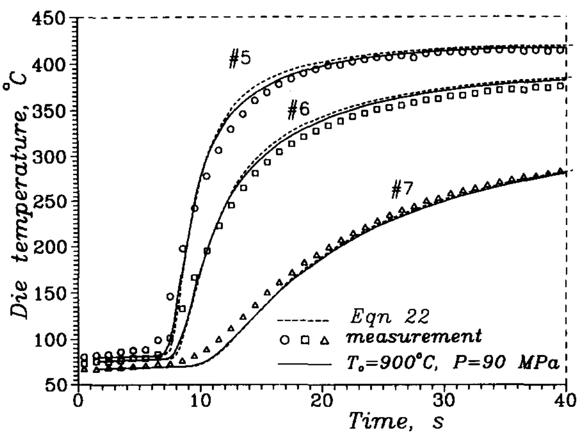


Fig. 14. Comparison of the die temperatures at the thermocouple locations computed using the calculated heat-transfer coefficients and eqn. (22).

results obtained with the originally computed heat-transfer coefficient while the dashed lines result from the use of eqn. (22). The corresponding distributions of the temperatures for the hot and the cold dies are shown in Figs. 10 and 11, respectively. Checking back to Fig. 6, it is noted that eqn. (22) and the original data differ by quite a substantial amount: differences of 25% are noted. It is observed, however, that the heat-transfer coefficients, as obtained from the empirical relationship, give distributions of the temperatures very close to those of the original results and to the actual measurements.

The second case is for a pressure of 90 MPa and a temperature of 900°C. Use of the predicted value of the heat-transfer coefficient in the model also gives very close approximations of the results. Surface temperatures are quite close and the temperature distributions of the dies are also comparable, see Figs. 12–14, respectively.

It appears that computations of the temperature distribution in the upsetting process depend on the correct value of the interfacial heat-transfer coefficient: this dependence is, however, not as pronounced as expected.

5. Conclusions

An objective technique for determining the interfacial heat-transfer coefficient in metal-forming processes has been proposed. The heat-transfer coefficient is treated as an unknown function of the time of contact, the pressure and the temperature and is determined using a least-squares approximation. The method makes use of the measured temperatures near the surface of an experimental die brought into contact with a second, hot, die. The finite-element method is employed to solve the resulting heat-transfer problem. The interface heat-transfer coefficient is obtained as a function of the time of contact, the interfacial pressure and the initial temperature of the hot die. Empirical relations are then developed, giving h as a function of the three parameters. Use of these relations in a predictive model of the process gives the distributions of the temperatures close to the measured values.

The interfacial heat-transfer coefficient is a strong function of the contact pressures, but it is not a very strong function of the temperature of the hot die. The calculated temperature distributions do not depend on the magnitude of the coefficient of heat transfer as strongly as expected, variations in h of 10% magnitude causing no more than 2–3% differences in the bulk or surface temperatures.

Acknowledgements

The financial support of the Natural Sciences and Engineering Research Council of Canada is gratefully acknowledged. The discussions with Professor M.M. Yovanovich of the University of Waterloo are greatly appreciated.

References

[1] J.C. Bruch and G. Zyvoloski, *Int. J. Numer. Meth. Eng.*, 8 (1974) 481–494.
[2] G. Comini, S. Del Guidice, R.W. Lewis and O.C. Zienkiewicz, *Int. J. Numer. Meth. Eng.*, 8 (1974) 613–624.
[3] J. Donea, *Int. J. Numer. Meth. Eng.*, 8 (1974) 103–110.
[4] R.W. Lewis, K. Morgan and O.C. Zienkiewicz, *Numerical Methods in Heat Transfer*, Wiley, New York, 1981.
[5] K.H. Huebner and E.A. Thornton, *The Finite Element Method for Engineers*, Wiley, New York, 1982.
[6] B.K. Chen, P.F. Thomson and S.K. Choi, *J. Mater. Process. Technol.*, 30 (1992) 115–130.
[7] C.M. Sellars, *Mat. Science Techn.*, 1 (1985) 325–332.
[8] P. Dadras and W.R. Wells, *ASME J. Eng. Ind.*, 106 (1984) 187–195.
[9] S.L. Semiatin, E.W. Collings, V.E. Wood and T. Altan, *Trans. ASME J. Eng. Ind.*, 109 (1987) 49–57.
[10] U. Klafs, Doctoral Thesis, Technical University of Hanover, 1969.
[11] P.R. Burte, Y.-T. Im, T. Altan and S.L. Semiatin, *Trans. ASME J. Eng. Ind.*, 112 (1990) 332–339.
[12] A.N. Karagiozis, M.Sc. Thesis, University of New Brunswick, 1986.
[13] M. Pietrzyk and J.G. Lenard, *Thermal-Mechanical Modelling of the Flat Rolling Process*, Springer, Heidelberg, 1991.
[14] M. Pietrzyk and J.G. Lenard, in: H.R. Jacobs (Ed.), *Proc. Nat. Heat. Transfer Conf.*, Houston, 1988, pp. 47–53.
[15] G.N. Vanderplaats, *Numerical Optimization Techniques for Engineering Design, With Applications*, McGraw-Hill, New York, 1984.
[16] H.E. Boyer and T.L. Gall, *Metals Handbook*, Desk Edition, ASM, Metals Park, 1985.